New Zealand Acoustics

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Features

Regulars

Cover Photo: Tom Roberts' Shearing the rams, 1890.

Public Domain

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Contributions to the Journal are encouraged, and may be sent directly to the Editor either by email, or by post c/o the Acoustical Society of New Zealand Incorporated, PO Box 1181, Auckland.

From the President and Editor

From the President

My word, here we are half way through 2011 already. The remainder of the year is shaping up to bring a great leap forward for our Society – well, two great leaps forward, to be more precise!

Firstly – as you see from the byline and contact details in this issue, the Society's name has been changed from the previous "New Zealand Acoustical Society" (NZAS) to the new "Acoustical Society of New Zealand" (ASNZ). This was done to avoid confusion with the New Zealand Audiological Society. The change has also resulted in a new logo being developed. So – fanfare and trumpets, please! – this is in fact a momentous occasion – and this is the inaugural edition of the ASNZ Journal.

Secondly – the new Membership regime is one step closer. The online membership ballot has now closed, and the results are in (thanks James Whitlock). A total of 58 responses were received, which is a great indication that Members are keen to have a say in their society.

The results of the ballot were: 54 voted YES and 4 voted NO. So, with a 93% majority – we are set to roll out the new membership format as of 1 July. To refresh your memory regarding the details, please visit the website at www.acoustics.ac.nz/membership.php. You will also find there the link to the membership form, which I encourage you to download and fill out to get the process happening.

This has been a long time in the making, and is the result of a significant amount of effort put in by a small, dedicated group of people. My personal thanks, and the thanks of the society

Members, go to those involved (you know who you are...)

Which reminds me. As I write this, I recall that I have been tasked with looking at the issue of Continuing

Publication Dates and Deadlines

New Zealand Acoustics is published quarterly in March, June, September, and December.

The Deadline for material for inclusion in the journal is 1st of each publication month, although long articles should ideally be received at least 2 weeks prior to this.

Advertising

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Society Membership

Membership in the Acoustical Society of New Zealand is open to anybody interested in acoustics. There are no entry requirements. Members receive benefits including;

- Direct notification of upcoming local events
- Regular mailing of Noise News International
- Reduced charges for local and national Society events
- Priority space allocation for trade stands at society events
- Discounted rates on selected acoustic products

To become a member of the society, visit www.acoustics.ac.nz or contact the Secretary.

Professional Development, as it might apply to our new Membership regime. Specifically in the first instance I will be collating a set of examples of CPD requirements and ongoing assessment from other professional societies, both Australian and international, to review in relation to the specific needs of our acoustical society.

As with the issue of Membership and our Rules of Conduct, the draft CPD provisions will be circulated for comment to all Members.

I would like also to acknowledge our Principal Editor, John Cater. John has volunteered to represent the Society as a non-voting observer at the annual International Institute of Acoustics and Vibration (IIAV) Board of Directors Meeting. However, before I express too much gratitude for this, I should point out that this year's IIAV conference (the eighteenth International Congress on Sound and Vibration (ICSV18)) is being held in Rio de Janeiro, Brazil. So I'm probably more envious that he has the opportunity to go to Rio than I am grateful for his time!

I'm sure John will bring back some interesting papers to share with us in future journal editions.

In the meantime, I'm looking forward to this journal issue to come out so that I can check the solution to Issue 1's cryptic acoustic crossword!

Best regards to all until the next issue.

Rachel Foster

Editor's Ramble

I am once again pleased to deliver a new issue of the journal, this time brought to you by the ASNZ.

This issue sees the return of the 'Upcoming Events' section, which I think is a useful part of this journal. I am concious that at the moment the list of events is in no way complete, and it should contain more information about activities in NZ; please contact me if you know of other events that you think are of interest to our members.

The issue also contains three technical papers on diverse subjects. The first has a distinctly Kiwi flavour and details some measurements of noise exposure for shearing workers. As you will know if you have ever been in a woolshed at shearing time, it is a noisy environment and one where hearing protection is definitely appropriate (but is not used). This is a work area that seems to have been overlooked in the past, perhaps because the workers are mainly itinerants.

As noted above, the Society has a new name and a new logo, these can be found for your viewing pleasure in an announcement on page 9.

The second research paper describes work about locally resonant materials. This is an exciting area of research being done in New Zealand and is part of the larger field of meta-materials, one of the 'hot topics' in applied physics at the moment. Two of the acoustic snippets that appear later in this issue also involve applications for meta-materials.

The final article is a review of international vibration standards (NZ currently has none), and contains recommendations for which standards might best be applied in the NZ context.

As promised this issue includes an all-new cryptic crossword and solutions to the clues presented in Issue 1. (Special Mention to Stuart Camp who was first to return the correct answers).

Finally, as usual, the issue ends with the restaurant CRAI ratings. This time the list is something of a tribute to the restaurants that will no longer appear, due to the effects of the Christchurch earthquakes.

The ASNZ Council is currently looking at ways to better deliver services to the society membership; this includes looking at what kind of information is printed in this journal. If you have any ideas, please get in contact (I am not getting much mail!).

As editor, I do have the opportunity for the 'last word', so I will use this to ask you to note that I have two technical papers at the ISCV18 Conference in Rio (one on jet noise, one on sound production in the vocal tract), so I won't be spending all my time at the beach!

Once again, thanks to all those that contributed. to this issue. Wishing you an enjoyable and educational read.

John Cater

Noise in the Shearing Industry

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A paper previously presented at ISSA 2010, 29-31 August 2010, Auckland

Abstract

The aim of this study was to assess the risk of noise induced hearing loss (NIHL) in shearing and investigate practicable control methods. Woolshed surveys included shed construction data, noise dosimetry and area noise sampling. The noise exposures from 40 personal measurements were all above the 85 dB(A) action level, lying in the range 86-90 dB(A). Shearers had the highest exposure, "near field" noise coming from the action of the cutting edge in the shearing comb, but also from downtubes and gears. Noise for sorters and pressers was contributed to by the stereo system (found in all shearing sheds). Lined sheds seemed to have slightly higher noise levels than unlined sheds. None of the shearing crews had hearing protection available. Redesign of the shearing equipment primarily the handpiece but also the downtubes and gears could potentially reduce the exposure by 2-3 dB(A) and possibly more. In the meantime shearing crews need to wear hearing protection and be subject to audiometric surveillance.

Introduction

Agriculture is a hazardous industry, with New Zealand studies identifying noise induced hearing loss (NIHL), low back pain, chemical related morbidity and mental health as being of concern. (Firth, Herbison et al. 2001) One study including arable, dairy, mixed and sheep farming shows that the latter had the highest noise exposures, a median level equivalent (Leq) of 86.8 dB(A), interquartile range 84.3-90.7). (Firth, Herbison et al. 2006) The main activities being carried out by those with exposures above the 85 dB(A) exposure standard were riding a motorcycle, driving heavy machinery and using hand held power tools, including shearing hand-pieces. "Grab" sampling has shown that noise levels in shearing sheds can be intense, with levels up to 97dB(A). (Occupational Safety and Health Service of the Department of Labour 1995). Most shearing is however carried out by contractors, so the burden of exposure is not primarily carried by farm owners or operators.

There are three distinct groups of workers in shearing sheds, the shearers themselves; wool sorters who collect the fleece from the shearers, sort and "skirt" it and the pressers who process the bales. The hazards of shearing have been described, and include manual handling and biological agents. The noise environment will be slightly different for the three occupational groups. The shearers are close to the shearing plant (the drive, including the "down tubes" and "elbows", generally the responsibility of the farm owner) and the hand pieces and combs which are their own responsibility (figure 1). The sorters work close to the shearers. The pressers operate various types of wool press, wooden and metal, variously actuated and some of which are noisier than others (Figure 2). The acoustic environment can be dominated by the "radio", usually a stereo system, and is modified by the construction of the shed. How these factors interact to influence the acoustic environment in shearing sheds has not however been fully described to date. The aims of this study were to carry out a noise survey in a sample of New Zealand Shearing Sheds, describe the noise exposure associated with shearing activities and help to define the acoustic environment.

Methodology

Study population and sampling

The New Zealand Shearing Industry Health and Safety Committee (the Committee) identified shearing contractors in the North and South Island of New Zealand to participate in the study. Each contractor was contacted by telephone and the purpose of the study explained. Each were then sent copies of employer and employee information sheets, and appointments were made for follow up on-site.

Questionnaire

An environmental assessment form was developed in association with the Committee, this included the number of people working in the shearing crew, type of shearing (normal full shear, crutching etc.) number of sheep sheared, hours of work, shed construction (materials used for frame, walls, roof and lining) and the type of woolshed equipment in use (shearing plant and wool press). The use of hearing protection was observed, as was personal stereo use.

Environmental survey

Noise dosimetry was carried out using Cirrhus Research doseBadge wireless personal noise dosimeters (Class 2 instruments) calibrated before and after measurement using the combined reader/calibrator unit. The Badges were mounted in the hearing zone of the participant, and the data downloaded to the reader and then to the doseBadge software for later analysis.

Grab samples were taken for specific activities using a Brüjel and Kjær type 2260 type 1 sound level meter to record A weighted sound pressure levels and octave band analyses.

These were video-taped for later analysis.

Analysis

The data was transferred manually to SPSS version 14. The analyses were initially descriptive, with, depending on the distribution of the noise levels within comparison groups, ttests or the equivalent non-parametric Mann Whitney U test used to test for "between group" differences.

Results

A total of 9 contractors were approached and all agreed to participate. Surveys were carried out in 16 sheds.

A total of 122 individuals were working in the sheds during the survey, the majority being shearers. The majority (80, 66%) identified their ethnic group as New Zealand Maori, with 42 (34%) New Zealand European. A total of 15,381 sheep had been shorn and as the survey was carried out in January (summer in New Zealand) the activities recorded were normal shearing. The median duration of the working day was 8 hours, with a minimum of 7 and maximum of 10.

Wood framed shed construction was most common, with only one steel framed shed represented. The vast majority of sheds (13, 81%) had corrugated iron walls, with 75% having a corrugated iron roof. Most sheds (11, 69%) were unlined and 5 (31%) had a wooden or plywood lining.

There was no hearing protection in evidence and only 2 individuals used personal stereos.

A total of 40 dosimetry measurements were made, 25 shearers, 9 wool pressers and 6 pressers. The distribution of the noise exposures had a reasonably normal distribution, with a mean of 89 dB(A), 95% confidence interval (CI) 88-90. Those for the individual occupations (shearers, wool sorters and pressers) were non-normal (as depicted in the box-plots, figure 3), and the medians and inter-quartile ranges are reported in table 1.

The shearers had the highest noise exposure, with pressers and sorters progressively less. The noise exposure common to all the occupational groups was the radio.

Table 1. Noise exposures of occupational groups.

Occupation	N	Mean Duration (h.mm)	Median Leq	Inter-quartile range	Min-max
Shearer	25	7.40	90	88.5-91.5-	84-93-
Sorter	9	6.72	87	85-89	85-91-
Presser	6	6.24	89	87-90	88-90-

A comparison of all the dosimetry readings taken in unlined sheds were compared with those in lined sheds. This showed median levels in unlined sheds of 88.5 dB(A) and lined sheds of 90 dB(A). An independent samples Mann-Whitney U test showed that this difference was significant ($p<0.05$).

The noise sources for shearers comes from the drive, transmitted through a down tube, through a set of gears in the "elbow" and through another drive tube to the hand piece, containing gears and the cutting comb. The noise for the shearers was near-field, the shearing hand piece being, at most, an arm's length away, with the elbow and drive tubes at varying distances according to posture and activity.

Impact noise was evident from movement of the sheep and the elbow striking the sides of the pen. Reductions in noise exposure of 1-2 dB(A) were noted on later video analysis of the area noise samples. There were two main designs of plant in use with roughly equal numbers (18:15) and 6 which were of miscellaneous design or were "missing data". There was no significant difference in noise levels between the two main design types.

Noise for sorters comes from the activities of the shearers, with whom they are in close proximity for about half of the time. They also use plastic scrapers to clean up the wool off the shearing board, producing impulse noise. Greater reductions in area noise monitoring were evident, with radio "intermissions" showing reduced levels in the order of 3 dB(A).

Sorters are exposed to impact noise from metal catches on metal wool presses and also help with penning the sheep, so they are near the shearers at times. The same comments apply to the background noise.

Discussion and Conclusions

The main source of noise for shearers was, unsurprisingly, the mechanical noise from the down-tubes, elbow and handpiece. The variation in the levels (2 dB or so) reflect differences in both hand piece and shearing plant design and maintenance, the former the responsibility of the shearer and the latter the farmer.

Further investigation by the Acoustics Research Group in the Department of Mechanical Engineering at the University of Canterbury (Submitted for peer review) has shown that there are significant differences in noise levels between models of shearing plant when new, with scope for design improvements in the down tubes, the gears and the handpiece to reduce rub, backlash and friction noise respectively.

The pressers had noise exposures somewhat similar in level to the shearers, but influenced by the fasteners and catches on the press.

The noise exposure of all the groups was affected by the music in sheds. The politics of the type of music listened to and the intensity at which it is played has yet to be investigated, but seems to be a function of the hierarchical social system within each shearing crew.

No one was observed to use hearing protection, but few individuals used personal stereos either.

Paradoxically, lined sheds seem to be noisier, but the lining will not be acoustically absorptive and those with no lining may transmit and thus diffuse more noise.

The survey does have limitations. It was a convenience sample, however there is no reason to suppose that the noise exposures would be any different as a result. Although the contractors were identified by the Committee, all shearers

use a standard range of hand pieces. The sheds were not however chosen to reflect a range of acoustic environments, the majority of them being wood and corrugated iron.

We also had small numbers leading to less precision in the estimates of the exposure of sorters and pressers, but to some extent this was due to the ratio of numbers of shearers sorters and pressers, generally in the order of 3 or 4:2:1 respectively. Because of the dynamic nature of the job, the near field exposures of the shearers and their proximity to one another it was also difficult to characterise the noise sources using area monitoring.

It is clear that there are opportunities for noise reduction at source. There are undoubtedly design modifications which can be made to hand pieces, however these will not be without cost, which will have to be borne by the shearers themselves.

Keeping the comb sharp puts less stress on the equipment and this will help, although we could not measure the effect.

The issues surrounding the music in sheds is likely to be a complex one, but one which could reduce the exposures significantly for all the groups, those more remote from the shearing potentially gaining more benefit. Education as to what levels are harmful, and some simple noise monitoring equipment to measure the noise and keep the listening level reasonable, would help.

Maintenance of the plant and the acoustic environment of the shed would also reduce exposure and are the responsibility of the farm owner or operator. Proper maintenance of the gear could be insisted upon, but the expense of acoustically treating sheds which are in use for a matter of weeks

Figure 1. Shearers and sorters.

Figure 2. Woolpress.

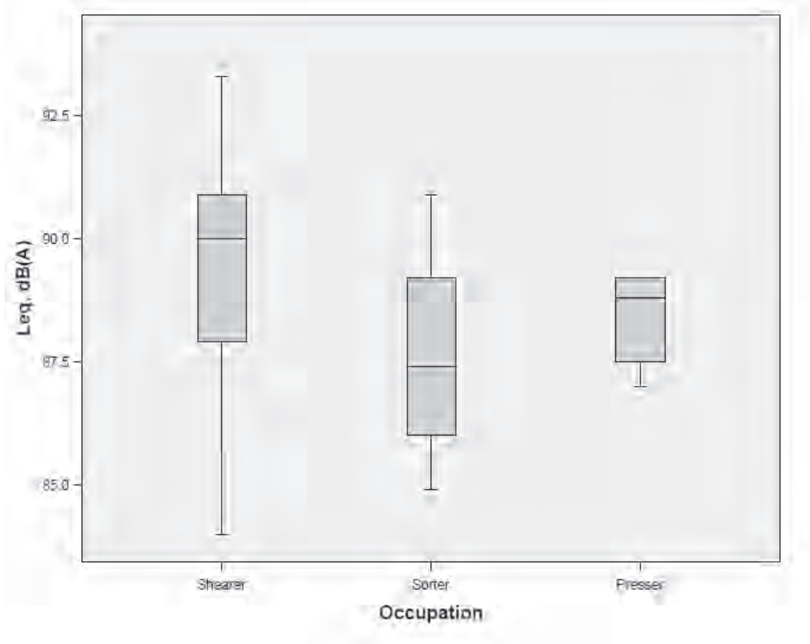

Figure 3. Box plots of noise levels for groups.

(or days) in the year might be more problematic.

This group of workers really do need to use hearing protection, which was not in evidence at all. Education and training would undoubtedly help, but the social cohesion and identity of this group might prove a problem in trying to promote attitudinal change.

We have not, as yet, been able to perform base-line (in the quiet) and compared this with monitoring (after noise exposure) hearing tests to detect temporary threshold shift (TTS).

Although there is debate surrounding the issue regarding whether TTS leads to Permanent Threshold Shift (PTS) it probably does indicate cochlear strain. Such evidence may strengthen the case for proactivity in the use of hearing protection.

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ACKNOWLEDGEMENTS

The study was funded by the Accident Compensation Corporation.

