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CONCEPT DESIGN THEORY OF SHOCK ABSORBING MUFFLERS FOR AIR-BORNE NOISE

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ABSTRACT

The paper considers acoustic analysis of the shock absorbing muffler within a rotary compressor. The internal space of the compressor is modelled as a combination of cavities and pipes. A simple one-dimensional impedance approach is used for the acoustic analysis in the low frequency range, with ignoring the effects of gas flow and temperature gradients that are closely related to power efficiency of the compressor. Using the similarity between the vibration isolator and the shock absorbing muffler, the source strength transmissibility is newly proposed as a performance measure of the muffler and its validity is supported by power analysis. Some important muffler design rules obtained are; (1) a muffler cavity and its opening throat should be used as a pair, (2) a long thin throat is desirable for high frequency noise isolation, (3) a large muffler cavity should be used with care since it shortens the working frequency range of the muffler. The rules were applied to redesign a compressor muffler currently in use, and a significant improvement was achieved by simply attaching a throat to the outlet holes of the muffler.

1. INTRODUCTION

One of the major noise sources in the air conditioning system is the compressor. In particular, the aerodynamic shock wave generated inside the compressor at every expansion cycle can radiate excessive noise via transmitting through the exterior shell. To reduce such air-borne noise, either installation of dissipative damping elements inside the compressor or use of a thicker shell may not be a good answer because the one may contaminate the refrigerant fluid and the other is expensive. Thus the general practice is to install reflective type muffler on the outlet hole of the compressor cylinder[1].

Studies on compressor mufflers are rare compared to those on their counterparts in the engine exhaust system [2]. In fact, design of a compressor muffler relies more on empirical trial and error procedures rather than analytical or numerical tools. The reason would reside on their exact but non-predictable performance measure, which is the radiation power of the compressor. Prediction of radiation noise requires an accurate three-dimensional coupled model of fluid dynamic, acoustic, and structural dynamic components of the compressor, completion of which is nearly impossible or very expensive.

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The motivation of this work is that the role of mufflers is conceptually not different from that of vibration isolators. Vibration isolators, which can be modeled as a parallel connection of a spring and a damper at low frequencies, are used for isolating high frequency shock vibration transmission[3]. Although its shapes are rather complicated, the muffler behaves in a similar manner as will be clear later. Thus, the source strength transmissibility, which is compatible to the velocity transmissibility in vibration isolators, is adopted as the performance measure of a muffler. One of the advantages of this new performance measure is that we now confine the problem to the muffler itself without considering the rest of acoustic and structural parts. Consequently, with a simple acoustic model of the muffler itself only, one is now able to predict the performance of candidate mufflers in the design stage so as to facilitate developing a high performance muffler.

2. PERFORMANCE MEASURE OF MUFFLERS

2.1 Structure of a rolling piston type rotary compressor

The structure of a rolling piston type rotary compressor is shown in Figure 1 where the refrigerant gas flow path is indicated by thick arrows. The gas from the suction pipe is compressed in the cylinder, and the compressed gas passes sequentially through the muffler, the 1st expansion chamber, air gaps, the 2nd expansion chamber, and finally is discharged through the discharge pipe[4].

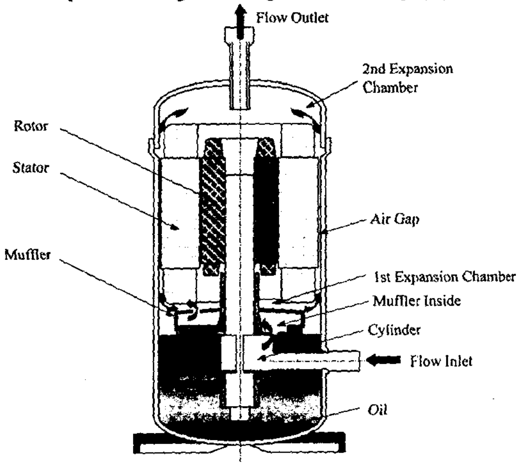


Figure 1. Structure of a rotary-type compressor where the blocked arrow indicates flow of a refrigerant fluid.

Figure 2 shows detailed drawings of the muffler where top and section views are shown together. As shown in Figure 2(a), the discharged compressed gas flow from the cylinder is divided into clockwise and counterclockwise directions, and is again discharged to the 1st expansion chamber through the two holes on the surface of the muffler. Due to the geometric interference of the valve, less amount of gas flows through the counter-clockwise path. Each path has a contraction area as shown in Figure 2(b), and the area contraction ratio(Sectional area B/Sectional area A) is about 18%. As will be discussed, the area contraction acts as a reflective muffler.

