# Mass-Air-Mass resonance cavity suppression using Helmholtz Resonators

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**ABSTRACT**

Densification of housing is leading to the construction of more medium-rise, multi-tenancy buildings. In order to minimise material and building cost but achieve the required structural performance, it is attractive to use lightweight construction methods based on materials such as plasterboard and light timber framing (LTF). LTF construction is traditional in NZ but where it is used for inter-tenancy walls in medium and high density housing we are finding that transmission of noise between dwellings is becoming a major problem.

In this exploratory research project we report on initial encouraging results of using Helmholtz Resonators (HR) to reduce the loss of insulation in lightweight cavity walls in the region of the mass-air-mass (MAM) resonance. This resonance results from the two wall leaves enclosing an air spring on which they can resonate. Often the wall insulation in this critical low frequency band is significantly poorer than if it were only a single leaf wall. Such walls, although meeting code requirements, can be subjectively undesirable when insulating against sounds from low frequency loudspeakers which are a ubiquitous feature of modern living. So far the work has comprised using 3D printed HRs tuned to the MAM resonance and coupled into an experimental cavity to demonstrate proof of principle.

## 1. INTRODUCTION

### 1.1. Problem

In a world of population densification, increased traffic flows and high power home entertainment systems, noise pollution is becoming a large problem. There is increasing concern in New Zealand and overseas about inadequate sound insulation in buildings and the implications for occupants' health and well-being both in the public and private sector. The problem is particularly evident in medium-high density housing, which are projected to become 30 percent of Auckland's housing by 2050 (Lyne, 2004).

Irritating acoustic intrusion frequently occurs at low frequencies, below 500kHz (eg; bass beat from music systems). Sound within this frequency range is often found in the work-place, and can cause loss of concentration and thus reduced productivity. Achieving effective isolation in this range is both challenging and expensive with conventional solutions, which require significantly increasing the density, mass or thickness of the partition through which the sound is transmitted. This introduces additional weight and costs as well as a reduction in usable floor space.

Resent work by Hall (Hall, 2013), highlighted the difficulties in providing adequate sound insulation at low frequencies. By using localised resonance metamaterials he was able to improve the sound transmission loss at low frequencies, but it was found that this method was difficult to manufacture in a cost effective manner.

### 1.2. Sound transmission loss

Sound insulation by a solid partition is the result of reflection of sound waves due to an impedance mismatch. Impedance in an acoustic sense is defined as the effort or force need to excite, or induce a velocity in a medium (Rayleigh,1945). The closer the impedance values are to each other, the more efficient the energy transfer will be. The key to improving sound insulation though a partition is to maximise the difference between these values.

Sound insulation properties of a solid partition increase as the frequency of the acoustic sound wave increases. This is because at lower frequencies there is an increase in the wave's vibration inertia and it takes more kinetic energy to excite a heavy mass.

The quantity most commonly used for expressing the performance of a partition's sound insulation is the sound transmission loss (TL) or sound reduction index (R). The sound reduction index (R) is the ISO unit standard measuring sound transmission loss through a partition (ISO 10140-4). The mass law is a commonly used approximation for predicting the ability of a single panel to reflect sound at frequencies below coincidence.

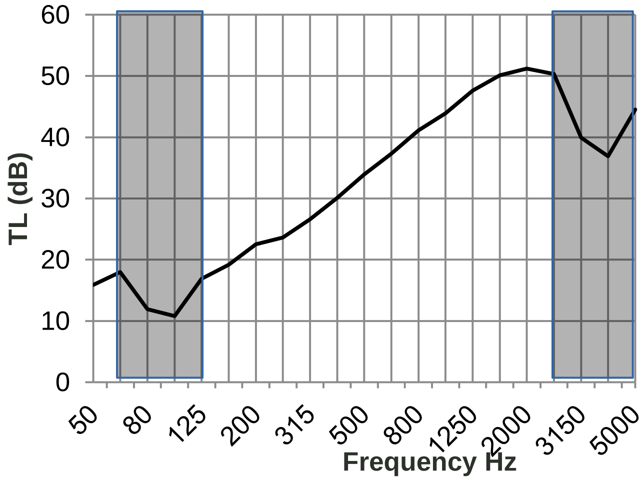



Figure 1. Sound transmission loss of a double leaf partition measured at the Acoustics Testing Service UoA