We would like to thank the New Zealand Shearing Industry Health and Safety Committee for their support and encouragement.

On 1st July 2011

The New Zealand Acoustical Society

officially became:

The Acoustical Society of New Zealand

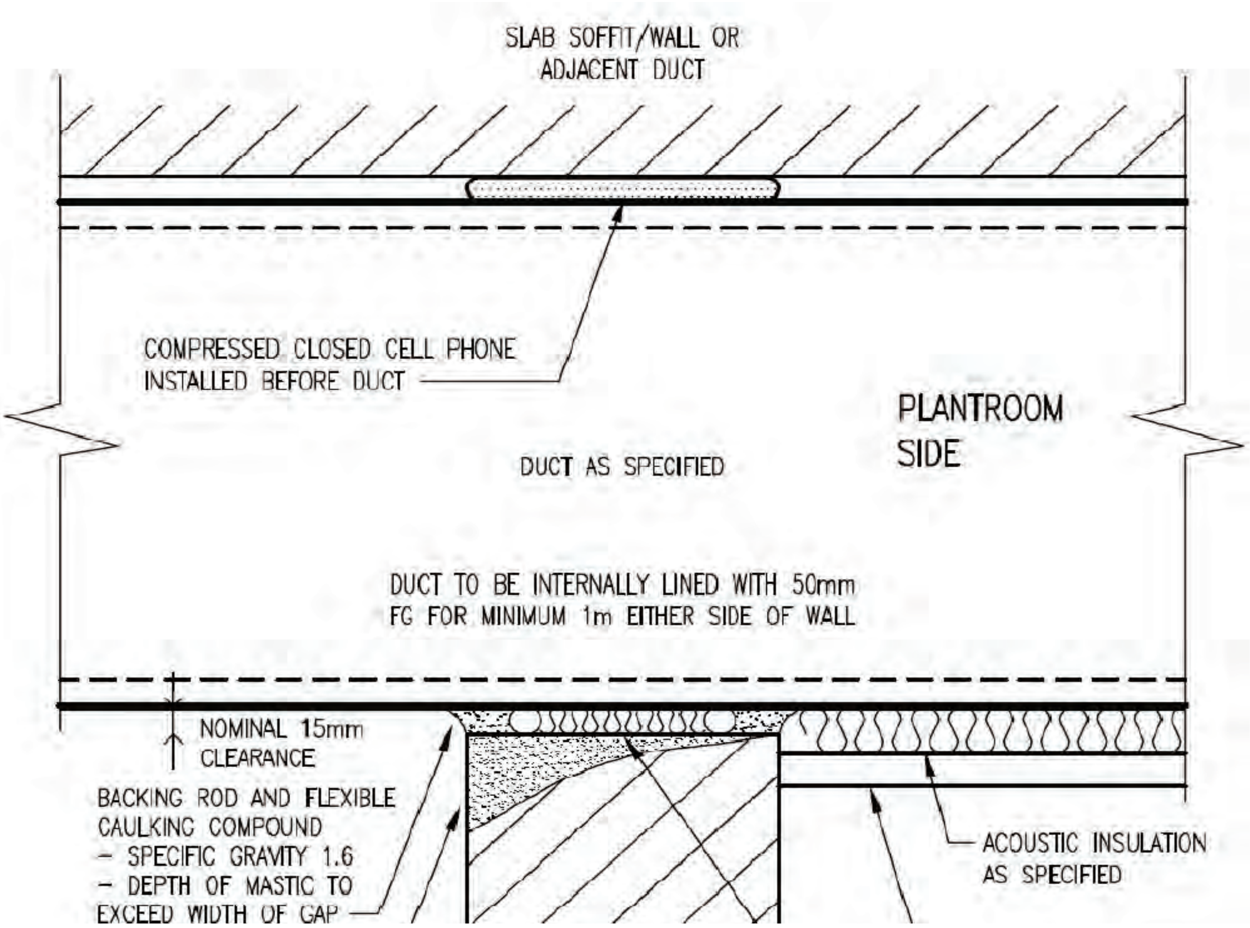
This change has been submitted and accepted by the Registrar of Incorporated Societies, along with our amended Rules and Rules of Conduct.

Thank you all for your support and encouragement in making this important change.

We warmly invite all members to apply to become a Member or Affiliate of the ASNZ.

Please see www.acoustics.ac.nz/membership for all details, documents, transition procedures and membership forms.

Distractions:
What's Wrong with this Acoustic Duct Penetration Detail?

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Modelling and Experimental Validation of Complex Locally Resonant Structures

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A paper previously presented at ISSA 2010, 29-31 August 2010, Auckland

Abstract

The increase in population worldwide has highlighted the inadequacies of sound insulation in buildings. The problem is particularly evident in medium-high density housing situations, which are projected to become 30% of Auckland's housing by 2050. This will have implications on occupants' health, productivity and quality of life. Prevention of sound transmission through walls and ceilings in the lower frequency range of human hearing is particularly important, but is a difficult problem. This problem provides an opportunity to ask the question: Can we design an acoustic insulation system that provides improved sound insulation performance over a conventional system, within this frequency range? This paper outlines an investigation into novel meta-materials known as Locally Resonant Structures. These structures can exhibit acoustic band gaps, or frequency ranges of unusually low sound transmission. One-dimensional mathematical models are used in conjunction with finite element analysis FEA to develop various locally resonant element concepts functional below 1kHz. Acoustic testing is then used to experimentally verify the performance of the elements through comparisons with modelling data. Various resonator elements have shown a peak effective mass up to fifty times greater than their rest mass. Locally resonant structures have increased peak transmission losses by as much as 40dB over that of a non-resonant structure of equivalent area density within the designated frequency range. These resonators can be distributed throughout the wall structure on a scale shorter than the wavelength of structural vibrations in the wall matrix. The resulting system has the potential to provide significantly higher transmission loss at low frequencies than conventional wall systems of similar size and weight. The longer term goal is to determine an effective design of local resonator that can be incorporated into a practical insulation system.

Introduction

Background

As the population density increases the power of domestic home entertainment systems grow and automation proliferates, so does noise pollution. There is increasing concern in New Zealand [1] and overseas [2], about inadequate sound insulation in buildings and the consequent implications for occupants' health and well-being both in the public and private sector.

Indications from recent studies [3], [4], [5], show growing dissatisfaction from residents regarding the acoustic performance of their accommodation, reflected by an increasing number of noise nuisance complaints [5]. The problem is particularly evident in medium-high density housing situations, which are projected to become 30% of Auckland's housing by 2050 [6].

Acoustic intrusion commonly occurs at frequencies below 1 kHz (i.e the bass beat from music systems) where human hearing has its highest sensitivity, but achieving effective insulation in this range with conventional solutions such as increasing the density or total mass of the partition is both challenging and expensive. This provides an opportunity to ask the question: Can we design an acoustic insulation system that provides improved sound insulation performance over a system with an equivalent mass density, within this frequency range?

This paper will focus on the development of locally resonant structures LRS. Simple analytical models of single- and multiresonant spring (linear) mass systems have been used to study important design trade-offs and response characteristics such as band width, band positioning and sound transmission loss during and after the frequency of localised resonance. New LRS specimens are then subjected to dynamic, plane wave impedance tube and diffuse field testing methods, to indicate the performance of the meta-material samples.

Sound insulation

Conventional sound insulation involves a barrier which reflects sound transmission energy. For plane waves travelling though a medium the quantity most commonly used for expressing the performance of a partition's sound insulation is the transmission loss TL or sound reduction index *R*. First defined in the 1950s [7], the sound reduction index is related to the transmission coefficient, t by:

$$R = 10\log_{10}\left(\frac{1}{\tau}\right) \tag{1}$$

The transmission coefficient is a frequency dependent fraction of the incident sound energy and the transmitted sound energy through a medium.

In the problem frequency range, reflection properties are dominated by the mass/area. This region may be approximated by the mass law equation:

$$R_{ml} = 10\log_{10}\left[1+\left(\frac{\pi f M \cos\theta}{\rho_a c_a}\right)^2\right] \tag{2}$$

where M is the mass per area, r_a, is the density of air, c_a, speed of sound, f , is the frequency, and q is the angle of incidence. The sound reduction index is maximised when sound is transmitted at normal incidence.

Meta-materials

Meta-materials are artificial materials engineered to provide properties which may not be readily available in nature. These inhomogeneous materials have a non-uniform composition. Klironomos et al [8] showed the presence of inhomogeneity in a material can influence the propagation of waves in periodic material structures. These materials have been developed by John et al [9] and Kushwaha et al. [10] in the areas of electromagnetics and acoustics respectively. Meta-materials can form band gaps enabling the material to prevent wave transmission in specific frequency ranges of electromagnetic, elastic or acoustic waves in any direction. There are two current mechanisms that can be used to create band gap materials [11]:

- Bragg scattering,
- Localised resonance

Analysis of large scale acoustic Bragg scattering was first realised in 1995 by Martinez-sala et al [12] where he described the sound transmission properties of a large open air sculpture in Madrid. This sculpture consisted of a periodic crystal-like arrangement of tall metal rods. Band gap behaviour of these structures is due to the phenomena of wave diffraction and interference created by the higher density rods acting as scattering reflectors. In order to create an acoustic band gap in the audible range using Bragg scattering, the internal structure of the material needs to be large. This is because for the existence of Bragg scattering, it is required that the lattice constant/ arrangement be a minimum of half the wave length of the incident sound wave [13]. For low frequency wavelengths in order of metres, this is simply to large to be practical for insulation applications.

Localised resonances were used to create band gaps in 2000 by Liu et al. [14] where a three component meta-material, including a host material with polymer coated rigid inclusions, was used to create localised resonances. Essentially the frequency of the band gap is dictated by the resonant frequency of the resonators and is independent of periodicity and symmetry. LRS use internal resonances to alter the effective properties of the material at different frequencies. One such property is the ability to inhibit sound transmission in a targeted frequency range. This was proved in 2000 Liu et al. [14] when a significant improvement in sound transmission loss was found between 200-1000 Hz in a selected 100Hz band using a unique LRS known by Liu et al. [14] as a locally resonant sonic material LRSM.

Theory

It has been shown by Milton et al. [15], Yao et al. [16], Huang and Sun [17, 18], Gang et al. [19], Calius et al. [20] that the essential features of LRS can be captured by spring-mass models. Figure 1 is a spring-mass model representation of a single resonance LRS. This model shows a mass attached to a spring mounted on a backing layer suspended on two more springs.

The point force F applied to the layer represents the pressure applied by a plane wave sound field on the structure. The response of the system shown in Figure 1 (where time dependence eiwt is assumed) can be represented as the

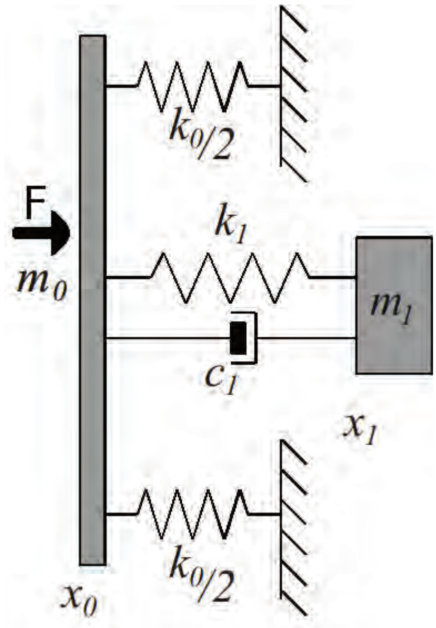


Figure 1: Spring-mass model.

stiffness, damping and mass matrix below:

$$\begin{pmatrix} F \\ 0 \end{pmatrix} = \begin{bmatrix} k_0 + k_1 - m_0\omega^2 + i\omega c_1 & -k_1 - i\omega c_1 \\ -k_1 - i\omega c_1 & k_1 - m_1\omega^2 + i\omega c_1 \end{bmatrix} \begin{pmatrix} x_0 \\ x_1 \end{pmatrix}$$

F is the total force, x is the displacement, m is the mass, c is the damping coefficient and k is the spring constant. By rearranging and solving the matrix and assuming no damping it is possible to obtain the systems effective mass (m_T) as [15]:

$$m_T = \frac{F}{a_{m_o}} = m_0 + m_1 \frac{\omega_1^2}{(\omega_1^2 - \omega^2)} \qquad (3)$$

In this equation F is the externally applied force and a_{mo} is the acceleration of the host/layer. w_1 is the resonant frequency of the spring k_1 and mass m_1 when attached to a rigid base and may be found using:

$$\omega_1 = \sqrt{\frac{k_1}{m_1}} \qquad (4)$$

By changing the spring stiffness k_1, or internal mass m_1 the resonant frequency

Figure 1: Construction of the floor structure.

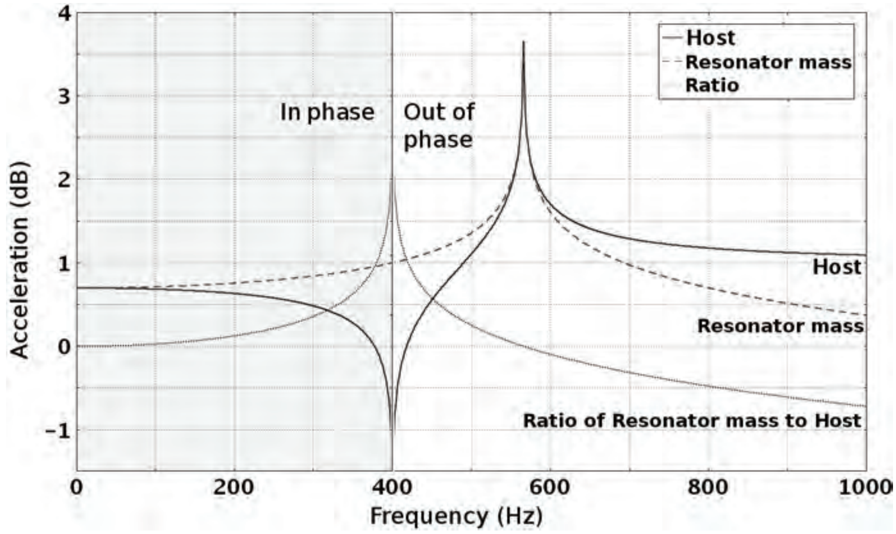


Figure 2: The logarithm acceleration of the host and resonator mass of the LRS in Figure 1 when f_1 = 400Hz. The plot also shows the acceleration ratio of the resonator mass to host material.

and the amplitude may altered. The relative acceleration and phase of the the layer/host material m_o and the resonator mass m_1 from Figure 1 (where the supporting springs, k_0 are neglected) may be seen in Figure 2.

It may be seen in Figure 2 and from analysis of equation 3 that at frequencies well below $f_1 = \omega_1/2\pi$ = 400Hz, the acceleration of the host material and resonator mass is close to equal, and the effective mass is approximately the sum of the components $m_T = m_0 + m_1$. At frequencies far above the resonance frequency, where $\omega >> \omega_1$, the acceleration of the resonator mass

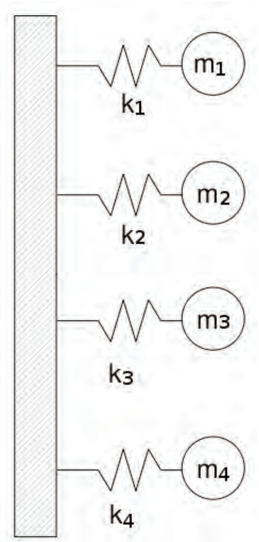


Figure 3: Resonators in parallel. These resonators have different mass and spring stiffnesses. Damping not shown for clarity.

approaches zero and the total effective mass becomes the host material only, where $m_T = m_0$.

The frequency range most relevant to this research are frequencies around resonance. As ω approaches ω_1 the resonator mass and host components are in-phase with each other. It can be seen in Figure 2 that the acceleration of the host material drops towards zero whilst the resonator mass acceleration is increasing.

At $\omega = \omega_1$ the ratio of the resonator mass and host acceleration is at a maximum and the host material is almost stationary. The host material has a large total effective mass and therefore sound transmission can be reflected well at this frequency. At frequencies immediately above resonance components become out-of-phase with each other and it may be seen in Figure 2 that the acceleration of both the host and resonator mass increases to a maximum. At this point the ratio of the two components is zero. The result of this high acceleration in the host material is a decrease in the effective mass to almost zero and an increase in transmission though the structure. By substituting m_T = 0 and rearranging equation 3 it can be seen that the frequency at which this occurs is described by [16]:

$$\omega = \omega_1\sqrt{\frac{(m_0+m)}{m_0}} \qquad (5)$$

The biggest draw backs found in LRS research to date are the inability to produce attenuation over a wide range of frequencies, the detrimental effects (immediately after resonance) on transmission loss from the peak acceleration of the the matrix material, and to a lesser extent, the limiting effect of damping.