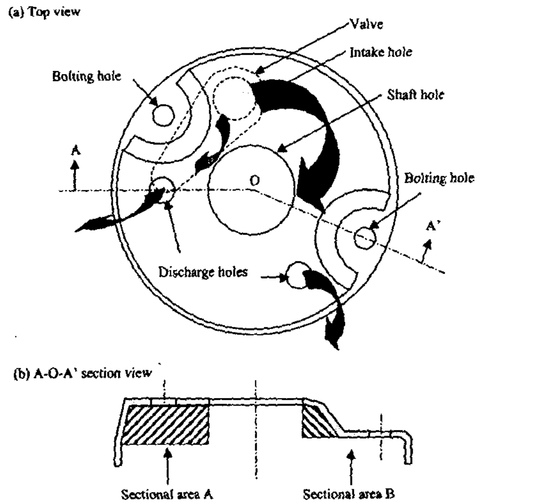


Figure 2. Detailed plots of the muffler; (a) top view, (b) Section view.

2.2 Air-borne noise transmission in a compressor

The acoustic power transmission circuit diagram of the compressor may be drawn as shown in Figure 3 where Π_0 is the active power input from the compressor cylinder(active source), Π_1 is the transferred power to the 1st expansion chamber through the muffler. The passively induced power Π_1 can be regarded as an active power from the viewpoint of the 1st expansion chamber. If it is assumed that there is no sound absorbing materials used inside the muffler, which means the muffler is purely reflective, there will no power consumption on the passage through the muffler so that Π_0 will be equal to Π_1[2]. Some portion of Π_1 will be dissipated through radiation Π_2 and the rest is transferred to the 2nd expansion chamber Π_3 through the air gaps. The power Π_3 will again be dissipated through radiation in the 2nd expansion chamber, and the rest is transferred to the discharge pipe. To sum up the discussions in equation form, they are

$$\Pi_0 = \Pi_1, \qquad \Pi_1 = \Pi_2 + \Pi_3 \qquad (1)$$

In order to reduce noise radiation, which is the main concern, a fundamental solution may be reducing the original power input Π_1 by use of a properly designed muffler. Suppose the outlet of the cylinder generating constant source strength Q is directly connected to the 1st expansion chamber without the muffler in between, and let the power input without the muffler be Π_x. Note that Π_0 defined above is the power when the same source strength Q is applied to the system with the muffler, while Π_x is for the one without it. Then, we may state that the objective of a muffler is *to reduce the power input with the muffler* Π_0 *with respect to the power input without the muffler* Π_x. This can be written in equation form as

$$\Pi_0 << \Pi_x \qquad (2)$$

Care should be taken here that the input power from a source of constant strength is dependent on the input impedance, and thus Π_x is generally different from Π_0.

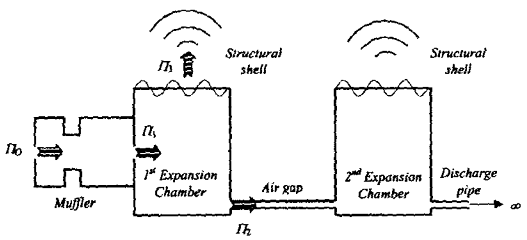


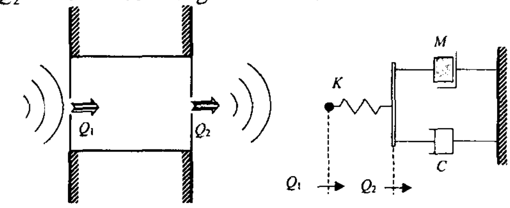
Figure 3. Power trasmission in a simplified acoustic model of the compressor

