THE DEVELOPMENT OF VACUUM ISOLATING PANELS FOR NOISE CONTROL APPLICATIONS

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1 INTRODUCTION

TIn the field of noise control, a commonly evoked acoustic principle is that of sound isolation through the use of massive materials. Whereby the more massive a material is the more suitable it is for attenuating sound. Consequently most high performance sound isolation measures are 'massive' by nature usually including a cavity to increase high frequency performance.

There is another principle of attenuation that can be explored, and that is to reduce the acoustic conductivity of the isolation medium. Not only does this lower the potential for longitudinal pressure waves to be transferred through the medium. To reduce the conductivity of the medium, this research proposes a vacuum. It is a well known principal that sound cannot exist in an absolute vacuum.

Qualitative Term	Pressure (Pascals)	Relative Percentage
Atmospheric Pressure	101.3 kPa	0%
Low Vacuum	100 kPa to 3 kPa	0.013% to 97%
Medium Vacuum	3 kPa to 100 mPa	97% to 99%
Outer Space	100 µPa to <3fPa	> 99.999%
Perfect Vacuum	0 Pa	100%

Table 1. Vacuum qualitative definitions

Could such a design derived from this principal serve as an effective method of noise control? [1] First a 'massive' sound isolating enclosure was needed as a control

2 BACKGROUND

One of the biggest challenges in designing any vacuum panel is to maintain structural integrity under the considerable pressures that all surfaces of the panel are subject, approximately 10 tonnes per square metre. The most obvious way of reinforcing a panel to withstand such forces is to physically brace the structure, creating bridging paths (coupling) between opposing faces, across the vacuum. While this bridging may prevent the panel from collapsing, it facilitates the transfer of sound (and heat) and this compromises the performance of the panel.

There is an established industry concerned with Vacuum Insulation Panels (VIPs), with a focus on energy conservation for application in the building industry. Consequently, the design parameters of such panels focus on minimising the thermal conductivity. Some of the design aspects that achieve this are at odds with maximising acoustic absorption and sound transmission loss. It would seem pertinent that considering a different design criteria to maximise sound transmission loss, might result in a significantly different design for a VIP. This paper will make a distinction of terms and call this a Vacuum Isolation Panel (VIsoP). There should however be an amount of applicable overlap in elements of the construction and materials which should be evaluated, e.g. seals, valves, films, pumps etc. would assist in design and testing of a VIsoP. Little research in assessing the sound attenuation performance of VIP panels has been found [2,3,4].

It was necessary to construct a conventional sound isolation enclosure, based on the mass principle, with the aim of achieving at least 6dB higher attenuation than the anticipated performance of the VIsoP, in all octave bands, in accordance with ISO 140-1:1998[5] . This will be measured with respect to a broadband noise source, sealed within the enclosure. To establish a baseline point of comparison for the subsequent tests of the various VIsoP designs, the noise source will be measured at 1 m from the open enclosure, in an anechoic chamber, see figure 1, the radiation direction characteristics were not measured. Once, the final panel is added modal resonances should be considered as the enclosure was of equal dimensions, 0.54m. Hence first order modal resonances occurred at 317 Hz (1st axial), 448 Hz (1st tangential).

Figure 1- open control enclosure

Figure 2 Squash balls panel under a vacuum

The design target was to try and reach the highest level of attenuation with respect to a workable mass. (The final control enclosure mass when fitted with the dampening cladding was approximately 50kg). To this end four enclosure designs were tested, see figure 3.

