A Study on the Analysis and Improvement of the Acoustic Characteristics of the Muffler with Complex Geometry

by

*Jin-Yong Mo, Man-Sun Huh, Jung-Rae Kim,
Heui-Jong Kang, and Sang-Kyoung Oh
Noise and Vibration Group,
SAMSUNG ELECTRONICS Co., Suwon, KOREA

ABSTRACT

An acoustic transmission loss analysis method for mufflers with complex geometry is developed using MSC/NASTRAN on the basis of acoustic-structural analogy and two-microphone method. In this study, mufflers with simple and complex shapes are analyzed using this method and compared with theoretical and experimental results for verification.

Applying this method to design of suction muffler of reciprocating compressor, we could obtain 4dB(A) total noise reduction, especially more than 10dB(A) noise reduction at 500Hz.

1. Introduction

The prediction of transmission loss(TL) in mufflers has been accomplished by plane wave theory[1] or acoustic four-pole parameter method[2] with FEM. However, plane wave theory does not provide a theoretical calculation for mufflers with complex geometry. It is the reason that the simple one-dimensional theory does not give an explanation of complex wave phenominum due to complicated geometry.

To overcome these defects, an acoustic four-pole parameter method was developed using FEM. The method provides the general solution with respect to the geometry and properties of the medium. But it takes too much time to calculate for each four-pole parameter of the muffler which have multi-inlet and multi-outlet.

Recently, an experimental method, wave seperation method[3][4] by two microphones is used to estimate the transmission loss.

In this paper, we developed a method to predict the transmission loss of mufflers with complex geometry. A method was developed by MSC/NASTRAN on the basis of acoustic-structural analogy[5] and wave seperation method. This new method not only has the advantages of the four-pole parameter method, but also it gives a direct answer for multi-inlet(or multi-outlet) muffler. This method was verified through comparison with theory and test. Applying the method to design a muffler of reciprocating compressor, we developed a new muffler which is more quiet with good performance.

2. Definition of transmission loss

Fig.1 shows a typical muffler that has a small diameter pipe on both sides and muffler body in middle.

The performance of the muffler is measured in terms of transmission loss defined as the ratio of the sound energy with the incident on the muffler body .to transmitted downstream into an anechoic termination. That is,

$$TL = 10\log \frac{A_i S_{AA}(f)}{A_o S_{CC}(f)} \tag{1}$$

where, $S_{AA}(f)$ and $S_{CC}(f)$ are the sound power of inlet and outlet.

Physically, the higher value of the TL is, the less sound wave is transmitted to the outlet.

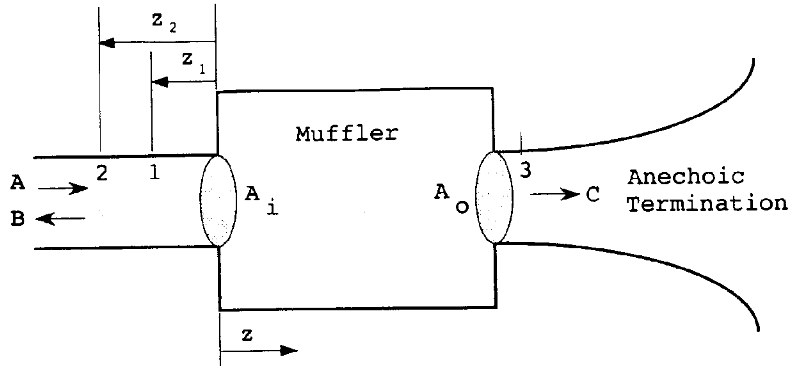


Fig.1 A view of typical muffler

where, A_i and A_o are the area of inlet and outlet,

3. Experimental method

Fig.1 shows a simple muffler. There exist a progressive sound wave A and a reflected sound wave B at the inlet of muffler. Therefore, to determine the TL according to eq.(1), one must seperate the progressive sound wave A, and calculate the input sound power $S_{AA}(f)$. The wave seperation has been done using two microphones. The method makes use of two microphones located at points 1 and 2. Fig.2 shows the experimental set up for the method.