It assumes that the partition element is infinite, that the panel has negligible bending and shear stiffness, that the partition element has no damping forces and that the incidence waves are plane waves at any angle of incidence

Residential housing construction in New Zealand often comprises of plasterboard panels separated by a low density wooden or steel framing. Within the cavity, a layer of sound absorption material is added to reduce the cavity resonances between the panels and improve the TL. This solution improves the TL of the partition over a solid partition of the same mass whilst improving structural rigidity and maintaining a low-weight low cost solution.

The double leaf panel fabrication technique does not follow the mass law prediction. Figure 1 shows a plot of the TL of a double leaf partition with a 90mm air gap. Two reductions in TL may be seen at 100 and 3500Hz. These are the mass air mass (MAM) resonance and coincidence regions.

### 1.3. Mass air mass resonance

The MAM resonance dip is a common problem found in most double leaf panel constructions when the air between the panels acts as a spring. The dip typically occurs between 60 and 100Hz depending on the depth of the cavity. At a certain frequency the air spring and panel arrangement reaches a resonance where the panels are oscillating out of phase with each other. This frequency may be calculated using equation 1 (Hall, 2013):

$$f_{MAM} = \left(\frac{1}{2\pi}\right)\sqrt{\frac{1.8\rho c^2(m_1 + m_2)}{d m_1 m_2}} \quad (1)$$

where $f_{MAM}$ is the frequency of mass air mass resonance, is the $\rho$ density of air, $c$ is the speed of sound, $m_1$ and $m_2$ are the mass of the panels respectively and $d$ is the cavity depth.

This problem is made worse when considering:

- Mass law behaviour of panels produces a poor insulation performance in this region
- The trend in popular music towards bass driven music,

A typical double-leaf gypsum system as shown in figure 2, has a 75mm-120mm air gap which may be filled with a fibrous sound absorption material which improves the TL above the mass air mass resonance region. This is due to its absorption properties and therefore suppression of the cavity resonances. However MAM there is no improvement in TL when absorption material added.

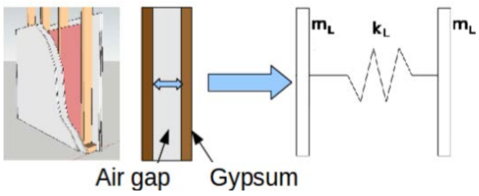



Figure 2. Mass air mass resonance system

### 1.4. Helmholtz resonators

The Helmholtz resonator (HR) consists of an open cylindrical neck volume connected to a larger chamber. When pressure waves move across the mouth of the neck, the volume of air present within the neck oscillates. The HR can be represented as a single degree of freedom (SDOF) spring-mass system, where the plug of air within the neck is thought of as the mass and the air inside the chamber acts as a spring.
The resonator's frequency of resonance may be calculated by equation 2 (Dodd, 1971):

$$f_o = \frac{C_o}{2\pi}\sqrt{\frac{A_{neck}}{V_{cavity} \cdot L_{effective}}} \quad (2)$$

Where $f_o$ is the resonance frequency, $C_o$ is the speed of sound in air, $A_{neck}$ is the neck area, $V_{cavity}$ is the cavity volume and $L_{effective}$ is the effective neck length.
The length of the tube determines the mass of air that is within the pipe. In order to determine the resonance frequency, an accurate length is required. However, for a pipe that is only open at one end and has a finite diameter, the acoustical length is slightly longer than its physical length. As a result an additional length known as the end correction factor is added using equation 3 (Dodd, 1971):

$$L_{effective} = L_n + 1.7d \quad (3)$$

Where $L_n$ is the actual length of the neck and $d$ is the neck diameter

HR's have been commonly used to attenuate sound in applications such as baffles, ducts and car exhausts. An impedance mismatch created by the resonators cause a reflection of the sound waves targeting a narrow band of frequencies.
Publications by J. Mason and F. Fahy in 1988, and Pieyrzko and Mao in 2005, discus the application of HR's inside a partition for absorption of sound within the cavity around the MAM resonance, however to date no commercially available system has been introduced to the construction market. This is understood to be due to the large chamber sizes needed to produce a frequency of resonance in the region of 100Hz. Publications (Jiménez, 2017 and Xiaobing, 2014) using rainbow and spiral shapes to reduce the size of a HR have inspired the team to develop a system for suppression of the MAM resonance resonance phenomenon using compact HR's over a wider band.