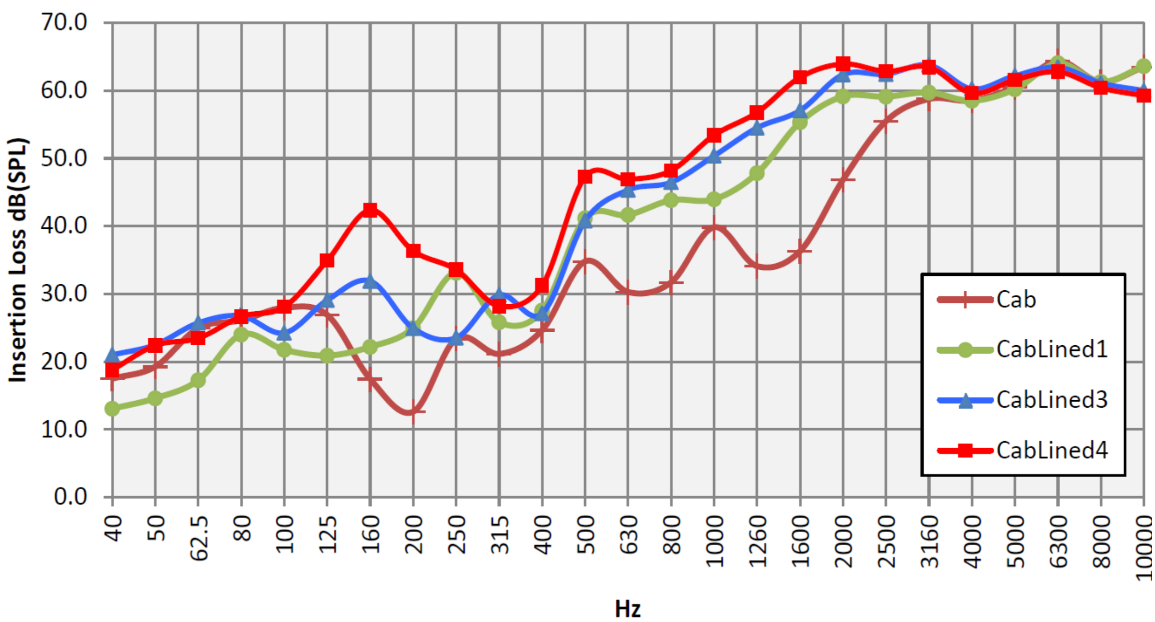

Figure 3 Insertion loss performance of evolving, ‘massive’, control enclosure prototypes

3 THEORETICAL VACUUM PANEL PERFORMANCE

In order to estimate the levels of vacuum necessary to achieve degrees of attenuation, a series of calculations were undertaken to determine the mechanical load the test panels would need to endure.

At partial vacuum there are less air molecules for the pressure wave to transfer its energy. So to adhere to the principle of energy conservation; when a pressure wave reaches the area of lower density, the energy would need to be suddenly dispersed over a much larger volume, or each particle would need to increase velocity. Neither are physically possible, and so instead, some of the energy simply reflects back from the boundary surface.

In the case of this experiment, the value of r_1, is the acoustic impedance, and is determined by the density of atmospheric air, r and the speed of sound c , r_1= r_1c_1. The reflection coefficient R is given by

$$R = \frac{reflected \quad pressure \quad amplitidue}{incident \quad pressure \quad amplitude}, \frac{r_2 - r_1}{r_2 + r_1}$$