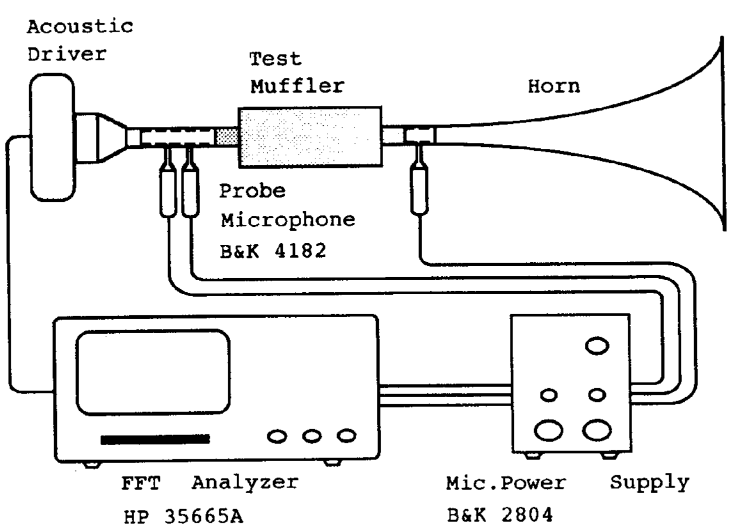


Figure.2 Experimental setup for sound wave seperation by means of the two-microphone method.

The detail measuring methods are as follows ;

After fixing the microphones at points 1 and 2, random-noise is given at the inlet of muffler by acoustic driver. Measuring the autospectrum(S_{11},S_{22}) and the cross-spectrum(C_{12},Q_{12}) with FFT analyzer. Input sound power $S_{AA}(f)$ is calculated by following eq.(2).[4]

$$S_{AA}(f) = [S_{11}+S_{22}-2C_{12}\cos k(Z_1-Z_2)+2Q_{12}\sin k(Z_1-Z_2)]/[4\sin^2 k(Z_1-Z_2)]$$
$$S_{BB}(f) = [S_{11}+S_{22}-2C_{12}\cos k(Z_1-Z_2)-2Q_{12}\sin k(Z_1-Z_2)]/[4\sin^2 k(Z_1-Z_2)] \qquad (2)$$

where, k is $2\pi f/a_o$, a_o is sound speed, Z_1 and Z_2 are the distance in Fig.1, C_{12} and Q_{12} are the real and imaginary part of the cross-spectrum. Finally, the autospectrum S_{33} is measured at point 3 where a horn is placed for the anechoic condition. The value S_{33} is equal to the output sound power $S_{CC}(f)$. Then, applying $S_{AA}(f)$, $S_{CC}(f)$ to eq.(1), one can get the transmission loss.

4. Method for analysis

Generally, for the situation of no mean flow and zero viscosity, the equation for fluid motion is similar to that of structural motion under an acoustic-structural analogy. MSC/NASTRAN, a general finite element computer program, gives a solution for the response of a fluid motion using the acoustic-structural analogy.

In this paper, a method for calculation of the transmission loss of muffler, applying experimental wave seperation skill to MSC/NASTRAN, was developed. To explain this method, let us consider a simple muffler as shown in Fig.3.

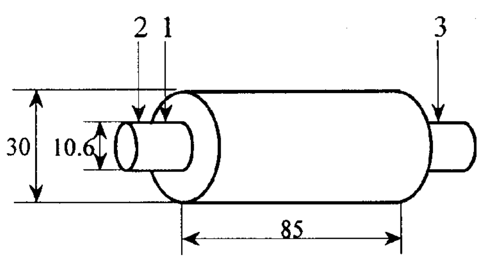


Fig.3 A simple muffler

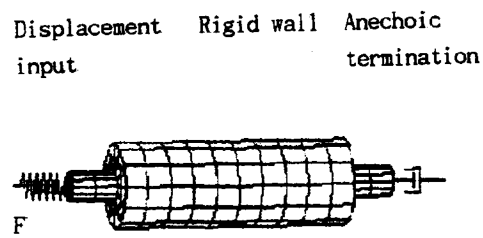


Fig.4 Finite element model and boundary condition.

The inner cavity of the muffler which is filled with air, was modeled with CHEXA and CPENTA elements(see Fig.4). At inlet boundary, the pressure(or displacement) is a known function of frequency. This condition was expressed by applying enforced motions that have a sine sweep at unit/Hz from 0Hz to 10kHz. In MSC/NASTRAN, the enforced motion was expressed by LARGE STIFFNESS METHOD.[6] The outlet condition is anechoic termination. The anechoic condition was represented by using a damping element with a damping coefficient

$$B = A/\rho a_o \tag{3}$$

where, A is the area associated with grid point, ρ is the fluid mass density, a_o is the sound speed.

