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

Acoustic attenuation analysis of circular dual-chamber mufflers with non-uniform flow

Huang, Hongpu[1]
School of Power and Energy Engineering, Harbin Engineering University
Harbin, Heilongjiang 150001, P. R. China
Guangxi Liugong Machinery Co., Ltd.
Liuzhou, Guangxi 545007, P. R. China

Ji, Zhenlin[2]
School of Power and Energy Engineering, Harbin Engineering University
Harbin, Heilongjiang 150001, P. R. China

ABSTRACT

The frequency-domain linearized Navier-Stokes equations (LNSEs) are employed to predict the acoustic attenuation performance of circular dual-chamber mufflers with non-uniform flow. The steady flow in mufflers is calculated by computational fluid dynamics (CFD) and then transferred to non-consistent acoustic mesh from CFD mesh. Finally, the finite element method of LNSEs is adopted to predict acoustic attenuation performance of muffler. This paper focuses on the effects of various configurations on acoustic attenuation performance of mufflers in the presence of flow, including (1) the number of inner tubes, (2) the length of inner tubes, and (3) the non-coaxial tubes. The numerical predictions show that the flow changes the acoustic attenuation performance of mufflers.

Keywords: Acoustic attenuation, circular dual-chamber mufflers, non-uniform flow
I-INCE Classification of Subject Number: 34

1. INTRODUCTION

In order to reduce the noise inside the pipeline system, the muffler with multi-chambers has been widely used as a common acoustic element. Among the multiple chambers mufflers, the dual-chamber mufflers generally have the wider acoustic attenuation dome and the fewer number of pass-bands in the curve of transmission loss[1].

In order to predict the acoustic attenuation performance of dual-chamber mufflers, researchers have presented different analytical and numerical methods, such as transfer matrix method[2], mode matching technique[3,4], boundary element method[5-7], finite element method[8], finite volume method[9]. Although the transfer matrix method is efficient in calculation, it is only suitable in the plane wave frequency range.

[1] hhpliuzhou@163.com
[2] jizhenlin@hrbeu.edu.cn

Therefore, the multi-dimensional analytic and numerical methods are required for the accuracy prediction of acoustic attenuation performance at higher frequencies considering the effect of high order modes. Selamet et al[4] employed a two dimensional (2-D) axisymmetric analytical approach based on the mode-matching technique to investigate the acoustic behavior of a circular dual-chamber muffler. Ji[6]developed a three dimensional substructure boundary element technique to predict the acoustic attenuation performance of multi-chamber mufflers. Liu et al[8] used the finite element method to predict the the acoustic attenuation performance of circular dual-chamber mufflers. Middelberg et al[9] applied the CFD software based on the finite volume method to determine the acoustic response of expansion chamber mufflers. The above-mentioned works[4,6,8,9] discussed the effect of various internal geometry, including the number of chamber, the length of inner tubes, as well as the orientation of inlet tube, on the acoustic attenuation performance of the mufflers. In the absence of flow, it was found that these effects change the acoustic attenuation performance of the mufflers drastically.

The researches on acoustic behaviors of multi-chamber mufflers with flow are very limited. Ji et al[7] applied the boundary element method to predict the acoustic performance of expansion chamber mufflers with uniform flow. For more complex inner geometry, the assumption of uniform flow inside mufflers may be not reasonable for the acoustic attenuation prediction. In order to consider the non-uniform flow inside mufflers, Liu and Ji [10] employed the time-domain CFD method to predict the acoustic attenuation performance of perforated tube mufflers. Their predictions agreed well with the measurements. However, the huge computation cost makes that the 3-D time-domain CFD method is not suitable for practical design of silencers. In recent years, some researchers have linearized all of equations governing fluid flow to describe sound propagation in the ducts with flow. Kierkegaard et al[11] found that this frequency-domain linearized Navier-Stokes equations (LNSEs) approach is efficient and accurate.