2.3 Mechanism of the shock-absorbing muffler

Figure 4(a) shows a schematic diagram of a shock-absorbing muffler, which is no more than a simple expansion chamber that lies between a noisy space on the left and the acoustic free space on the right. There are holes in the chamber so that noise from the noisy space is transferred to the acoustic free field. The noisy space could represent the compression cylinder while the quiet space represents the expansion chamber. The shock input acting on the left hole is assumed to be of constant source strength Q_1, and this source induces source strength of Q_2 on the right hole that is now radiating sound to the acoustic free space. The role of the muffler is buffering the shock Q_1 so as to emit a less shock Q_2 to the acoustic system of concern(acoustic free field). In this sense, the role of the muffler is conceptually very similar to that of the vibration isolator. In fact, as shown in Figure 1(b), a lumped parameter model for the muffler shows a similar representation, except it has a sky-hook damper. The spring K represents the muffler, and the mass M and damper C correspond reactive and resistive part of radiation impedance, respectively[5]. Thus, we may suggest *transmissibility* T as a performance measure of the muffler, which is given by[3]

$$T = \frac{Q_2}{Q_1} \tag{3}$$

where again Q_1 is the source strength input to the muffler, and Q_2 is the source strength to the acoustic free field.



(a) Acoustic shock muffler
(b) Lumped parameter modeling

Figure 4. Acoustic shock muffler and its simplified model

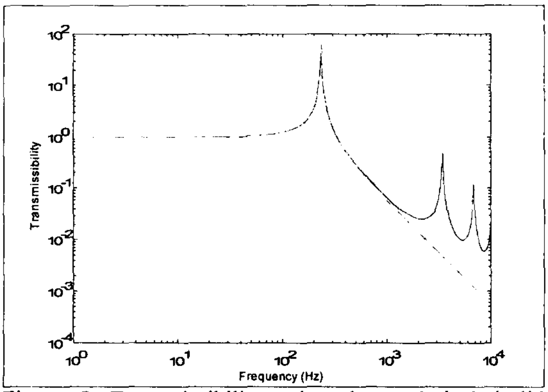


Figure 5. Transmissibility using the analytical (solid line) and the lumped parameter (dashed line) models shown in Figure 4(a,b), respectively.

The transmissibility for the muffler model is shown in Figure 5 where the analytical and the approximated lumped parameter models are compared in the same plot. As can be seen, the lumped parameter model closely resembles the analytical model in the low frequency range under the first expansion chamber resonance, which is the second peak in the plot. Interpretations of the plot can be made similarly as those for the vibration isolator [3]. The muffler isolates high frequency sound transmission while trading off the low frequency sound transmission. It is clear that the muffler works in a similar way as a vibration isolator.

2.4 Validation of the transmissibility as a performance measure of a muffler

Consider the power with and without the muffler in place in Figure 4. In absence of the muffler, the active source of constant source strength Q_1 will directly act at Terminal 2 so that the radiated power without the muffler is given by[5]

$$\Pi_x = \frac{1}{2}|Q_1|^2 \operatorname{Re}(Z_2) \tag{4}$$

where Z_2 is the acoustic impedance of the terminal 2. On the other hand, the radiated power with the muffler is

$$\Pi_O = \frac{1}{2}|Q_2|^2 \operatorname{Re}(Z_2) \tag{5}$$

Thus, the power reduction ratio of with and without the muffler can be written as

$$\frac{\Pi_O}{\Pi_x} = |T|^2 \tag{6}$$

Equation (6) can be conveniently used *since the transmissibility of a muffler offers the information on how much power is reduced by using the muffler.* If the transmissibility of muffler is a half, for example, the power reduction ratio becomes a quarter so that three quarters of the original input power in absence of the muffler are reduced. Therefore, the transmissibility given in equation (3) can be used as a convenient performance measure to index the quality of mufflers. To the author's best knowledge, transmissibility has never been used to value the performance of a muffler.

3. IMPEDANCE APPROACH TO PIPE SYSTEMS

3.1 Impedance approach to a single pipe system

Consider a one-dimensional acoustic pipe of length L and cross sectional area S as shown in Figure 6. The volume velocities at each end are denoted as Q_1 at $x = 0$ and as Q_2 at $x = L$, and sound pressures are similarly denoted as p_1 and p_2, respectively. For a given impedance at Terminal 2, the impedance at Terminal 1 is given by [5]

$$Z_1 = Z_o \frac{Z_2 \cos kL + jZ_o \sin kL}{Z_o \cos kL + jZ_2 \sin kL} \tag{7}$$

where the pipe characteristic impedance is $Z_o = \rho c / S$ in which S is the pipe scross-sectional area, ρ is fluid density, and c is sound speed. The volume velocity *transmissibility* T can be written as

$$T = \frac{Q_2}{Q_1} = -\frac{Z_t}{Z_d + Z_2} \tag{8}$$

where the direct and transfer impedances of Terminal 1 with setting $Q_2 = 0$ are $Z_d = -jZ_o \cot kL$ and $Z_t = jZ_o \csc kL$, respectively.

