Research Article

Study of Transmission Loss on Muffler

[1,2]Wang Bing, [1]Wang Yongjuan and [1]XU Cheng
[1]School of Mechanical Engineering, Nanjing University of Science and Technology,
Nanjing, 210094, China
[2]School of Mechanical Engineering, Huahai Institute of Technology, Lianyungang, 222005, China

Abstract: Transmission loss is the main method which measure silencer sound deadening capacity. The most effective noise elimination technique of compensator is muffler. Muffler is equipment which organizes sound transmission and permits airflow pass and must satisfy definite muffling amount and frequency property. The study builds model and theoretically studies and analyzes the internal sound problem and contrasts between FEM and transmission matrix method and computes transmission loss. On the basis of experimental verification, combining model analysis with sound pressure distribution graph, analyze muffling property deeply.

Keywords: FEM, muffler, transmission loss, transmission matrix method

INTRODUCTION

Noise especially strong impulse noise, serious harm to the human body, the most harm auditory is organ damage and hearing loss, as well as adverse effects on the nervous, digestive and circulatory systems.

The installation of the muffler muzzle is effective muffler technology of muzzle noise. Muffler is to prevent the sound propagation and allows airflow through an apparatus, which is the main technical measure to reduce wind noise. Muzzle silencer must meet the requirements of acoustics, mechanics and usability. In the acoustic aspect, meet some of the muffler quantity and frequency characteristics. Complex muffler major research is FEM, BEM, the acoustic transfer matrix method. FEM is the most commonly used numerical method. Peat firstly introduced FEM into acoustic at the 1991 meeting of the Fifth International (Peat, 1991). Kirby applied FEM to transmission loss calculation (Kirby, 2001). Denia further applied FEM to the case of the uniform fluid flow field, in addition to these, researchers also study on the muffler from a different perspective (Denia and Selamet, 2007). Its unique advantages of the finite element method for solving muffler silencer performance and other issues, it is to overcome the traditional methods for the structure of the muffler of restrictions and directly solve within the muffler fluctuation equation, fully consider its structure, medium, temperature influence in the reasonable

selection of the boundary conditions in the case of high precision. In 1973, American scholar Papadakis took the lead with acoustic filter theory to guide the design of resistance muffler and make use of lumped parameter approximation algorithm to analyze muffler components, the method using only the wavelength is much larger than the geometric dimensions of the muffler, for low-frequency approximate calculation (Papadakis and Fowler, 1973). In 2005, Omid published a classic paper plane wave theory, using plane wave theory to analyze the acoustic characteristics of the case of no airflow muffler (Omid *et al.*, 2005).

In 2003 Bilawchuk published an article using equivalent circuit of the transfer matrix method for the calculation of the acoustic characteristics of the muffler, in the same period (Bilawchuk and Fyfe, 2003), Ahmet studied acoustic transfer matrix in the presence of mean flow. In the seventies (Ahmet and Alinur, 2005), Wu proposed theory of acoustic wave propagation in the presence of the airstream (Wu and Zhang, 1998), but still did not consider the influence of the temperature gradient. In the eighties, with further study of the airflow and temperature, more comprehensive discussion of the use of the method to calculate the design muffler are proposed. Domestic scholars also do a lot of research on transfer matrix and applied muffler acoustic performance into computing. This study will examine the acoustic characteristics of the muffler under the factors combined effects of flow rate, temperature and thermal viscous effects, to calculate the transmission loss.

Corresponding Author: Wang Bing, School of Mechanical Engineering, Nanjing University of Science and Technology, Nanjing, 210094, China

THE SILENCER ACOUSTIC PERFORMANCE EVALUATION PARAMETERS

The acoustic performance' good or bad mainly measure from its size and noise elimination frequency. Its evaluation parameters mainly include insertion loss, transmission loss and noise reduction amount.

Insertion loss: Insertion loss is the difference value of acoustic power level before and after installed from the tube mouth outwardly radiated sound power level of the difference, if the measurement point of the two measurements are the same, i.e., the insertion loss can also be expressed for the installation of the muffler of the measurement points before and after the acoustic pressure the level of the difference.

