B&W UNIVERSAL

INDUSTRIAL NOISE SERIES

PART IX: REACTIVE SILENCERS

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INTRODUCTION

Reactive silencers are basically chamber and tube type units that may include some packing materials for middle and high frequency performance. These types of silencers are principally used on reciprocating engines (pistons) or other equipment having significant impulse type sound energy. The chamber and tubes are designed (tuned) to the principal impulsive frequencies that need to be attenuated. The attenuation is caused by the chambers and tubes that create internal reflective sound fields that reduce the sound energy. Because these units are tuned for each application, any changes will affect performance and includes connecting ducting, especially tail pipes. Figure 1 shows a combination reactive silencer having a small absorptive pack section. Note the internal pass tubes and chambers.

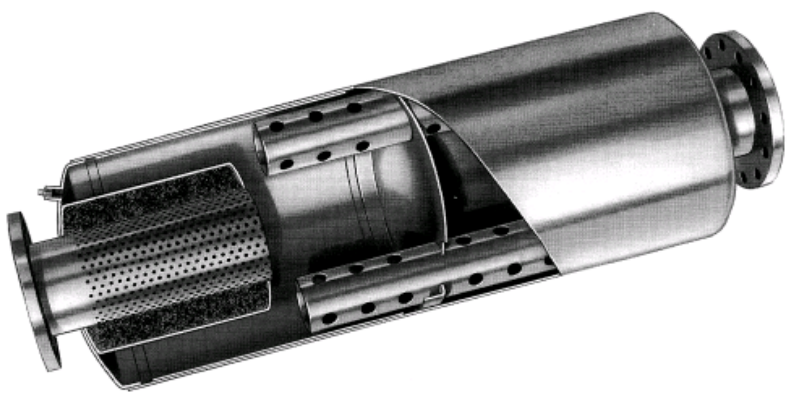

Figure 1 – A typical reactive silencer

Reactive silencers have been around for over 100 years and are still the most challenging and demanding of any type of silencer to develop. Early mufflers were primarily designed to mitigate the exhaust noise from stationary engines and then automobile and marine engines. Stationary engines were usually constant speed whereas automobile and marine engines operate at various speeds (rpms). The automobile and marine mufflers were primarily designed to quiet the engine while at idle. Somewhere along the process mufflers were stuffed with absorptive insulations to reduce the higher frequency noises and to make them a bit more broadband in attenuation. Broadband means to more effective in reducing noise over a wider frequency spectrum.

The acoustical performance of reactive silencers is usually specified in terms of insertion loss (IL) or transmission loss (TL), a more appropriate metric. The following two paragraphs are from page 374 in Beranek and Ver,[1] and include (embellishments) for clarity:

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[1] A.G. Galaitas and I.L. Ver, "Passive Silencers and Lined Ducts," Chap. 10 in *Noise and Vibration Control Engineering*, edited by L.L. Beranek and I.L. Ver (John Wiley & Sons Inc. NY, 1992)

The IL is the most appropriate indicator of a silencer's performance because it is the level difference of the acoustical power radiated from the un-silenced and silenced system. It is easy to quantify from pre- and post silencing data and can be costly to perform. IL is hard to predict because it depends on knowing the impedances (acoustical properties) of the source (engine) and termination (stack or tail pipe) which vary from one application to another. By contrast, the TL is easy to predict but is only an approximation of the silencer's actual performance because it does not account for the source (engine) impedance and models all silencer's outlets with anechoic terminations. The TL and IL become identical in the case when the silencer termination is anechoic (all sound exiting the silencer is perfectly absorbed – no exit or termination affects).

The ultimate selection of an evaluation criterion is based on the trade-offs between the desired accuracy in the predictions and the amount of available resources ($). For example, the source impedance required for IL predictions can be determined experimentally but is too costly and time consuming. As a result, the silencer design is generally based on predicted TL, with a clear understanding of the associated approximations. The final evaluation can only be verified by actual field-testing the IL.

What this means is performance of a reactive silencer cannot be guaranteed without extensive development work. On small units it may be worth the risk as it just means swap out the deficient one for a new one that might work but this approach is risky for a large, complex unit. There is little reason for us to take on onerous risks and liabilities in order to provide a silencer that we can only estimate the expected performance at best.

STANDARD SILENCERS

Just selecting a silencer out of the catalog is not an acceptable practice for reciprocating engine applications unless it is a replacement unit. Figure 2 shows an analysis of a large combination, three-chambered silencer. Note the dips at 40 and 50 Hz; these are called nulls where either the TL or IL makes a dip. All reactive silencers have these dips or nulls and can be most pronounced. A null is like a “short circuit” allowing the acoustic energy to be transmitted through the silencer. In this application, this silencer cannot be used on an engine that has a significant engine order or firing frequency near 40 or 50 Hz; otherwise silencer performance would be most disappointing. Now on variable speed engines, such as those used in automotive and marine applications, this mainly becomes important at idle speed where most all such silencers are designed to quiet the engine at idle.