## 2. AIM

To conduct preliminary experimentation on the feasibility of improving the sound transmission loss of a double leaf partition at the mass air mass resonance using multiple HR's confined to the volume of a single partition stud. Inspired by the Rainbow Trapping Absorber [5] we aim to tune multiple resonators over a specified frequency range in order to reduce the sound transmitted across a wider bandwidth.

## 3. EXPERIMENTAL SETUP

### 3.1. Wall segment

A sample unit of a double leaf partition was fabricated to simulate a section of a typical intertenancy partition found in New Zealand. The unit consisted of two 430x430x6mm acrylic panels placed on either side of a 90 x45mm wooden stud frame.

Due to variability in fixing conditions the acrylic panels were simply placed on the studs creating free boundary edge conditions. Acrylic panels were used as its transparency allowed measurements to be taken on both panels using a laser vibrometer while under excitation. Figure 3 shows the sample partition, it can be seen that the separating studs surround the entire sample creating an air cavity between the two acrylic panels. Holes may be seen in the centre of each stud, which were drilled after initial control testing to insert the HR necks.

Figure 3. Partition segment sample

### 3.2. Excitation device

A 430x430mm sound box was used as the excitation method for this experiment.

A loud speaker is enclosed within the sound box as an excitation source to generate a frequency sweep of sound over the desired frequency range. The wall segment was placed on top of the sound box which has a rubber seal to prevent excess sound loss through the edges of the sample. The sound box was placed within the Acoustic Testing Service's listening room to avoid any room mode interference. A frequency sweep was generated through a loudspeaker.

Figure 4. Sound box

### 3.3. Measurement method

A *Polytec PDV-100* Portable Digital Vibrometer mounted on a tripod was used to measure the velocity of upper and lower acrylic panels of the partition sample. Effectively a higher velocity will mean higher amplitude which correlates to higher transmittance through the sample. The velocity was measured at a number of points on the panel face, and this was repeated for each sample.

It could be satisfactorily assumed because of the small cross section of the sound box compared with the wave-length of the frequencies of interest, the panels would be subjected to plane wave normal incidence so single point measurement would give an accurate indication of the overall performance of the sample. The advantage of using this method is that the effects of room modes on the system are negligible.
The laser vibrometer may be seen in figure 5. Each measurement was taken three times, at each frequency with good correlation between results.

Figure 5. Laser vibrometer mounted on tripod

## 4. SAMPLES

HR samples were fabricated using additive manufacturing. As shown in figure 6, the printer was a fused deposition modelling (FDM) printer model *"Prusa i3 mk2s"*.
Equation 2 was used to determine the dimensions that the HR would require based on a prescribed resonance frequency. Using these dimensions an STL file of the HR design generated using *Creo Parameterics 3.0*. Samples were then printed using Poly-lactic-acid (PLA) filament.

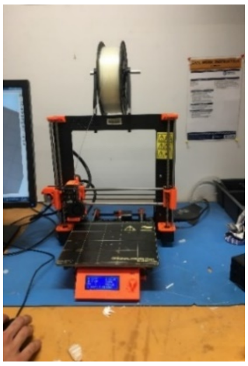


Figure 6. *Prusa i3 mk2s* FDM 3D printer

It was found that the roughness on the inside of the resonator neck produced by the 3D printer increased the amount of damping in the resonator to an unacceptable level. It was decided instead to print only the chamber of the HR and use smooth PVC piping for the neck inserted in to the flange of the chamber. This had the added benefit of being able to fine tune the HR frequency of resonance by twisting the tube to extend or detract the pipe. The HR samples may be seen in Figure 7 and 8.

