Noise Control Potential of Vacuum Isolation Panels

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ABSTRACT

In the field of noise control, a commonly evoked acoustic principle is that of sound isolation through the use of massive materials. This paper explores the potential of a vacuum to provide noise control measures. The primary aim is to quantify the relationship between the acoustic attenuation potential of a custom built Vacuum Isolation Panel and a degree of vacuum. Through doing so, this research provides an understanding of the magnitude of physical forces such designs would need to endure to offer any perceivable level of noise reduction. This in turn provides valuable data for future feasibility studies into the design and commercial viability of product development this area. The results showed that the relationship between the quantities is exponential, but only begins to be perceivable above a 90% vacuum level and thereby, 90% mechanical load.

Keywords: Sound, Isolation, Transmission I-INCE Classification of Subjects Number(s): 33

1. INTRODUCTION

A principle commonly evoked in sound isolation measures is mass principle. Whereby the more massive a material is (and usually of high density) the more suitable it is for attenuating sound. These are sometimes referred to as mass barriers. Consequently most high performance sound isolation measures are 'massive' by nature.

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; it lowers the acoustic impedance of the transmission layer. This creates a dramatic impedance mismatch with the surrounding media, causing reflection. Much like a person in a fish-tank trying to hear an alarm bell ringing in an adjacent fish tank, where the two tanks are separated by air, a medium of lower acoustic impedance than water.

To reduce the conductivity of the medium, this project proposes a vacuum. It is a well known principal that sound cannot exist in an absolute vacuum. What is not well understood, empirically, is to what level sound can exist at levels of partial vacuum; and when it comes to applying this principal to noise control, to what degree will sound conduct through a panel, or full enclosure, where the cavity of the panels is in a vacuum state? Are other principals such as coincidence and resonance rendered irrelevant also? Could such a design derived from this principal serve as an effective method of noise control?

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 document will make a distinction of terms and call this a Vacuum Isolation Panel (VIsoP).

Having made that distinction, while the design of a VIsoP may differ considerably from a conventional VIP, there would be an amount of applicable overlap in elements of the construction and materials that should be evaluated. Knowledge of common components such as seals, valves, films, pumps etc. would assist in design and testing of a VIsoP.

Acoustic performance testing of a VIP would also be of value, both as a point of comparison to the

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VIsoP; and to the VIP industry, as it appears there has been little research is assessing the sound attenuation performance of such panels, Tenpierik (1).

One of the largest difficulties faced by any Vacuum Panel, is to maintain structural integrity under the considerable pressures that all surfaces of the panel are subjected to. At a total vacuum, the each surface of the panel is subject to 14.7 psi, or approximately 10 tonnes per square meter. 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). One can then appreciate why the nature and amount of structural reinforcement compromises the performance of the panel; the level of compromise in the design is likely to be determined by the desired performance of the panel.

2. VACUUM

The term vacuum refers to an area of space containing no matter. This is of course one end of a continuum, the opposite of which might be an area of infinite density, such as a black hole.

In everyday usage the term describes an area of air that has a lower pressure, and therefore lower density, than atmospheric conditions. It can be described using qualitative terms such as low, medium, high, ultra high and so on. Each of these terms corresponds with a range of pressures that can be expressed in a variety of units.

The SI unit of pressure is the pascal (symbol Pa), but vacuum is usually measured in torrs, named for Torricelli, an early Italian physicist (1608–1647). A torr is equal to the displacement of a millimetre of mercury (mmHg) in a manometer with 1 torr equaling 133.3223684 pascals above absolute zero pressure (2).

As a vacuum is measured in units of pressure, this document could express specific measured vacuum pressures in Pa. However, due to the full range of vacuum being considered, an easier way to express the pressure is to express it as a percentage, relative to atmospheric pressure on earth, at sea level. That is to say, the boundaries of 0% to 100% vacuum will be a linear progression between 101.3Pa and 0kPa.

