THE INFLUENCE OF BAFFLE ORIENTATION ON SOUND ATTENUATION OF DISSIPATIVE SILENCERS

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1. INTRODUCTION

Noise transmission in HVAC systems is often controlled with the installation of "dissipative silencers" in the air-ducts. For rectangular ducts, the preferred arrangement consists of several baffles. The greater the length of the baffle, the greater amount of acoustic energy absorbed. Two other parameters control the sound absorption: the thickness of the baffle and the size of the air space between the baffles (1,2,3). Another geometric parameter, namely the ratio of the cross-sectional area to the perimeter also influences sound absorption. Although this has been known for a considerable period of time, no one seems to have examined this effect in a systematic manner to date. This experimental study reports some interesting results for ducts with square cross-sections and un-conventional baffle arrangements.

2. METHOD

The noise reductions afforded by 10 different fiberglass baffle configurations were measured. Each configuration consisted of rigid, acoustic grade fibreglass board cut to appropriate size. These were installed at one end of a 122cm long 30cm square steel duct. An ILG fan was used as a sound source. This arrangement is not as sensitive to radiation loading are loudspeakers that are often used in model experiments. The test section was attached to a 91cm long duct that terminated in a reverberant space. The insertion loss was taken to be the difference between the sound pressure level with and without silencer elements.

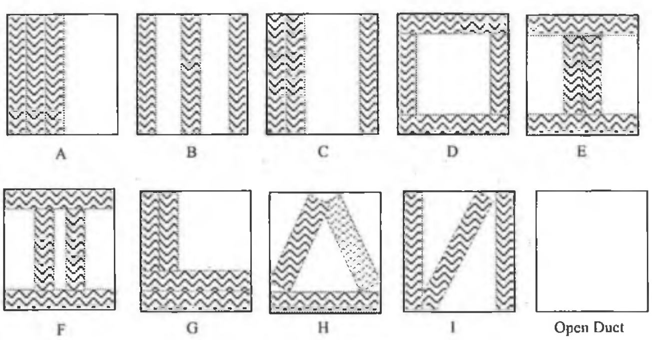

Figure 1. Schematic of test configurations

3. RESULTS

All measured data exhibited poor attenuation at low frequencies, rising to a maximum, and then decreasing again at higher frequencies. The measured data was very repeatable; deviations from the mean were within +/- 0.5 dB.

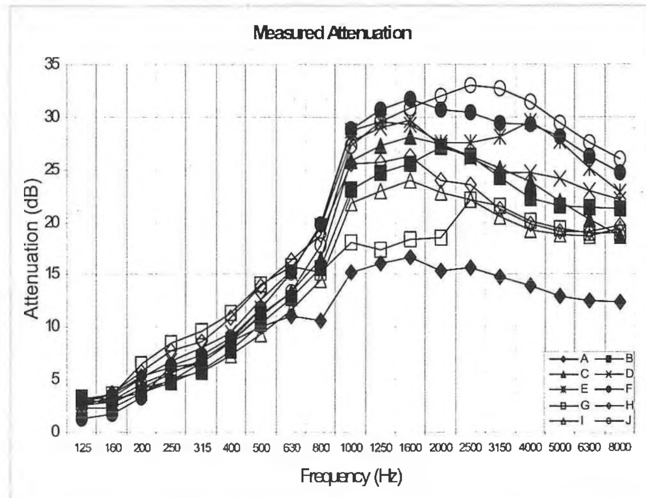

Figure 2. Measured Insertion Losses

For design purposes it is important to select the frequency (f_{peak}) where maximum attenuation (IL_{max}) is required. This is because the attenuation band of most silencers is not easily widened. Suppose the maximum attenuation L_0 occurs at f_0., and the levels and frequencies in the neighboring bands are (L_1,$0.8f_0$) and (L_2, $1.25f_0$). Introducing $z=(f/f_0 -1)$ one can generate three equations for the coefficients of the quadratic: $L(z)=a+bz+cz^2$. The quadratic has a maximum at $z=-0.5b/c$ from which $IL_{peak}=L_0-0.25b^2/c$, and $f_{peak}=f_0(1-0.5b/c)$ follow.

Table 1

	A	B	C	D	E	F	G	H	I
fmax (Hz)	1623	2117	1681	1653	3838	1708	2620	1697	2603
ILmax (dB)	17.0	27.4	28.2	29.7	29.7	31.9	22.3	23.9	33.0
Dh	0.20	0.12	0.20	0.20	0.14	0.10	0.20	0.11	0.11
λ/Dh	1.03	1.32	1.00	1.01	0.65	1.96	0.64	1.87	1.17
$\mathcal{L}$/Dh	2.95	4.92	2.95	2.95	4.43	5.91	2.95	5.60	5.35
Ao/A	0.50	0.50	0.50	0.44	0.44	0.44	0.44	0.54	0.49
Ad/Ao	1.00	0.50	1.00	1.00	0.50	0.33	1.00	0.33	0.46
Pl/Pd	0.33	0.80	0.67	1.00	0.67	0.75	0.50	0.62	0.76