Figure 7. Example of a printed HR

Figure 8. Example of printed HR's mounted in wall unit

## 5. RESULTS

### 5.1. Control partition segment

Measurement was first made on the partition unit with no resonators without absorption material in the cavity. These results were used to determine the mass air mass resonance of the system and therefore the targeted frequency region for designing the HR. By measuring the velocity over the surface of the upper panel, we were able match the expected mode shapes of the panel which were computed in *COMSOL multiphysics*, and thus determine with strong certainty the exact frequency of the MAM resonance of the system. Figure 9 shows the control samples velocity magnitude in dB for the centre point of the panel after excitation. The MAM resonance was determined to be 138Hz shown by a peak in velocity.

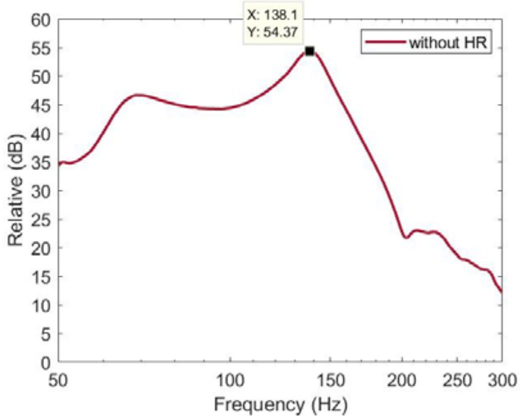



Figure 9. Velocity magnitude of control without HR's

### 5.2. Single resonance frequency

Four HR's were designed and fabricated with a frequency of resonance at 138Hz. The neck of each resonator was then placed through the wooden stud so the aperture was positioned near the centre of the cavity. Figure 10 shows the resulting velocity magnitude at the centre of the outmost panel with one resonator added. Figure 11 shows the resulting velocity magnitude at the centre of the outmost panel with 1-4 resonators added. It can be seen that there is a reduction in the velocity and hence sound transmitted through the partition at the frequency of resonance. There is a large difference between 1 and 2 resonators with a smaller improvement seen between 2 and 4 resonators.

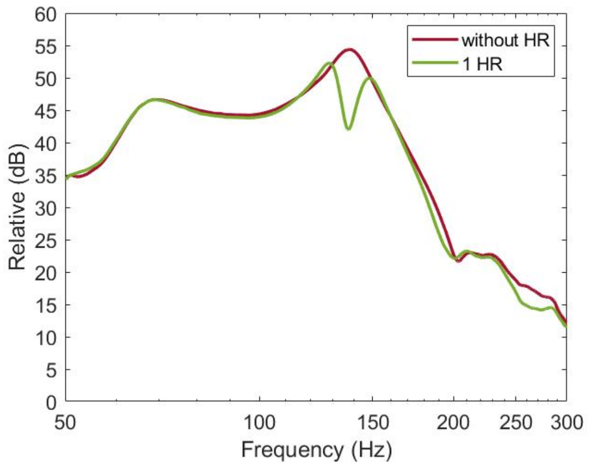



Figure 10. Velocity magnitude of sample with 1 HR

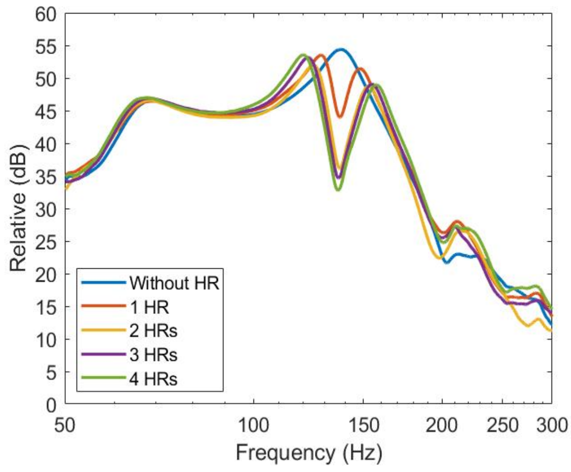



Figure 11. Velocity magnitude of sample with 1-4 HR's

### 5.3. Multiple resonance frequencies

Four HR's were then fabricated with different frequencies of resonance between 125- 156Hz. The difference in resonance frequencies was achieved by varying the chamber size and the neck length. The resonance frequencies were chosen in order to evenly spread the effect of the HR's over the entire MAM resonance bandwidth. Whilst not optimised, the spread of resonance frequencies was reasonably even. The neck of each resonator was then placed through the wooden stud so the aperture was positioned near the centre of the cavity.