LRS performance, including the relative magnitude of these drawbacks, is strongly affected not only by the characteristics of the resonator itself, but more importantly by the way these local resonators are connected together to form the LRS. The realization of useful LRS-based applications depends on the combination of cost-effective materials and processes with modelling tools that enable design, analysis and optimization. A modelling-driven building block approach is being used to develop LRS designs, with experimental verification at every level. The locally resonant unit represented schematically in Figure 1 provides the basic building blocks from which groups of resonant units are integrated to form layers which are combined to form panels.

This paper presents a modelling methodology that predicts the transmission loss of a variety of resonator arrangements, together with initial measurements of sound transmission loss in an impedance tube and a full-scale room-to-room test facility. This modelling methodology is then used to explore the sensitivity of LRS performance to design parameters, with particular attention to broadening the transmission loss bandwidth and reducing detrimental effects outside this frequency band.

Methodology

It is well known that complex mechanical systems can be represented by a combination of a large enough

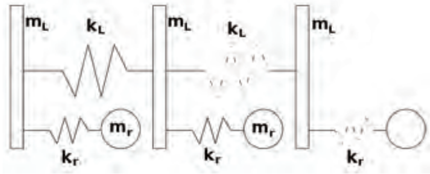


Figure 4: Resonators in Series. The spring stiffness, and mass of the layers are constant. All resonators have the same stiffness and mass. Damping is not shown for clarity.

number of singledegree- of-freedom SDOF subsystems such as the one depicted schematically in Figure 1 [21].

Modelling

A program ComScptv1 has been developed in Comsol script to calculate the normal incidence transmission loss of a system of layers and resonators using springs and masses coupled in various ways. The program enables the user to change the number of layers, the number of resonators per layer and the mass and spring stiffness of each component. For each springmass arrangement ComScptv1 calculates the stiffness, damping and mass matrix. ComScptv1 may then find the displacement (*x*) at an arbitrary point. The effective mass (mt) of the system is found using the relation:

$$m_T = \frac{F}{\ddot{x}} = \frac{F}{-w^2 x} \qquad (6)$$

The effective mass per area (M_T) may then be calculated using:

$$M_T = \frac{m_T}{S} \qquad (7)$$

where S is the surface area of the layer/ panel. Assuming q = 0 for normal incidence plane waves and pfM_T/ $r_a c_a$ >> 1 for most well insulating walls, the sound reduction index (R_0) of the

system is then found from:

$$R_0 = 20\log_{10}\left[\frac{\pi f M_T}{\rho_a c_a}\right] \qquad (8)$$

Transmission loss typically varies with angle of incidence. When predicting the sound reduction index for a sample subjected to diffuse field transmission (R_d) at frequencies in the mass controlled region, equation 9 [22] was used, where R_0 is plane wave simulation or experimental impedance tube sound reduction index results and k is the wavenumber.

$$R_d = R_0 - 10\log_{10}\left[\ln\left(kS^{\frac{1}{2}}\right)\right] + 20\log_{10}\left[1 - \left(\frac{\omega}{\omega_1}\right)^2\right] \qquad (9)$$

Three resonator configurations have been analysed using this modelling method. The first configuration is shown in Figure 1. This is a single resonance frequency system. Resonators are added to the system in parallel, but all resonators have the same resonant frequencies.

Parallel configuration

The second configuration, shown in Figure 3, incorporates four resonators in parallel. Each resonator has a different frequency.

Series configuration

The final arrangement first realised in [16] and shown in Figure 4 includes a number of resonant layers in series. Each resonator has its own mass (m_r) and spring stiffness (k_r). The resonators are attached to layers with mass (m_L). The layers are arranged in series, and

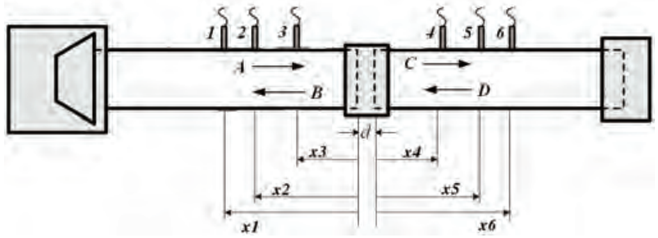


Figure 5: Impedance tube.

separated by a inter-layer coupling spring with stiffness (k_L). In this situation any spring stiffness and mass may be altered to provide different performance characteristics. In this configuration the ComScptv1 keeps a constant system mass by changing the mass of each resonator and layer depending on the number of units in the system. Multiple high TL bands may also be created in a single structure by designing systems of layers in series with a constant inter-layer coupling spring stiffness. For example in a 9 layer system it is possible to group 3 lots of 3 layers together by changing the inter-layer coupling spring stiffness of each group to be equal.

Experimental methods

Two different experimental methods were used to generate performance data and validate the modelling approach. Laboratory scale evaluations were performed using an impedance tube, which is suitable for testing single units or small groups of resonators, and measuring the transmission and reflection of predominantly plane waves. Full-scale measurements were performed between reverberation rooms, which were suitable for testing relatively large specimens consisting of many resonators under diffuse sound field conditions.

Plane wave testing

The impedance tube was adapted for normal incidence transmission loss measurements and designed to conform to the European Standard ISO 10534-2:2001(E). The dimensions of the tube shown in Figure 5 are based around the B & K Type 4026 impedance tube. The resonator units being tested were housed within a 100mm diameter hollow cylinder made from medium density fibre MDF, which could contain several resonators. Alternative resonator designs were attached to a backing plate that represented the matrix material of the LRS.

The cylindrical LRS sample was suspended on two rubber rings between two parts of the impedance tube. A loud speaker generates plane wave sound that propagates down the first tube. Part of the signal is transmitted through the sample which is measured in the second tube using three microphones. Microphones 1, 2 and 3 are used to find the transmitted side complex wave constants A and B whilst microphones 4, 5 and 6 are used to find the receiving complex wave constants C and D.

The pressures found from each B&K 4190 microphone may be written as the equations shown below:

$$
\begin{aligned}
P_1 &= Ae^{j(\omega t - kx_1)} + Be^{j(\omega t + kx_1)} \\
P_2 &= Ae^{j(\omega t - kx_2)} + Be^{j(\omega t + kx_2)} \\
P_3 &= Ae^{j(\omega t - kx_3)} + Be^{j(\omega t + kx_3)} \\
P_4 &= Ce^{j(\omega t - kx_4)} + De^{j(\omega t + kx_4)} \\
P_5 &= Ce^{j(\omega t - kx_5)} + De^{j(\omega t + kx_5)} \\
P_6 &= Ce^{j(\omega t - kx_6)} + De^{j(\omega t + kx_6)}
\end{aligned}
$$

The constants (A,B,C,D) are then found by solving the complex equations using the least squares determinant method. The receiving side of the impedance tube is in anechoic conditions where reflections (*D*) are assumed to be near 0 and hence the transmission coefficient is found to be near the ratio of A to C. The transfer coefficient may be found from $t = (AC-BD)/(AA-DD)$. Where t is the transmission coefficient. When t is applied to $R = 20\log10(1/|t|)$ the sound reduction index may be found.

Diffuse field testing

The room-to-room testing facility shown in Figure 6 used for full-scale diffuse field testing was designed to ISO 140-3. Two reverberation rooms (202 and 208 m^3) are used to measure the sound reduction index of the samples. The test specimen is placed so as to fill the adjustable gap between two well-insulated sliding doors that separate the two rooms. A broadband pink noise source signal is then placed in one of the rooms. The spatial average sound pressure and reverberation times RT in the emitting and receiving rooms is then measured using 1/2" B&K 4190 and 4165 microphone. The process is then repeated with the noise source in the other room. Data was processed in third

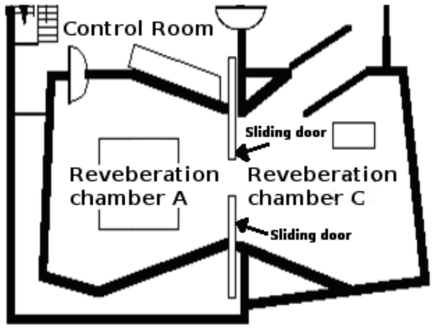


Figure 6: The University of Auckland acoustics research centre (ARC) room to room testing facility.

octaves.

In order to study the frequency response in more detail than 3rd octaves, the spectrum was found by calculating the Power spectral density (square of the magnitude of the Fourier transform of the signal) from the raw time domain pressure signals. The narrow band R_T was found by interpolating the 3rd octave R_T results. The absorption area of the receiving room was found using:

$$A = \frac{0.163V}{T_{60}} \quad (10)$$

where T_{60} is the reverberation time, V is the volume of the receiving room. The level difference (δL) of the specimen was then calculated from:

$$\delta L = 10\log_{10}[P_0] - 10\log_{10}[P_1] \quad (11)$$

where P_0 is the incident sound power and P_1 is the radiated sound power. Under the assumption of diffuse sound fields in the transmitting and receiving rooms the actual sound reduction index of the specimen may be found using:

$$R_d = \delta L + 10\log_{10}\left[\frac{S}{A}\right] \quad (12)$$

where S is the area of the wall specimen. The single-layer panel consisted of 252 resonators attached to a 2.65 x 0.95 x 0.01m plasterboard matrix layer using Loctite 401 adhesive.

The accelerations perpendicular to the panel plane were also measured while the panel was subjected to pink noise . For each measurement two PCB A353 B65 accelerometers were attached at any 2 of 9 different positions on the back of the panel using wax. As well as the amplitude of the acceleration as a function of frequency at various locations, these measurements also allowed the phase difference to be calculated between adjacent resonators. This testing method gives in-site into the sound insulating performance of large scale meta-material samples under a diffuse field.

Results

Single frequency locally resonant structures The features of a single frequency LRS have been modelled and are shown in Figure 7. There is a large increase in transmission loss at 400Hz which occurs at the resonant frequency (ω_1) of the resonator. At

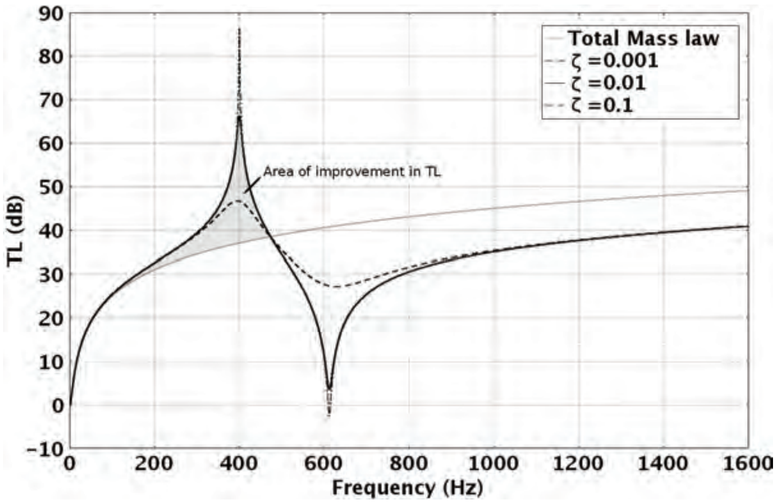


Figure 7: Comparison of transmission loss verse frequency of a single resonant frequency LRS showing the effect of increasing the damping factor z from 0.001 to 0.1.

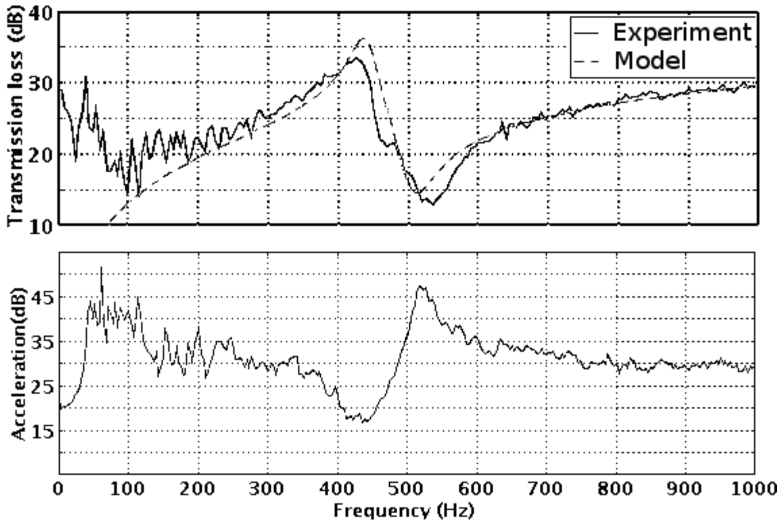


Figure 8: Frequency comparisons of diffuse field transmission loss and typical panel acceleration for a large single resonant frequency LRS panel.

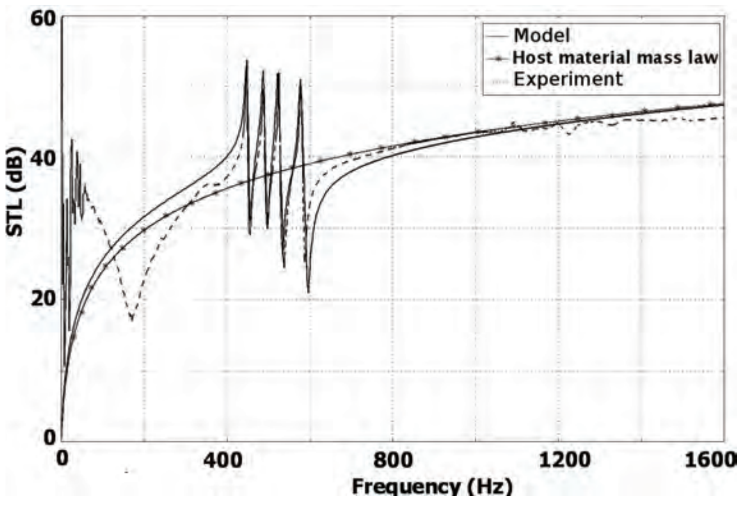


Figure 9: Comparison of transmission loss verses frequency of modelling and experimental results for a LRS with four nonidentical resonator units arranged in parallel. The host mass law is also shown.

this frequency the host material has its lowest acceleration magnitude and the LRS has a high effective mass. w_1 may be manipulated by changing the mass and stiffness of the resonator spring and mass. The large dip in transmission loss soon after this peak is the result of a high acceleration magnitude of both the host and resonator mass and therefore a low total effective mass. The effect of damping in a single resonant frequency arrangement is also shown in Figure 7. It can be seen that as damping is increased there is a smoothing effect on the resonant peak. Higher damping lowers the maximum peak sound reduction index, but also reduces the sound reduction index dip.

The results for a single-layer panel with 252 nominally identical resonators arranged in a system essentially like Figure 1 are given in Figure 8. The top graph shows the experimental transmission loss obtained from full-scale testing of the panel and the transmission loss for the equivalent diffuse field spring mass model over the frequency range of interest approximated by equation 9. It is clear that there is a good match between the model and experiment above 200 Hz. At frequencies below 200 Hz the uncertainty in the measurements increases significantly as the wavelength starts to approach the panel and reverberant room dimensions and the panel mounting resonance is reached.

This effect rapidly escalates at 100 Hz and below. The bottom graph gives the acceleration of the panel at its centre over the same frequency range. This graph confirms the acceleration of the host/panel material is at a minimum when the transmission loss is at a maximum and visa-versa.

It has been observed from single resonance modelling and experimental analysis that not only is there a transmission loss performance gain at around w_1, but before this frequency there are also transmission loss performance benefits. These are indicated by the shaded region in Figure 7. If w_1 is raised in frequency (by increasing the spring stiffness), both the sound reduction index performance gains over mass law and frequency band width increases.

Parallel multi frequency systems

The results of parallel arrangements of non-identical resonators on a single layer are presented in this section. The predicted and measured plane wave sound transmission loss for an LRS with four non-identical resonator units arranged in parallel configuration similar to Figure 3 are compared in Figure 9. The LRS consisted of four individual resonators spaced at resonant frequencies of 40Hz apart, each with equal mass, but differing in spring stiffness. The results from impedance tube testing provide experimental verification of the spring mass model, with excellent agreement near the resonant region and above. The resonance of the compliant rubber suspension system that holds the LRS test specimen in the impedance tube is responsible for the additional transmission loss peak and valley observed in the test data below 200 Hz.

Figure 10 shows the sound reduction index results when modelling 15 resonators in parallel. All 15 masses are of equal weight, but their frequency of resonance is defined by the spring stiffness of the resonator. The resonant frequencies have been spaced 15 Hz apart, and a small amount of damping z= 0:01 has been added to create a more realistic representation of the material. Also included in this plot is a single resonator with the total mass of the sum of all 15 resonators with the same damping factor. The mass law of the host material and total system is also shown. It can be seen that the parallel arrangement approximately doubles the bandwidth of attenuation above mass law with some reduction in TL.

