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REACTIVE ATTENUATORS - USES AND ABUSES

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1. INTRODUCTION

In this paper an absorptive attenuator is assumed to have no reactive effects. For this to be the case there must be no changes of section throughout the inlet pipe (this term is assumed to include ducts), the attenuator, and the discharge pipe, and can only be achieved with a cylindrical attenuator without a pod whose bore is the same as that of the associated pipes.

Conversely a reactive attenuator is assumed to have no absorptive materials but achieves its results by changes in cross-sectional area in the pipes or by the introduction of impedance tubes. A Helmholtz resonator is a particular type of reactive attenuator but will not be discussed in this paper.

Most conventional attenuators such as splitter attenuators or cylindrical attenuators with a central pod are a combination of absorptive and reactive elements.

Absorptive attenuators naturally have a poor low frequency response, generally falling off below 500 Hz unless very thick absorptive materials are used. Also if attenuator pressure drops are to be minimised low airway velocities are necessary which normally results in bulky attenuators. In some environments such as encountered in refrigeration systems conventional absorptive materials such as Rockwool, glass fibre and foams are liable to be broken down by reaction with the refrigerant with the result that particles of the material will either block up the system or cause a compressor to seize up. Even with clean air systems large amplitude low frequency pulsations such as those generated by low speed reciprocating air compressors may cause the absorptive material to break up so that one finishes up with a pile of dust going through the system.

Absorptive attenuators also have a broad-band frequency response and will not selectively attenuate specific pure tones such as a fan blade frequency or an engine 'firing' frequency.

Reactive attenuators are particularly useful when low frequency pure tones require to be attenuated and the most common type is that used in automotive exhaust systems.

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2. EXPANSION CHAMBERS

The general theory of expansion chambers is well covered in a 1952 paper by Davis *et al* (1) and some of their results for a single expansion chamber are shown in Figure 1. From these results it can be seen that the law of diminishing returns applies to increases in chamber area and the attenuation at a particular frequency is very dependent on chamber length, being zero when the chamber is an integer multiple of the frequency half-wavelength. It will also be observed that the correlation between measured and theoretical results deteriorates at higher frequencies, and this is even more noticeable with more complex configurations involving multiple chambers and connecting tubes. In this particular paper by Davis the effect of flow velocity is mentioned but not allowed for in the calculations; if flow velocities in excess of Mach 0.5 are expected there are numerous papers (e.g. (2)) by Professor P.O.A.L. Davies of I.S.V.R. and others to which reference could be made.

The simultaneous equations which require to be solved for two expansion chambers with a connecting tube are shown in Figure 2 and it can be seen that the determination of the attenuation of A_5 relative to A_1 requires some rather hairy calculations involving complex numbers and the writer has found that a computer spreadsheet is very useful in this respect.

A particular application involving a centrifugal fan in an air conditioning unit is shown in Figure 3. On this unit very tight in-duct sound power levels had been specified and there was an even stricter limitation on space available for attenuators and allowable pressure drop. It had been shown that the 250 Hz octave band was most critical and therefore only the calculations for this frequency band are shown in this paper.

Standard catalogue data for splitter attenuators made by Woods of Colchester is shown in Figure 4; these particular attenuators were chosen because they had 'squared' ends as distinct from 'bull-nose' and it was thought that it would therefore be easier to separate the absorptive and reactive components of the attenuation. This was done by extrapolating the curves down to zero length and assuming that the slope 'm' of the curve represented the absorptive component; in the particular case shown for a 100 mm airway this is 0.0152 dB/mm of splitter length. A cross-correlation was then carried out for various airway widths.

Figure 5 shows the target in-duct sound power level curve (specified in third-octaves) for the unit shown in Figure 3 and the fan manufacturer's levels in octaves. The following calculation for the inlet end of the unit was carried out for the 250 Hz octave band:

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L_w of fan	94.0 dB
L_w target (sum of ⅓-octaves)	75.3 dB
Attenuation required	18.7 dB
Attenuation by absorption	9.1 dB
Attenuation by reaction	11.2 dB
Total attenuation	20.3 dB

This assumed that the only absorption was from the 600 mm long splitters; although the plenum chambers were lined with 25 mm thick Rockwool the effect of this at 250 Hz was thought to be negligible.