Transmission loss: Transmission loss means sound power level value difference at the entrance and the outlet, which is $TL = L_{T1} - L_{T2}$, in which, L_{T1}, L_{T2} respectively, is sound power level of the entrance of the incident and sound power level of the muffler outlet at (no reflective end).

Under normal circumstances, the muffler inlet duct cross-sectional area and environment is the same. So above equation simplifies to:

$$TL = 20 \log \left| \frac{T_{11} + T_{12}/Y + T_{21}Y + T_{22}}{2} \right|$$

Y-impedance, $Y = c/S$, *c*-sound velocity, *S*-cross-section area of the pipe.

Muffler transmission loss is the nature of the muffler itself, only is the relevant with the structure of the muffler and has nothing to do with the velocity of sound in the outside of the muffler, so transmission loss of the evaluation parameters is used for theoretical studies, it's difficult to measure in practice.

Since insertion loss is only need to be measured before and after the muffler is installed in the number of a fixed measuring point SPL and easy on-site measurement, therefore it is the most commonly used performance evaluation target of the muffler, but it is not only concerned with the performance of noise reduction, but also related to impedance of the outlet end and is not easy to theoretical predictions. On the contrary, the transmission loss of the theoretical prediction is relatively easy, because the characteristics of the transmission loss is only the reaction of the transmission characteristic of the muffler itself, without the influence of sound source piping system and outlet tail pipe, that is the sound source the outlet end of the impedance contents, means that the transmission loss is the approximation of the muffler actual performance; but if the sound source and the end of the muffler to be seen as no echo, transmission loss and insertion loss should be equal.

TRANSMISSION MATRIX

For an acoustic unit, if the acoustics amount of the unit front portion is input, the acoustics amount of the system behind portion is output, it can be established transfer function based on the transfer matrix:

$$\begin{pmatrix} P_1 \\ U_1 \end{pmatrix} = \begin{pmatrix} A & B \\ C & D \end{pmatrix} \begin{pmatrix} P_2 \\ U_2 \end{pmatrix} = [T] \begin{pmatrix} P_2 \\ U_2 \end{pmatrix}$$

where,
P_1, P_2 = Sound pressure of input and output of muzzle
U_1, U_2 = Sound volume velocity of input and output of muzzle

For the complex structure of the muffler, it can be decomposed into simple combination of elements, the total transfer matrix is the accumulation of each unit transmission matrix.

Considering thermal viscous effect rigid straight tube:

$$[T_1] = \frac{e^{-jK_d kl}}{h^+ + h^-} \begin{bmatrix} T_{11} & j2 \sin K_c kl \\ j2h^+ h^- & T_{22} \end{bmatrix}$$

while,

$$T_{11} = (h^- + h^+) \cos K_c kl + j(h^- + h^+) \sin K_c kl$$
$$T_{22} = (h^+ + h^-) \cos K_c kl + j(h^+ - h^-) \sin K_c kl$$
$$K^{\pm} = K_0/(1 \pm K_0 M)$$
$$h^{\pm} = K_0(1 + ((1-j)/s)\sqrt{2(1 \pm K_0 M)})$$
$$M = v_0/c$$

M = Mach number
v_0 = Straight tube gas velocity

$$K_d = \frac{K_0^2 M}{1 - K_0^2 M^2} \quad K_c = \frac{K_0}{1 - K_0^2 M^2}$$
$$K_0 = 1 + \frac{1-j}{\sqrt{2}} \frac{\gamma - 1 + \sigma}{\sigma s}$$

while,
γ = Heat transfer ratio:

$$\sigma^2 = \mu C_p / K$$

σ = Prandtl coefficient
C_p = Isobaric specific heat capacity
μ = Shear viscosity coefficient
K = Thermal conductivity coefficient, while approximated muzzle muffler Prandtl number 0.5
k = wave number
$k = 2\pi f/c$, *c*-sound velocity, *f*-frequency

Expansion chamber: The acoustic focus of the study of the expansion chamber lies in the sectional portion mutations, the same as the processing method of the expansion chamber with inner tube, simple expansion chamber can be seen as the case of the expansion chambers of the inner tube in the band when the length of the inner tube is zero, so the expansion chamber with inner tube is the object of study.