Figure 12 shows the resulting velocity magnitude at the centre of the outmost panel with one and two resonators added with different frequencies of resonance. Figure 13 shows the resulting velocity magnitude at the centre of the outmost panel with 1-4 resonators added all with different frequencies of resonance. It can be seen that there are four dips, or reductions in the velocity representing each HR and the overall velocity reduction was seen over a wider band width.

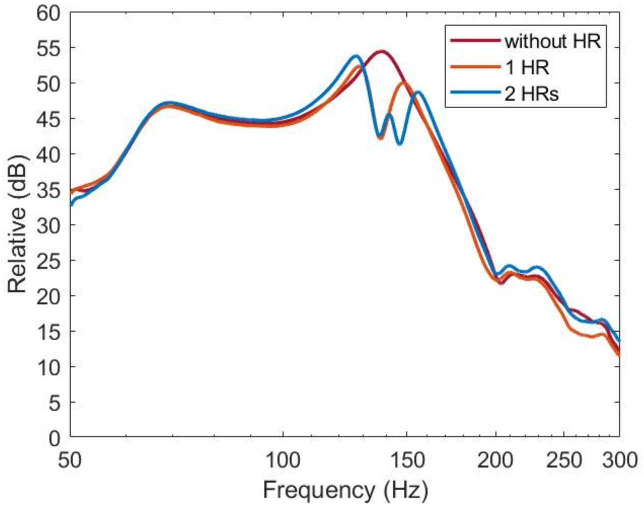



Figure 12. Velocity magnitude of sample with 1-2 differently tuned HR's

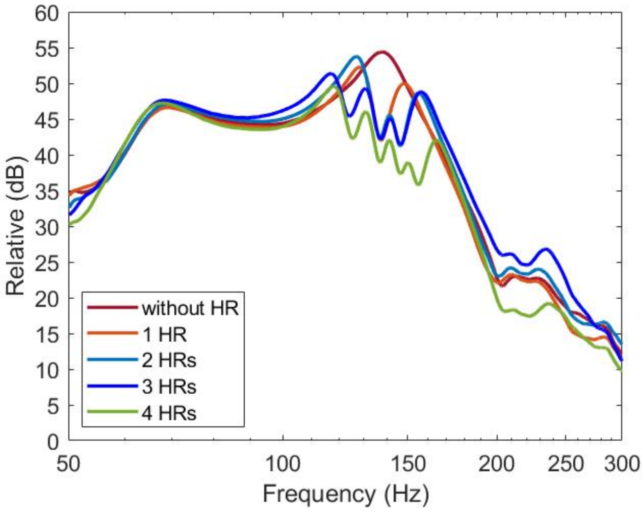



Figure 13. Velocity magnitude of sample with 1-4 differently tuned HR’s

Figure 14 shows the resulting velocity magnitude difference between 4 HR tuned and different frequencies of resonance with and without fibrous absorption material within the cavity. As expected there is very little difference.

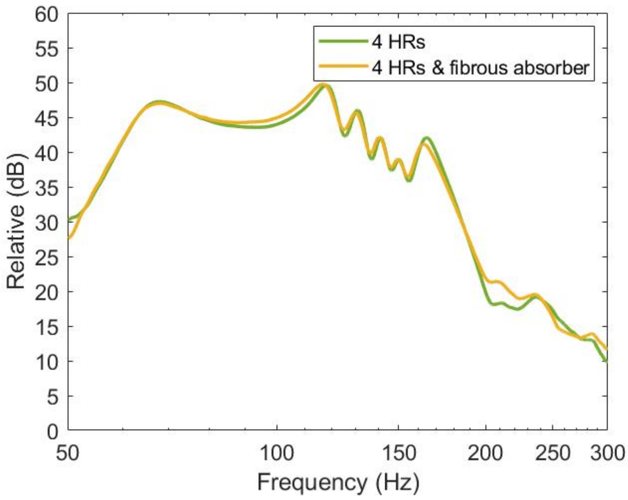



Figure 14. Velocity magnitude of sample with 4 differently tuned HR’s with and without absorption material