The objective of the present study is then (1) to employ a numerical method based on the frequency-domain linearized Navier-Stokes equations to predict the acoustic attenuation performance of circular dual-chamber mufflers with non-uniform flow; (2) to investigate the effects of various configuration, including the number of inner tubes, the length of inner tubes and the non-coaxial tubes on acoustic attenuation performance of the mufflers. The results of this study provide a deeper insight into understanding the dual-chamber mufflers in the presence of non-uniform flow.

2. NUMERICAL APPROACH

In this paper, LNSEs are formulated in the frequency domain and the harmonic variation is assumed. The method considers the dependent variable $A(\boldsymbol{x},t)$ as the sum of mean flow part $A_0(\boldsymbol{x})$ and acoustic perturbation $A_1(\boldsymbol{x})e^{\mathrm{j}\omega t}$. To linearize all of equations governing fluid flow and neglect the higher order acoustic variables, the frequency domain LNSEs can be written as

$$\mathrm{j}\omega\rho_1 + \nabla\cdot\left(\rho_0\boldsymbol{u}_1 + \rho_1\boldsymbol{u}_0\right) = M \tag{1}$$

$$\rho_0\left(\mathrm{j}\omega\boldsymbol{u}_1 + \left(\boldsymbol{u}_0\cdot\nabla\right)\boldsymbol{u}_1 + \left(\boldsymbol{u}_1\cdot\nabla\right)\boldsymbol{u}_0\right) + \rho_1\left(\boldsymbol{u}_0\cdot\nabla\right)\boldsymbol{u}_0 = \nabla\cdot\boldsymbol{\sigma}_1 \tag{2}$$

$$\boldsymbol{\sigma}_1 = -p_1\mathrm{I} + \mu\left(\nabla\boldsymbol{u}_1 + \left(\nabla\boldsymbol{u}_1\right)^T\right) + \lambda\left(\nabla\cdot\boldsymbol{u}_1\right)\mathrm{I} \tag{3}$$

$$\begin{aligned}&\rho_0 C_p\left(\mathrm{j}\omega T_1 + \left(\boldsymbol{u}_0\cdot\nabla\right)T_1 + \left(\boldsymbol{u}_1\cdot\nabla\right)T_0\right) + \rho_1 C_p\left(\boldsymbol{u}_0\cdot\nabla\right)T_0\\&-\alpha_0 T_0\left(\mathrm{j}\omega p_1 + \left(\boldsymbol{u}_0\cdot\nabla\right)p_1 + \left(\boldsymbol{u}_1\cdot\nabla\right)p_0\right) - \alpha_0 T_1\left(\boldsymbol{u}_0\cdot\nabla\right)p_0 = -\nabla\cdot\left(-k\nabla T_1\right)\end{aligned} \tag{4}$$

$$\rho_1 = \rho_0\left(\beta_T p_1 - \alpha_0 T_1\right) \tag{5}$$

where, the subscripts ‘0’ and ‘1’ represent background mean flow part and acoustic perturbation.

To implement the finite element method of LNSEs, a weak formulation is created from the Equation 1-5 by multiplication with test function, respectively and followed by integration over the domain. With stokes’ divergence theorem, the integrated formulation reduces all second order derivatives to first order derivatives. The boundary conditions of simulation can be found in the boundary integration term.

Figure 1 shows the overview of simulation conditions for predicting acoustic behavior of mufflers. The subscripts ‘+’ and ‘-’ represent the forward moving wave and backward moving wave in the sound propagation. The superscripts ‘u’ and ‘d’ denote the upstream and downstream of muffler, respectively. Only plane waves are assumed to be present in the ends of upstream and downstream. Therefore the relation between sound pressure and particle velocity is expressed by Equation 6. For the evaluation of acoustic behavior of muffler, the transmission loss (TL) in Equation 7 is used.