Table 1 - vacuum definition

Qualitative Term	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 99.9%
High vacuum	100 mPa to 100 nPa	99.9% to 99.999%
Ultra high vacuum	100 nPa to 100 pPa	-
Extremely high vacuum	<100 pPa	-
Outer Space	100 µPa to <3fPa	-
Perfect vacuum	0 Pa	100%

With respect to the qualitative terms used to describe vacuum ranges, one can see that a ‘low’ vacuum encompasses almost all the range, yet there are several more terms allocated to describe the top 3% of (vacuum) pressures. This is suggestive of the importance, or the significance of difference in the very small changes in pressure at the lowest end of the spectrum. The pressures explored, and indeed the instrumentation used in this research only describes pressures of a ‘low’ vacuum.

3. PERFORMANCE ESTIMATIONS

In order to estimate the levels of vacuum necessary to achieve degrees of attenuation, a series of calculations were made. It was hoped this would provide a starting point to understand the levels of mechanical load the test panels would need to endure in order provide a target level of attenuation.

A pressure wave needs a medium through which to be transmitted. A (partial) vacuum by definition needs to be less dense than atmospheric pressure; and therefore has 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 its velocity. Neither are physically possible, and so instead, some of the energy simply reflects back from the boundary surface.

The reflection coefficient is defined as the ratio of the reflected and incident amplitudes, and this is given by the difference of the acoustic impedances divided by their sum, shown below as equation (1)

$$R = \frac{reflected\ \ pressure\, amplitidue}{incident\ \ pressure\, amplitude} = \frac{r_2 - r_1}{r_2 + r_1} \qquad \textbf{(1)}$$

In the case of this experiment, the value of r_1, is the acoustic impedance, equation (2), and is determined by the density of atmospheric air (*p*), and the speed of sound (*V*).

$$r_1\, or\; z_1 = \rho_1 V_1 \qquad \textbf{(2)}$$

One might think that the value for the speed of sound is important, as indeed it varies with temperature. However, the speed of sound (*c*)is related to pressure divided by density, given in equation (3).

$$c = \sqrt{\gamma \cdot \frac{p}{\rho}} \qquad \textbf{(3)}$$

Changing the temperature, changes the density of the medium and not the pressure, so the speed of sound varies. However applying a vacuum to a gas, reduces both the pressure and density proportionally, and so the speed of sound does not change. In which case, the selected value for *c* has no bearing on the reflection coefficient in this case. All that changes as the vacuum increases, is the acoustic impedance reduces, and the reflection coefficient between the surfaces increases.

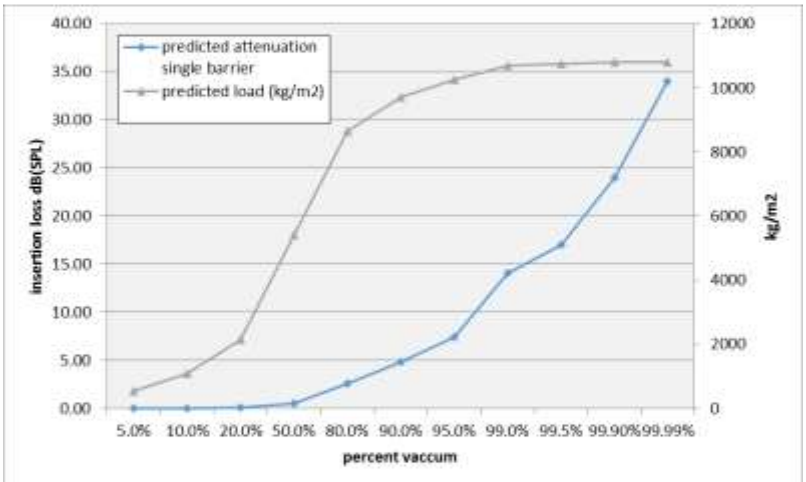

Figure 1 - predicted insertion loss and mechanical load

It can be seen from this series of predictions that; the pressure must be reduced by approximately 80% before an audible difference is expected to be perceived (2.5 dB); by which time the panel is already subjected to nearly 9000kg/m^2

If this were the only finding, one might think the application of a vacuum panel to be of little practical use. However, as the vacuum increases above 80%, the projected attenuation continues to rise exponentially, while the rate of pressure increases 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 greater attenuation.