## 6. DISCUSSION

From the results present in this paper it is clear the using HR’s has reduced the velocity of the outer transmitting panel of the system at the MAM resonance. Both single and multiple resonance systems reduce the transmittance of the control sample by 21.5 and 15.5 dB respectively. There was a large decrease in velocity once the system was changed from one resonator to two resonators tuned to the same frequency. There was only a small difference in performance when two additional resonators were added to the system tuned to the same resonance frequency.

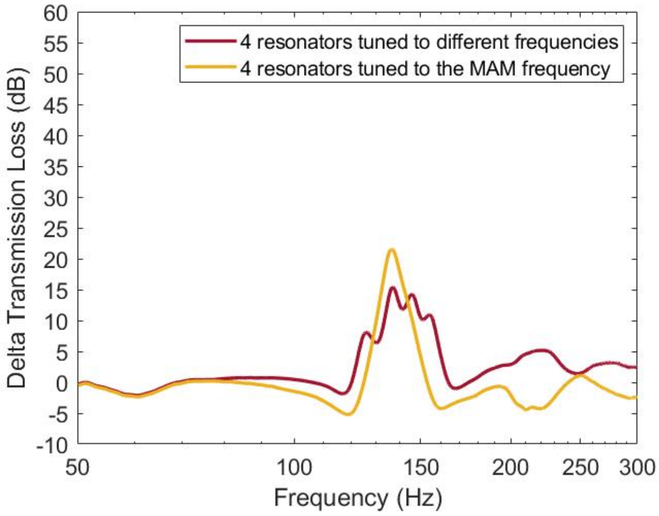



Figure 15. Improvement in TL from adding 1 vs 4 differently tuned HR's

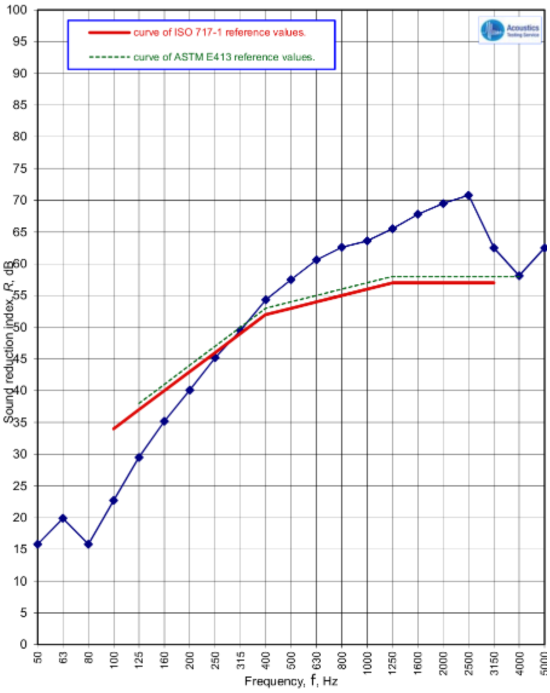
When comparing the single and multi-resonance systems where there are HRs tuned to different frequencies over a band of 30 Hz, there is clearly a widening of the effected frequency band of reduced vibration. This can be clearly seen in figure 15 which shows the improvement in transmission loss over the control sample. However the spread of the resonance frequencies means that a lower peak transmission loss is achieved.

It is believed that this is a necessary trade off, and it is more valuable to have an even spread of improved performance over a wider band, thus removing the MAM resonance than a sharp peak in improved transmission loss. It is important to note that when changing the aperture placement of the HR within the cavity unit there was no noticeable change, and application of fibrous absorption material did not change the MAM resonance region effect.