For the reaction attenuation the source area S_1 was taken as the fan inlet flange which expanded in to the fan chamber S_2, followed by the splitter airways representing the 'connecting tube' S_3 in the equations, then another plenum chamber S_4 with a number of ducts feeding in to this chamber, the area of these ducts being summated to give the inlet area S_5. No allowance was made for changes in direction of the air flow at the unit inlets or the fan inlets. From Figure 5 and summing the third-octaves comprising the 250 Hz octave it can be seen that the measured L_w was 69.0 dB representing an actual attenuation of 25 dB. It was calculated that about 2 dB less attenuation would be available on the outlet side of the unit. The higher sound power levels at higher frequencies on the outlet side shown in Figure 5 were thought to be due to poor air distribution through the outlet splitters giving rise to high velocities in the top sections of the airways and hence higher levels of regenerated sound.

It is also of interest to note that the estimated pressure drop of these splitter attenuators was 26 Pa with an air flow of 5.11 m^3/s and this was also approximately the measured value.

3. $\lambda/4$ ATTENUATORS

From Figure 1 it can be seen that with a single expansion chamber a maximum attenuation is achieved when the length of the chamber is a quarter-wavelength ($\lambda/4$) but unfortunately this will give poor attenuation at integer multiples of the corresponding frequency. One way to overcome this problem is to have tubes protruding in to the expansion chamber (see Figure 6) for a length corresponding to $\lambda/4$ for the undesirable frequencies. In this particular example which was designed for the author by Professor P.O.A.L. Davies (3) medium-speed diesel main engines running at 420 rev/min on a passenger ship were giving a noise level on the games deck of 84 dB(A) - which was not very popular, especially as there were also high noise levels from the engine room supply fans!

The lowest frequency of interest was 28 Hz and there were also other frequencies which were multiples of engine speed and the silencer was designed particularly to tune out these frequencies. There was a limitation on the size of the silencer as it had to be 'shoe-horned' in to the existing funnel casing, and in the end the bottom of the silencer was elliptical and the top circular with an inclined axis as shown. Unfortunately after the silencers had been built it was decided to run the engines at 460 rev/min and therefore optimum acoustic performance was not obtained; even so a useful attenuation of 17 dB(A) was measured. The good correlation between predicted and measured attenuation is shown in the lower part of Figure 6.

This type of attenuator has also been successfully used for minimising the pulsations on the delivery side of screw compressors used in refrigeration plant. These compressors typically run at about 3000 rev/min on 50 Hz supply (3600 rev/min on 60 Hz supply) and if they have four lobes on the main rotor generate pulsations at 200 Hz, 400 Hz, 600 Hz and 800 Hz and higher harmonics. It has been found useful to use a two-chamber attenuator with two $\lambda/4$ tubes in each chamber (four in total) tuned to these four frequencies. Care must be taken to design for the velocity of sound in the particular refrigerant being used - remembering that it is operating in the superheated phase. It should be noted that there is quite a difference in the velocity of sound in common refrigerants such as R22 and ammonia. Tables are available for the velocity of sound in some refrigerants in the saturated phase but the author has generally found it necessary to calculate the superheated velocity of sound by using thermodynamic tables and the following equation:

$$c = V\sqrt{-\left(\frac{\partial P}{\partial V}\right)_s}$$

where
c = velocity of sound / m/s
P = pressure / Pa
V = specific volume / m^3/kg
S = entropy / kJ/(kg K)

In automotive systems this type of attenuator can be put to particularly good use and a computer program is available which compensate for high gas velocities (Mach No. $\rightarrow$ 1) and will also optimise the design of the attenuator over a given engine speed range. Using this program Professor Davies and his colleagues (4) developed a compact 'folded' silencer for a military vehicle where space was very restricted and a low noise level was required.

4. $\lambda/2$ ATTENUATOR

This type of attenuator (Figure 7) has been included just for completeness as the author only knows of one attenuator which has been designed on this principle. The problem was a very low frequency (7.2 Hz) air pulsation from the funnel of a diesel-engined ship and this was seriously affecting the accommodation on the ship. With a wavelength of about 75 m a $\lambda/4$ type of attenuator was obviously not practical and a low-pass filter type (see next section) would have created too much back pressure on the exhaust system. The solution was to split the gas stream in to two paths (a gas diverter is depicted in the bottom of the silencer) so that half of the flow went through the central 'straight-through' pipe and the other half through a helix - of the same cross-sectional area as the central pipe - which had a length a half-wavelength longer than the central pipe. When the two gas streams met at the top of the silencer the pulsations were therefore 180° out of phase and cancelled each other out.