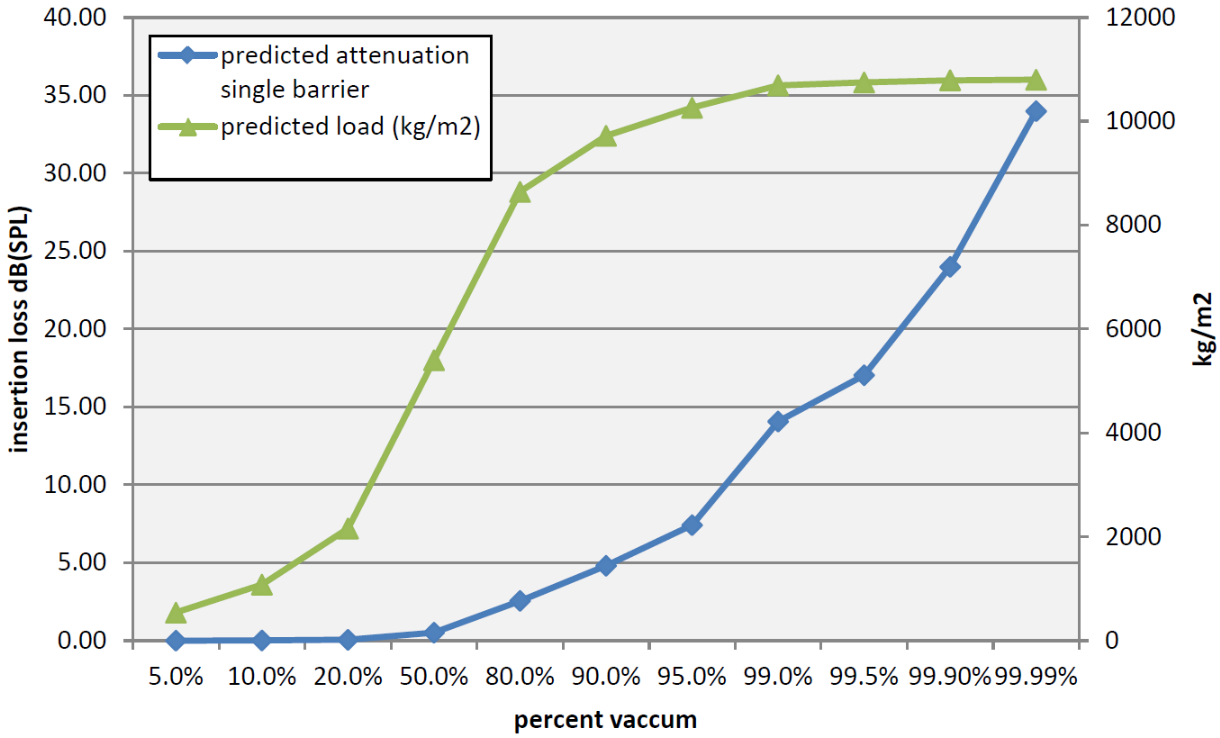

Figure 4 Theoretical insertion loss performance compared to pressure load under varying vacuums

It can be seen from the series of predictions that; the pressure must be reduced by approximately 80% before an audible difference is expected to be perceived (3 dB); by which time the panel is already subjected to nearly 9000 kg/m$^{2.}$

As the vacuum increases above 80%, the projected attenuation continues to rise exponentially, whilst the rate of pressure continues to increase linearly. This means that if the panel can withstand the load induced from 80% vacuum, it has only 20% more load to bear to potentially realise magnitudes more attenuation.

4 BRIDGED VACUUM PANEL DESIGNS

The load on the vacuum panel has to be countered. Two solutions were purposed: Mechanical braced panels using flat faced panels with structural elements in the cavity; and internally resistant panels which has the advantage of reducing coupling. The simplest design is a bridged structure and hence these were the first designs prototyped.

Figure 5 Lattice structure panel

Figure 6 Point load panel structure

Both latticed and point load designs maintained an 80% vacuum with a 3150 Kg load across the panel, see figure 2. Insertion loss measurements were taken under anechoic conditions using a pink noise source generating 100 dB(lin) over a 10 second period using a class 1 sound level meter 10 cm from the vacuum panel face.