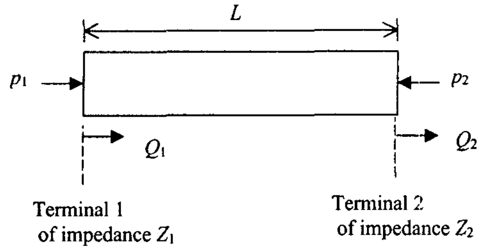


Figure 6. A one-dimensional pipe with two terminals whose impedance boundary conditions are Z_1 and Z_2.

3.2 Impedance approach to a multiple pipe system

A. Serial connection

Now consider a multiple pipe system as shown in Figure 7 where two pipes are connected in series. The transmissibility between Q_3 and Q_1 is given by

$$T_{13} = Q_3/Q_1 = T_{12}T_{23} \tag{9}$$

where $T_{12} = Q_2/Q_1$ and $T_{23} = Q_3/Q_2$. A general serially connected multiple-pipe system consisting of more than two pipes can be analyzed similarly.

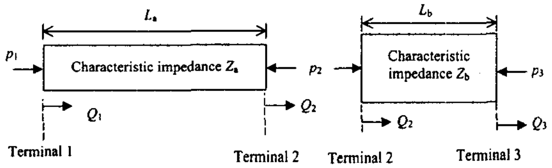


Figure 7. A two-pipe system connected in series

B. Parallel connection

Parallel connections of pipes are often practiced in muffler design, as shown in Figure 8. The impedance at Terminal 1, $Z_1 = p_1/Q_1$, can be written as

$$\frac{1}{Z_1} = \frac{1}{Z_{1A}} + \frac{1}{Z_{1B}} \tag{17}$$

where the impedances of the two branches are $Z_{1A} = p_1/Q_{1A}$ and $Z_{1B} = p_1/Q_{1B}$, respectively.

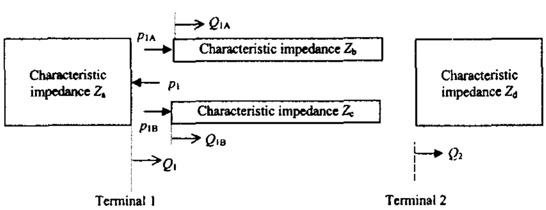


Figure 8. A two-pipe system connected in parallel

4. CONCEPT DESIGN OF A MUFFLER

4.1 A single muffler

Figure 9 shows a multiple pipe system that may be regarded as a simple model of a muffler-cavity-pipe system where the system is of total length L and outer radius R, and inner radius r. It should be noted that the muffler radiates now to a cavity, instead of the acoustic free field to which was the case given in Figure 4. This type of muffler more closely resembles the compressor muffler, whereas the model in Figure 4 corresponds to the automobile muffler.

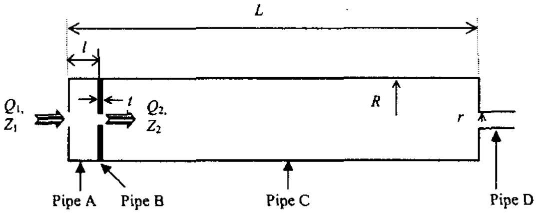
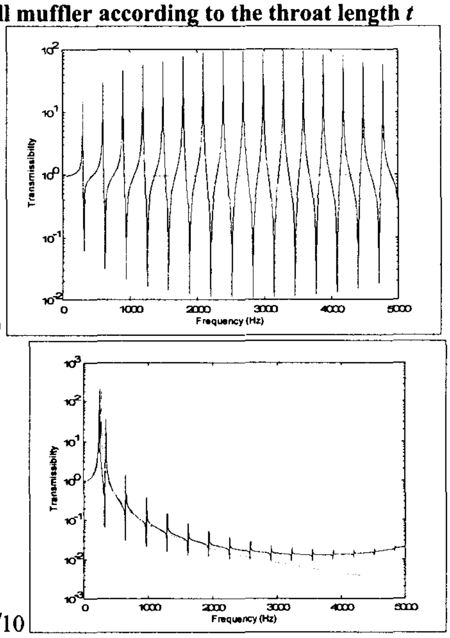


Figure 9. A single muffler-cavity-pipe system

A. A small muffler according to the throat length *t*



(a) $t = 0$

(b) $t = l/10$

Figure 10. Transmissibility of a single muffler according to the throat length t; (a) $t = 0$ and (b) $t = l/10$.