4. DESIGN

4.1 Control Enclosure design

As it was not practical to test a vacuum panel according to any standard, a new test procedure had to be devised. In an attempt to emulate a transmission suit, the concept was to measure the insertion loss of an enclosure. The idea was to construct an enclosure according to conventional principals of absorption, mass insulation and vibration isolation.

If the sound source was fully enclosed within a six sided cube, all of the same materials and construction, then one could quantify the insertion loss of any and all of the six panels by performing measurements with one panel removed, see figure 2, compared to the panel fitted, see figure 3.

Figure 2 - open control enclosure

Figure 3- sealed control enclosure

The design target was to try and reach the highest level of attenuation with respect to a workable mass. (The final control enclosure, fitted with mass dampening cladding weighed approximately 50kg.) At this stage of the project, as construction of VIsoP prototypes had not yet commenced, there were no expectations or even estimations as to the level of attenuation a vacuum panel might produce. Instead, performance numbers from commercial sound isolation booth products were pursued as a reasonable design target, see figure 4

4.2 Test and Calibration Procedure

A powered loudspeaker was installed inside the innermost box. Power and line cables had been prewired to the outside of the rear of the enclosure with the necessary holes sealed. An external signal generator delivered a pink noise spectrum to the loudspeaker. A calibrated class1 SLM was mounted directly in front of the loudspeaker at a fixed distance, and the output volume was altered until it produced 100dBZ at a distance of 1m (from the loudspeaker). Then the components of the sixth, open face of the enclosure were re-fitted and sealed, and the test was repeated. The difference in level represented the insertion loss of the panel. Below 60Hz, and above 1250Hz, there was not enough difference between the attenuated levels and background noise to accurately quantify the attenuation; in some instances performance may have been undervalued. While not ideal, this does not invalidate relative comparisons between panels.

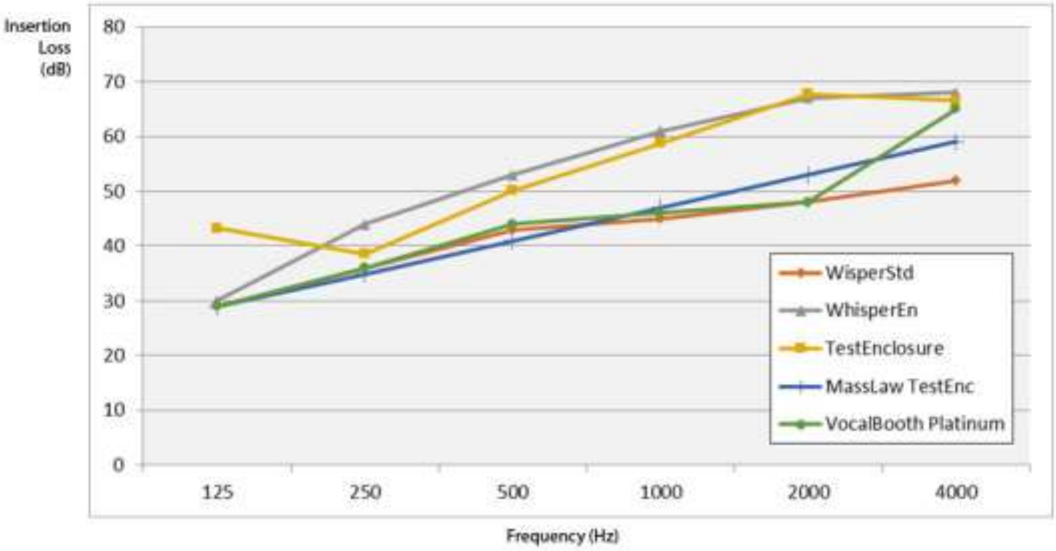

Figure 4 - comparison to commercial isolation booths

To demonstrate how well the final control enclosure met the design target, Figure 4, shows the insertion loss as compared to the performance of leading commercial grade sound isolation booths. Published performance measurements for most noise control products tends to be given in octave bands from 100Hz to 4000kHz. Additionally, the theoretical performance of the mass used to create the enclosure is plotted. (See MassLaw TestEnc series Figure 4) The increase in performance of the control enclosure, over its constituent mass can be attributed to the gains from the design elements of decoupling and double-leaf construction.