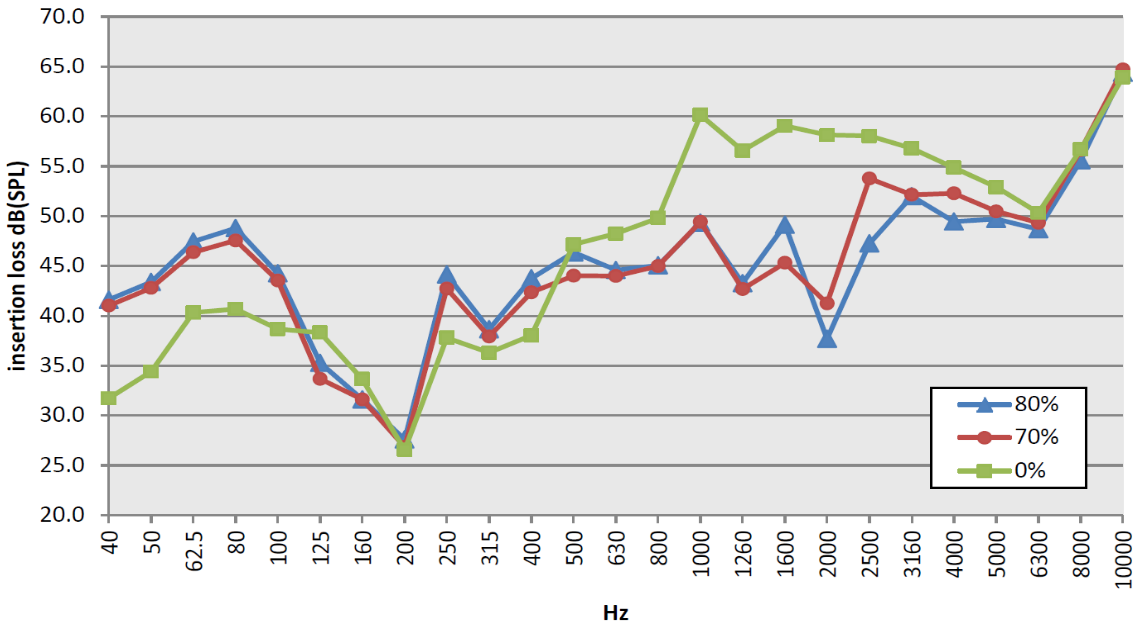

Figure 7 Insertion loss performance of the flat lattice vacuum isolating panel under vacuum.

While their low frequency attenuation (80Hz) was impressive an 8 dB increase over the air filled panel at frequencies less than 100 Hz, see figure 7. However, the A-weighted attenuation performance degraded as the vacuum increased which resulted in a distinctly and immediately noticeably subjective increase in loudness as the lattice panel was vacuumed. This centered around 2kHz, which was particularly audible. It is also interesting to note that the frequencies that worsened under vacuum correlate with the findings of Maysenholder, as noted by Tenpierik[2].

5 UNSUPPORTED VACUUM PANEL DESIGNS

To avoid the structural bridging a parabolic shape was pursued. This was formed from two differing arc lengths; edge to edge and corner to corner.

Figure 8 Layered Epoxy and fibreglass panel

Figure 9 With added metal reinforcement frame

After several redesigns the convex VIsoP sustained the highest of vacuum, 95%, offering the cleanest set of results with minimal physical coupling. Figure 10 shows sets of results with varying degrees of vacuum, compared to the theoretical mass law (ML) based predictions, all compared with the heavy control cabinet panel which was used as a baseline.