The sound transmission loss for a LRS with 8 different resonators is shown in Figure 11. Note the damping factor of z = 0:05 is significantly higher than for the

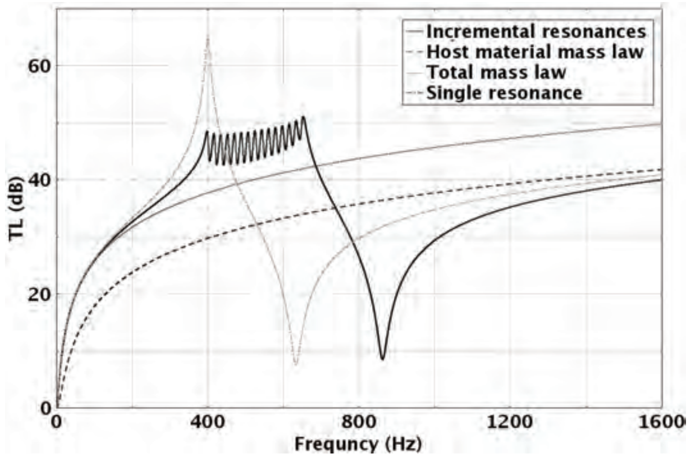


Figure 10: Comparison of single frequency and 15 frequencies LRS. The latter has 15 resonant frequencies generated by resonators arranged in parallel. The total system mass is constant with a damping factor z = 0.01.

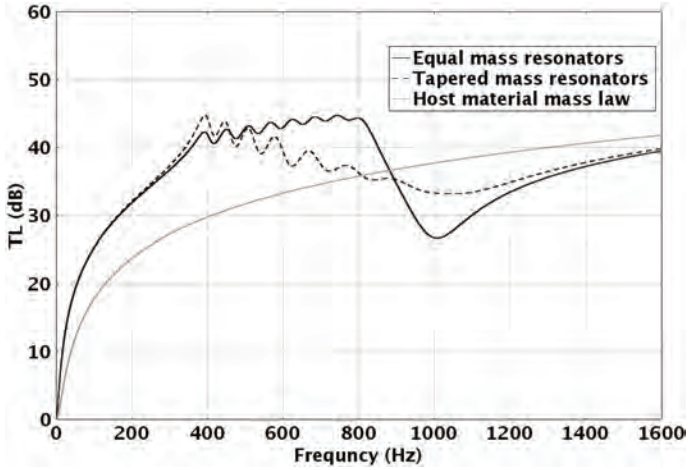
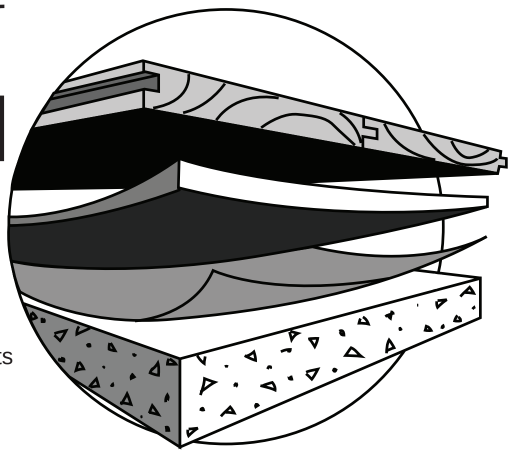


Figure 11: Comparison of transmission loss verses frequency for parallel LRS with equal mass resonators and tapered mass resonators. Damping z = 0.05 is used.

EMBELTON

IMPACTAMAT

FLOOR ISOLATION



TYPICAL PERFORMANCE CHARACTERISTICS

The table below gives IIC ratings based on tests of various surface treatments Ref. ASTM E989 using an Impactamat resilient interface on a 100mm thick concrete structural floor.

FLOOR SURFACE TREATMENT (Floating Floor Construction)	Impactamat			Overall IIC Rating	IIC Improvement over bare slab	Ref.fig.
	Construction	Type	Thickness			
Loose lay timber veneer flooring with thin foam bedding layer	full cover	750	5mm	47-50	18-20	1
Direct bond 19mm block parquetry	full cover	900	5mm	45-49	18-20	2
Direct bond 10mm ceramic tiles	full cover	750	5mm	44-46	13-15	2
Particle board or strip timber battens supported at nom. 450 x 450 centres with acoustic absorption	pads 75 x 50mm	750	10mm	52-60	21-30	3
Double layer bonded 12mm ply with bonded parquetry, supported at nom. 300 x 300 centres (sports floor)	pads 75 x 50mm	750	10mm	52-57	21-27	4
50mm reinforced concrete slab or 25 mm slab with 20mm bonded marble/slate/ceramic tile	full cover	750	10mm	58-63	27-32	6
50mm reinforced concrete slab	full cover	750	15mm	59-64	28-33	5
100mm reinforced concrete slab	full cover	750	15mm	60-65	29-34	5

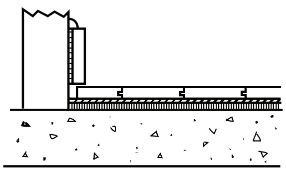


Fig. 1 Timber loose lay floating floor

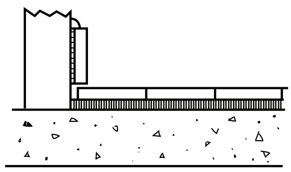


Fig. 2 Direct bond parquetry or ceramic tiles

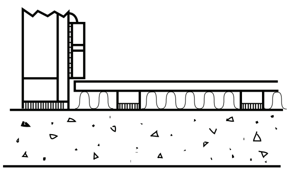


Fig. 3 Timber strip floor on battens with isolated frame wall

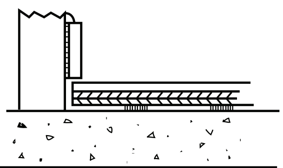


Fig. 4 Sports floor

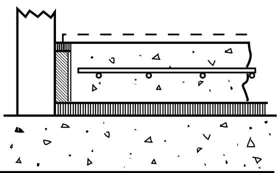


Fig. 5 Concrete slab

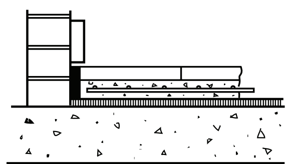


Fig. 6 Marble/slate ceramic tiles with thin reinforced slab

IMPACTAMAT by EMBELTON features two main environmental properties: it is recycled and it reduces noise pollution.
Indeed, it is made from 100% recycled natural rubber recovered from tyres, granulated and reconstituted as a solid mat (various sizes are available upon request).

IMPACTAMAT is a flexible material manufactured as a preformed sheet bound together with a flexible binder.
It is a low cost impact absorbing layer for covering hard earth or concrete in outdoor applications or as an underlay for in-situ cast or pre-cast concrete floors where noise isolation is required (rubber underlay, acoustic insulation, door mats, playground and sports surfaces, industrial floor tiles etc.).

VIBRATION CONTROL LTD.

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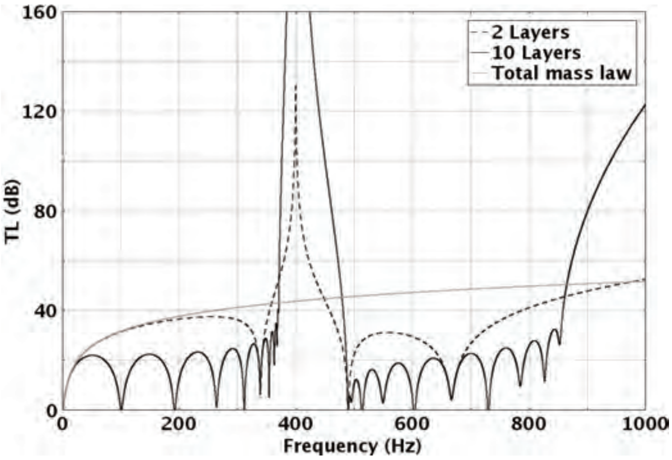

Figure 12: Comparison of transmission loss verses frequency for 2 and 10 layer LRS. Inter-layer coupling spring and resonator spring stiffnesses are all constant and a damping factor z of 0.001 is used.

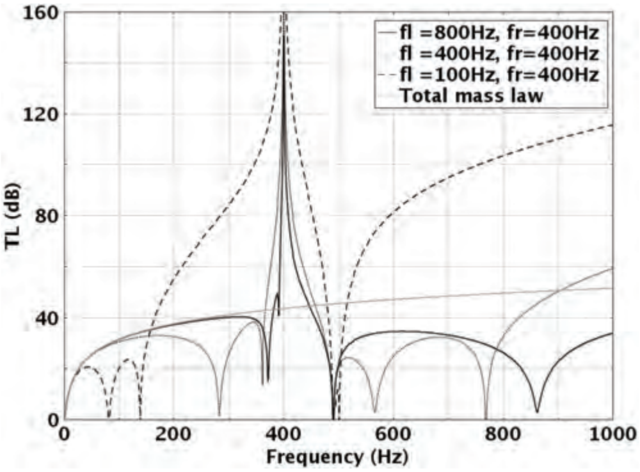

Figure 13: Comparison of transmission loss verses frequency of a 3 layer LRS. The frequency of resonance of the resonators (f_r) remains constant while the inter-layer coupling frequencies of resonance (f_l) are tuned to resonant frequencies from 400-800Hz by adjusting k_L. the model uses a damping factor z of 0.001.

similar system whose response is given in Figure 10. The plot also shows a more gradual reduction in transmission loss that avoids the development of a dip in transmission loss around 950Hz. This is due to the use of mass tail-off where the mass of the highest frequency resonator is lowered from 0.03 to 0.004 relative to the lowest.

Series systems

Systems consisting of series-coupled resonators are investigated in this section. The predicted transmission loss for seriescoupled systems are shown in Figures 12-15. Figure 12 shows the predicted transmission loss for a multilayer system similar to Figure 4. The plot shows the sound reduction index as a function of frequency for increasing numbers of layers. The total mass of the system is held constant as the number of layers is increased. A low damping factor of 0.001 is applied throughout the arrangement. There is high attenuation present between approximately 400 and 500Hz. As the number of layers increases the band gap increases in attenuation, however the width of this region remains similar. With an increase in the number of layers there is an increase in the number of transmission loss dips before and after the band gap.

Figure 13 shows sound transmission loss of a three layer system with variation in the inter-layer coupling spring stiffness, k_L. As k_L is reduced, while the spring stiffness of the resonators, kr remains constant, the frequency band width of high attenuation increases by a factor of 3 while the peak attenuation remains constant. The band width is dictated by the ratio of k_L to k_r and k_L must be lower than k_r for a wide attenuation band to occur.

A three layer system with the addition of damping is shown in Figure 14. High damping was applied (z = 0:1) to the interlayer coupling springs. Damping in the resonator springs remained low (z =0:001). Transmission loss dips before and after the band of high TL are significantly reduced without effecting the high TL band magnitude.

An example of a system with multiple band gaps is shown in Figure 15. This system consists of a total of 9 layers created by coupling 3 groups of 3 layers. The inter-layer coupling springs in each group are tuned to the same resonant frequency f_l and resonators in each group are tuned to the same resonant frequency f_r. For the first group f_l=400Hz and f_r=500Hz, for the second group f1=800Hz and fr=900Hz and for the last group f_l=1200Hz and f_r=1300Hz.

Series Parallel systems

Series and parallel arrangements may be combined to incorporate their different performance characteristics. Figure 16 shows the performance of a 15 layer system. The inter-layer coupling spring stiffnesses are constant throughout the system. The resonator resonant frequency increases with layer number in increments of 15Hz via changing the resonator spring stiffness. The band of large transmission loss has been widened to between approximately 400 and 650Hz compared with 400 and 500Hz for the conventional 10 layer series system shown in 12 with the same total system mass.

Figure 17 shows 8 layers of parallel arranged resonators in series. There are 15 parallel resonators on each layer. Each layer has the individual performance shown in Figure 10. When the layers are combined in series with a constant inter-layer coupling spring stiffness, there is an increase in transmission loss over the equivalent single layer parallel arrangement for the same frequency range (between 400 and 650Hz).

Discussion

The results obtained for single-frequency LRS through both experimental observations and analysis indicates that when w <= w_1, where w_1 is the resonant frequency of the resonator, the LRS transmission loss is equal or greater than that of a homogeneous material with the equivalent mass area density

At frequencies approaching ω_1, the host material and resonator mass are in phase with each other. It may be seen by modelling shown in Figure 2 that the acceleration of the resonator mass gradually increases in this frequency region while the host material reduces in acceleration. At the resonant frequency of the resonator mass, experimental results shown in Figure 8 indicate the acceleration of host material becomes zero and the LRS has large effective mass. The result of this is a high transmission loss which is clearly indicated in Figure 7 and 8 around 400 Hz. It is interesting to note that at this frequency the resonator mass has not yet reached its peak acceleration. At frequencies above this area the motion of the host material and resonator mass are out of phase with each other. It may be seen in Figure 7 and Figure 8 that at 600 Hz there is a significant drop in transmission loss. This feature is the result of a peak in acceleration of both the resonator mass and host material shown in Figure 2 and Figure 8. At this point the effective mass of the LRS becomes zero. The frequency of zero effective mass may be calculated using equation 5. At much higher frequencies the resonator mass acceleration approaches zero and the effective mass of the LRS is approximately equal to the mass of the host material.

The comparisons between modelling and experimental results given in the top graph of Figure 8 and Figure 9, together with the experimental results obtained by Yao et al.[16], confirm that systems of linear spring-mass models

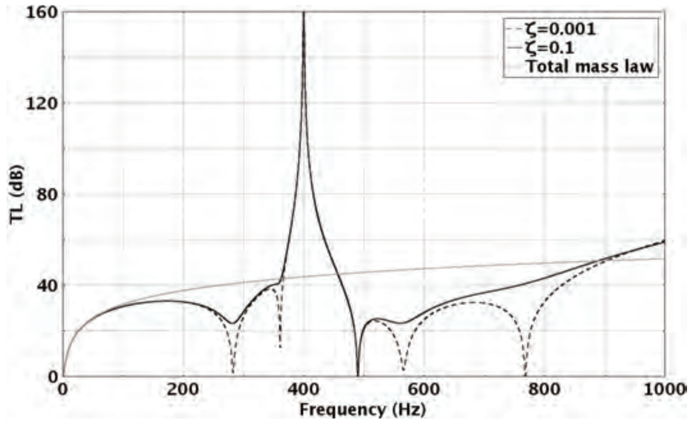

Figure 14: Comparison of the effect damping in the inter-layer coupling springs has on transmission loss verses frequency of a 3 layer LRS. The damping factor ζ in the inter-layer coupling springs ranges from 0.001 to 0.1.

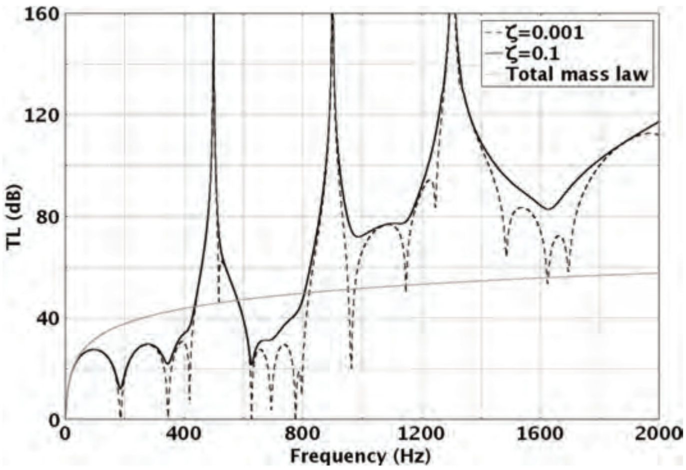

Figure 15: Comparison of transmission loss verses frequency for a 9 layer multi-band-gap LRS. 3 layer groups tuned at 3 different frequencies. There is a damping factor ζ of 0.001 and 0.1 in the inter-layer coupling springs.

such as those represented schematically in Figures 1 through 4 can be used to obtain an accurate estimate of LRS sound transmission behaviour, whether it is in an impedance tube or between reverberant chambers.

The key question for practical applications is how to maximize the frequency band over which the LRS is effective while also achieving a large enough increase in sound attenuation within that frequency range. Ideally the LRS will have a multiplicity of resonances at frequencies that are so close together that the resonant peaks overlap. This can be achieved by designing systems of resonant units with incremental closely spaced frequencies and an appropriate amount of damping. Other researchers [23] [24] [25] have approached this problem by constructing multilayer LRS where each layer has a single resonance frequency, that is the resonators in any given layer are all tuned to the same frequency, but this frequency is different from layer to layer. The limitation inherent in this approach is that increasing the system's bandwidth requires additional layers, each layer increases the thickness of the system, and building applications impose practical limits on the total thickness.

So the question was whether sets of resonant units with different but closely spaced resonant frequencies arranged in a single layer could produce a similar effect. The experimental and model results presented in this paper demonstrate that this is the case. When resonators are placed in parallel with only a small amount of damping, Figure 10 shows that the band of increased attenuation becomes several times wider than for the previous single resonator arrangement, but has a significantly lower peak magnitude. Similar to the single-frequency LRS, there is also a significant drop in the transmission loss after resonance due to a drop in the effective mass of the structure. Eventually the LRS curve asymptotically approaches the host material mass law. Note that the merging of the separate resonant peaks in this parallel system is strongly influenced by the amount of damping, as is easily seen by comparing Figures 9, 10 and 11.