$$p^{+} = \frac{p_1 + \rho_0 c_0 v_1}{2} \tag{6}$$

$$TL = 20\log 10\left(\left|\frac{p_u^{+}}{p_d^{+}}\right|\right) \tag{7}$$

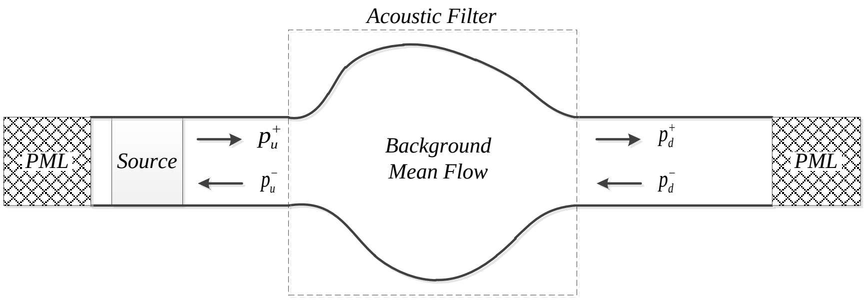

Figure 1. Overview of simulation conditions of muffler.

As an acoustic excitation, the mass source term M in LNSEs is applied over the specific domain of upstream of muffler. In this paper, Gaussian pulse function is employed to set the source distribution along the axis direction. The magnitude of the function reaches its maximum in the middle position of the source domain and equals zero at edges. Perfect matched layers (PMLs) are arranged at the upstream and downstream of muffler, which is employed to absorb the reflected sound from ends. The slip wall boundary conditions are employed as bellow

$$\boldsymbol{u}_1 \cdot \boldsymbol{n} = 0 \tag{8}$$

where, $\boldsymbol{n}$ is normal direction of walls.

3. RESULTS AND DISCUSSION

In this section, the seven configurations of dual-chamber mufflers in Figure 2 will be employed to investigate the effect of internal geometry and inlet orientation on acoustic attenuation performance of mufflers. The chambers have a diameter of D=152.9 mm and the length of l_{s1}= l_{s2}= 141.2 mm. The inlet and outlet tubes have the diameter of d=49.3 mm. The extended length of outlet tube into the right chamber is l_o=25 mm and the extended lengths of inner tubes into the chambers are l_{c1}=25 mm and l_{c2}=59 mm, respectively. The rest structural dimensions of dual-chamber mufflers are shown in Table 1. For the configurations C and D, the extended length of inlet tube into the left chamber is l_e=47.2 mm. For the configuration G, the extended lengths of shorter inner tube are l_{c3}=12.5 mm and l_{c4}=29.5 mm, respectively.

In this paper, some background mean variables in LNSEs is spatial-dependent, such as $\boldsymbol{u}_0$, p_0, μ. The rest background mean variables, such as T_0 and ρ_0, are set as homogeneous inside mufflers. The maximum size of the acoustic mesh of mufflers is less than 4 mm. The background mean variables inside silencer are calculated first by the commercial software ANSYS Fluent with k-ε turbulence model and then transferred from CFD mesh to non-consistent acoustic mesh. The LNSEs can be solved by using the commercial software COMSOL Multiphysics to evaluate the acoustic attenuation performance of mufflers.

In order to validate the accuracy of the LNSEs method, the numerical prediction (blue curve) and the published measurement[4] of transmission loss for the configuration A are shown in Figure 3. The comparison between these results exhibits a good agreement. There are some high spikes in the transmission loss curve. According to the theory of transfer matrix method, frequencies of the spikes correspond approximately to the extended lengths of tubes into the chambers. Considering the length and end-correction of extended tubes into the chambers, the resonance frequencies near 1150 Hz and 2550 Hz are verified by the theory of quarter-wave resonator. The resonance frequencies near 1600 Hz may correspond to the theory of Helmholtz resonator. In order to validate the guess, the extended length is adjusted to 20 mm. The predicted result (green curve) in Figure 3 shows that the resonance frequency is shifted to higher frequency. It is attributed that the volume of the Helmholtz resonator decreases due to the reduction of extended length of outlet tube.

Figure 5 shows the transmission loss of the seven configurations of dual-chamber mufflers with non-uniform flow. As the inlet Mach number is M=0.2, there is obviously numerical error in the low frequency range, especially for those asymmetric configurations B, F and G. Figure 4 shows the amplitude of sound pressure and reflected sound pressure along the axis of outlet tube of the configuration B at the frequencies of 100Hz, 1000Hz and 2000Hz. The reflected sound pressure is obtained by using the plane wave decomposition with mean flow in the outlet tube. It is found that the PML absorbs inefficiently the reflected sound in the low frequency range. This causes the inaccurate calculation of transmission loss. However, the quick fluctuation of curve does not affect the analysis of acoustic performance. Comparing the acoustic attenuation performance of mufflers with end-inlet and side-inlet, the similar acoustic behavior could be found.