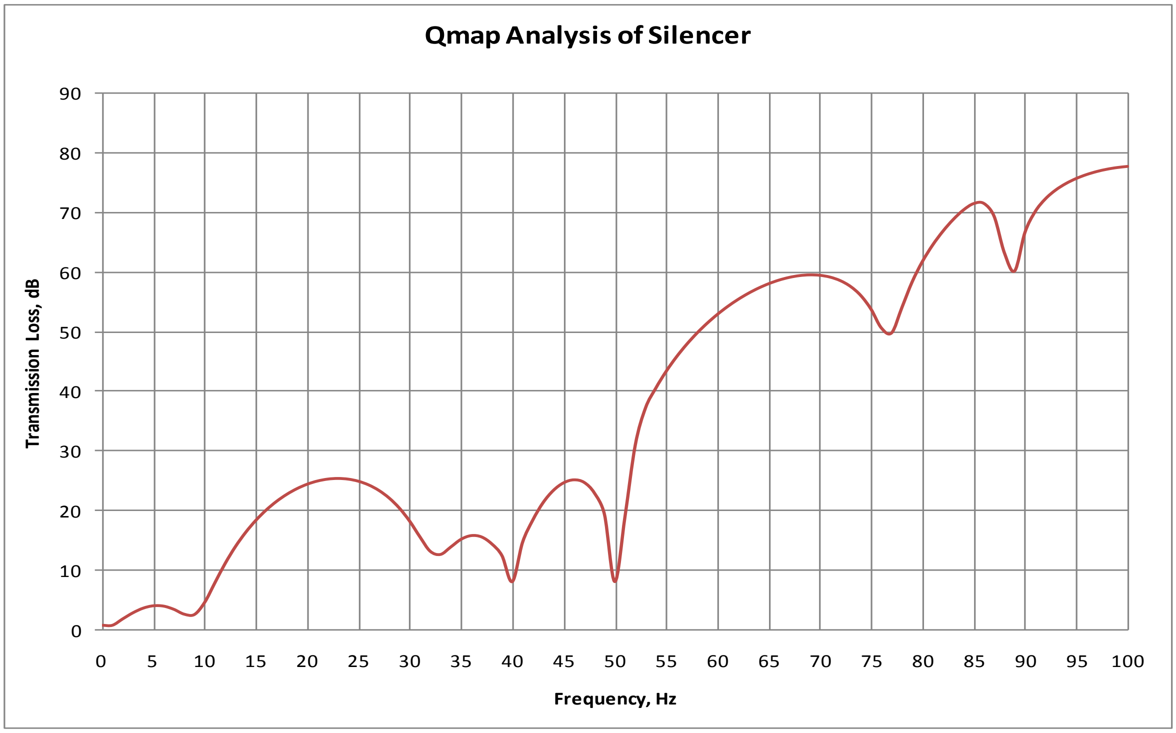

Figure 2 – Analysis of a Three-Chamber Reactive Silencer

The importance of obtaining engine data and thoroughly analyzing the application cannot be over emphasized. When the supplier of the silencer is not provided with engine data then it becomes a guess as to what the principal engine orders may be.

ENGINE DYNAMICS

As just mentioned, the challenging aspect in designing a reactive silencer is identifying the engine dynamics that create the sound energy, that is, the sound power level and the engine frequencies that need to be attenuated. The most common error is assuming it is only the firing frequency and perhaps a harmonic that need attenuation but frequently it is multiple harmonics that can include one-half rotation orders (1½, 2½, 3½, 5½, etc.) that can be significant.

So where do all these tones come from? The exhaust manifold may be considered as a type of side branch resonator that generates these multiple tones as the exhaust gas blows out the cylinder and into the exhaust manifold; think about a long tube with multiple ports, otherwise known as a musical instrument like a clarinet.

Figure 3 illustrates an exhaust manifold for a six-cylinder engine.

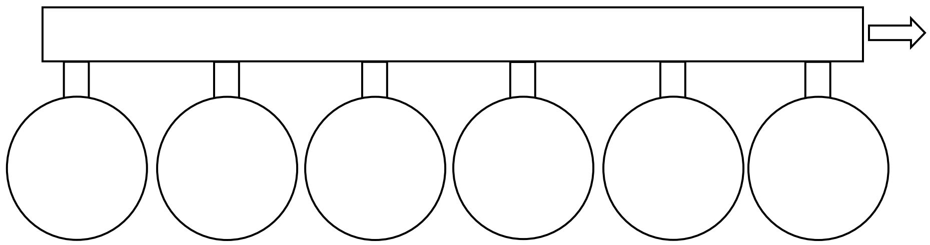

Figure 3 – Engine Exhaust Manifold (6 Cylinder)

The size of the valve, size and spacing of the exhaust ports, size of the manifold, and engine firing sequence all affect the harmonics and their amplitude. V-bank engines with cross-connected exhaust manifolds create even more harmonics. Performing exhaust measurements is only way to identify the principal engine tones. Octave band data is not adequate because one octave band can have multiple tones and one-third octave is only slightly better. The engine maker needs to clearly identify the principal tones that require attenuation.