Figure 10 shows transmissibility results according to changes of the muffler throat length where solid and dashed lines denote analytical and lumped parameter models, respectively. Although not exact, however, the simple lumped parameter model approximates the analytical one reasonably well. It is interesting that in Figure 10(a) the muffler with just a partition ($t \approx 0$), not a throat, does not work properly. This feature is quite striking since, when this was used to isolate sound radiation to the free field in Figure 4, a large amount of high frequency sound isolation was achieved. The reason is that the ambient air of the free field in Figure 4 acts as a mass(mass loading effect), while the cavity in Figure 9 acts as a spring. If we add a throat, which acts as mass loading, the muffler now works as a noise isolator as shown in Figure 10(b). The peaks in Figure 10(b) occur at the resonance frequencies of Pipe C, and the frequency range considered is below the first resonance frequency of the muffler cavity, Pipe A. Figure 10(b) clearly demonstrates that a throat of moderate length should be incorporated into a compressor muffler. In addition, it is clear that the behaviour of the muffler is similar to that of a vibration isolator. The transmissibility trend according to the amount of mass was also investigated and is shown in Figure 11 where the amount of mass was controlled by the throat length. As shown in Figure 11, the performance of high frequency sound isolation improves as the amount of mass increases.

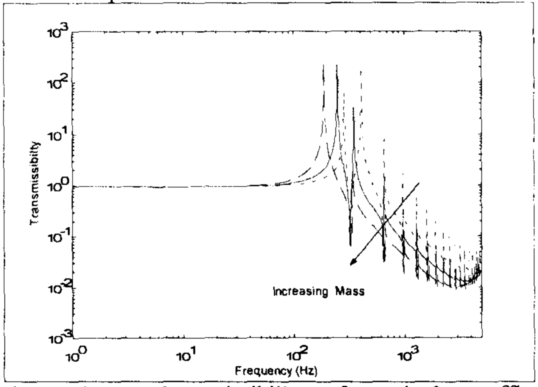


Figure 11. . Transmissibility of a single muffler according to changes of the throat length t

B. large muffler of cavity length $l = L/5$

Now consider a larger muffler having a larger cavity volume, Pipe A in Figure 9. Figure 12 shows the transmissibility for the muffler of cavity length $l = L/5$ and throat length $t = l/10$. The other design parameters were kept the same. The dominant peaks at around 1500 Hz, 3000 Hz, and 4500 Hz in Figure 12 are due to resonance frequencies of the muffle cavity. Thus, it can be stated that resonance inside a muffler should be avoided as much as possible. However, this design parameter is strictly constrained by the power efficiency of the rotary compressor.

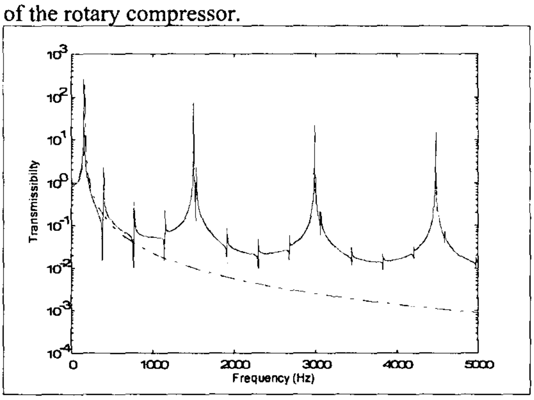


Figure 12. Transmissibility of a single muffler with a larger muffler cavity volume; cavity length of $l = L/5$ and throat length of $t = l/10$.

4.2 A double muffler

This section considers a double muffler as shown in Figure 13 where the large muffler of cavity length l considered in the previous section is partitioned in the middle. The transmissibility Q_2/Q_1 of this double muffler(solid line) is compared with that of the large single muffler(dashed line) in Figure 14. It is clear that use of a double muffler by partitioning the cavity of a single muffler offers better isolation than a single muffler. However, this partitioning significantly decreases the buffering volume of the discharged gas Q_1, thus may be against the power efficiency of the compressor.