This particular problem was encountered in 1958 and these days it is possible that an attempt would have been made to overcome it with the use of 'active' noise control - if active noise generators could be designed which could withstand the high temperature (454 C) and corrosive effects of the exhaust gases and the sea-going environment.

5. LOW-PASS FILTER ATTENUATORS

The design of passive electrical filters is covered in a number of textbooks and a simple low-pass filter is shown in Figure 8 which consists of a series reactance (or choke) and a parallel capacitance. The shape of the amplification curve is depicted with the so-called cut-off frequency (f_c) at the point where the attenuation is 3 dB.

The acoustical analogy of the electrical circuit is also shown where the acoustical reactance ($\rho L_t/A$) consists of a comparatively small diameter pipe in the gas stream and the acoustical capacitance ($V/\rho c^2$) is an expansion chamber of a particular volume.

The author tends to use this type of attenuator for fairly low-speed and medium-speed reciprocating air compressors and refrigeration compressors where the wavelength of the sound in the fluid is rather long for a $\lambda/4$ type attenuator. An example of this is given in the figure where the machine was an 8-cylinder compound refrigeration compressor with six cylinders on the low stage and two on the high stage with the capability of unloading one of the latter, thus giving a once per revolution fundamental frequency of 24.8 Hz and hence $\lambda/4$ of 1.92 m.

It is generally accepted that it is advisable for the volume of the expansion chamber to be at least ten times the swept volume of a cylinder of the machine and in practice it has been found worthwhile to first of all determine the diameter and length of the expansion chamber - often to fit in to the space available and using standard sizes of tube - and then to calculate the dimensions of the impedance tube - again using standard sizes of tube and keeping in mind additional pressure drop.

In the case shown the high gas pulsations were not only causing high noise levels but they were also severely damaging the internals of an oil separator about two metres from the compressor in the delivery line. It was also fortunate that it was possible to measure the magnitude of the pulsations in the refrigerant both just after the compressor and before the oil separator so that the performance of the attenuator could be measured, and it was found that a 39 dB attenuation had been achieved at the fundamental frequency and 42 dB at twice this frequency. If this chamber's dimensions are compared with those in Figure 1 it can be seen that a plain expansion chamber would only be expected to give about 25 dB attenuation at the fundamental frequency and therefore it can be assumed that the extra 14 dB is largely due to the low-pass filtering effect.

6. ATTENUATOR 'ABUSES'

The most common misapplication of reactive attenuators is to position them at some distance from the source of the pulsations in which case the attenuator can easily be a half-wavelength from the source at a particular frequency and give rise to amplification of the pulsation. An oil separator, condenser or other pressure vessel in a compressor delivery system can act as an effective attenuator and protect other items downstream of it but, if badly located as was the case for the machines in Figures 7 and 8, the effects can be damaging. Conversely the author has deliberately designed in some $\lambda/4$ tubes in to 'close-coupled' oil separators so that their natural attenuation could be improved.

Similarly cases have been known when a low-pass filter attenuator has been designed, or a $\lambda/4$ attenuator, and the designer has not noted that the length of the expansion chamber has been a half-wavelength or multiple for an important frequency with consequent disaster. Examination of Figures 3, 6 and 8 will show that this effect has been considered in these cases.

Another effect which is not so easy to design against are the natural frequencies of the body of the attenuator, especially at higher frequencies.

In refrigeration compressors it is common practice for the compressors to utilise the crankcase of the compressor as a suction expansion chamber and therefore it is rarely necessary to incorporate additional attenuators in the suction line. However on the delivery side modern compressors have very small incorporated expansion chambers and there is an increasing necessity to fit attenuators in the delivery line. Also with refrigeration compressors fairly large quantities of lubricating oil are mixed with the delivery gas and on reaching an expansion chamber this oil separates out; care must then be taken in the design of any attenuator to self-drain this oil to prevent the attenuator becoming 'choked'.

There are a number of 'catalogue' pulsation dampers on the market, particularly for the refrigeration industry, but data on the attenuation they give in normal working conditions, particularly at specific frequencies, is not generally available. Unfortunately the manufacturers of these devices, which generally just consist of a number of 'untuned' small expansion chambers, do not advise that they should be fitted close to the compressor although the 'bible' of the refrigeration industry (5) - which like the other Bible is not often looked at for guidance - does give some useful recommendations.

7. REFERENCES

1. Davis, D.D. *et al.* Theoretical and experimental investigation of mufflers with comments on engine-exhaust muffler design. *National Advisory Committee for Aeronautics (NACA) Report 1192.* 6 October 1952.