DESIGN BASICS

Reactive silencers are generally the preferred method for passively attenuating low frequency sound from reciprocating engines. Reactive silencers are generally a series of interconnecting chambers of varying lengths with internal pass tubes and the performance is determined principally by the geometry of the unit. The length of the chamber and pass tube typically controls the tuning (frequency) and the change in diameters controls the amount of attenuation; the greater the change in diameters the greater the attenuation. Multiple chambers can be acoustically coupled to further enhance low frequency attenuation but again the available length can limit the performance. Now, reactive silencers are tuned to discrete engine frequencies thus a “standard” silencer will not have the same performance on other engines.

It is imperative that engine data be provided that includes not only the exhaust flow rate and temperature but the size of the engine, number of cylinders, operating mode: 2 or 4 cycle, and rpm. It is critical to define the design point because as mentioned, a reactive silencer is designed for fixed known frequencies. It is preferable to have one-third octave band sound power levels from 12.5 Hz though 10 kHz. This is necessary to correctly quantify the low frequency sound power levels of the engine. Octave band levels are too broad to be of benefit and some OEMs only provide octave band levels starting with the 63 Hz band thus the sound power levels of the fundamental firing frequencies are unknown.

The tuning of the chamber and pass tubes is based on ¼ wavelength reflections and impedance mismatches where changes in cross area occur as exhaust flow enters and exits chambers through smaller diameter pass tubes. The ¼ wavelength is determined by the gas temperature entering the silencer; higher gas temperature cause an increase in the sound of speed (c) creating a larger wavelength, λ.

The speed of sound at atmospheric pressure may be estimated by,

$$c = 49.03\sqrt{(T\ °F + 460)}\ \text{ft/s} \qquad (3a)$$

$$c = 20.05\sqrt{(T\ °C + 273)}\ \text{m/s} \qquad (3b)$$

Where, T is the temperature in either Fahrenheit or Celsius as appropriate. If the gas is under pressure then the speed of sound must be calculated based on the pressure (P) and density of the gas (ρ). The symbol, √(…) means to calculate the square root of the argument (…).

$$c = \sqrt{(1.4P/\rho)}\ \text{m/s or ft/s} \qquad (4)$$

The ¼ wavelength is then,

$$\lambda/4 = c/(4\cdot f) \qquad \text{feet or meters depending on } c \qquad (5)$$

where, f is frequency (Hz). The following table lists ¼ wavelengths at approximately 700°F (370°C) to illustrate the quarter wavelengths at various frequencies.

Table I - ¼ Wavelengths at 700° F / 375° C

Frequency (Hz)	16	20	25	31.5	40	50	63
¼ wavelength (ft)	26.1	20.9	16.7	13.3	10.5	8.37	6.66
¼ wavelength (m)	7.96	6.38	5.10	4.05	3.19	2.55	2.03

A review of Table I illustrates the challenge in tuning a silencer for multiple frequencies. Also, an allowance for the acoustical wave to expand and contract through the chambers must be taken into account and causes additional length.

The base attenuation is a function of the change in diameters. A classical simple in-line expansion chamber provides the following transmission loss,

$$TL = 10 \log [1 + ¼\,(m - 1/m)^2 \sin^2(kL)]\ \text{dB} \qquad (6)$$

where m is the area ratio (large/small), k is the wave number (ω/c) and L is the length of the chamber (ω= 2πf). The argument of the sin function becomes unity for a tuned chamber where kL = 1 at that particular frequency (f). The following table lists TL values for five expansion ratios.

Table II – TL of Expansion Chamber for kL = 1

m	5	10	20	40	80
TL, dB	8	14	20	26	32

A study of papers and texts about reactive silencer design and development all state that testing is the only method to assure performance; theory only gets you close. Laboratory testing cannot duplicate field conditions and scaling small system performance to full scale is just as problematic. Chapter 5 in Munjal[2] cites a litany of challenges in performing laboratory testing and provides some design guidelines in chapter 8 but Munjal's work is primarily associated with small engines hence small silencers. Many experts at universities cite that present day theoretical models for small silencers may not be adequate for very large reactive silencers.

PERFORMANCE CRITERIA

It is critical to clearly state the requirements of the silencer to be supplied and understand what is being specified and how it will be measured for compliance. The following information should be provided as a minimum:

a. Acoustical criterion or criteria.

b. Flow rate, operating temperature and pressure loss limits.

c. Engine data and operating conditions.

- rpm (fixed or variable speed application)
- number of cylinders
- dual or combined silencer for V bank engines
- 2 or 4 cycle

d. Machine acoustical data – must be frequency based.

- Identify significant engine orders

e. Connection data, orientation, and size limits including weight.

f. All specifications and standards related to the performance requirements.

g. All specifications and standards related to measurements.

[2] M.L. Munjal, *Acoustics of Ducts and Mufflers*, John Wiley & Sons, Inc. 1987

Computer models can only estimate performance. It takes developmental testing and iterative designs to arrive at a design. The only way to verify performance is by testing and development on the actual engine having an identical exhaust (or inlet) system arrangement.

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