Section sudden contraction structure: The cross-sectional area of the upstream pipe of the cross-sectional structure of sudden contraction is S_1, the downstream duct cross-sectional area of the tube is S_2, the length of tube is l, the airflow of tube resonator is regarded as no speed.

Sudden contraction of the pipe cross-sectional area of the pipe make airflow destroyed, generates complex flow including whirlpool sound field, in this case also becomes very complicated and will provoke many higher order wave here because the theoretical restrictions can only be approximate to study the changes in the sound field:

$$[T]_3 = \begin{bmatrix} 1 & 1 \\ h_1^+ & -h_1^- \end{bmatrix} [A]^{-1} [B] \begin{bmatrix} 1 & 1 \\ h_2^+ & -h_2^- \end{bmatrix}^{-1}$$

$$[A] = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix} \quad [B] = \begin{bmatrix} B_{11} & B_{12} \\ B_{21} & B_{22} \end{bmatrix}$$

$$A_{11} = S_1(1/z + (2M/z - 1)h_1^+ + M(M/z - 1)g_1^+)$$
$$A_{12} = S_1(1/z - (2M/z - 1)h_1^- + M(M/z - 1)g_1^-)$$
$$A_{21} = 1 + e_1^+ + Mh_1^+$$
$$A_{22} = 1 + e_1^+ - Mh_1^-$$

$$B_{11} = S_2[1/z + (2M/z - 1 + (\gamma - 1)\xi M^2(M/z + 1))h_2^+ + M(M/z - 1)g_2^+]$$
$$B_{12} = S_2[1/z - (2M/z + 1 + (\gamma - 1)\xi M^2(M/z - 1))h_2^+ + M(M/z - 1)g_2^-]$$
$$B_{21} = 1 + e_2^+ + M(1 + \xi)h_2^+$$
$$B_{22} = 1 + e_2^+ - M(1 + \xi)h_2^-$$

$$z = p_3 / \rho_0 c u_3$$
$$h^\pm = K_0(1 + ((1 - j)/s)\sqrt{2(1 \pm K_0 M)})$$
$$e^\pm = \frac{1-j}{\sigma s}\sqrt{2(1 \pm K_0 M)}$$
$$g^\pm = 1 - ((1 - j)/s)((\gamma - 1)/\sigma \sqrt{2(1 \pm K_0 M)})$$

$\zeta =$ Local pressure loss coefficient caused by sectional mutations.

ORIGINAL MODEL

Muffler is a six-cavity and one tube structure, as into the left entrance, right end, the structure shown in Fig. 1 (Wang *et al.*, 2010).

The muffler is a resistance muffler, the pipe is the use of different shapes and sizes, resulting in impedance mismatch of the acoustic propagation, make it reflection at its junction cross-section mutation in the acoustic characteristics, part of the acoustic wave to the reflected by the sound source direction, leaving part continues to propagate forward, to achieve the purpose of noise reduction:

$$T = [T_1]_1[T_2]_2[T_3]_3[T_4]_1[T_5]_2[T_6]_3$$
$$[T_7][T_8]_2[T_9]_3[T_{10}]_1[T_{11}]_2[T_{12}]_3 \tag{1}$$

- Rigid straight tube substructure
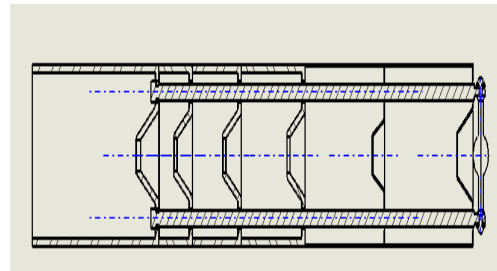- Intubation Section sudden contraction substructure