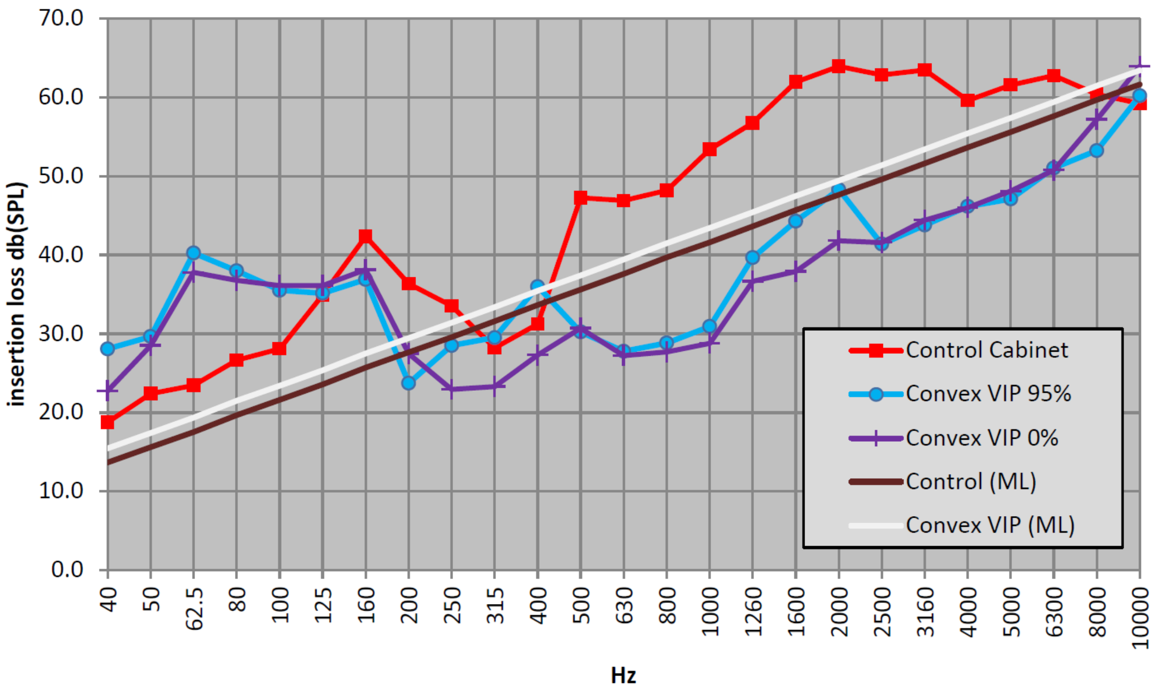

Figure 10 Insertion loss performance of the unsupported vacuum panel compared to the mass law

There are a number of interesting comparisons in figure 10. It can be seen that the VIsoP, whether under vacuum or not, performs better than the control enclosure up to 125 Hz, but then not as well for the most of the remaining spectrum; despite the VIsoP being more massive, 8.3 kg than the control panel, 6.7 kg. There was a clear deviation is at 500Hz where the VIsoP underperforms the mass law until reconverging at 10 kHz. The panel under vacuum performed 5 dB better between

250 and 400 Hz and again between 1600 and 2000 Hz than when filled with air. Unfortunately, due to the failure of the experimental methodology to account for enclosure resonance, noise reduction, comparisons of each panel to the theoretical performance of their constituent mass cannot be considered accurate. However a theoretical comparison based on one and two leaf prediction assuming infinite reflections was possible, see figure 11.