One method of shaping the transmission loss curve and reducing the transmission loss dip around 1000 Hz is to use different masses in the resonators tuned to different frequencies. Figure 11 shows that by tapering off the weights of the parallel resonator masses gradually from the bottom to the top over the LRS high TL band and with the addition of damping, the result is a smoothing effect over the TL dip after localised resonance. It has been already shown that practical methods of implementing these design features are possible and will be studied in future work.

Single-frequency and parallel multi-frequency LRS both consist of a single reflective layer with a specific effective mass spectra. In a series arrangement a sound wave will interact with multiple reflective layers each with there own effective mass frequency spectra. The result is a band gap of high attenuation. As the number of layers in the series increases, the magnitude of this attenuation approaches infinity, an effect that can be seen in Figure 12. It is important to note that the width of the band gap, when measured between its shoulders, does not change significantly with the number of layers in series. Figure 13 demonstrates that this bandwidth is actually determined by the stiffness of the coupling between the layers in series. It is evident from these results that the inter-layer coupling spring has to be of equal stiffness or softer than the resonator spring for a band gap to fully develop, and the softer the inter-layer coupling relative to the resonator, the wider the band width.

In a multilayer system there are multiple transmission loss dips outside the transmission band gap, and their number increases with the number of layers. Because these dips are the result of layer movement within the structure, increasing the damping factor significantly in the inter-layer coupling (z = 0:1-0:5) reduces the adverse effects of these transmission loss dips while having no visible effect on the sound reduction index performance within the band gap. It is clear in Figures 16 and 17 that combinations of series and parallel arrangements is what yielded systems with the largest amount of attenuation over large band widths. Therefore, by designing a LRS system with the following elements

- Several layers in series
- Offset spring stiffness between resonators and layers
- Damping between each layer

a stop band filter response is effectively created that has a wide enough bandwidth to be suitable for practical applications.

Experimental verification of the model predictions has yet to be conducted. Implementations of this series-parallel LRS are still at the design stage due to the complexity of the problem when considering practical constraints such as structural integrity, construction materials and cost.

Conclusion

Locally resonant structures (LRS) exhibit a significant improvement in sound attenuation over what can be achieved with a homogeneous material of similar mass, albeit in a limited frequency band width. The magnitude of improvement is strongly dependent on the ratio of host resonator mass, the damping of the resonator mass and the overall design of the LRS. To guide design, a modelling approach was developed based on systems of interconnected single-degree-of-freedom linear spring, mass and damper units, each of which represents one or a group of identical locally resonators.

Good correlation was obtained between modelling and various experimental methods showing that this modelling approach can be used to estimate both plane wave and diffuse field transmission through an LRS in the frequency domain of interest.

Different LRS system configurations were analysed using this modelling method for the purposes of widening the frequency band of improved transmission loss, increasing the magnitude of transmission loss in this band and reducing adverse effects outside this band.

Parallel systems were shown through modelling and testing to produce transmission loss gains over a much wider frequency range, with a reduction in the transmission loss dips at other frequencies by tapering the resonator mass distribution and damping across frequencies. Series resonators developed very high peak transmission loss, leading to transmission band gaps or stop bands. These complex designs need more detailed modelling and further experimental analysis to develop practical implementations.

The realization of LRS applications requires the use of modelling to optimise the geometry, material properties, performance and cost of the materials, as well as to understand the tolerances of these variables. The ideal final outcome would be a cost-effective method for the fabrication and implementation of a locally resonant meta-material that satisfies the qualities described previously.

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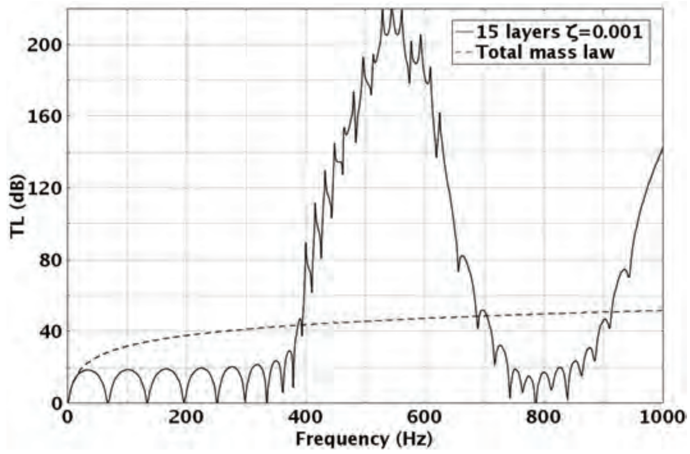

Figure 16: Comparison of transmission loss verses frequency of a 15 layer LRS. As the layer number increase so does the frequency of resonance of the resonators. Resonators are tuned at increments of 15Hz by changing the resonator spring stiffness. The inter-layer coupling spring stiffness remains constant throughout the system. A damping factor *z* of 0.001 is used.

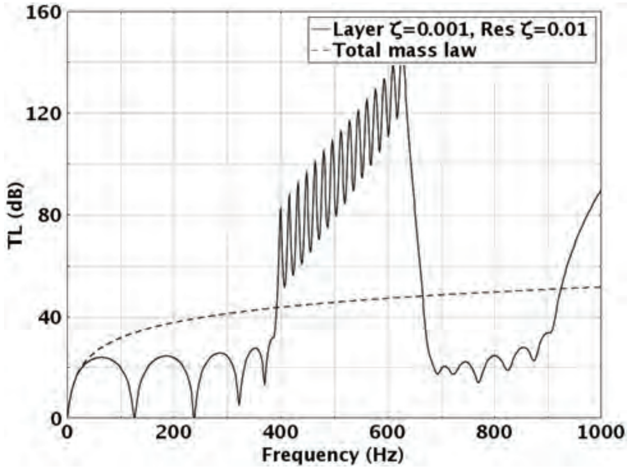

Figure 17: Comparison of transmission loss verses frequency of a 8 layer LRS with 15 parallel resonators on each layer. The parallel resonators on each layer are tuned at incremental resonant frequencies of 15Hz while the inter-layer coupling spring stiffness remains constant throughout the system. A damping factor *z* = 0:1 in the resonators is used.

A Review of the Adoption of International Vibration Standards in New Zealand

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Abstract

The New Zealand Standards authority has no current vibration standards relating to human exposure since withdrawing its adoption of the ISO 2631 series in 2005. Notwithstanding this, ISO 2631-2:1989 continues to be implemented by local government and other requiring authorities. This paper conducts a review of the major environmental vibration standards (i.e. human exposure and building damage) currently in use around the world, with a view to recommending a fresh suite of standards for adoption in New Zealand. The rating systems of a selection of human response standards are compared and used in a practical study of truck vibration.

Introduction

Environmental vibration assessments generally consider two factors: building damage risk and human response. These two facets are linked as peoples' sensitivity to vibration in buildings can be exacerbated by concern over damage to their building. The most prevalent sources of environmental vibration are associated with construction activities, blasting (for construction or quarrying purposes) and transportation (i.e. road and rail traffic).

This paper has been prompted by the fact that New Zealand has no current environmental vibration standards, and so local government and requiring authorities have been relying on the adoption of international standards.

A number of relevant international vibration standards are reviewed. The criteria and methodologies in these standards have been assessed in order to determine a current and practicable suite of standards recommended for adoption in New Zealand.

In addition, a comparison of human response standards is used in a practical comparison case study of truck vibration, to assess the equivalency between standards.

Standards Relating To Human Response To Vibration

Human response standards specify criteria in terms of comfort, quality of life and working efficiency for human receivers, using adverse comment as a test for limits of acceptability (a similar procedure as used for the determination of acceptable environmental noise limits (e.g. NZS 6802, 2008)).

During construction activities, the level of tolerance, particularly from residents and office tenants tends to relate to concern over possible building damage to their building structure. For this reason, building damage risk is usually the primary focus during construction assessments, but the human response element still needs to be managed.

Human response standards generally utilise a range of calculation methods, weightings and rating curves, therefore it is difficult to directly compare and contrast them based on the criteria alone. In the 'Equivalency Study' section below, a comparison of four Standards is undertaken whereby measured truck drive-by data is processed and rated by each standard, according to its criteria for residential sensitivity.

The standards in the following sections relate to measurement and evaluation of human response to vibration in buildings.

ISO 2631-2:1989

This International Standard ISO 2631-2:1989 "Evaluation of human exposure to whole-body vibration – Part 2: Continuous and shock-induced vibration in buildings (1 to 80 Hz)" was superceded in 2003 (refer next section) and the assessment criteria were removed from the 2003 version due to international criticism.

However, the 1989 version is still referenced (due to its assessment criteria) by a number of legislative and requiring authorities, including:

- Auckland City District Plan: Isthmus Section under sections 8.8.1.6, 8.8.3.9 and 8.8.10.9 Vibration in Buildings (Business Zones)
- Auckland City District Plan: Central Area Section under section 7.6.5.1 Vibration in buildings affecting comfort or amenity.
- Waitakere City District Plan under Living Environment Rules section 14.1, and Working Environment Rules section 10.1
- The New Zealand Transport Authority's Environmental Plan

The criteria contained in the Standard are multiplying factors of a frequency weighted base curve (expressed as both acceleration and velocity) which are designed to represent magnitudes of approximately equal human response with respect to human annoyance and/or complaints about interference with activities. The Standard states that compliance with vibration levels "at these values no adverse comments, sensations or complaints have been reported."

Annex A of the Standard contains a table of multiplying factors which are applied to the base curve to produce

Table 1: Ranges of multiplying factors applied to base curves for human response criteria in ISO 2631-2:1989. Source: (International Organisation for Standardisation, 1989).

Place	Time	Continuous or intermittent vibration	Transient vibration with several occurances per day
Critical working areas	Day Night	1	1
Residential	Day	2 to 4	30 to 90
	Night	1.4	1.4 to 20
Office	Day Night	4	60 to 128
Workshop	Day Night	8	90 to 128

"satisfactory magnitudes of building vibration with respect to human response" in different building types, as shown in Table 1:

Transient vibration is defined as "a rapid buildup to a peak, followed by a damped decay... it can also consist of several cycles of approximately the same amplitude, providing that the duration is short (i.e. less than 2 seconds)." The continuous and intermittent criteria cover other typical vibration sources including traffic and rail, which the Standard classifies as intermittent.

ISO 2631-2:2003

In April 2003, the 1989 version of ISO 2631-2 was superseded and the revised version was significantly different. It removed the guidance values (Table 1 above) citing international criticism and that the range of values measured in human response tests were too widespread for an International Standard.

Furthermore the New Zealand Standards authority withdrew its vibration standard NZS/ISO 2631 series (which was identical to ISO 2631:1989) in April 2005.

Arguably, this eliminates ISO 2631-2:1989 as a current and valid standard for determining the effects of vibration on building occupants, therefore despite its continued use and reference by local government and requiring authorities in New Zealand, it is considered inappropriate as a standalone reference document.

Note that the methodology in ISO 2631-2:2003 for measurement of building vibration is still valid, but conducting a vibration measurement according to this Standard is questionable if the subsequent assessment must then refer to a different standard for criteria.

AS 2670-2:1990

This Australian Standard is identical to ISO 2631-2:1989, and despite the ISO Standard having been superseded, it still holds a current status, as of June 2010. This may be an oversight by Standards Australia, or it could be an intentional endorsement of the old Standard over the new.

This Standard could potentially be adopted in New Zealand in order to, in essence, retain the ISO 2631-2:1989 criteria. However, this would ignore the deliberate action of the ISO retracting the Standard, and is not recommended.

ANSI S2.71-1983 (R 2006)

The American Standard ANSI S2.71-1983 (R 2006) "Guide to the Evaluation of Human Exposure to Vibration in Buildings" is similar to ISO 2631-2:1989 also. The weighting base curves match (with very few exceptions) the ISO curves, and the recommended criteria are very similar if slightly more stringent.

Unlike ISO 2631-2:1989, this Standard includes tentative modification factors for frequency of occurrence and event duration, which are used to adjust the criterion curves. However, the application of these factors is not clear. For the frequency of occurrence factor, for instance, there is no indication as to whether it applies to continuous or intermittent vibration and applying them to a vibration source with a large number of discrete events (e.g. traffic) may result in unachievably strict criteria.

The Standard was developed in 1983, but was revised in 2006; the introduction in the revision indicates that it merely contains "provisional recommendations on satisfactory magnitudes" which are a "compromise between the available data and the need for recommendations which are simple and suitable for general application." It would seem therefore to be aligning with ISO 2631-2:2003 in retreating somewhat from the earlier version without, in this case, actually superseding it.

DIN 4150-2:1999

The German Standard DIN 4150-2:1999 "Structural Vibration – Part 2: Human exposure to vibration in buildings" is from the same suite of Standards as DIN 4150-3:1999 which assesses building damage (refer section on DIN 4150-3:1999 below).

The Standard uses unique descriptors for vibration velocity data, which is band limited to 1-80Hz, weighted and normalised according to the specifications in another German Standard (DIN 45669-1, 1995) to produce values in terms of KB(t). This parameter is time-averaged to produce $KB_t(t)$ values which are then further averaged in 30 second blocks to produce the KB_{FTm} rating value. The maximum KB_{Fmax} signal is also used as a rating value.

The guideline values for human exposure in dwellings (A_u, A_o and A_r) are obtained through the use of a flow-diagram which contains tests for the calculated KB_{FTm} and KB_{Fmax} values. These guideline values are further modified according to whether they are short-term, generated by road traffic, rail traffic or construction work.

In general, the Standard appears to be comprehensive but very complicated, and unfamiliar in the context of New Zealand experience with vibration standards. The calculation of the KB_{FTm} from measured vibration waveform data requires statistical programming as it cannot be calculated using 'standard' tools such as Microsoft Excel.

It is understood that there are special software packages available to undertake the calculations, but these are not available in New Zealand. It is therefore considered that this Standard is too complex to be easily adopted in New Zealand.

NS 8176E:2005

The Norwegian Standard NS 8176.E:2005 "Vibration and shock – Measurement of vibration in buildings from landbased transport and guidance to evaluation of its effects on human beings" specifically addresses vibration effects from rail and road traffic. It purports to have been developed to fill a requirement for a transport-specific vibration standard, stating in its introduction that the recommended limits in ISO 2631-2 – presumably the 1989 version – "are not adequate for vibration from transport".

It is referenced in the NZTA Environmental Plan and has been successfully adopted in a number of large Auckland roading projects.

The Standard outlines the requirements for measuring equipment, and outlines a measurement procedure which requires a minimum of 15 heavy vehicle 'passings' (i.e. train, tram or heavy road vehicles (gross weight greater than 3500 kg)). The maximum velocity values v_i of each of these passings is recorded with a slow time-weighting in 1/3 octaves between 0.5Hz and 160 Hz. There is provision for acceleration values also, however the application is identical so for the purposes of this description, velocity will be used.

The values for each pass are weighted according to the W_m weighting curve (ISO 2631-2, 2003), and the mean and standard deviation of the 15 passings is calculated. The mean and standard deviation are then combined (assuming a log-normal distribution) to provide a statistical maximum value $v_{w,95}$. Specification of the statistical maximum value implies that there is about 5% probability for a randomly selected passing vehicle to give a higher vibration value. Note that this is of a similar nature to the percentile levels adopted in NZ for noise but would be expressed as an L_5 i.e. the percentile is inverted.

Appendix A of the Standard contains exposure-effect curves for annoyance and disturbance which look at the relationship between measured $v_{w,95}$ levels and percentage of people affected. This is a very useful resource which can assist in predicting and quantifying vibration effects. It is similar to Shultz curves (Shultz, 1978) for noise but may not have been as thoroughly tested to determine the veracity of the curves.

Appendix B of the Standard gives guidance classification of dwellings in relation to their sensitivity to vibration. The four classes of dwelling and corresponding statistical maximum values are as follows:

"**B.3 Guidance vibration classes**

The statistical maximum value for weighted velocity (or acceleration) shall not exceed the limits specified in Table B.1 [refer Table 2]

- B.3.1 Class A: Corresponds to very good vibration conditions, where people will only perceive vibration as an exception.
- *NOTE Persons in Class A dwellings will normally not be expected to notice vibration*
- B.3.2 Class B: Corresponds to relatively good vibration conditions.
- *NOTE Persons in Class B dwellings can be expected to be disturbed by vibration to some extent*
- B.3.3 Class C: Corresponds to the recommended limit value for vibration in new residential buildings and in connection with the planning and building of new transport infrastructures.
- *NOTE About 15% of the affected persons in Class C dwellings can be expected to be disturbed by vibration.*
- B.3.4 Class D: Corresponds to vibration conditions that ought to be achieved in existing residential buildings.
- *NOTE About 25% of persons can be expected to be disturbed by vibration in class D dwellings. An attempt should be made to meet class C requirements, but Class D can be used when the cost-benefit considerations make it unreasonable to require class C."*

Class C relates to about 15% of receivers being disturbed by vibration, and Class D relates to about 25%. These recommendations are based on the large scale exposure-effect studies in Appendix A of the Standard. The studies were conducted in fourteen areas of Norway, with residents' reactions to vibration from road traffic, railways, underground and trams.

Scandinavian countries are generally recognised for maintaining a high living-standard, so it is considered that the survey outcomes may be relatively conservative in terms of residents' responses to environmental vibration effects.