2. Alfredson, R.J. and Davies, P.O.A.L. Performance of exhaust silencer components. *J. Sound Vib.*, 1971, 15(2), 175-196.

3. Davies, P.O.A.L. Exhaust system silencing. *Proc. Imas 73*, 1973, 4/59-4/65.

4. Davies, P.O.A.L. and Alfredson, R.J. Design of silencers for internal combustion engine exhaust systems. *Instn. Mech. Engrs*, Automobile Division conference volume Vibration and Noise in Motor Vehicles, 1971, C96/71.

5. Guide and Data Book. *American Society of Heating Refrigerating and Air-Conditioning Engineers (ASHRAE).* 1970 edition.

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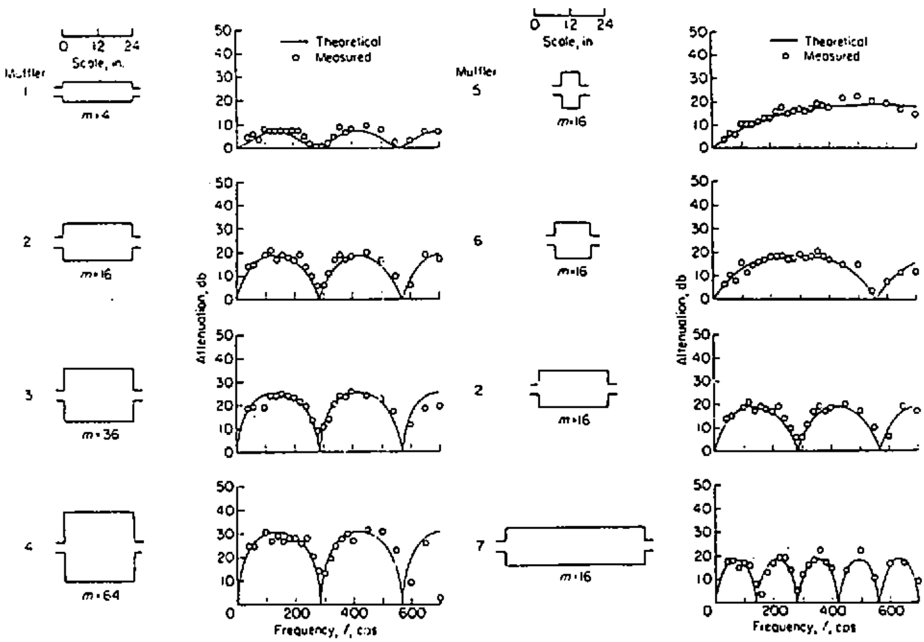
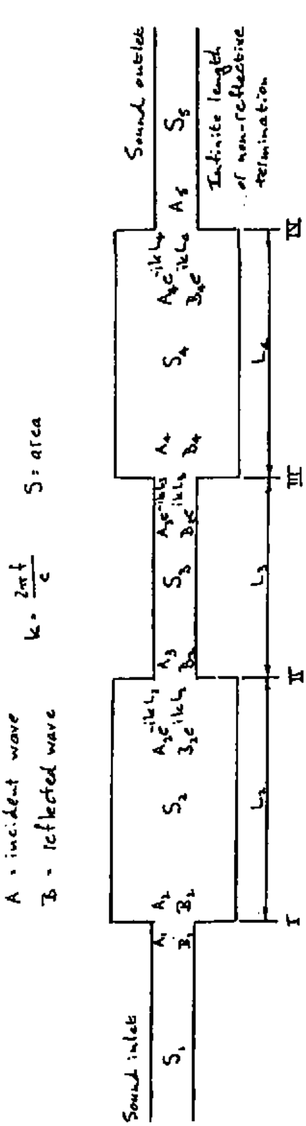


(a) Effect of expansion ratio *m*

(b) Effect of length

Figure 1. Attenuation of single expansion chambers.