4. DISCUSSION

Conservation of energy suggest that

$$p^2(x)A_d/\rho c = p^2(x+dx)A_d/\rho c + \alpha p^2 P_l dx/\rho c$$

Here A_d is the duct area, P_l the lined perimeter of the duct and α a measure of the amount of energy absorbed. The differential form is has a solution $p^2(x)=p^2(0)e(-\alpha P_l x/A_d)$. Noting that $10\log[p^2(0)/p^2(x))]$ is the attenuation, it follows that $IL=bP_l\mathcal{L}/A_d$ which can be expressed in terms of geometric parameters of the air passage: $IL=B\ P_l/P\mathcal{L}/D_h$, where D_h is the hydraulic diameter the open air-passage. Now B is not likely to be a constant, as the attenuation depends on the amount of energy dissipated in the sound absorbing material. The absorption coefficient of fibrous material is a function of the thickness of the material. The factor B takes the form: $60\ t_{eff}^{\ 0.65}$; $t_{eff}=(A/Ao-1)D_h\ P/P_l$ when IL is a maximum (IL_{max}).

The theory of dissipative silencers suggests that the peak frequency and the inter-baffle spacing (h) are related $B=\frac{2hf_{peak}}{c}$ where c is the speed of sound and f_{peak} is the frequency for which peak attenuation is realized. The present configurations differ from the typical baffle arrangement, and an effective inter-baffle spacing h_{eff} is required. A review of the measured data shows that the attenuation curves are not all self- similar. This suggests that it may not be possible to extend the concept of h_{eff} to non-parallel baffle configurations. It is likely that h_{eff} will correlate with the hydraulic diameter D_h and square root of the individual air-duct cross-sections. In fact the data for both fall into three distinct groups. and $c/(D_h f_{peak})$ falls into three distinct groups

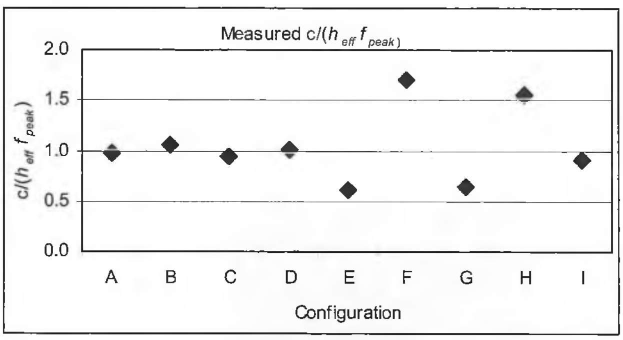

Figure 3. Pattern of $c/(h_{eff}f_{peak})$ ratios

Configurations that are of the parallel baffle type group tend to take on values near unity, whereas configurations with more complex geometry take on higher or lower values. The results are shown in figure 3 use the parameter

$$heff = 0.5(D_h + A_d^{\ 1/2})$$

In view of the limited number of configurations, it is not possible to establish a trend. It does appear that configurations that contain multiple passages with different wall treatment (Configurations F and H) have $c/(h_{eff}\, f_{peak})$ ratios greater than unity. Those with L-shaped lay-out (E,G) have $c/(h_{eff}f_{peak})$ less than unity.

5. SUMMARY

Several silencer configurations suitable for installation in rectangular ducts have been tested. It was found that the peak insertion loss could be described by a general expression that takes into account several geometrical parameters that describe the configuration:

$$IL_{max} = 60\ (A/A_o\text{-}1)P_l/P^{\ 0.35}\ L/D_h\ (\text{dB})$$

This value occurs at

$f_{peak}=c/h_{eff}$; $heff = 0.5(D_h+A_d^{\ 1/2})$: Similar ducts
$f_{peak}=1.5c/h_{eff}$; $heff = 0.5(D_h+A_d^{\ 1/2})$: L shaped baffles
$f_{peak}=0.6c/h_{eff}$; $heff = 0.5(D_h+A_d^{\ 1/2})$: different ducts

All silencer configurations tested had the same length, making it impossible to separate out end-effects, which are due to diffraction. As the magnitude of the end effects is normally small (2-4 dB), one would expect that the above formalism will not change significantly, when this effect is accounted for.

6. ACKNOWLEDGEMENTS

The author would like to thank Mr. A. Della Penna President of Coolbreeze Air Conditioning and Heating Inc. for making available the test facility and instrumentation, and Dr. Sharbaf Ebrahimi for his guidance and encouragement. This paper is based on Experimental Investigation of the Effect of the Configuration of Fibreglass Baffles on Sound Attenuation in a Rectangular Duct, an extended essay in partial fulfillment of the requirement of the author's International Baccalaureate (2001).

7. REFERENCES

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