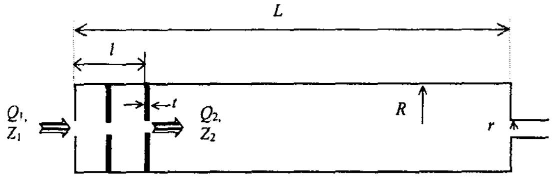


Figure 13. A double muffler system where $l = L/5$ and $t = l/10$

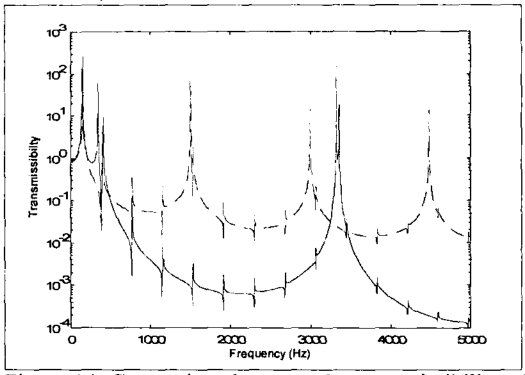


Figure 14. Comparison between the transmissibility of a double muffler(solid line) shown in Figure 13 and the single muffler in Figure 12.

5. APPLICATION TO THE COMPRESSOR MUFFLER

The compressor model currently in use, as shown in Figure 3, was analyzed and the transmissibility is shown in Figure 15. Unlike the cases discussed in Section 4, no excellent sound isolation trend is observed. Figure 15 indicates that the original model may not be a good design as far as noise isolation performance is concerned. Thus, we considered some design parameter changes to improve noise isolation performance, which is attaching a throat at the muffler outlet.

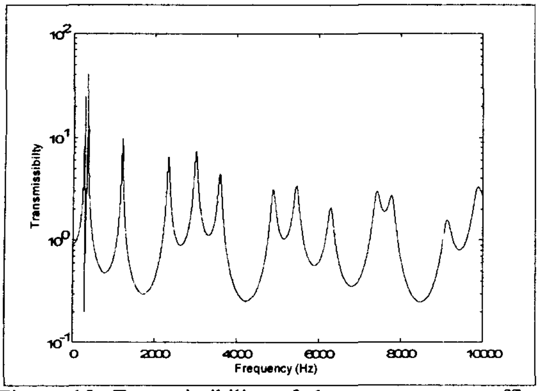


Figure 15. Transmissibility of the compressor muffler shown in Figure 3.

Figure 16 compares the transmissibility characteristics of the models with(solid line) and without(dashed line) a throat at the muffler outlet. The dashed line repeated after Figure 15 denotes for the original muffler which has no throat. The throat length was set to be 10 mm and its cross-sectional area was the same as that of outlet holes of the muffler. It is clearly demonstrated that use of cavities and throats as pairs significantly enhances the noise isolation performance.

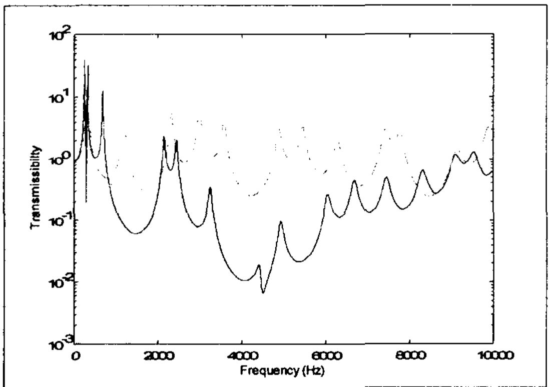


Figure 18. Comparison of the transmissibility characteristics between the models with(solid line) and without(dashed line).

6. CONCLUSIONS

The paper has considered acoustic analysis of the shock absorbing muffler within a rotary compressor. The internal acoustic space of the compressor was modeled as a combination of cavities and pipes. A simple one-dimensional impedance approach has been used for the acoustic analysis in the low frequency range with ignoring the influences of gas flow and temperature gradients, which are closely related to power efficiency of the compressor. Using the similarity between the vibration isolator and the shock absorbing muffler, the source strength transmissibility was newly proposed as a performance measure of the muffler and its validity was supported by power analysis. Some important muffler design rules obtained are; (1) a muffler' cavity and its opening throat should be used as a pair, (2) a long thin throat is desirable for high frequency noise isolation, (3) a large muffler cavity should be used with care since it shortens the working frequency range of a muffler. The rules were applied to redesign a compressor muffler currently in use, and significant improvement of the performance was achieved by only attaching a throat to the outlet holes of the muffler.

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