$m_I = \frac{S_2}{S_1}$ $\quad$ $m_{II} = \frac{S_2}{S_3}$ $\quad$ $m_{III} = \frac{S_4}{S_3}$ $\quad$ $m_{IV} = \frac{S_4}{S_5}$

$A_1+B_1 = A_2+B_2$ $\quad$ $A_2e^{-ikL_2}+B_2e^{ikL_2} = A_3+B_3$ $\quad$ $A_3e^{-ikL_3}+B_3e^{ikL_3} = A_4+B_4$ $\quad$ $A_4e^{-ikL_4}+B_4e^{ikL_4} = A_5$

$A_1-B_1 = m_I(A_2-B_2)$ $\quad$ $m_{II}(A_2e^{-ikL_2}-B_2e^{ikL_2}) = A_3-B_3$ $\quad$ $A_3e^{-ikL_3}-B_3e^{ikL_3} = m_{III}(A_4-B_4)$ $\quad$ $m_{IV}(A_4e^{-ikL_4}-B_4e^{ikL_4}) = A_5$

I	II	III	IV
$2A_2 = A_1\left(1+\frac{1}{m_I}\right)+B_1\left(1-\frac{1}{m_I}\right)$	$2A_3 = A_2(1+m_{II})e^{-ikL_2}+B_2(1-m_{II})e^{ikL_2}$	$2A_4 = A_3\left(1+\frac{1}{m_{III}}\right)e^{-ikL_3}+B_3\left(1-\frac{1}{m_{III}}\right)e^{ikL_3}$	$A_5 = A_4e^{-ikL_4}+B_4e^{ikL_4}$
$2B_2 = A_1\left(1-\frac{1}{m_I}\right)+B_1\left(1+\frac{1}{m_I}\right)$	$2B_3 = A_2(1-m_{II})e^{-ikL_2}+B_2(1+m_{II})e^{ikL_2}$	$2B_4 = A_3\left(1-\frac{1}{m_{III}}\right)e^{-ikL_3}+B_3\left(1+\frac{1}{m_{III}}\right)e^{ikL_3}$	$A_5 = \left(A_4e^{-ikL_4}-B_4e^{ikL_4}\right)m_{IV}$
$A_1 = \frac{1}{2}\left\{(1+m_I)A_2+(1-m_I)B_2\right\}$	$A_2 = \frac{1}{2}\left\{\left(1+\frac{1}{m_{II}}\right)A_3+\left(1-\frac{1}{m_{II}}\right)B_3\right\}e^{ikL_2}$	$A_3 = \frac{1}{2}\left\{(1+m_{III})A_4+(1-m_{III})B_4\right\}e^{ikL_3}$	$A_4 = \frac{1}{2}\left(1+\frac{1}{m_{IV}}\right)A_5e^{ikL_4}$
$B_1 = \frac{1}{2}\left\{(1-m_I)A_2+(1+m_I)B_2\right\}$	$B_2 = \frac{1}{2}\left\{\left(1-\frac{1}{m_{II}}\right)A_3+\left(1+\frac{1}{m_{II}}\right)B_3\right\}e^{-ikL_2}$	$B_3 = \frac{1}{2}\left\{(1-m_{III})A_4+(1+m_{III})B_4\right\}e^{-ikL_3}$	$B_4 = \frac{1}{2}\left(1-\frac{1}{m_{IV}}\right)A_5e^{-ikL_4}$

Figure 2. Equations for double expansion chambers.