With these boundary conditions, an analysis was performed by SOL 108 in MSC/NASTRAN. Completing the analysis, one gets sound pressure at each nodal point in the form of a conjugate. If we assume the value of the sound pressure at points 1,2,or 3 in Fig.3 as equation (4)-(6)

$$S_1(f) = a(f) - ib(f) \tag{4}$$

$$S_2(f) = c(f) - id(f) \tag{5}$$

$$S_3(f) = e(f) - ig(f) \tag{6}$$

then, the autospectrum and cross-spectrum are

$$S_{11}(f) = a(f)^2 + b(f)^2 \tag{7}$$

$$S_{22}(f) = c(f)^2 + d(f)^2 \tag{8}$$

$$S_{12}(f) = C_{12}(f) + Q_{12}(f) \tag{9}$$

where, $C_{12}(f) = a(f)c(f)+b(f)d(f)$, $Q_{12}(f) = a(f)d(f)+b(f)c(f)$

put these values into eq.2, the sound power $S_{AA}(f)$ is calculated, and the $S_{CC}(f)$ is given by

$$S_{CC}(f) = S_{33} = e(f)^2 + g(f)^2 \tag{10}$$

Now, one can get the transmission loss using eq.(1).

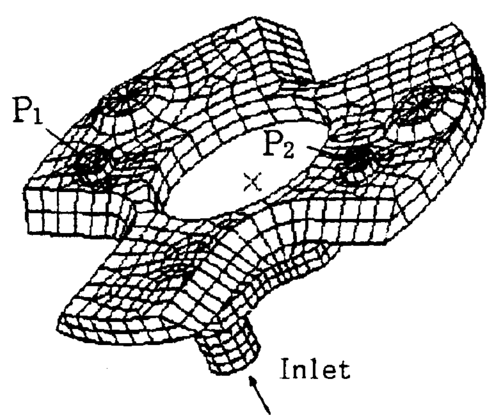


Fig.5 A finite element model of a muffler with complex shape

Secondly, consider a muffler with multi-outlet to verify this method. Fig.5 shows the finite element model of the muffler of a rotary compressor. The analysis has been done by the same method that was applied to the simple muffler. But in this case, as there are two output port(P_1,P_2,in Fig5), two noise transmission paths exist. So, TL are obtained at each outlet.

5.Results

To verify the analysis results, the transmission loss of a simple muffler(see in Fig.3) is calculated by MSC/NASTRAN and compared with plane wave theory and experimental method. The theoretical results are obtained from eq.(11).

$$TL = 10 \log[\ 1+1/4(M - 1/M)^2 \sin^2 kl] \tag{11}$$

where, M = 8 ; the ratio of inlet, outlet area to that of muffler area. l = 85 mm ; the length of muffler body. Fig.6 shows the theoretical results, and Fig.7 shows the comparison between MSC/NASTRAN and experimental method. As shown the comparison, we obtained a good agreement between the methods.

