UNIVERSAL
1925 Hwy. 51/138 W● PO Box 411●Stoughton, WI 53589

INDUSTRIAL NOISE SERIES

Part VIII

REACTIVE SILENCERS

Elden F. Ray

August 11, 2010

REACTIVE SILENCERS

1. 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 creating an internal sound field that "blocks" the acoustic sound wave from continuing to propagate through the silencer and out the exhaust or inlet end. Because these units are tuned for each application, any change will affect performance. Figure 1 shows a combination reactive silencer having a small absorptive pack section. Note the internal pass tubes and chambers.

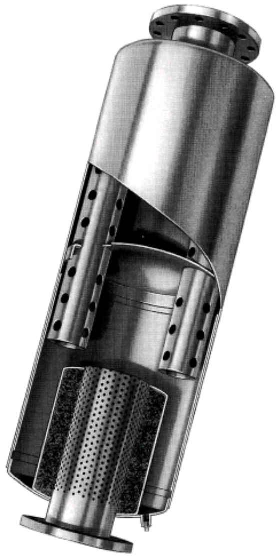

Figure 1 – A typical reactive silencer

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:

> 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.

2. STANDARD SILENCERS

Just selecting a silencer out of the catalog is not an acceptable practice for reciprocating engine applications. The importance of obtaining engine data and thoroughly analyzing the application cannot be over emphasized. The following figure shows the analysis of a standard silencer that was sold based on its published insertion losses. The catalog listed an insertion loss of 34 dB in the 63 Hz band and the customer's requirement was only 20 dB so should be good to go. Shortly after commissioning the site received complaints and field measurements showed the 63 Hz band as only achieving a noise reduction of four decibels. A review of the engine data showed it had a firing frequency (harmonic) at 50 Hz and a detailed analysis of the silencer showed there was a null at that frequency as shown in Figure 2. A null is like a "short circuit" allowing the acoustic energy to be transmitted through the silencer.

---

[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)

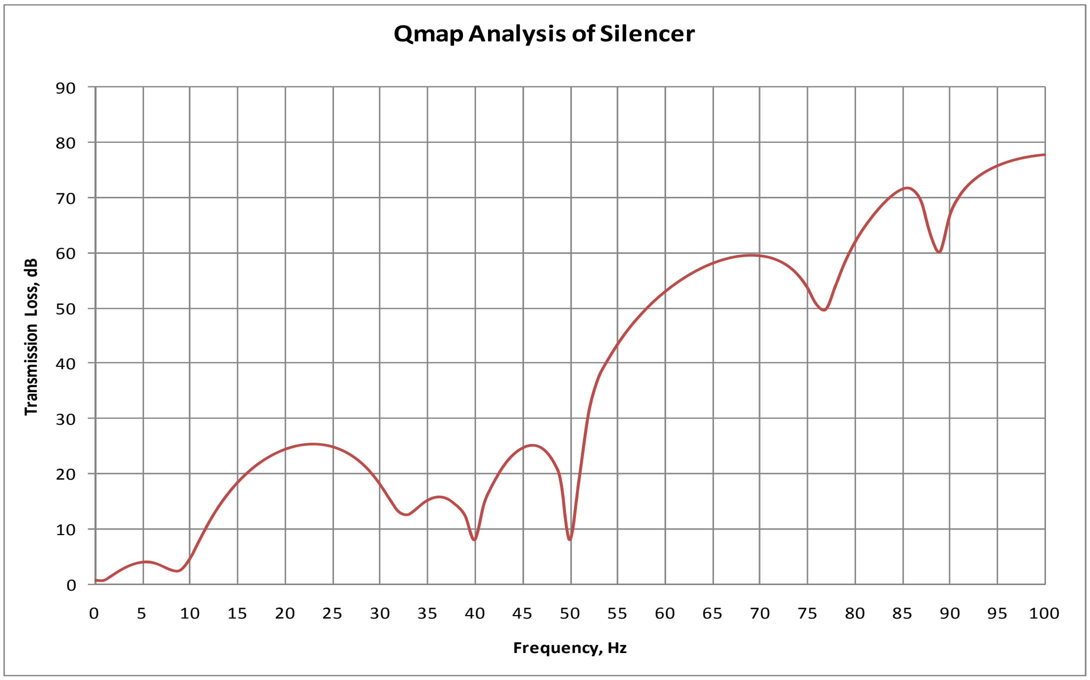

Figure 2 – Analysis of a 3 Chamber Reactive Silencer

On an octave band basis, there was "no problem" but on a discrete frequency basis there was a big problem. This situation demonstrates that engine data and installation details are necessary to thoroughly understand the application and to ensure the silencer performs as expected.

3. ENGINE DYNAMICS

A 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 (exhaust ports), otherwise known as a saxophone or other similar horned instrument. The exhaust valve rapidly opens with the piston pumping exhaust gas into the manifold all along its length thus creating multiple tones. Figure 3 illustrates a common exhaust manifold with six engine cylinders dumping into it.

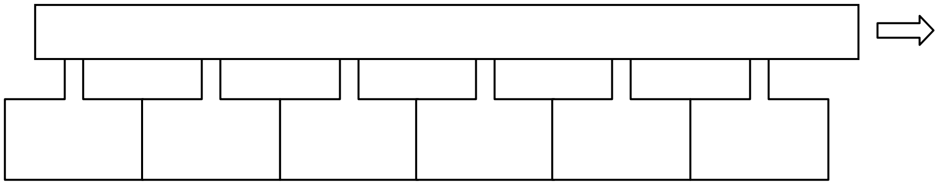

Figure 3 – Engine Exhaust Manifold (6 Cylinder)
Lower row are cylinders with exhaust ports connecting to a common manifold.

The size and spacing of the exhaust ports, size of the manifold, and engine firing sequence all affect the harmonics and their amplitude. 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.

4. 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

$$c = 49.03\sqrt{(T\ ^\circ\mathrm{F} + 460)}\ \mathrm{ft/s} \tag{3a}$$

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

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).

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

The ¼ wavelength is then,

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

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,

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

where m is the area ratio (large/small), k is the wave number (ω/c) and L is the length of the chamber ($\omega = 2\pi f$). The argument of the sin function becomes unity for a tuned chamber where $kL = 1$ at that particular frequency. 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 problems in performing laboratory testing and provides some design guidelines in chapter 8 but Munjal's work is primarily associated with

---

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

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.

Analytical computer models can only estimate performance. It takes developmental testing and iterative designs to arrive at a design that might work. The only way to verify performance is by testing on the actual engine having an identical exhaust (or inlet) system arrangement. The length of connecting duct work also affects performance.