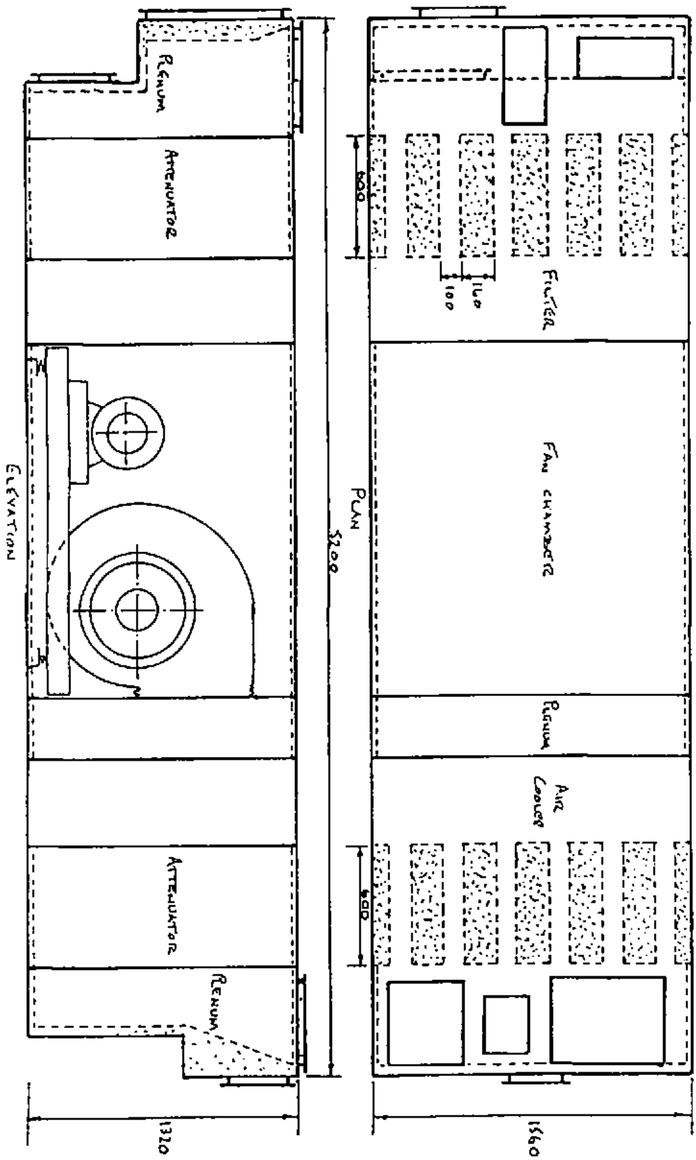


Figure 3. Arrangement of Air Conditioning Unit.

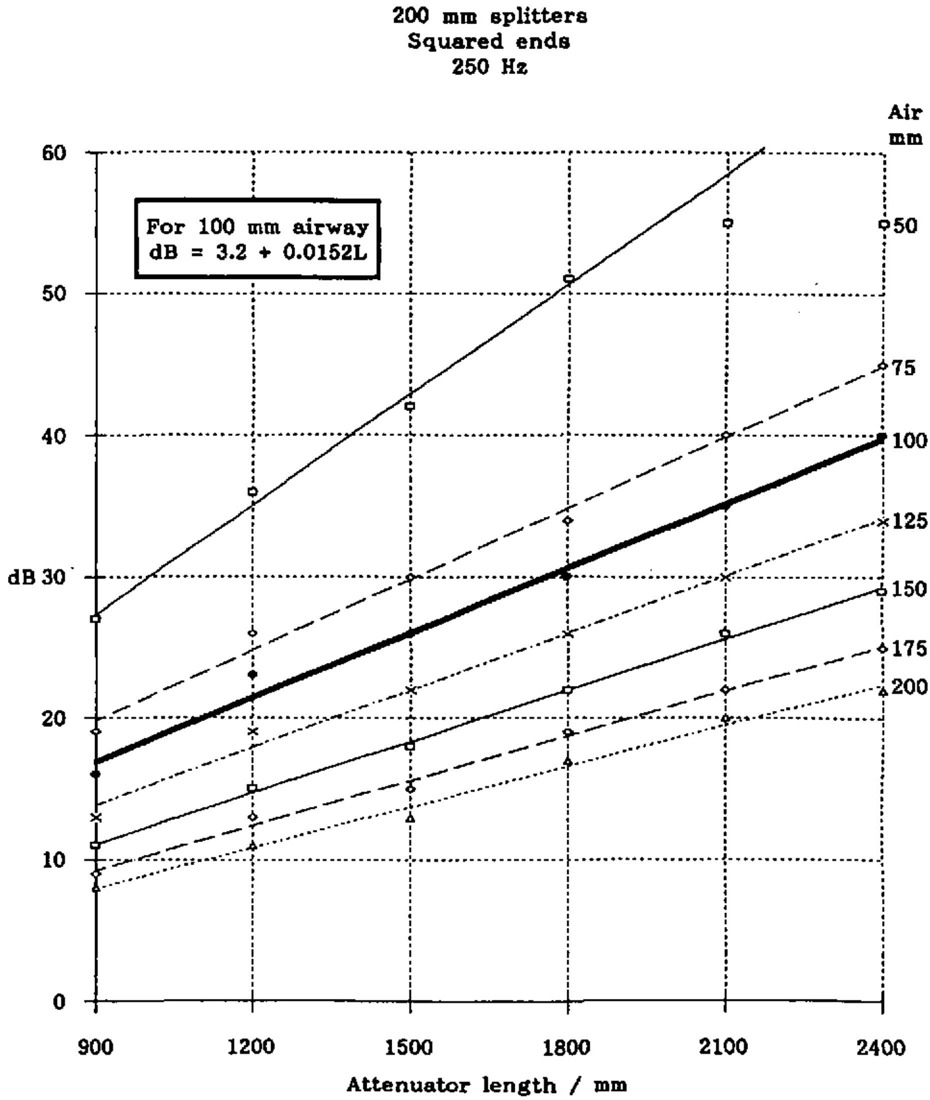


Figure 4. Woods splitter attenuators.