Table 1. Structural dimensions of various configurations of dual-chamber mufflers. (Units: mm)

Dimension	Configurations						
	A	B	C	D	E	F	G
d_i	49.3	34.8	49.3	34.8	28.5	49.3	34.8
δ	0	42.5	0	42.5	42.5	42.5	42.5
l_i	59	59	47.2	47.2	59	59	59

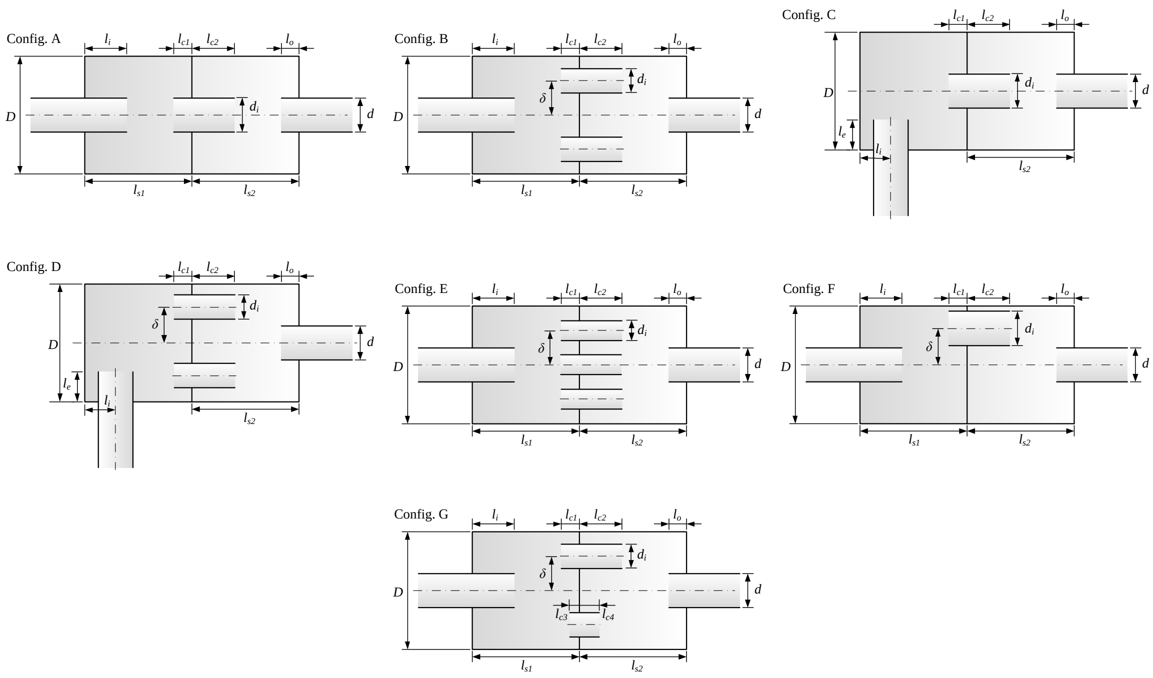

Figure 2. Various configurations of dual-chamber mufflers.

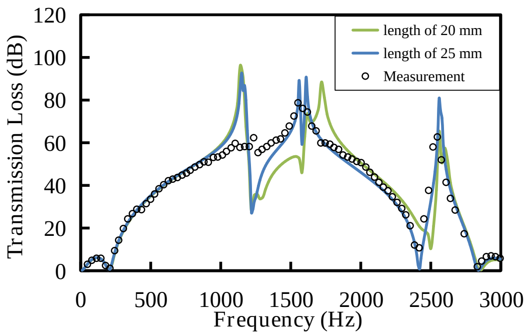

Figure 3. Transmission loss of the configuration A of dual-chamber mufflers without flow:the extended lengths of outlet tube into the right chamber are 20 mm and 25 mm.