Area *S* of test specimen: 11.95 $m^2$
Mass per unit area: $kg/m^2$
Air temp in the test rooms: 23 °C
Air humidity in test rooms: 65 %
Source room volume: 208 $m^3$
Receiving room volume: 202 $m^3$

| Frequency *f* Hz | *R* One-third octave dB |
|---|---|
| 50 | 15.8 |
| 63 | 19.9 |
| 80 | 15.8 |
| 100 | **22.7** |
| 125 | **29.5** |
| 160 | **35.2** |
| 200 | **40.1** |
| 250 | **45.2** |
| 315 | **49.5** |
| 400 | **54.3** |
| 500 | **57.5** |
| 630 | **60.6** |
| 800 | **62.6** |
| 1000 | **63.6** |
| 1250 | **65.5** |
| 1600 | **67.8** |
| 2000 | **69.5** |
| 2500 | **70.8** |
| 3150 | **62.5** |
| 4000 | **58.1** |
| 5000 | 62.5 |

Notes: 1. #N/A = Value not available.
2. **Bold** values are used to calculate STC and $R_w$.
3. Words in ***Blue Italic*** in the description are manufacturers brand names.




Rating according to ISO 717-1 $R_w$ $(C;C_{tr})$ = 53 ( -4 ; -12 ) dB
$C_{50\text{-}3150}$ = -8 dB $C_{tr,\ 50\text{-}3150}$ = -19 dB
$C_{50\text{-}5000}$ = -7 dB $C_{tr,\ 50\text{-}5000}$ = -19 dB
$C_{100\text{-}5000}$ = -3 dB $C_{tr,100\text{-}5000}$ = -12 dB

Rating according to ASTM E413 -87
**Sound Transmission Class = 54 dB**

Figure 16. Measured TL of a standard double leaf partition

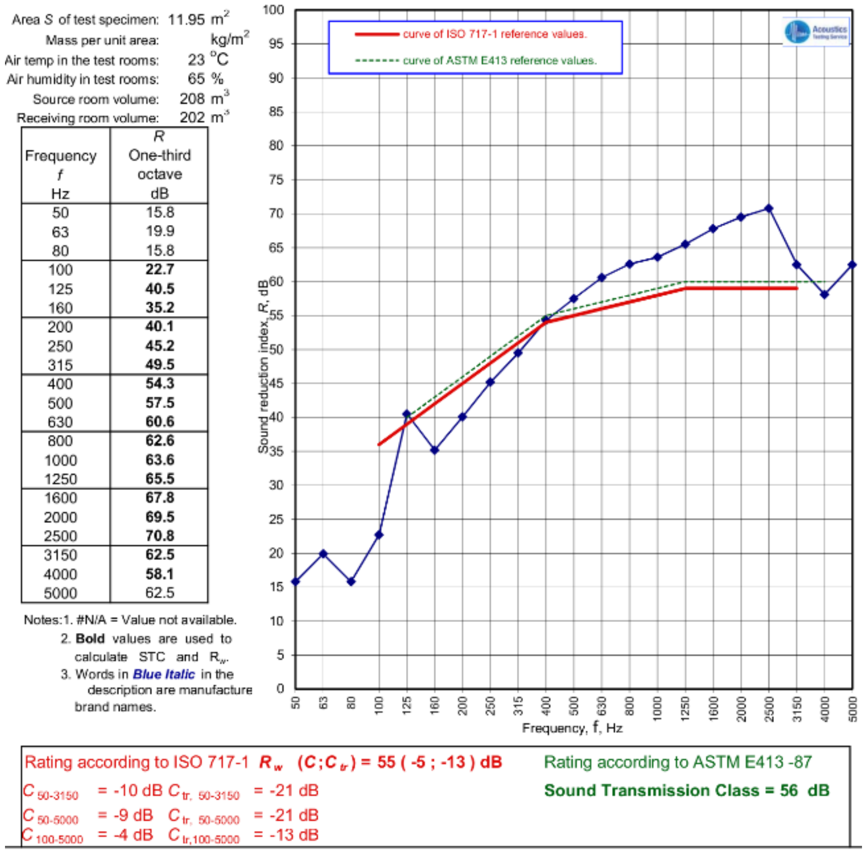



Figure 17. Theoretical TL improvement when applied to a standard double leaf partition.