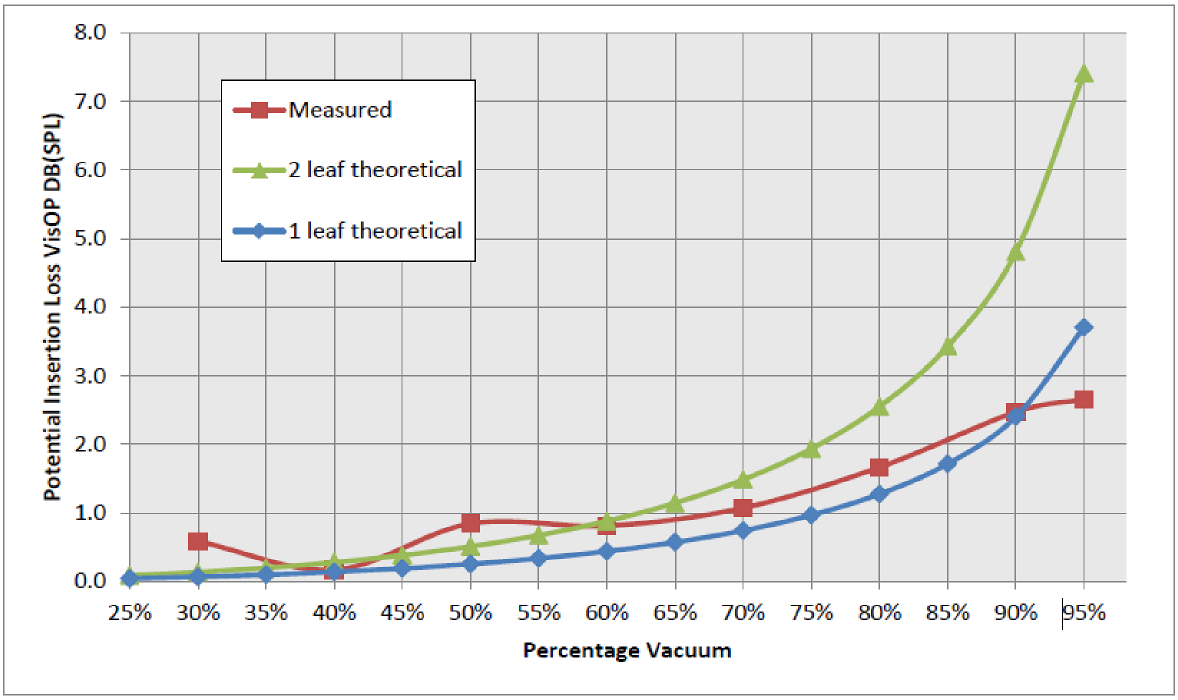

Figure 11 Potential insertion loss of the VIsoP compared to 1 and 2 leaf predictions under vacuum

Based on the measurement performance of the VIsoP a simplified empirical formulae was developed to predict the insertion loss of a panel under vacuum

$$Vac_{IL} = -10 \log\left(1 - (0.7Vac\,\%)^2\right)$$

or extrapolated to the full range of vacuums

$$Vac_{IL} = 10 \log\left(e^{(0.25\,Vac\,\%)}\right) + 0.25$$

Given the real world complications present in any VIsoP, requiring at least two leafs by definition, and thereby two acoustic impedance changes, it seems reasonable to assume that the panel could not perform as well as the theoretical predictions, but better than a single leaf barrier with just one impedance change.

6 CONCLUSIONS

A number of new results were found both positive and negative as regards lightweight vacuum isolating panels. It can at least be concluded after a great deal of prototype development that it is extremely difficult to construct a panel with an internal cavity that is capable of resisting atmospheric pressures; made even more difficult if trying to do so from lightweight materials and manual fabrication processes. It was found that a VIsoP design that incorporates direct physical coupling as a means of resisting the external inward pressure is unlikely to outperform a non-coupled design, and furthermore is likely to perform worse under vacuum than when at equivalent pressure (0% vacuum).

It has been observed within the limitations of the employed experimental methodology that sound attenuation potential of a custom VIsoP, while exponential, is unlikely to yield a perceivable (broad spectrum) insertion loss until at least 90% vacuum level is reached, by which time 90% of the mechanical load (atmospheric pressure) will have been endured. Finally, it was established that low frequency insertion loss of a lightweight panel can be increased using VIsoP with a 95% vacuum.

7 REFERENCES

1. S. Walters, Noise control potential of vacuum isolating panels, MSc Thesis, London South Bank University, (2012).
2. M J Tenpierik, Vacuum insulation panels applied in building constructions. PhD Thesis, Delft University of Technology, Delft, (2009).
3. R. Baetens, B Jelle, J. Thue,M J Tenpierik, S. Grynning, S. Uvlokk, A. Gustavsen, Vacuum insulation panels for building applications: A Review and Beyond, Energy and Buildings, 42(2), 147-172, (2010).
4. W Maysenholder, Sound Transmission Loss of Vacuum Insulation Panels, Fraunhofer Institute for Building Physics, Stuttgart, Germany, (2008).
5. BS EN ISO 140-1:1998 Incorporating Amendment No.1, Mar 23, (2005).