In can be seen that the control enclosure performed as good as one of the most expensive commercial offerings. However based on the enclosure dimensions, there was a strong axial mode established when the test enclosure was sealed that reduced the attenuation potential at 330Hz, evidenced by the performance dip seen in the 250Hz octave. After several days of modifications and retesting, it was concluded that the performance of the control enclosure was as good as practically possible for the chosen surface density.

4.3 Test Panel Design

In order to pursue the measurable relationship between levels of vacuum and levels of attenuation, one has to consider the mechanics involved. It is immediately apparent that when a cavity is evacuated, the bounding surfaces tend towards collapse due to the increasing pressure differential, between the inside and outside of the panel.

This can be countered in a number of ways, but for the purposes of this research, the panel design has been divided into two types.

- Mechanical braced/bridged/coupled

This involves flat faced panels, of lightweight and flexible material. In order to counteract the inward pressure, structural elements of reinforcement will be contained between the panels within the cavity.

- Inherently/Internally resistant/non coupled

This design requires a much stiffer, stronger material; likely to be heavier and formed in a manner that it resists inward pressure.

Contemplating the level of potential pressure that could be applied to the test panel of approximately 0.54m^2, at 100% vacuum, a total of 3150kgs of pressure would be applied evenly across both sides of the panel surfaces. This would seem to be an extraordinary amount of mechanical load, and so one might opt for an extremely strong and stiff material to construct the panel from; like sheet steel – however there are limits of practicality that must be considered. In particular:

- For the purposes of this independent research, an individual needs to be able to safely

handle the panel.
- The costs of the material and construction need to be affordable.
- The inherent mass of the panel needs to be sufficiently lower than the surrounding enclosure such that attenuation attributed to the vacuum can be measured.

4.3.1 Strut Braced Panel

Foreseeing that there would be difficulties encountered trying to engineer a non-coupled panel, work progressed in parallel with a flat, coupled panel. It was expected that there would be less mechanical challenges to overcome on the way to enduring high levels of vacuum, however this would be at the expense of sound transmission caused by the coupling. Considering that, prototypes progressed, each with increasing amounts of physical coupling, as the physical forces of high vacuum became apparent. Several prototypes did not progress into the acoustic testing lab, as their failings were obvious within the construction workshop.

Eventually, a lattice made from interlocking struts was used, as the number of struts could be varied, altering the size of the matrix. A 4x4 arrangement was chosen as it offered the minimum amount of structural support needed, with the least amount of surface area interconnecting the panels, see figure 5. Another sandwich layer of soft materials (foam, cardboard) was placed between the outer panes and the inner lattice, in an attempt to reduce the mechanical coupling. Custom rubber seals were used to space and seal the two panes, forming a rubber frame that would reduce flanking transmission through the frame. Due to the modular nature of this design; and given that it had to preserve the ability to be disassembled, modified and the re-sealed, a combination of semi permanent sealant, and duct tape was used. Consequently, vacuum pressures of approximately 80% were the upper limit of what was attainable.

Figure 5 - strut braced panels

Figure 6 - point braced panels

4.3.2 Point Braced Panel

Testing from the Strut Braced panel lead to the development of a point based bridged panel; whereby the panel cavity contained evenly positioned rubber balls that contacted the two panes at their ‘polar’ points. The acoustic principles behind the intention of the design were:

- that the surface area contacting each pane was reduced
- there was inherent damping in the rubber balls,
- given the balls were air filled, the rigidity of the coupling was expected to be reduced as compared to the solid struts.