A useful method for measuring the application value of these results is to investigate the difference the system presented in this paper would make on a traditional partitions sound transmission loss. Results were applied to a known standard double leaf partition values previously measured in the acoustics laboratory chambers at the University of Auckland.

Figure 16 shows the measured data, figure 17 shows the adjusted results. The test results have been modified at the 125 Hz 1/3 octave band by reducing the level by 11 dB. This 11 dB is not the reduction when the acrylic panel is excited by a tone at 136 Hz (which showed a level drop of approx. 22 dB i.e. a loudness reduction to less than a quarter of the original loudness) but by the reduction measured when the panel is excited by the sum of tones over the range 126-146Hz. Exciting the panel by a range of tones over one fifth of an octave was felt to be a fairer test of when the panel would be excited by broad band noise. The reduction by 11 dB is a reduction to less than half the loudness of the original.

The Sound transmission class (STC) (ISO10140-4) was seen to improve from 53 to 55. This means the HR system was theoretically able to convert a failing partition to a passing partition conforming to the New Zealand building code by adding negligible mass to the system. We feel this energy dissipation method through absorption within the cavity provides a solution to a problem that will only increase in relevance as society moves into the future.

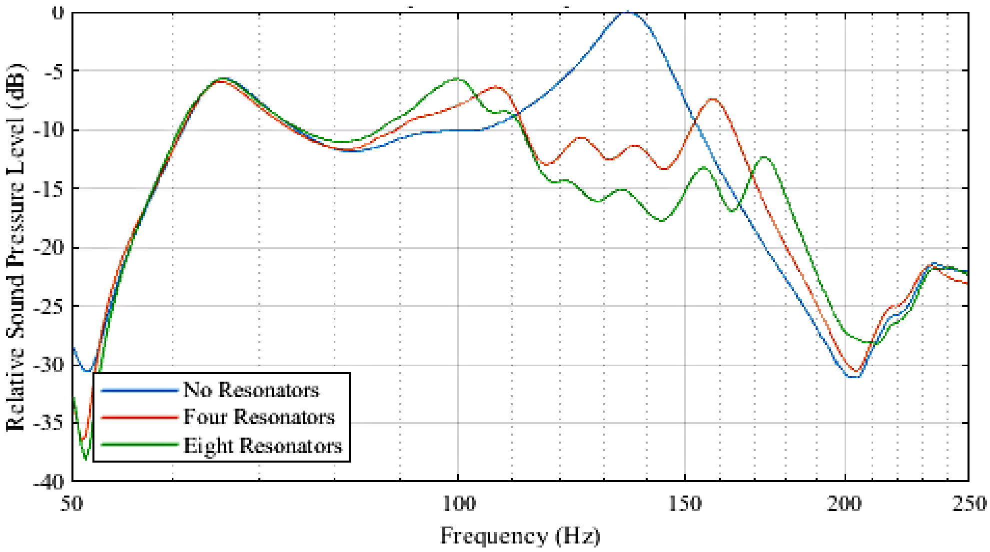


Figure 18. Measurement of several helholtz membrane resonators in a wall cavity unit

Since the completion of this paper, further work has been carried out using membrane HR to lower the frequency of resonance whilst maintaining a compact design. A flexible membrane is incorporate into one surface of the HR cavity. As shown in Figure 18, this method is effective and will enable easier incoperation of the resonator system into a partition.

## 7. CONCLUSION

It was shown that HR's are a light weight compact passive solution to the MAM resonance phenomenon that plagues the sound transmission loss performance of double leaf partitions around the world.

Significant gains were found in reducing the transmission of a test sample through the use of both single and multiple resonance HR's systems when the aperture of the HR was placed within the cavity whilst the chamber was kept outside of the partition unit.

The HR provide a means to beat the mass law. The team hopes to incorporate the required HR into stud or nog designs before large scale testing, and is looking for industry collaborations.

## ACKNOWLEDGMENTS

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