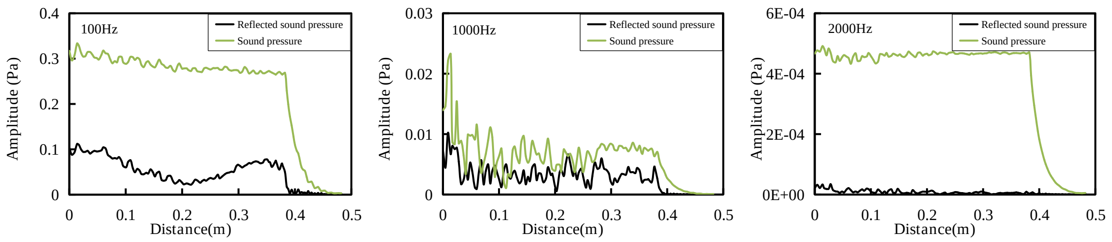

Figure 4. Amplitude of sound pressure and reflected sound pressure along the axis of outlet tube of the configuration B at the frequencies of 100Hz, 1000Hz and 2000Hz(inlet Mach number M=0.2).

With the increase of flow velocity, the acoustic attenuation performance of most configurations has changed. These effects are mainly manifested in the following three aspects: (1) the shift of resonance frequencies, (2) the decrease of amplitude of TL at resonance frequencies and the increase of amplitude of TL at some frequencies, (3) the decrease of amplitude of TL in the low frequency range for the configurations A and E. The first aspect is attributed to that (a) the end-corrections of extended tubes into the chambers do not remain the same when the flow velocity changes, (b) the different inner geometries of the configurations affect resonance frequencies in the range from 1230 Hz to 2430 Hz, which correspond to the theory of Helmholtz resonator. Because the acoustic impedance may change compared to the case without flow, it may cause the change of TL amplitude. The last aspect may be due to the obvious discrepancy of the inner geometry between the configurations A/E and the rest. The configurations A/E have the coaxial inlet, outlet and inner tubes.