Of course the mechanical pressure of the vacuum needed to be resisted, and there it was expected that the balls would expand according the very same pressure differential that would cause the panels to push in on them on the upper and lower points. In order to protect the circumference of the balls within the panel from deforming and potentially exploding; a rigid band was fixed, enveloping each ball to prevent expansion in that plane, see figure 6.

4.3.3 Convex Panel Construction

The first prototype came from the idea that a convex surface is structurally more resistant to inward pressure, than a flat surface. It seemed the simplest shape that would achieve this might be a shallow, four-sided pyramid. It was intended that a base mould be made, and then using a material known as HIPS (High Impact PolyStyrene) be heat treated and vacuum formed over the mould to create the shape.

After the initial mould was made from a dense blue-foam, it was decided that the flat faces of the four sided pyramid present points of obvious structural weakness, and so a parabolic shape was pursued instead. Forming a 3-dimensional parabolic shape, of two differing arc lengths; edge to edge and corner to corner, was extremely time consuming. Due to the chosen tooling process, the corners had to be separated and refitted prior to moulding. These small imperfections in this base mould proved to be extremely problematic in all but the last prototype. Had this stage employed a CNC tooling machine, forming a geometrically perfect CAD object, many complications and mechanical failures would have been avoided.

The fifth and final prototype constructed for this research was to reconstruct the fourth prototype, with the addition of a metal reinforcement frame, embedded within a fibreglass construction, see figure 7. It was also apparent that the sealant around the panel edges needed to be more robust, able to resist delamination when under strain.

Figure 7 - welded steel reinforced frame

A steel frame was welded together, containing a half-inch square bar perimeter, and quarter-inch parabolic struts to form the convex shape. The idea being that in order for the panel to deform and transition to inversion, the struts would have to effectively have to lengthen, pushing against the square boundary frame.

The rubber edge seals were further prepared with a mild sanding to the edges in contact with the fibreglass so that they would create stronger bonds with the sealant. They were then permanently fixed in place using liquid epoxy resin, to increase the design’s ability to resist delamination when subjected to flexural stain.

5. PANEL PERFORMANCE

5.1 Measurement Procedure

As there is no relevant standard to govern this type of testing, the following considerations determined the principles of the method:

- One of the first limitations, was that due to the necessary use of a vacuum pump, it was not possible to conduct the tests within the anechoic chamber. If the tests were concerned with the absolute sound reduction index of the panels, this would have presented a significant problem. Given that intention of the measurement was to quantify the insertion loss; having an anechoic environment was not crucial.
- Before each test, a baseline or ‘before’ sample was taken, including environment background noise and the spectrum of pink noise signal as produced by the test system, using the same measurement procedure at the same fixed distance.
- A pink noise generator fed a signal to a powered loudspeaker within the enclosure.
- The measurement microphone was placed 10 cm in front of the open panel.
- A calibrated sound level meter (SLM) was use to give realtime readings allowing the amplitude of the sound source to be adjusted to 100 dBZ.
- Using an acoustics measurement software package, WinMLS, and an external microphone; this combined measurement system was then calibrated in the same position to read 100 dBZ.

The use of a graphical 1/3 octave band analyser became the tool of choice to conduct these measurements as the instant visual representation of the measurements became apparent and more crucial in later tests. While this measurement system falls outside of industry certification, separate testing against a certified class 1 SLM showed it offered sufficient accuracy for these tests.

As the testing occurred across several months, and across different locations within the test facility, each round of measurements required the calibration and baseline process to be performed as described. After the calibration and baseline measurements were established. The next configuration to be measured was the specific test panel at atmospheric pressure; or 0% vacuum. This would serve as a second baseline to which measurements taken at increasing levels of vacuum would be compared.

The test panel was mounted into the open cube and held tightly in place with ratchet straps. The mitred rubber edges enabled a good seal, although at times, tape and putty were used to fill any perceivable gaps.