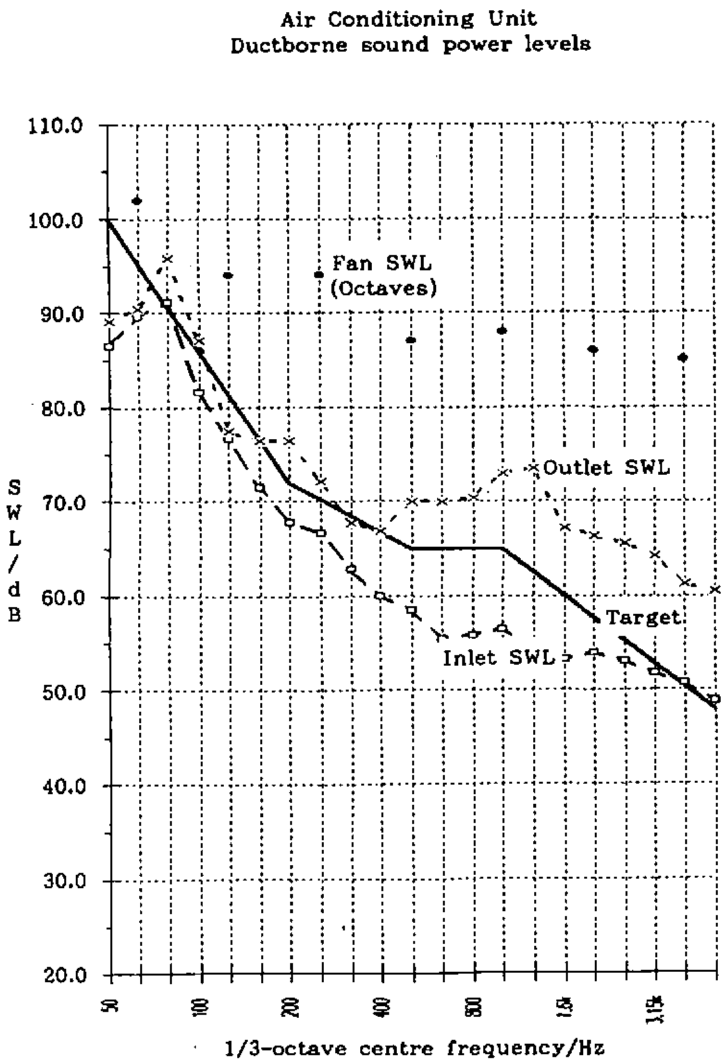


Figure 5. Attenuation of absorptive/reactive attenuators.

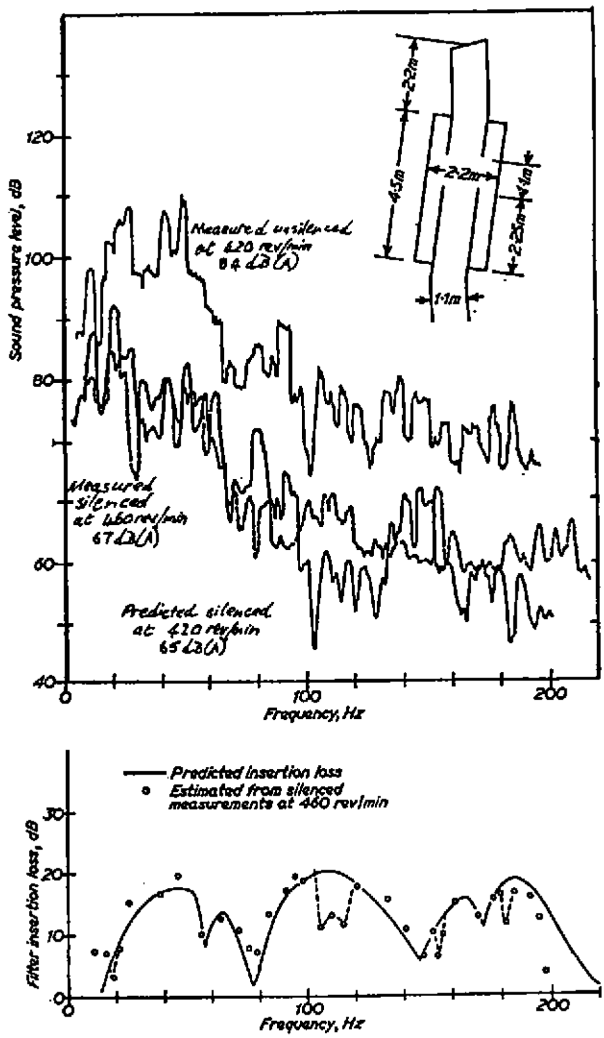
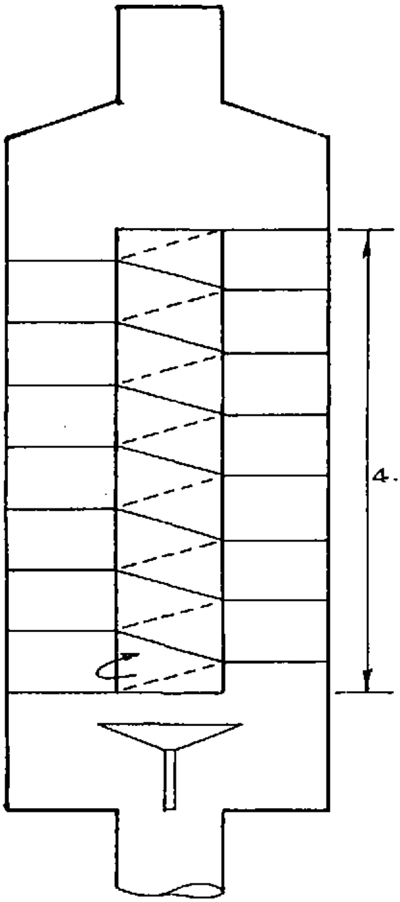