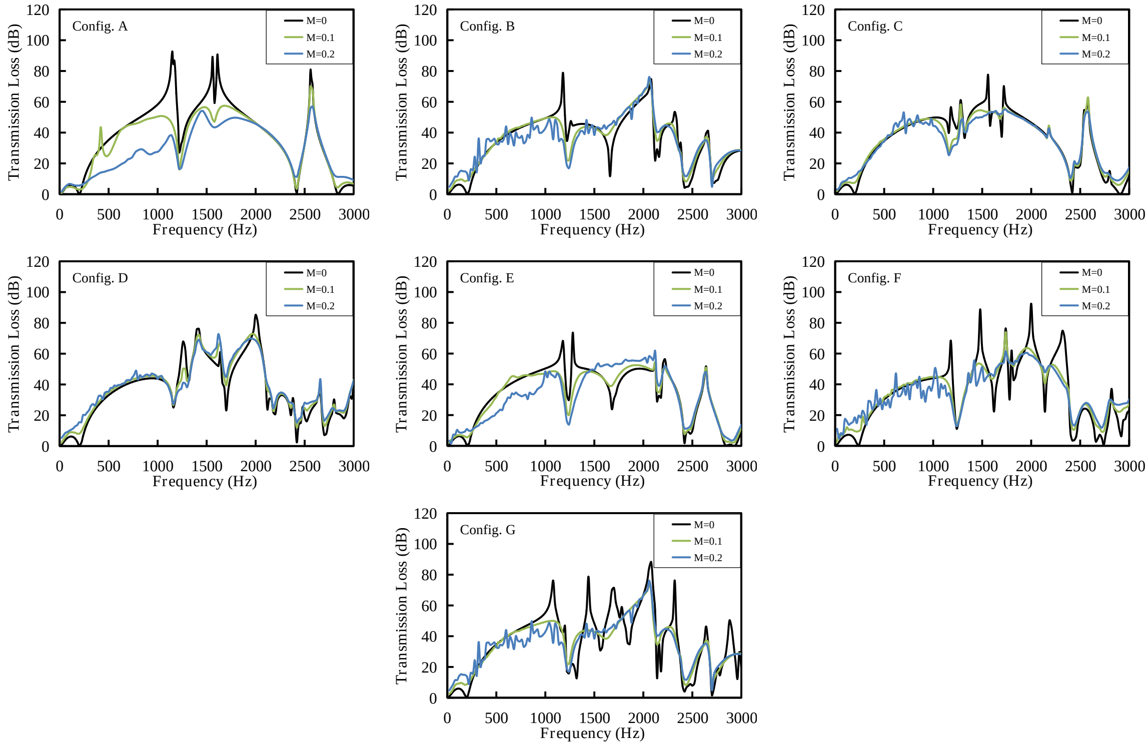

Figure 5. Transmission loss of various configurations of dual-chamber mufflers with non-uniform flow

4. CONCLUSIONS

In the present study, the frequency-domain linearized Navier-Stokes equations are employed to predict the acoustic attenuation performance of circular dual-chamber mufflers with non-uniform flow. As the Mach number M=0.2, the numerical error is visible compared to the cases of M=0 and 0.1. However, the quick fluctuation of curve does not affect the analysis of acoustic performance.

The effects of inner geometry and flow velocity have been investigated. The changes of inner geometry of the various configurations shift the resonance frequencies of mufflers. The end-corrections of extended tubes into the chambers and the acoustic impedance are affected by the flow velocity inside mufflers. The two aspects also change the acoustic attenuation performance of mufflers. In the low frequency range, the acoustic attenuation performance of the coaxial configurations is drastically affected by the high Mach number flow.

5. ACKNOWLEDGEMENTS

The authors acknowledge the support of the research grant 11674076 from National Natural Science Foundation of China.

6. REFERENCES

1. M.L. Munjal, “*Acoustics of ducts and mufflers*”, John Wiley & Sons(2014)

2. M.L. Munjal, “*Plane wave analysis of side inlet/outlet chamber mufflers with mean flow*”, Applied Acoustics 52.2 (1997): 165-175

3. A. Selamet, Z.L. Ji, “*Acoustic attenuation performance of circular expansion chambers with extended inlet/outlet*”, Journal of Sound and Vibration 223.2 (1999): 197-212

4. A. Selamet,F. D. Denia, and A. J. Besa, “*Acoustic behavior of circular dual-chamber mufflers*”, Journal of Sound and Vibration 265.5(2003):967-985

5. A. Selamet, Z.L. Ji, “*Case study: Acoustic attenuation performance of circular expansion chambers with extended end-inlet and side-outlet*”, Noise Control Engineering Journal 48.2(2000):60

6. Z.L. Ji, “*Acoustic attenuation performance analysis of multi-chamber reactive silencers*”, Journal of Sound & Vibration 283.1(2005):459-466

7. Z.L. Ji, Q. Ma, and Z.H. Zhang, “*Application of the Boundary Element Method to Predicting Acoustic Performance of Expansion Chamber Mufflers With Mean Flow*”, Journal of Sound and Vibration 173.1(1994):57-71.

8. C. Liu, Z.L. Ji, and Z. Fang, “*Numerical analysis of acoustic attenuation and flow resistance characteristics of double expansion chamber silencers*”, Noise Control Engineering Journal 61.5(2013):487-499.

9. J.M. Middelberg, T.J. Barber, S.S. Leong, K.P. Byrne and E. Leonardi, “*Computational fluid dynamics analysis of the acoustic performance of various simple expansion chamber mufflers*”, Proceedings of acoustics(2004)

10. C. Liu, Z.L. Ji, “*Computational Fluid Dynamics-Based Numerical Analysis of Acoustic Attenuation and Flow Resistance Characteristics of Perforated Tube Silencers*”, Journal of Vibration and Acoustics 136.2(2013):021006.

11. A. Kierkegaard, S. Boij, G. Efraimsson, “*A frequency domain linearized Navier-Stokes equations approach to acoustic propagation in flow ducts with sharp edges*”, The Journal of the Acoustical Society of America 127.2 (2010): 710-719