5.2 Measurement Results

5.2.1 Flat Panels

One of the outcomes that was hoped for, before any measurements were conducted, was that there might be an obvious perceivable attenuation to the sound as the vacuum level increased inside the test panel. This affect was most obvious in the flat panel designs, however the audible effect represented a worsening in performance as the vacuum increased. There was a distinct and immediately noticeably **increase in loudness** as the panel was subjected to vacuum. As the atmospheric pressure pushed inward on the panels, what was a reasonably good insulator, was degraded as the outer panels coupled with the internal grid structure.

As this degradation was mostly centred around 2kHz, it was understandable that the effect 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 Baetens et al (3); where the testing of commercial VIPs (filled with core material) showed a poor result in the same frequency ranges when a vacuum was applied.

At frequencies below 100Hz though, the panel performance increased steadily as the vacuum increased, such that particularly for its mass, this construction performed extremely well. It was unfortunate that higher levels of vacuum could not be measured. Due to the nature of the silicon sealant, only 80% vacuum was achievable. At such pressure, the sealant contracted, and pinholes around the edges prevented further increase in the vacuum. Variations to the cross-sectional area and configuration of the bracing, yielded no appreciable difference in results, often causing unsuitable panel deformation.

5.2.2 Ball braced panel

While this panel was expected to perform better than the strut braced panel, a similar effect was observed. As the vacuum increased the insertion loss of the panel degraded.

As with many experimental ideas, the reason behind their failure or success becomes immediately obvious once tested. The equal and opposite forces acting on the rubber balls, changed their properties away from a flexible material with damping, to a stiff and inflexible material. They behaved as though made of steel, as equal and opposite forces held them totally rigid in all directions. This explained why under vacuum as they ‘stiffened’, the acoustic coupling between the panel’s panes increased.

5.2.3 Convex Panel

The convex panel sustained the highest levels of vacuum and was expected to offer the ‘cleanest’ set of results in that the design was intended to minimise the physical coupling.

The simplest quantity that would indicate whether the panel had performed in the way that was anticipated, was to observe the total sound power levels decrease from the 0% vacuum reference, as the vacuum increased towards 100%. While this trend appeared slight, nevertheless, it could be observed immediately as a trend.

5.2.4 Measured relationship compared to Theory / Predictions.

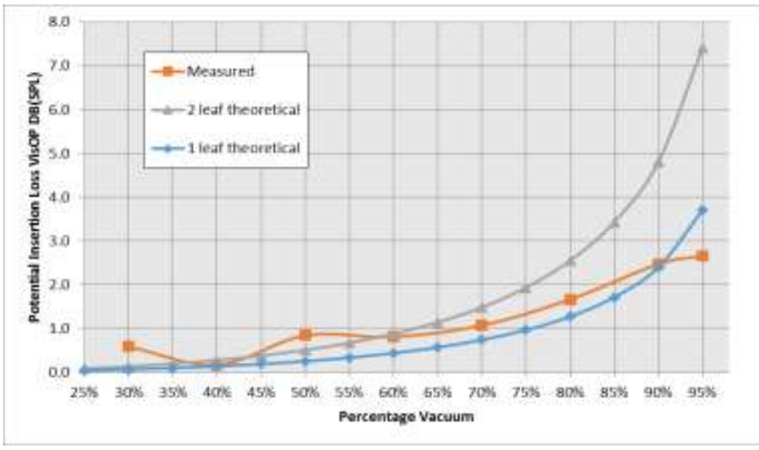

Figure 8 - final relationship, measured VS predictions

Figure 8 demonstrates perhaps the single most concise answer to the posed research question. The relationship (shown as Measured from the convex panel) between level of attenuation and the degree of vacuum, offered by a vacuum insulation panel, appears to be exponential. Furthermore, despite some anomalous data inherent in the methodology, the relationship is positioned approximately between the calculated theoretical level of attenuation of a 1 leaf and a 2 leaf panel.