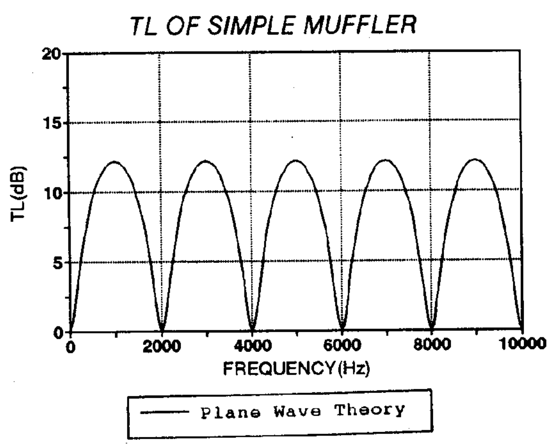


Fig.6 Calculation of TL

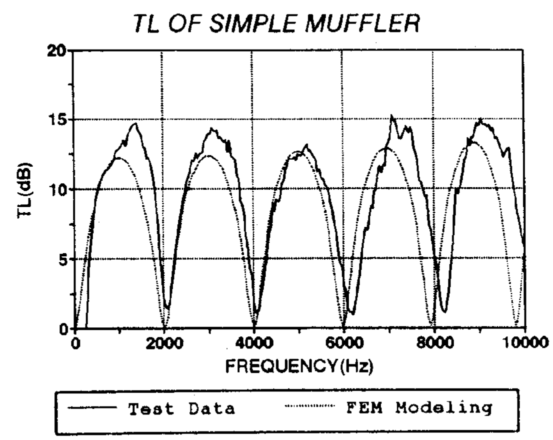
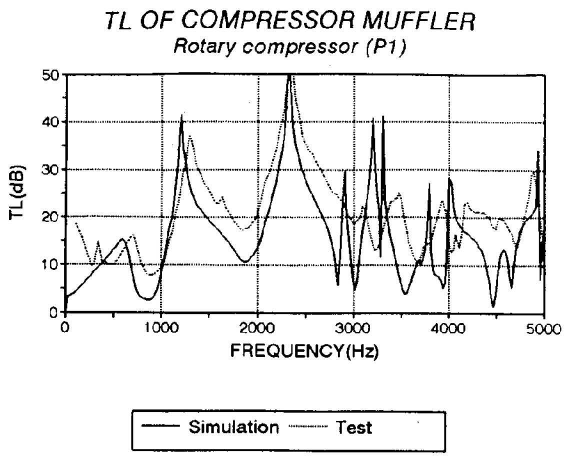


Fig.7 camparision of TL between FEM and test

In case of the muffler with the complex geometry, the TL at P_1 and P_2 are calculated, and compared with that of test as shown in Fig.8 and in Fig.9. The comparison shows that the results between the two methods has good agreement below the frequency of 2500 Hz.



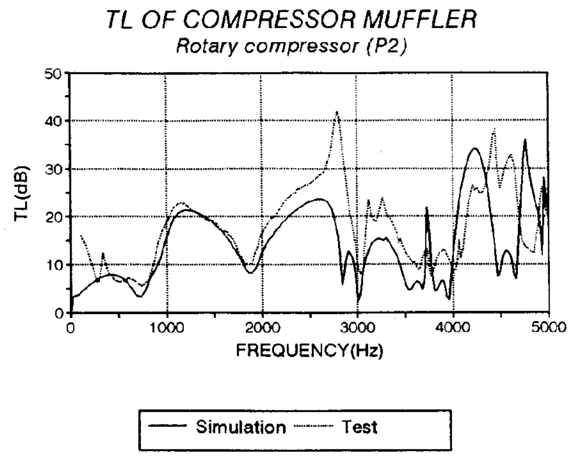
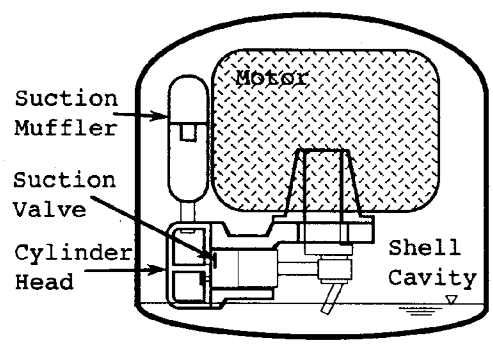


Fig.8 Comparis on of TL at P_1 Fig.9 comparison of TL at P_2

6. Application

We applied this method to the design of muffler in a reciprocating compressor. Fig.10 shows the schematic diagram of general reciprocating compressor.



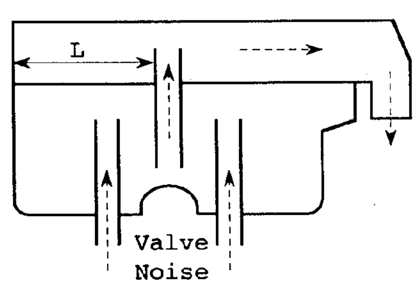


Fig.10 A view of reciprocating compressor Fig.11 muffler section

In this compressor, changing the refrigerant gas, we confronted a problem that the noise in the vicinity of 500Hz was incresed on operation. After many experiments, we found out that the noise was generated by the resonance of the shell cavity due to excitation of suction valve vibration. That is, as shown in Fig.11, the valve noise which is generated and passed to the suction muffler excites the cavity that has a resonance frequency of 500Hz.

There are two ways to solve this problem. One way is to enlarge the volume of the cavity so that the frequency of resonance is away from

500Hz. But this method has many limitations. The other method is to modify the suction muffler to enhance the performance of noise reduction at 500Hz. We chose the second one. To check the performance of this original muffler, the author calculated the TL of it. Fig.12 shows the finite element model. The results is plotted by solid line in Fig.13.