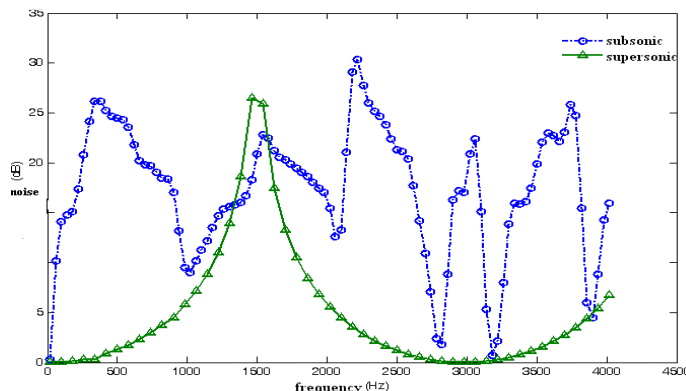Fig. 1: Resistance mufflers structure diagram

Fig. 2: The transmission loss of the transfer matrix method for calculation of subsonic and supersonic

- The expansion cavity sudden expansion substructure
- Using substructural acoustic transfer matrix, the transmission loss is shown in Fig. 2:

$$TL = 20 \log \left| \frac{T_{11} + T_{12} / Y + T_{21}Y + T_{22}}{2} \right|$$

Y-tail pipe acoustic radiation impedance, $Y = c/S$, while *c*-sound velocity, S-cross-sectional area of the pipe. Using Eq. (1) derivation of each sub-structure of the acoustic transfer matrix, work out the transmission loss program. The application can be compiled from the muffler transmission loss in the range of 20-4000Hz.

As can be seen from Fig. 2, the average noise reduction is about 15dB in the range of 20-4000Hz, unstable, has passing frequency, but supersonic passing frequency is small, the average noise reduction is about 10dB. The greater the airflow velocity, the more noise reduction peak decreased.

FEM MODEL

Modeling principles: Establish muffler sound field finite element model due to the complexity of the muffler sound field structure must be a reasonable assumption and the sound field boundary plus the appropriate boundary conditions, the sound pressure of the muffler inlet and export is calculated and thus obtain the muffler transmission loss. This study use Solid works to create a geometric model of the muffler, use Gambit to build FEM model and Hypermesh mesh, SYSNOISE sound field analysis and calculation of the transmission loss of the muffler. Thinking the muffler internal as non-coupled sound field, that ignored the role of fluid and structure coupling, so only the muffler internal cavity modeling.

Model mesh density must be sufficiently detailed to be able to distinguish the highest frequency, unit side
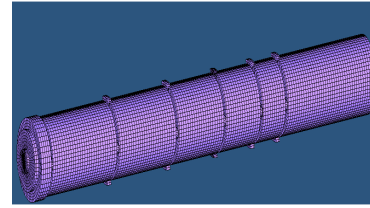

Fig. 3: FEM model of muffler

length selected to ensure the calculation accuracy, the relationship of minimum side length L_e and the highest calculated frequency f_{max} is $L_e = 0.09 \, c/f_{max}$.

Human ear audible frequency range is 20Hz-20000Hz, 1000-4000Hz frequency range of the human ear is sensitive. the higher the calculated frequency, the smaller the unit edge length, the greater the number of the unit, for the calculation of the performance of the silencer of the muffler itself, namely, transmission losses, it's necessary to get incidence sound pressure at the entrance, the sound pressure of the nonreflected wave at the outlet. Therefore, the entrance to increase the unit velocity incentive and sound absorption coefficient 1 of export is applied for all absorbing boundary. The model shown is Fig. 3 and transmission loss is in Fig. 4.

Figure 4 is the graph of the subsonic transmission loss of the two calculation methods and experimental, the map can be seen in the entire frequency range, results using FEM calculation are in good agreement with the experimental results show that FEM can accurately predict the performance of the resistance muffler.