6. CHALLENGES AND INACCURACIES

There were a number of challenges that needed to be considered in the testing methodology:

6.1 Insertion Loss or SRI?

The custom testing methodology of this research was not aiming to emulate a transmission suite, where leafed constructions can be quantified for their sound reduction potential in absolute terms. Instead, a quantification of the insertion loss was performed; which is certainly valid for stating in absolute terms to what degree the insertion of the test panel attenuated the specific source in that specific enclosure. Such measurements are only valid for comparison between panel types (and vacuum levels within a given panel)

6.2 The testing environment; varied background noise.

Due to health and safety regulations applicable in the acoustics lab, the vacuum pump could not be operated inside the lab. This meant that the panel measurements were conducted in the main acoustic lab area, with the vacuum pump outside. It was therefore impossible to ensure a constant level of background noise, and meant that due to the reverberant nature of the environment, any frequencies where a test panel out-performed the control enclosure, the accuracy of such a measurement was further reduced. However, the lab environment was consistently under 30 dBA when the experiments were undertaken.

6.3 Unstable pressures

One of the main inaccuracies during the measurement was the rate of change in pressures once the vacuum pump was disengaged. Depending on the panel design and the quality of the seals, changes in pressure of up to 5% (absolute) would occur across the ten second measurement period. This was somewhat averaged by disengaging the pump and starting the measurement partly above the desired pressure.

To reduce this inaccuracy, the apparatus would need a flow adjustment control, and the pump would need to be running constantly (in an acoustically isolated environment) during the measurement.

6.4 Unattainable pressures

As was outlined above, due to the difficulties with seals on the panels, there was a limit to the (low) pressures that could be reached. Perhaps the most difficult challenge though, was the engineering of a non coupled (convex) panel that could withstand the pressures.

The final day of testing was difficult indeed, as was the case with each prototype, it was completely unknown as to how much pressure it could structurally endure. It was only practical to take sets of measurements in 10% increments, and then up to 95%. After which point, the next attempted marker was 98%, at around which point, the panel imploded in spectacular fashion.

6.5 Unwanted sound artifacts in the measurements

Accurate measurement of the high frequency octave bands was compromised by two elements; the air leakage from the panel during the measurements, and after the vacuum pump was disengaged.

In the fibreglass/resin design, as the panel reached new levels of vacuum, there were occasional cracks and pops as the strain increased on the material. In most cases such an occurrence was immediately obvious in the measurement, and the data set was discarded. As with the other panels, for each measured pressure, an average of 3 measurements was taken where possible.

7. Conclusions and Potential

It can at least be concluded after a great deal of prototype development that it is extremely difficult to construct a non-coupled 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.

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 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. While the potential gains beyond this level of vacuum would indeed be desirable, the challenges of constructing such a design remain significant.

With respect to the theoretical performance detailed in this research, the measured performance of the custom VIsoP was positioned between that of a one and two leaf infinite reflection barrier.

The measured performance of the custom VIsoP can be simplified to the empirical formula (4): But a full range, extrapolated relationship could be estimated by formula (5):

$$\boldsymbol{Vac_{il} = -10\log(1-(0.7\times Vac_{\%})^2)} \tag{4}$$

$$\boldsymbol{Vac_{il} = 10\log\left(e^{(0.25\times Vac_{\%})}\right)+0.25} \tag{5}$$

Where $\boldsymbol{Vac_{il}}$ is the insertion loss to be gained from the vacuum, $\boldsymbol{Vac_{\%}}$ is the percentage expressed as a coefficient.

Having provided a fundamental understanding of the relationship between vacuum pressure and attenuation potential, it is clear that to utilise the sound isolation potential of a vacuum, any such VIsoP needs to be engineered to withstand full atmospheric pressure if it is to be of any significant value. This becomes increasingly challenging as VIsoP surface area increases.

While the prototypes in this research where unable to reach desirably high levels of vacuum, and the weight of the panels was comparable to the ‘massive’ conventional noise control surface densities; it remains a distinct possibility, that such panels could be engineered given sufficient resources.

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