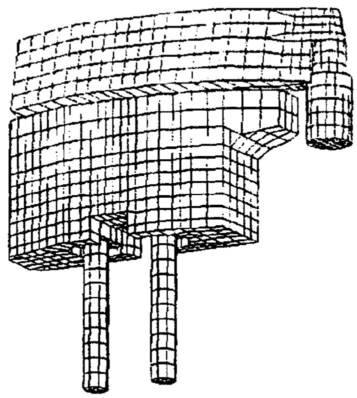


Fig.12 Finite element model of muffler in reciprocating compressor

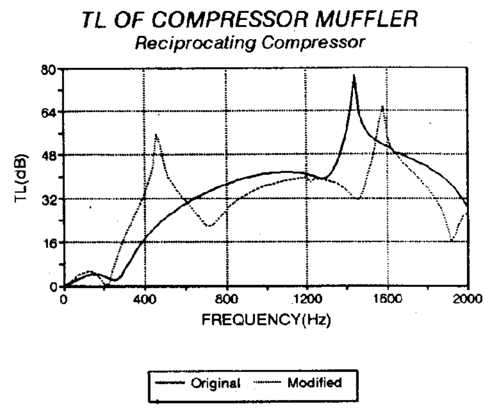


Fig.13 Comparis on of TL

One knows that the TL of the original muffler has a low level at 500Hz. To enhance the TL at this range, the author has simulated many structures of a new muffler, and has decided the structure of the muffler as in Fig.14. Its finite element model was represented in Fig.15.

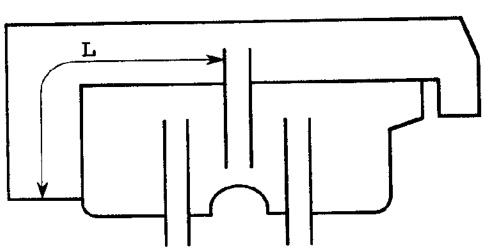


Fig.14 The shape of modified muffler

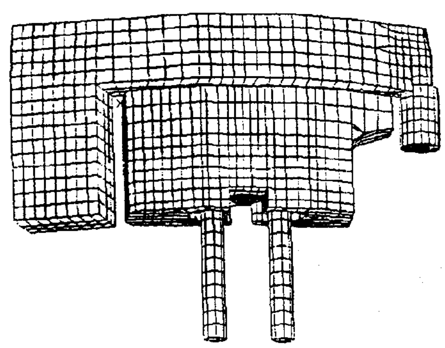


Fig.15 Finite element model of modified muffler

The calcualtion of TL was shown by dotted line in Fig.13. From the results, one can know the TL of modified muffler have a great increasement at the frequency range of 500Hz. It is because the resonance frequency of the muffler is shifted to 500Hz.

To examine the effect of noise reduction of the modified muffler, we made a sample muffler, applied to the reciprocating compressor, and measured the noise level on operation. The results are compared in Fig.16.

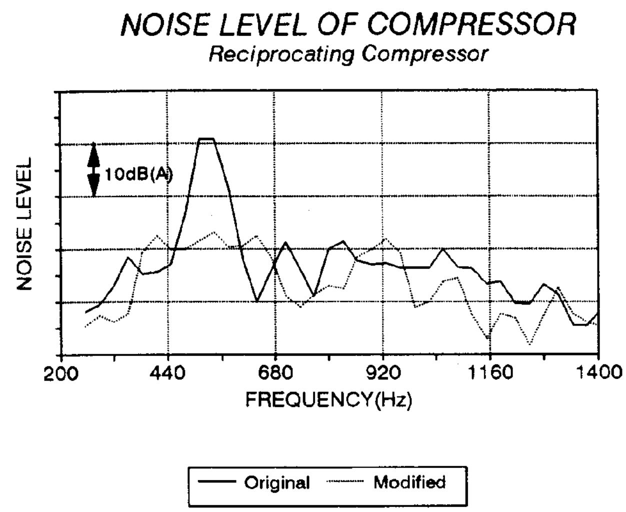


Fig.16 Comparison of noise level between the original muffler and that of the modified

As shown in Fig.16, when using the modified muffler in reciprocating compressor, the noise reduction is more than 10dB(A) at 500Hz, in total, 4 dB(A).

7. Conclusions

A method to predict the transmission loss was developed using MSC/NASTRAN. To verify the method, it has been applied to mufflers with simple and complex geometry. The method has a good agreement with theory and experimental methods. Applying this method to the design of a muffler in a reciprocating compressor, we developed a new muffler which is more quiet with good performance.

8. References

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