Slightly lower than the experimental results in the high frequency domain calculation results, which can be summarized as finite element calculations ignore the viscous damping effect of the medium. And using the transfer matrix method, it can be seen that the transmission loss curve near the low-frequency the calculated results of both the difference is small, but to
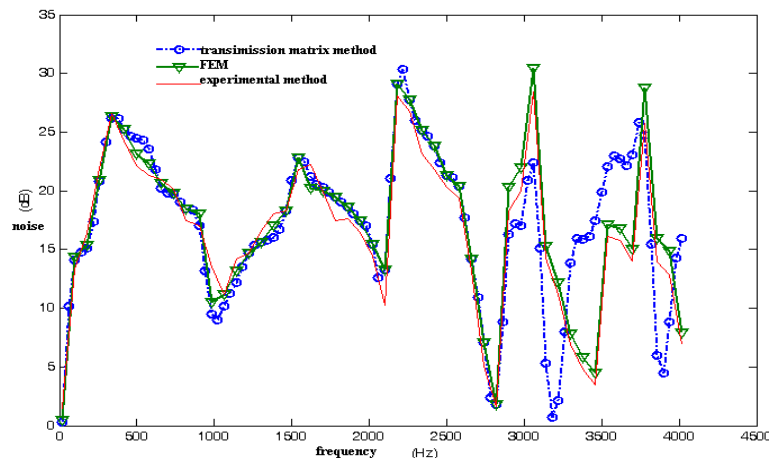

Fig. 4: Transmission loss diagram of subsonic

the high frequencies, the results vary widely, the results obtained using the transfer matrix method only at low frequencies the results are accurate and with the advent of high-frequency mode wave and then the results of the calculation using the transfer matrix method is not credible and passing frequency at the high-frequency domain is more.

CONCLUSION

- This study predict transmission losses to muzzle muffler, using transmission matrix method for subsonic and ultrasonic speed projectile carried out the analysis and calculation, average noise reduction at the range of 20-4000Hz is 15dB, unstable, having passing frequency, but passing frequency of supersonic is very small, the average noise reduction is 10dB. The greater the airflow velocity, the more noise elimination peak decreased.

- Compared the finite element method and the transfer matrix method with experimental results. In the entire frequency range, results using FEM calculation are in good agreement with the experimental results show that FEM can accurately predict the acoustic performance of resistance muffler. The results show that using the transfer matrix method only at the low-frequency results are accurate and with the advent of high-frequency mode wave and then using the transfer matrix method for the calculation of the result is not credible and the passing frequency is much in high-frequency domain.

REFERENCES

Ahmet, D. and B. Alinur, 2005. Wiener-hopf approach for predicting the transmission loss of a circular silencer with a locally reacting lining. Int. J. Eng. Sci., 43: 398-416.

Bilawchuk, S. and K.R. Fyfe, 2003. Comparison and implementation of the various numerical methods used for calculating transmission loss in silencer systems. Appl. Acoust., 64: 903-916.

Denia, F.D. and A. Selamet, 2007. Acoustic attenuation performance of perforated dissipative mufflers with empty inlet/outlet extensions. J. Sound Vib., 302: 1000-1017.

Omid, Z., A. Mehdizadeh and P. Marius, 2005. A three-dimensional finite element approach for predicting the transmission loss in mufflers and silencers with no mean flow. Appl. Acoust., 66: 902-918.

Papadakis, E.P. and K.A. Fowler, 1973. Ultrasonic attenuation by spectrum analysis of pulses in buffer rods: Method and diffraction corrections. J. Acoust. Soc. Am., 53: 1336-1343.

Peat, K.S., 1991. A transfer matrix for an absorption silencer element. J. Sound Vib., 146: 353-360.

Wang, B., W. Yongjuan and C. Xu, 2010. Experimental study on muzzle noise suppression of minor-caliber bullet. Nanjing Ins. Sci. Tech. Nat. Sci., 35(4): 160-163.

Wu, T.W. and P. Zhang, 1998. Boundary element analysis of mufflers with an improved method for deriving the four pole parameters. J. Sound Vib., 217(4): 656-668.