BS 6472-1:2008

The British Standard BS 6472-1:2008 "Guide to evaluation of human exposure to vibration in buildings – Part 1: Vibration sources other than blasting" is not widely adopted in New Zealand, but has advantages in the assessment of operational vibration effects due to its dose-response metric Vibration Dose Value (VDV).

VDV is calculated from the frequency-weighted vibration acceleration (weighted according to the W_b or W_d curves for vertical and horizontal acceleration respectively), which is integrated over the day or night time period. Table 1 of the Standard (refer

Table 2: Guidance classification of dwellings with the upper limits for the statistical maximum value for weighted velocity $v_{w,95}$ or acceleration $a_{w,95}$ after NS 8176.E:2005 Source: (Norsk Standard, 2005)

Type of vibration value	Class A	Class B	Class C	Class D
Statistical maximum value for weighted velocity $v_{w,95}$ (mm/s)	0.1	0.15	0.3	0.6
Statistical maximum value for weighted acceleration $a_{w,95}$ (mm/s^2)	3.6	5.4	11	21

Table 3: Vibration dose value ranges which might result in various probabilities of adverse comment within residential buildings, after BS 6472-1:2008. Source: (British Standards, 2008)

Place and time	Low probability of adverse comment $ms^{-1.75}$	Adverse comment possible $ms^{-1.75}$	Adverse comment probable $ms^{-1.75}$
Residential buildings 16 h day	0.2 to 0.4	0.4 to 0.8	0.8 to 1.6
Residential buildings 8 h night	0.1 to 0.2	0.2 to 0.4	0.4 to 0.8

Table 3) contains VDV ranges which may result in adverse comment in residential buildings, as shown in Table 3.

There is however some controversy surrounding the use and usability of VDV. For continuous vibration (such as motorway traffic), the "estimated VDV" metric eVDV is recommended in place of VDV. The correlation between VDV and eVDV for the same data set is variable, and relies heavily on the event period used in the calculation.

The Institute of Acoustics (UK) has undertaken comparison studies of the two parameters (Greer et.al., 2005), and concludes that eVDV is generally a reliable estimate of VDV provided the crest factors for transient signals are calculated correctly, and that the constant 1.4 in the eVDV equation is not necessarily correct and should be derived for a given signal (e.g. a value of 1.11 should be used for a sinusoidal signal).

This Standard is not known to have been adopted in New Zealand.BS 6472-2:2008

The British Standard BS 6472-2:2008 "Guide to evaluation of human exposure to vibration in buildings – Part 2: Blast-induced vibration" contains PPV criteria for human response to blasting, as well as prediction methods utilising scaled distance. It is not widely adopted in New Zealand.

The recommended criteria are shown in Table 4.

When compared with ISO 2631-2:1989 and NS 8176.E:2005, the recommended criteria in both BS 6472 Parts 1 and 2 are very lenient. For instance, it can be seen that a vibration level of 10 mm/s PPV (which is double the DIN 4150-3:1999 standard for residential building damage risk between 1-10Hz) is considered satisfactory in terms of human response. This is possibly due to the limitation of no more than 3 blasting events per day, however the allowable magnitude of each event is considered to be the primary consideration for blasting because of the potential for startle effect and disturbance with each blast.

Similarly, the British Standards criteria for building damage (BS 7385-1, 1990 and BS 7385-2:1993) are significantly less stringent than those in the commonly adopted DIN 4150-3:1999, so it appears that British Standards for environmental vibration in general are comparatively lenient.

This Standard is not known to have been adopted in New Zealand, but is referenced by Australian Standard AS 2187.2:2006 "Explosives – Storage and use, Part 2: Use of explosives".

It is possible that the lenient approach taken by the British Standards is defendable through underlying research, and the other standards commonly applied in NZ are overly stringent. However, immediate adoption of such lenient criteria into a large project, for instance, may be at odds with society's expected control of vibration effects, and the marked relaxation in vibration controls would be difficult to justify.

It is recommended that further research and investigative use of the British Standards are undertaken to gain experience in the methodologies therein. This will allow an informed assessment of their benefits (or otherwise) over the proposed suite of standards.

BS 5228-2:2009

The British Standard BS 5228-2:2009 "Code of practice for noise and vibration control on construction and open sites – Part 2: Vibration" is a comprehensive and voluminous standard covering many aspects of prediction, measurement, assessment and control of vibration from construction works.

In terms of vibration criteria this standard contains references to, and reiterates the criteria from BS 6472 (human response) and BS 7385 (building damage).

However Annex B of the Standard addresses human response to construction vibration and suggests that BS 6472 may not be appropriate. It states:

> *"BS 6472, as stated, provides guidance on human response to vibration in buildings. Whilst the assessment of the response to vibration in BS 6472 is based on the VDV and weighted acceleration, for construction it is considered more appropriate to provide guidance in terms of the PPV, since this parameter is likely to be more routinely measured based on the more usual concern over potential building damage. Furthermore, since many of the empirical vibration predictors yield a result in terms of PPV, it is necessary to understand what the*

Table 4: Maximum satisfactory magnitudes of vibration with respect to human response for up to three blast vibration events per day. Source: (British Standards, 2008)

Place	Time	Satisfactory magnitude PPV mm/s
Residential	Day	6.0 to 10.0
	Night	2.0
	Other times	4.5
Offices	Anytime	14.0
Workshops	Anytime	14.0

Table 5: Guidance on the effects of vibration levels. Source: (British Standards, 2009)

Vibration level (PPV)	Effect
0.14 mm/s	Vibration might just be perceptible in the most sensitive situations for most vibration frequencies associated with construction. At lower frequencies, people are less sensitive to vibration
0.3 mm/s	Vibration might just be perceptible in residential environments
1.0 mm/s	It is likely that vibration of this level in residential environments will cause complaint, but can be tolerated if prior warning and explanation has been given to residents
10 mm/s	Vibration is likely to be intolerable for any more than a very brief exposure to this level

consequences might be of any predicted levels in terms of human perception and disturbance. Some guidance is given in Table B.1 [refer Table 5]

The use of PPV is a pragmatic approach to construction vibration assessment and the criteria in Table B.1 are considered suitable for assessment of human response to construction vibration effects. Furthermore, the criteria have a reasonable correlation with DIN 4150-3:1999 in terms of the level of concern expected with regard to building damage.

It is noted that the primary issue relating to construction vibration is damage to buildings and although people may become concerned at levels above 1 mm/s PPV, in the context of a project this effect can be managed through communication with concerned residents and other mitigation strategies outlined in they project's construction management plan.

Discussion - Human Response Standards

To summarise the vibration standards for human response, the ISO 2631-2:1989 Standard which is traditionally applied in New Zealand is no longer considered suitable, as it was replaced in 2003 by an informative only standard. The NZ adoption of this Standard (NZS/ISO 2631-2:1989) was therefore withdrawn by Standards New Zealand.

A comprehensive review has been undertaken of relevant international standards from the UK, Europe, United States and Australia. The US and Australian Standards are aligned with the ISO 2631-2:1989 and may therefore be deemed inappropriate by association.

The Norwegian Standard NS 8176.E:2005 is an attractive alternative for use in traffic and rail assessments, as it contains a statistical approach to vibration events and community response relationships, has a history of successful implementation in major roading projects, and is referenced by the NZTA Environmental Plan. Furthermore, no known adverse effects have been reported for vibration on projects for which this Standard was applied.

However because it addresses only transportation vibration, another standard is needed to assess the effects of other vibration sources. Blasting and Construction are considered to be the other relevant vibration-inducing activities relating to environmental works, and it is considered that the human response criteria for both these operations are addressed by the British Standard BS 5228-2:2009, Appendix B.

The British Standard BS 6472-1:2008 contains an attractive methodology involving the use of Vibration Dose Value (VDV), which considers the period of exposure to vibration as well as the vibration level. However, the criteria in this Standard are considered to be too lenient and further investigation would be required to rationalise this before it could be considered for adoption in New Zealand.

Equivalency Study

To compare and contrast the human response standards, a dataset of truck drive-by measurements were assessed against the standards contained in the above section – NS 8176.E:2005, BS 6472-1:2008, ANSI S2.71-1983 (R 2006) and ISO 2631-2:1989. The Australian Standard AS 2670-2:1990 is assessed by proxy because it is identical to the ISO standard. The German Standard DIN 4150-2:1999 is considered too complicated to be easily adopted for use in New Zealand.

The purpose of the comparison is to investigate how each standard rates the same vibration dataset, and shows the equivalency of their criteria with respect to one another.

To ensure a clear vibration signal, the measurement location was selected adjacent to a road with high heavy vehicle numbers and a dilapidated surface – the entrance to a quarry in South Auckland.

The measurements were undertaken on 13th January 2010 using an Instantel Minimate Plus vibration meter with tri-axial geophone. The meter was positioned 25 metres from the closest lane of the two lane road (one lane in each direction). The geophone was fixed to the ground with ground-spikes and weighted with a sandbag.

The general geology of the site was provided by Beca Limited. The ground comprised medium-dense gravel, clayey silt and stiff to very stiff silty clay.

Fifteen truck passes were measured in accordance with the NS 8176.E:2005 Standard, as well as an ambient measurement i.e. with no traffic on the road. The vibration levels of the truck passes were considerably higher than the ambient measurement.

This comparison confirms that the British Standard BS 6472-1:2008 is significantly more lenient than the other three standards i.e. it considers that the measured data readily complies with the night-time residential criterion, whereas the other three standards indicate some annoyance and/or exceedance of their night-time residential criteria.

The subjective impression during the measurements was that vibration from the truck passes were detectable, but would not be considered excessive in any way.

Table 6: Comparison of the assessment outcomes of four human response standards.

	NS 8176.E :2005	BS 6472-1 :2008	ISO 2631-2 :1989	ANSI S2.71-1983 (R 2006)	
	$v_{w,95}$ mm/s	Class	VDV ($ms^{-1.75}$)	Multiplying factor	Multiplying factor
Vibration Level	0.18	C	0.016	4	4
Assess-ment	Complies with criterion for existing dwell-ings. Approx. 12% may be moder-ately/highly annoyed		Readily com-plies with residential night-time criterion. Low probability of adverse comment	Complies with residential daytime criterion, but exceeds night-time criterion	

It is noted that the ISO and ANSI standards do not contain an averaging method for multiple vibration events, so the rating is based on the worst truck pass. However, all but two of the 15 truck passes would comply with the daytime criterion but not the night-time, so the assessment in Table 6 for these standards generally represents the entire dataset.

Standards Relating To Building Damage Risk

The following standards relate to measurement and evaluation of the effects of ground-borne vibration on building structures.

DIN 4150-3:1999

The use of German Standard DIN 4150-3 "Structural vibration – Part 3: Effects of vibration on structures" is widespread in New Zealand and it has a history of successful implementation in projects involving construction activities and/or blasting. Two versions of the standard – the current 1991 version, and the earlier 1986 version – are referenced in several local government and other requiring authorities, as follows:

The earlier 1986 version of the standard is referenced in:

- Auckland City District Plan: Isthmus Section under section 8.8.2.7 Noise and Vibration arising from Blasting
- Auckland City District Plan: Central Area Section under section 7.6.5.2 Noise and vibration from explosive blasting of pile driving.
- Waitakere City District Plan under section 13.1(c) regarding blasting in quarry areas
- The NZTA Environmental Plan (see Section 3.4.1 below)

The 1999 version is referenced in:

- Auckland City District Plan: Hauraki Gulf Islands Section (Proposed 2006) under section 4.6.3 Noise and vibration from blasting or pile driving for construction activities.

The Standard adopts the Peak Particle Velocity (PPV) metric and gives guideline values which, "when complied with, will not result in damage that will have an adverse effect on the structure's serviceability."

The guideline values are different depending on the vibration source, and are separated on the basis of short-term and long-term vibration. The standard defines short-term vibration as "vibration which does not occur often enough to cause structural fatigue and which does not produce resonance in the structure being evaluated". Long-term vibration is defined as all other types of vibration not covered by the definition of short-term vibration.

In general, the short-term vibration definition would be applied to activities which follow the form of a single shock followed by a period of rest such as blasting, drop hammer pile-driving (i.e. non-vibratory), dynamic consolidation etc. All other construction activities would be considered long-term. Traffic may be categorised as either, depending on the nature of the vibration i.e. vibration from consistent (but rough) road surface may be long-term, whereas a road with a bump in the pavement may generate a short-term vibration event.

The criteria for short-term and long-term vibration activities, as received by different building types, are summarised in Table 7. This table is a combination of Tables 1 and 3 of the Standard.

The standard also contains criteria for buried pipework of different materials and the effects of vibration on floor serviceability, as well as guidelines for measurement of vibration in buildings i.e. placement and orientation of the transducers. It should be noted that these criteria are designed to avoid superficial damage to buildings i.e. cracking in plaster. Significantly greater limits would be applied for damage to structural foundations.

Table 7: Summary of Building Damage criteria in DIN 4150-3:1999. Source: (Deutsch Institut für Normung, 1999)

Type of Struc-ture	Short-term vibration				Long-term vibration
	PPV at the foundation at a frequency of:			PPV at horizontal plane of highest floor	PPV at horizontal plane of highest floor
	1 - 10Hz	10 -50Hz	50 -100Hz	At any frequency	At any fre-quency
Commercial, Industrial	20	20-40	40-50	40	10
Residential, School	5	5-15	15-20	15	5
Historic, Sensitive	3	3-8	8-10	8	2.5

To address this range in the effects on buildings, it is considered appropriate to adopt a statistical analysis methodology for assessing damage risk due to vibration. There is precedence for this approach in Section 8.8.2.7e of the Auckland City District Plan – Isthmus Section for blasting, and Section A10.3.1 of the Whangarei District Plan, although actual criteria of each differ slightly.

Table 8 proposes a statistical analysis methodology for short-term and long-term vibration, based on the limits contained in DIN 4150-3:1999.

BS 7385-1:1990 – ISO 4866:1990(E)

The British Standard BS 7385-1:1990 “Evaluation and measurement for vibration in buildings – Part 1. Guide for measurement of vibration and evaluation of their effects on buildings” is identical to ISO 4866:1990(E) “Mechanical vibration and shock – Vibration of buildings – Guidelines for the measurement of vibration and evaluation of their effects on buildings”, therefore it adopts the ISO standard and reproduces it in full (hence the two standards in the title).

ISO 4866:1990(E) establishes the basic principles for carrying out vibration measurements and processing data. In conjunction with BS 7385-2:1993, its scope is similar to that of DIN 4150-3:1999, but it addresses several aspects in greater detail than the German Standard.

The Standard contains a formula (rather than guidelines) for establishing whether the source is continuous (long-term) or transient (short-term), addresses the influence of soil attenuation, the structural response of different building types for various sources, measurement and reporting procedures, and a comprehensive building classification.

Another useful section contains a description of building damage categories, as follows:

“Cosmetic
The formation of hairline cracks on drywall surfaces, or the growth of existing cracks in plaster or drywall surfaces; in addition, the formation of hairline cracks in mortar joints of brick/concrete block construction
Minor
The formation of large cracks or loosening and falling of plaster or drywall surfaces, or cracks through bricks/concrete blocks
Major
Damage to structural elements of the building, cracks in support columns, loosening of joints, splaying of masonry cracks etc.”
This Standard is not known to be adopted in New Zealand.

BS 7385-2:1993

The second part of the BS 7385 series – BS 7385-2:1993 “Evaluation and measurement for vibration in buildings – Part 2. Guide to damage levels from groundborne vibration” sets vibration limits based on an extensive review of international case histories. The introduction states that despite the large number of UK case studies involved in the review, “very few cases of vibration-induced damage were found”.

The criteria, also in PPV, are contained in Table 1 of the Standard, refer Table 9

These criteria relate predominantly to transient vibration, and the standard suggests that the criteria “may need to be reduced by up to 50%”, especially at low frequencies. Notwithstanding this, the criteria are 3 to 10 times higher (i.e. less stringent) than those in DIN 4150-3:1999.

Note that there is no consideration for historic or sensitive structures in the above table. This is addressed in Section 7.5.2 of the Standard which states:

"7.5.2 Important buildings

Important buildings which are difficult to repair may require special consideration on a case-by-case basis. A building of historical value should not (unless it is structurally unsound) be assumed to be more sensitive."

Note that 'peak component particle velocity' refers to the maximum PPV of the three orthogonal axes (longitudinal, transverse or vertical), also known as peak vector sum (PVS).

This approach to historic structures is quite different to that of the DIN 4150-3:1999 Standard which is less definitive with its definition of such buildings and more stringent in its criteria.

Discussion – Building Damage Standards

To summarise the building damage Standards, the British building damage Standards (BS 7385-1:1990 and BS 7385-2:1993) are more comprehensive and detailed in their scope than DIN 4150-3:1999, and can be considered 'current' (having received endorsement from the recent BS 5228-2:2009 standard.

However, they are significantly less stringent than the German Standard DIN 4150-3:1999 and there is concern that their criteria may be too high, and may allow damage to building structures.