Figure 6. Reactive (λ/4) attenuator for ship's main engines.



Pipe diameter	813	mm
Silencer diameter	2743	mm
Helix pitch	533	mm
($7\frac{1}{2}$ turns)		
Engine speed	108	rev/min
(4 cylinders)		
Cylinder frequency	7.2	Hz (= f)
Exhaust temperature	454	C
	728	K

$\frac{1}{2}$-wavelength $= 10.025\sqrt{K}/f = 37.6$ m

Helix length = straight length + $\frac{1}{2}$-wavelength
= 4.1 m + 37.6 m
= 41.7 m

Figure 7. Reactive ($\lambda/2$) attenuator for ship's main engine.

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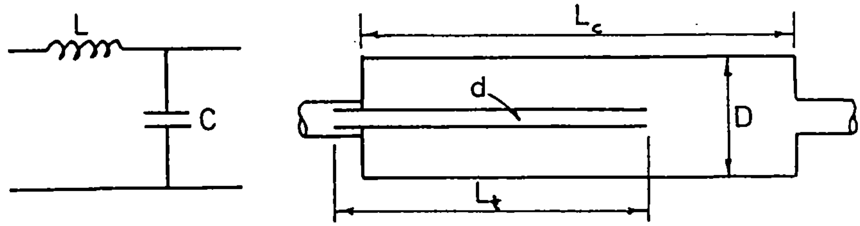

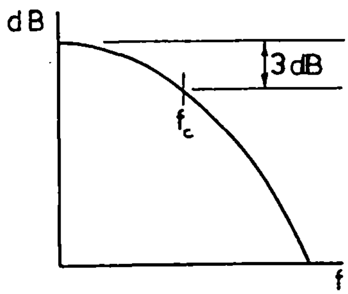

$L \equiv \rho L_t/A$

$C \equiv V/\rho c^2$

ρ = density

c = velocity of sound

f_c = cut-off frequency

$$A = \frac{\pi d^2}{4} \qquad V = \frac{\pi D^2 L_c}{4}$$

$$\omega_c = 2\pi f_c = \frac{2}{\sqrt{LC}} = \frac{2c}{\sqrt{\frac{L_t V}{A}}}$$

Cylinder 5" bore x 4" stroke
Swept volume per cylinder 1.287E-3 m^3

Chamber diameter 209.3 mm   Chamber length 750.0 mm
Chamber volume 25.8E-3 m^3 i.e. > 10 x swept volume

Velocity of sound (R22) 190.0 m/s
Cut-off frequency f_c 12.4 Hz = 50% x 1488 rev/min / 60

Tube diameter 27.3 mm   Tube length 540.0 mm

Wavelength for 24.8 Hz 7.66 m

Measured attenuation of gas pressure pulsation

39.4 dB in 31.5 Hz octave
42.2 dB in 63 Hz octave

Figure 8. Low-pass filter attenuator for reciprocating compressor.