The German Standard has a record of successful implementation in a number of major Auckland projects. Its criteria are more conservative than BS 7385-2:1993, but has not been found to be overly restrictive. It therefore affords adequate protection for building structures, and addresses the concerns of building occupants by setting a reasonable limit.

The adoption of a statistical approach to the implementation of DIN 4150-3:1999 is considered pragmatic, and promotes comprehensive monitoring and assessment of vibration activities such as construction works.

Australia does not have a National Standard for vibration building damage however it is understood that DIN 4150-3:1999 is widely adopted. Similarly, there is no American National Standard addressing building damage from vibration.

The DIN 4150-3:1999 standard is therefore considered most suitable to assess and quantify the risk of building damage from vibration.

Summary of Standards

A number of international environmental vibration standards have been reviewed with a view to informing the adoption of a relevant suite of standards to address environmental vibration effects relating to building damage and human response.

Due consideration has been given to those standards which have a successful history of implementation in New Zealand, and are recognised by authorities such as Auckland City Council, Auckland Regional Council, Waitakere City Council and NZ Transport Agency.

In lieu of the superceded ISO 2631-

Table 8: Long-term vibration criteria after DIN 4150-3:1999.

Vibration Duration (as defined in DIN 4150-3:1999)	Statistical Analysis Methodology
Short-term (e.g. blasting, drop-hammer piling, dynamic consolidation)	Activities shall be conducted so that 95 % of a measured activity on the foundation of any residential building shall produce PPVs not exceeding the limits specified in Table 3 of DIN 4150-3:1999 and 100 % of the measured events shall not exceed twice the limits specified in the same table.
Long-term (e.g. most other vibration sources)	For the measurement of long-term activities, PPVs shall be recorded at one second intervals. The total assessment period shall be sufficient to ensure a representative sample of the activity is recorded.

2:1989 Standard, traditionally adopted in New Zealand for assessing human response, the following Standards are recommended:

- Norwegian Standard NS 8176.E:2005 for human response to traffic and rail vibration.
- British Standard BS 5228:2009 for human response to construction vibration

NS 8176.E:2005 has been successfully implemented in a number of major Auckland projects, and aligns well with the rating criteria of ISO 2631-2:1989. Furthermore the straightforward calculation procedure and data relating to population annoyance are beneficial. It is more stringent than BS 6472-1:2008 but has shown to be practicable in New Zealand applications.

BS 5228-2:2009 gives guidance values for human response in terms of Peak Particle Velocity (PPV) which is directly applicable to construction and blasting operations.

For the assessment of building damage risk, the following Standard is recommended:

- German Standard DIN 4150-3:1999 for building damage risk relating to all vibration sources

DIN 4150-3:1999 is widely recognised and successfully implemented in New Zealand.

These recommendations provide a robust and, for the most part, familiar approach to assessment of environmental vibration.

It must be said that the suite of British Standards is an attractive option, as it is comprehensive and offers a complete range of vibration assessment tools with robust methodologies. However, in the context of New Zealand's implementation of vibration standards, its criteria are considered too lenient. This is not to say that the criteria are wrong, but an abrupt change from the current standards to less stringent criteria may cause alarm and consternation over the possible effects.

It is recommended that further investigations of the British Standards are undertaken in a New Zealand context, in an attempt to rationalise and qualify the differences between them and other relevant Standards, such as those assessed herein.

References

ANSI S2.71-1983 (R 2006) "Guide to the Evaluation of Human Exposure to Vibration in Buildings", American National Standards Institute, 2006

AS 2187.2:2006 "Explosives – Storage and use, Part 2: Use of explosives", Standards Australia, 2006

AS 2670-2:1990 "Evaluation of human exposure to whole-body vibration – Part 2: Continuous and shock-induced vibration in buildings (1 to 80 Hz)", Standards Australia, 1990

BS 5228-2:2009 "Code of practice for noise and vibration control on construction and open sites – Part 2: Vibration", British Standards Institute, 2009

BS 6472-1:2008 "Guide to evaluation of human exposure to vibration in buildings – Part 1: Vibration sources other than blasting", British Standards Institute, 2008

BS 6472-2:2008 "Guide to evaluation of human exposure to vibration in buildings – Part 2: Blast-induced vibration", British Standards Institute, 2008

BS 7385-1:1990"Evaluation and measurement for vibration in buildings – Part 1. Guide for measurement of vibration and evaluation of their effects on buildings", British Standards Institute, 2008

BS 7385-2:1993 "Evaluation and measurement for vibration in buildings – Part 2. Guide to damage levels from groundborne vibration", British Standards Institute, 1993

DIN 4150-2:1999 "Structural vibration – Part 2: Human exposure to vibration in buildings", Deutsches Institute für Normung, 1999

DIN 4150-3:1999 "Structural vibration – Part 3: Effects of vibration on structures", Deutsches Institute für Normung, 1999

DIN 45669-1:1995 "Mechanical vibration and shock measurement – Part 1: Measuring equipment", Deutsches Institute für Normung, 1995

Greer, R., Thornley-Taylor, R. et al. "ANC round robin VDV measurement exercise analysis of eVDV data", Acoustics Bulletin, Mar/April 2005

ISO 2631-2:1989 "Evaluation of human exposure to whole-body vibration – Part 2: Continuous and shock-induced vibration in buildings (1 to 80 Hz)", International Organisation for Standardisation, 1989

NS 8175:2005 "Sound conditions in buildings – Sound classes for various types of buildings", Standards Norway, 2005

NS 8176.E:2005 "Vibration and shock – Measurement of vibration in buildings from landbased transport and guidance to evaluation of its effects on human beings", Standards Norway, 2005

NZS 6802:2008 "Acoustics – Environmental Noise", Standards New Zealand, 2008

Schultz, T. J. "Synthesis of social surveys on noise annoy-ance", Journal of the Acoustical Society of America, 64, pp 377-405, 1978

Table 9: Transient vibration guide values for cosmetic damage in BS 7385-2:1993. Source: (British Standards, 1993)

Line	Type of building	Peak component particle velocity in frequency range of predominant pulse	
		4 – 15Hz	15Hz and above
1	Reinforced or framed structures, Industrial and heavy commercial buildings	50mm/s at 4Hz and above	
2	Unreinforced or light framed struc-tures, Residential or light commercial type buildings	15mm/s at 4Hz increasing to 20mm/s at 5Hz	20mm/s at 15Hz increasing to 50mm/s at 40Hz and above

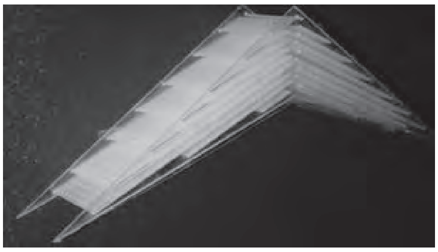
One-Way Sound

A group of researchers has followed up earlier work to show that nonlinear sound-bending materials are physically possible.

In a recent paper (Phys. Rev. Lett. 106, 164101, 2011) , a group from Italy and Singapore have found solutions for non-linear sound transmission that correspond to physically realisable properties. The solutions are for layered materials and the transmission process is formally called asymmetric wave propagation.

The calculations concerned a layered nonlinear, nonmirror-symmetric model. The soutions show that waves with the same frequency and incident amplitude impinging from left and right directions have very different transmission coefficients. This effect arises even for the simplest case of two nonlinear layers and is associated with the shift of nonlinear resonances. This idea has been referred to as a kind of one-way mirror for sound.

To direct sound waves, the researchers propose alternating layers of linear and strongly nonlinear materials asymmetrically. The group of physicists demonstrate mathematically that such an acoustic meta-material has the potential to be built one day. This was not previously known.



Acoustic cloaking device from Duke University. *Image Credit: BBC.*

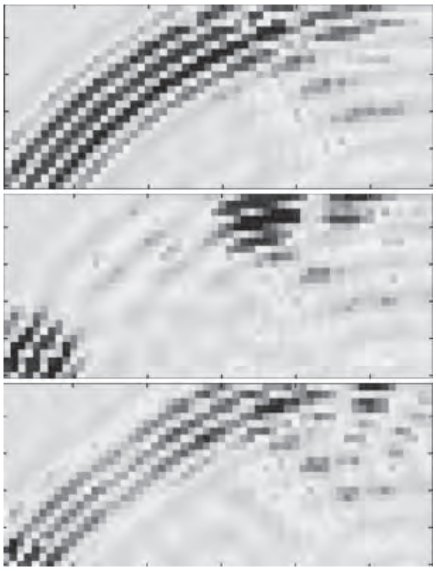
Adapted from: http://news.discovery.com/tech/one-way-sound-walls-acoustics-110505.

Acoustic Cloaking

Meanwhile, a group at Duke University in the USA have been experimenting with layered plastic meta-materials. Steve Cummer published early ideas about acoustic cloaking and has since led a team constructing physical cloaking devices.

So-called ‘invisibility cloaks’ usually work by bending light or microwaves in ways that would not normally be possible, through the use of layered meta-materials. The device at Duke University uses many of the same principles for sound waves.

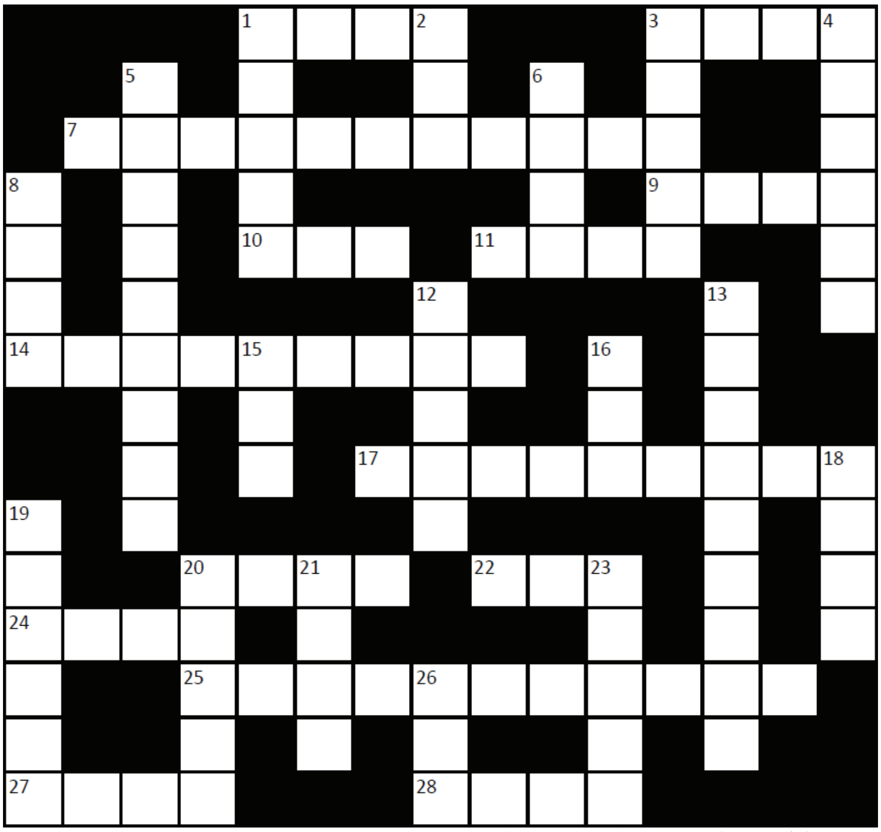
The device consists of stacked sheets of plastic peppered with holes whose arrangement and size redirect the incident sound waves. The geometry resonates at frequencies that either absorb or reflect sound waves, so it both blocks and contains them — anything underneath the stack of sheets does not experience the incident sound, and reflections can not be used to locate something coated with the stack. This works in air for audible frequencies between one and four kilohertz.



Acoustic reflections from a surface (top), from an object on it (middle) and from a cloaked object (bottom). *Image Credit: BBC.*

Source: BBC News

Acoustic Crossword #2



CLUES ACROSS

1. A drummer does this for adjustable valves (4)
3. It's said in relief softly, before cut (4)
7. Never barter about live description (11)
9. Bass, for example, split without first tap (4)
10. Records backwards measured quantity (3)
11. Church after mother, at speed (4)
14. See 5 Down
17. A way inside supports for reference material (9)
20. Wave of holy transgression to the east (4)
22. Remove the French and note from humble drone (3)
24. Mean free, for example, inside spa that's warm (4)
25. Rat inside gin, note I rearranged for hearing process (11)
27. Thrash out the rhythm (4)
28. A type of field close by (4)

CLUES DOWN

1. They offer no significant noise attenuation, in terms of metre established (5)
2. Look, it sounds like a weighting, for instance (3)
3. Black note (5)
4. See 15 Down

5Dn, 14Ac, 20Dn. Ringing in the ears, which can happen during an earthquake (9,9,5)

6. Mixed traffic flow statistics (4)
8. Silent note about good man (4)
12. Loud stringed instrument... or woodwind? (5)
13. It is felt by six, with bulkhead portion (9)

15Dn - 4Dn. Underdog (3-6)

16. Initially, a quite radical direction for acoustic treatment (3)
18. Aquatic mammal blocks a noise leak (4)
19. Terrific! Beranek's first description of the Christchurch Town Hall's acoustics (6)
20. See 5 Down
21. It isn't a direction. Write it down (4)
23. A mixed brass instrument. Without Ben, it's a source of vibration (5)
26. Impulsive noise source in big underwear (3)

Crossword solution in the next issue.

Crossword submitted by:

Calm Jokes with help from A Shy Mallard

2011

24-28 July, London, UK. 10th International Congress ICBEN (International Commission on Biological Effects of Noise).
http://www.icben2011.org/

27 - 31 August, Florence, Italy. International Conference of the International Speech Communication Association.
http://www.interspeech2011.org

04 - 07 September, Osaka, Japan. Internoise 2011
http://www.internoise2011.com

14 - 15 September, Glasgow, Scotland. Acoustics IOA Conference
http://www.ioa.org.uk

26 - 28 October, Cáceres, Spain. TECNIACÚSTICA'11.
http://sea-acustica.es

31 October - 04 November, San Diego, Cal., USA. 162nd Meeting of the Acoustical Society of America.
http://asa.aip.org/meetings.html

16 - 18 November, Cardiff, Wales, UK. Reproduced Sound
http://www.ioa.org.uk

2012

20 -25 March, Kyoto, Japan. IEEE International Conference on Acoustics, Speech, and Signal Processing.
http:// www.icassp2012.com

13 - 18 May, Hong Kong, China. Joint meeting of the 183rd meeting of the Acoustical Society of America, 8th meeting of the Acoustical Society of China, 11th meeting of Western Pacific Acoustical Conference and Hong Kong Institute of Acoustics.
http://acoustics2012hk.org

02 - 06 July, Edinburgh, UK. 11th European Congress on Underwater Acoustics
http://www.acua2012.com

19 - 22 August, New York, N.Y., USA. Internoise 2012.
http://www.internoise2012.com.

Crossword Solution #1

Sound Snippets: Military Matters

Faster than a speeding bullet?

U.S. Army forces in Afghanistan are now receiving more than 13,000 gunshot detection systems. These systems use acoustic sensors to locate the source of gunfire and are designed for use by individual soldiers.

"We're really trying to ensure that every Soldier is protected," said Brig. Gen. Peter Fuller, Program Executive Officer Soldier.

The Individual Gunshot Detector, or IGD is made by QinetiQ North America (a British defence technology company) and consists of four small acoustic sensors and a small display screen attached to body armor that shows the distance and direction of incoming fire.

The sensors are approximately the size of a deck of cards, and use the sound generated by supersonic gunfire to alert Soldiers to the location and distance to the hostile activity.

"When you get fired on, instead of trying to figure everything out, you will have technology to assist you in knowing what happened and where the shot was coming from," said Lt. Col. Chris Schneider, product manager for Soldier Maneuver Sensors.

The entire IGD system, procured by PEO Soldier and the Army's Rapid Equipping Force, weighs less than one kilogram.

The idea is to strategically disperse the systems throughout small, infantry units to get maximum protective coverage for platoons, squads and other units on the move. In the future, the Army plans to integrate this technology with its Land Warrior and Nett Warrior systems. These are network-situational-awareness systems for dismounted units, complete with a helmet-mounted display screen that uses GPS digital-mapping-display technology, Fuller said.

Adapted from: US Amry News Service 2011.

Image showing the IGD System worn by a soldier. *Credit: PEO Soldier.*

Squirrel Detector Sought

The United States Air Force is on the hunt for a new detector. One that is rugged— able to withstand extreme temperatures, blistering 80-kph winds and barren desert conditions. But it is not meant to detect dangerous insurgents, powerful explosives or undercover spies. Its target is the Mojave ground squirrel.

The isolation and secure nature of military zones in the USA means that they serve as habitats for more than 300 endagered and threatened plant and animal species, including ground squirrels.

Ground squirrel. *Image: Public Domain.*

Under the US Endangered Species Act, the Department of Defense is required to maintain and protect the home of these species.

To reduce the cost of manual monitoring of large areas of defence land, the US Air Force is looking for some help from acoustic technology, as the service has announced in a recent call for research proposals.

The challenge is to create a sensor array and recording system that can detect, distinguish and store, many hours of animal sounds. These sounds would need to be matched to individual species and the sources of the sound pinpointed in space.

Databases of digital "acoustic fingerprints" are already available for many birds and the proposed research seeks to extend these databases to include the many small mammals living in military zones.

It is envisioned that the data could be used to create a species-specific activity map for a region that could also show changes over the course of a year and help military planners make appropriate choices about scheduling activities in these zones.

This information would also be of use to scientists wanting to track animal migrations and local movements and lead to more efficient animal protection and management.

Adapted from: Lena Groeger, Wired Magazine, 2011.

CRAI Ratings

This issue is in remembrance of restaurants in Christchurch which have to be removed from our list because they no longer exist. 16 venues have so far been lost, while the status of others still inside the CBD is still unknown.

Auckland

215, Dominion Rd	(1)	HHHH½
Andrea (form. Positano), Mission Bay	(1)	HHH
Aubergine's, Albany	(1)	HHHH½
Backyard, Northcote	(1)	HH
Bask, Browns Bay	(1)	HHH
Bay (The), Waiake, North Shore	(1)	HHHHH
Bolero, Albany	(1)	HHHH
Bouchon, Kingsland	(1)	HH
Bowman, Mt Eden	(1)	HHHH½
Bracs, Albany	(1)	HHHH
Brazil, Karangahape Rd	(1)	HHH
Buoy, Mission Bay	(2)	HHHH½
Byzantium, Ponsonby	(1)	HHH
Café Jazz, Remuera	(1)	HHHH½
Carriages Café, Kumeu	(1)	HHHH
Charlees, Howick	(1)	HHHHH
Cibo	(1)	HHHHH
Circus Circus, Mt Eden	(1)	HH
Cube, Devenport	(1)	HH
Del Fontaine, Mission Bay	(1)	HHHHH
Deli (The), Remuera	(1)	HHHH
Delicious, Grey Lynn	(1)	HHHHH
De Post, Mt Eden	(1)	HH
Dizengoff, Ponsonby Rd	(1)	HH
Drake, Freemans Bay (Function Room)	(1)	HH
Eiffel on Eden, Mt Eden	(1)	HH
Eve's Cafe, Westfield Albany	(1)	HHH½
Formosa Country Club Restaurant	(1)	HHHHH
Garrison Public House, Sylvia Park	(1)	HHHH½
Gee Gee's	(1)	HHH
Gero's, Mt Eden	(9)	HHH
Gina's Pizza & Pasta Bar	(1)	HHH½
Gouemon, Half Moon Bay	(1)	HH
Hardware Café, Titirangi	(1)	HHHHH
Hollywood Café, Westfield St Lukes	(1)	HH½
IL Piccolo	(1)	HHHH
Ima, Fort Street	(1)	HHHH
Jervois Steak House	(1)	HHH
Kashmir	(1)	HHHH
Khun Pun, Albany	(2)	HHHHH
Kings Garden Ctre Café, Western Springs	(1)	HH
La Tropezienne, Browns Bay	(1)	HH
Malaysia Satay Restaurant, Nth Shore	(1)	HHHHH
Mecca, Newmarket	(1)	HHHHH
Mexicali Fresh, Quay St	(1)	HH
Mezze Bar, Little High Street	(16)	HHHH
Monsoon Poon	(1)	HHHHH
Mozaike Café, Albany	(1)	HH
Narrow Table (The), Mairangi Bay	(1)	HHHH½
One Red Dog, Ponsonby	(1)	HHH
One Tree Grill	(1)	HHH
Orbit, Skytower	(2)	HHHH
Patriot, Devonport	(1)	HHH½
Pavia, Pakuranga	(1)	HHHHH
Prego, Ponsonby Rd	(2)	HH
Remuera Rm, Ellerslie Racecourse	(1)	HHHHH
Rhythm, Mairangi Bay	(1)	HH
Rice Queen, Newmarket	(12)	HHHH
Sails, Westhaven Marina	(2)	HHHHH
Scirocco, Browns Bay	(1)	HHH
Seagers, Oxford	(1)	HHHH
Shahi, Remuera	(1)	HHH½
Shamrock Cottage, Howick	(1)	HH
Sidart, Ponsonby	(1)	HHHH½
Sitting Duck, Westhaven	(1)	HHH½
Sorrento	(1)	HH½
Stephan's, Manukau	(1)	HHHHH
Tempters Café, Papakura	(1)	HHHHH
Thai Chef, Albany	(1)	HHHHH
Thai Chilli	(1)	HHHHH
Thai Corner, Rothesay Bay	(1)	HHHHH
Tony's, High St	(1)	HHH
Traffic Bar & Kitchen	(1)	HH
Umbria Café, Newmarket	(1)	HHHH½
Valentines, Wairau Rd	(1)	HHHHH
Vivace, High Street	(2)	HH½
Wagamama, Newmarket	(1)	HHHH½
Watermark, Devonport	(1)	HH
Woolshed, Clevedon	(1)	HH½
Zarbos, Newmarket	(1)	HH
Zavito, Mairangi Bay	(1)	HH H

Arthur's Pass

Arthur's Pass Cafe & Store	(1)	HHH½
Ned's Cafe, Springfield	(1)	HHHH

Readers are encouraged to rate eating establishments which they visit by completing a simple form available on-line from www.acoustics.ac.nz, or contact the Editor.
Repeat ratings on listed venues are encouraged.

H Lip-reading would be an advantage. HH Take earplugs at the very least. HHH Not too bad, particularly mid-week.
HHHHA nice quiet evening. HHHHHThe place to be and be heard. (n) indicates the number of ratings.

CRAI Ratings (cont.)

Ashburton

Ashburton Club & MSA	(1)	HHHH½
Robbies	(1)	HHH
RSA	(1)	HHHH
Tuscany Café & Bar	(1)	HHH

Bay of Plenty

Alimento, Tauranga	(1)	H½
Imbibe, Mt Maunganui	(1)	H½
Versailles Café, Tauranga	(2)	HH

Blenheim

Raupo Cafe	(1)	HH

Bulls

Mothered Goose Cafe, Deli, Vino	(1)	HH

Cambridge

GPO	(1)	HHHHH

Christchurch

3 Cows, Kaiapoi	(1)	HHHH
Abes Bagel Shop, Mandeville St	(1)	HHHH
Alchemy Café, Art Gallery	(1)	HHHHH
Anna's Café, Tower Junction	(1)	HHHH
Arashi	(1)	HH
Azure	(2)	HHH
The Bog	(1)	HHHHH
Becks Southern Ale House	(11)	HHHH½
Buddha Stix, Riccarton	(1)	HHHH
Bully Haye's, Akaroa	(1)	HH
Café Bleu	(1)	HHH
Cashmere Club	(1)	HHHHH
Chinwag Eathai, High St	(8)	HH
Christchurch Casino	(1)	HH
Christchurch Museum Café	(1)	HHHH
Cobb & Co, Bush Inn	(1)	HHH
Coffee Shop, Montreal Street	(1)	HH
Cookai	(3)	HH½
Costas Taverna, Victoria Street	(1)	H½
Coyote's	(6)	HHH
Decadence Café, Victoria St	(1)	HHHHH
Drexels Breakfast Restaurant, City	(1)	HHHH½
Drexels Breakfast Restaurant, Riccarton	(1)	HHHH
Elevate, Cashmere	(1)	HHH
Fava, St Martins	(1)	HH
Foo San, Upper Riccarton	(1)	HHH½
Fox & Ferrett, Riccarton	(1)	HHHHH
Freemans, Lyttleton	(9)	HHH½
Gloria Jean's, Rotheram St	(1)	HHHH
Golden Chimes	(1)	HHHHH
Governors Bay Hotel	(1)	HHHH
Green Turtle	(1)	HHHH
Harpers Café, Bealey Ave	(1)	HHHHH
Hari Krishna Café	(1)	HHH
Holy Smoke, Ferry Rd	(1)	HH
Indian Fendalton	(2)	HH
Joyful Chinese Rest., Colombo St	(1)	HHHHH
Kanniga's Thai	(1)	HHH
La Porchetta, Riccarton	(4)	HH½
Little India	(2)	HHHHH
Lone Star, Riccarton Road	(6)	HHH
Lotus Heart, Colombo Street	(1)	HHHH
Lyttleton Coffee Co, Lyttleton	(1)	HHHH
Manee Thai	(6)	HH½
Mexican Café	(6)	HHH
Oasis	(1)	HHHH½
Old Vicarage	(2)	HHH½
Phu Thai, Manchester Street	(1)	HHH
Portofino	(3)	HHHHH
Pukeko Junction, Leithfield	(1)	HHHH
Red, Beckenham Service Centre	(1)	HHHH
Red Elephant	(1)	HHHH
Retour	(1)	HHH
Riccarton Buffet	(2)	HHHH½
Robbies, Church Corner	(2)	HHHH½
Route 32, Cust	(1)	HHHH
Salt on the Pier, New Brighton	(6)	HHH½
Santorinis Greek Ouzeri	(1)	HH
Scarborough Fare	(1)	HH
Speights Ale House, Tower Junction	(1)	HHHH
Tap Room	(9)	HHH
The Bridge, Prebbleton	(1)	HHHHH
The Bicycle Thief	(1)	HHHH½
The Sand Bar, Ferrymead	(2)	HHH½

A crowd watches as the Lone Star Restaurant in Manchester Street is demolished

The Vault, Cashel Mall	(1)	HHHH
Tokyo Samurai	(1)	HHHHH
Tutto Bene, Merivale	(2)	HH
Untouched World Cafe	(1)	HHHHH
Wagamama, Oxford Terrace	(6)	HHH
Waitikiri Golf Club	(1)	HH
Waratah Café, Tai Tapu	(1)	HHH

Clyde

Old Post Office Cafe	(1)	HHHHH

Dunedin

A Cow Called Berta	(1)	HHH½
Albatross Centre Cafe	(1)	HHHHH
Bennu	(1)	HHHH
Bx Bistro	(1)	HHHH
Chrome	(1)	HHHH½
Conservatory, Corstophine House	(1)	HHHHH
Fitzroy Pub on the Park	(1)	HHHHH
High Tide	(2)	HH
Nova	(1)	HHHHH
St Clair Saltwater Pool Cafe	(1)	HHHH½
Swell	(1)	HH
University of Otago Staff Club	(1)	HH

Gore

Old Post	(1)	HHH
The Moth, Mandeville	(1)	HHHHH

Greymouth

Cafe 124	(1)	HHH

Hamilton

Embargo	(1)	HHHHH
Gengys	(1)	HH
Victoria Chinese Restaurant	(1)	HHHHH

Hanmer Springs

Laurels (The)	(2)	HHHHH
Saints	(1)	HHHH½

Hastings

Café Zigliotto	(1)	HHH

Havelock North

Rose & Shamrock	(1)	HHH½

Levin

Traffic Bar & Bistro	(1)	HH

Masterton

Java	(1)	HH

Matamata

Horse & Jockey	(1)	HHHHH

Methven

Ski Time	(2)	HHH

Napier

Boardwalk Beach Bar	(2)	HHHHH
Brecker's	(1)	HHHHH
Café Affair	(1)	HH
Cobb & Co	(1)	H½
Duke of Gloucester	(1)	HHHH½
East Pier	(1)	HH
Estuary Restaurant	(1)	HHHHH
Founder's Cafe	(1)	HHHHH
Napier RSA	(1)	HHHHH
Sappho & Heath	(1)	HH

Nelson/Marlborough

Allan Scott Winery	(1)	HHHHH
Amansi @ Le Brun	(1)	HHHHH
Baby G's, Nelson	(1)	HHHHH
Boutereys, Richmond	(1)	HHHH
Café Affair, Nelson	(1)	HH
Café on Oxford, Richmond	(1)	HHH
Café Le Cup, Blenheim	(1)	HHH
Crusoe's, Stoke	(1)	HHH
Cruizies, Blenheim	(2)	HHHH½
Grape Escape, Richmond	(1)	HHHHH
Jester House, Tasman	(1)	HHHHH
L'Affaire Cafe, Nelson	(1)	HH
Liquid NZ, Nelson	(1)	H½
Lonestar, Nelson	(1)	HHHH
Marlborough Club, Blenheim	(1)	HH
Morrison St Café, Nelson	(1)	HH½
Oasis, Nelson	(1)	HHHHH
Rutherford Café & Bar, Nelson	(1)	HHHHH
Suter Cafe, Nelson	(1)	HH
Verdict, Nelson	(1)	HH
Waterfront Cafe & Bar, Nelson	(1)	HHH
Wholemeal Trading Co, Takaka	(1)	HHHHH

New Plymouth

Breakers Café & Bar	(1)	HHH
Centre City Food Court	(1)	HHHH
Elixer	(1)	HHHH
Empire Tea Rooms	(1)	HHHH½
Govett Brewster Cafe	(1)	HH
Marbles, Devon Hotel	(1)	HHH
Pankawalla	(1)	HHHHH
Simplicity	(1)	HHH
Stumble Inn, Merrilands	(1)	HHH
Yellow Café, Centre City	(1)	HHH
Zanziba Café & Bar	(1)	HHH

Oamaru

Riverstone Kitchen	(1)	HHHHH
Star & Garter	(1)	HHH
Woolstore Café	(1)	HHHH

CRAI Ratings (cont.)

Palmerston North

Café Esplanade	(2)	HHHH
Chinatown	(1)	HHHH
Coffee on the Terrace	(2)	HHH
Elm	(1)	HHHH½
Fishermans Table	(1)	HHHHH
Gallery	(3)	HHHH
Rendezvous	(1)	HH½
Roma Italian Restaurant	(1)	HHH
Rose & Crown	(1)	HH
Tastee	(1)	HHH
Thai House Express	(1)	HHHHH
Victoria Café	(1)	HHHH

Queenstown

Bunker	(1)	HHHH
The Cow	(1)	HHH
Sombreros	(1)	H
Tatler	(1)	HHHH
Winnies	(1)	HHHHH

Rotorua

Cableway Rest. at Skyline Skyrides	(1)	HHHHH
Lewishams	(1)	HHH
Woolly Bugger, Ngongotaha	(1)	HHH
Valentines	(1)	HHHHH
You and Me	(1)	HHHHH
Zanelli's	(1)	HH

Southland

Lumberjack Café, Owaka	(1)	HHHHH
Pavilion, Colac Bay	(1)	HH
Village Green, Invercargill	(1)	HHHHH

Taihape

Brown Sugar Café	(1)	HHHH½

Taupo

Burbury's Café	(1)	HHH
Thames		
Thames Bakery	(1)	HHH
Waiheke Island		
Cortado Espresso Bar	(1)	HHHH
Cats Tango, Onetangi Beach	(1)	HHHH

Timaru

Fusion	(1)	HHHHH

Wanganui

3 Amigos	(1)	HHH½
Bollywood Star	(1)	HHH½
Cosmopolitan Club	(1)	HHHH
Liffiton Castle	(1)	HH½
RSA	(1)	HHH½
Stellar	(1)	HHHH½
Wanganui East Club	(1)	HHHH

Wellington

162 Café, Karori	(1)	HHHHH
180o, Paraparaumu Beach	(1)	HH
88, Tory Street	(35)	HH
Anise, Cuba Street	(1)	HH
Aranya's House	(1)	HHHHH
Arbitrageur	(2)	HHH
Arizona	(1)	HH
Astoria	(2)	HHH
Backbencher, Molesworth Street	(1)	HHH
Bordeaux Bakery, Thorndon Quay	(1)	HH
Buzz, Lower Hutt	(1)	HH½
Brewery Bar & Restaurant	(5)	HHHH
Carvery, Upper Hutt	(1)	HHHHH
Chow	(1)	H½
Cookies, Paraparumu Beach	(1)	HHH½
Cosa Nostra Italian Trattoria, Thorndon	(1)	HHHH
Gotham	(6)	HHH½
Great India, Manners Street	(2)	HHHHH
Habebie	(1)	HH
Harrisons Garden Centre, Peka Peka	(1)	HHHH
Hazel	(1)	HH
Katipo	(1)	HHHHH
Kilim, Petone	(4)	HHHH½
La Casa Pasta	(1)	HHHH½
Lattitude 41	(3)	HHHH
Legato	(1)	HH
Le Metropolitain	(1)	HHHHH
Loaded Hog	(5)	HHHH½
Manhatten, Oriental Bay	(1)	HHHH
Maria Pia's	(1)	HHH
Matterhorn	(1)	HHH
Mungavin Blues, Porirua	(1)	HHHHH
Olive	(1)	HHHHH
Original Thai, Island Bay	(1)	HHHH
Palace Café, Petone	(1)	HH½
Parade Café	(1)	HH
Pasha Café	(1)	HHHH
Penthouse Cinema Café	(2)	HHH½
Pod	(1)	HH½
Rose & Crown	(1)	HHHHH
Shed 5	(1)	HH
Siem Reap	(1)	HH
Speak Easy, Petone	(1)	HH
Speights Ale House	(1)	HH
Sports Bar Café	(1)	HHHH
Stanley Road	(1)	HHH
Stephan's Country Rest., Te Horo	(1)	HHHHH
Windmill Café & Bar, Brooklyn	(1)	HH