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SPECIAL ISSUE

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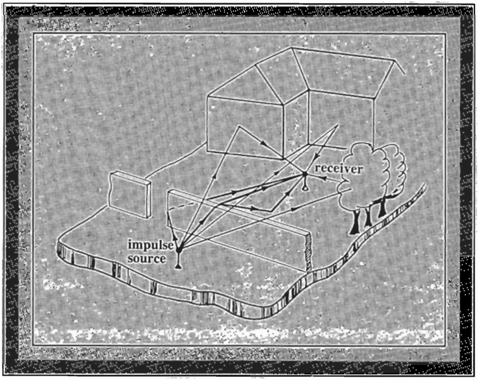

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Acoustics Australia

COVER ...
Ray paths and time trace obtained across a residental fence using an impulse source (see article by Charles Don).

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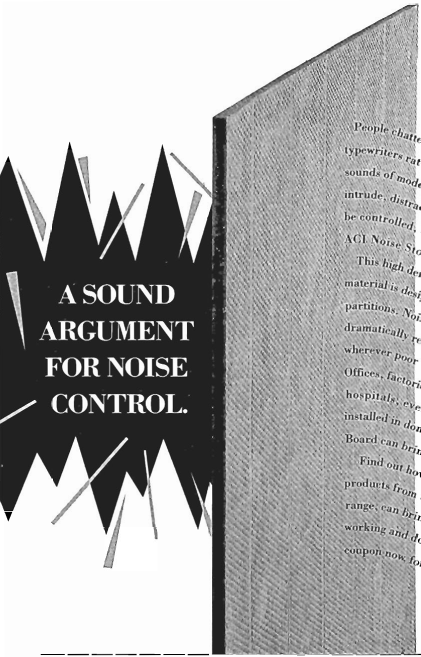
A SOUND ARGUMENT FOR NOISE CONTROL.

EDITORIAL 1

Dr. Stephen Samuels
President, Australian Acoustical Society

A WORD OF WELCOME FROM THE PRESIDENT

The practice of the science and art of acoustics in Australia covers a range of fields that is almost as diverse as the country is large. Few, if any, areas have been or are being left untouched. Broadly speaking, practitioners are to be found in both the corporate and government sectors, in various industries, in research institutions, in consultancy offices and in academe. Work of a high technical quality, frequently at technology's leading edge, has been consistent and prominent attributes of the efforts of Australian acousticians. Ample evidence of these observations is to be found in the excellent papers published in this edition of Acoustics Australia. Indeed they also exemplify the range of fields to which I have referred.

Exciting progress is being made, despite the economic difficulties which have beset Australia, like so many other countries. Some economic pundits have been recently suggesting that the light at the end of the present recessionary tunnel is beginning to glimmer. While this may or may not be true, the reality is that times have been rough in the last year or so and the acoustics fraternity in Australia has certainly felt the pinch. However like so many of our friends around the world, we pride ourselves on being able to tough it out in such times. In the Australian world of acoustics we seem to have managed to keep the recession-based casualty rate to a minimum.

Organisation and running of major international conferences is no small undertaking, particularly under these present economic circumstances. As President of the Australian Acoustical Society it is therefore with some pride that I, on behalf of all members of the Society extend a warm welcome to all our visitors. This special edition of Acoustics Australia has been purposely published at a time to coincide with both WESTPRAC IV and INTERNOISE 91. I trust it will add to the enjoyment of your visit and further your understanding of acoustics practice in Australia. I wish all delegates to the Conferences well and trust that your involvement is most successful and enjoyable, both technically and personally.

EDITORIAL 2

Howard Pollard
Chief Editor

This issue completes 10 years of production by the present editorial team. It is the 58th issue since the commencement of the old Bulletin in 1972. Until 1978 the New South Wales Division had the responsibility for the production and the Editorial Committees included Peter Knowland, John Irvine, Ted Weston, Ferge Fricke and Richard Heggie. In 1979 the Victoria Division took over the responsibility and the Chief Editor was Robin Alfredson followed by Rob Law in 1980 and Don Gibson in 1981. In 1982 the production was moved back to New South Wales with Howard Pollard as Chief Editor and Marion Burgess as Associate Editor.

In April 1985 the name of the journal was changed from Bulletin of the Australian Acoustical Society to Acoustics Australia. In the current regime, the Chief and Associate Editors are assisted by a number of active editors including Neville Fletcher, Dennis Gibbings, John Dunlop and Paul Dubout.

The current brief to the editors from the Council is to continue producing a quality technical journal that serves both as a reflection of acoustical activities in Australia and as a medium for news and product information of interest to members. A steady stream of articles from acousticians, both within and outside Australia, together with frequent expressions of interest from other countries, suggests that we are fulfilling a useful function.

There have been criticisms, of course, some who want more readable articles, some who want more relevant technical information, those who would prefer a newsletter rather than a technical journal, those who think we should use cheaper paper (whatever that is), those who want more colour illustrations, etc.

There is always room for improvement and new ideas. We would be pleased to hear from any member who has an unfulfilled yen to assist in the editing or who has positive ideas on the future direction of Acoustics Australia. I would like to thank all those who have assisted so ably in the production of this publication including our panel of consulting editors and referees; Leigh Wallbank, Business Manager; and the Staff of Cronulla Printing Co.

EDITORIAL 3

Colin Hansen
Organiser, Special Topic

Active control of noise and vibration is well on the way to becoming a billion dollar industry by the end of the decade. However, many of the technical problems associated with the implementation of active control systems in complex environments are yet to be solved, guaranteeing plenty of work for the numerous research groups that have been spawned in the past ten years. The extent of the research activity in this area is evidenced by the attendance of 300 delegates at a special conference on this subject held at Virginia Polytechnic Institute and State University in Virginia, U.S.A. in April, 1991. Over eighty papers on various aspects of the subject were presented by authors from the U.S.A. and a large number of other countries.

At the present time commercial systems are available for reducing noise in air handling ducts and automotive exhausts and systems are currently being developed to reduce aircraft interior noise, automobile interior noise, and submarine radiated noise to mention just a few.

In Australia, the largest group undertaking research in this area is the Mechanical Engineering Department at the University of Adelaide. Since the beginning of 1988, the group, led by Dr. Colin Hansen, has received approximately $900,000 in funding from the Sir Ross and Sir Keith Smith Fund, the Australian Research Council, the University of Adelaide, DSTO, Materials Research Laboratories, the Department of State Development and the Australian Electricity Supply Industry Research Board. Others in Australia working in this field include Drs. I Shepherd and F. La Fontaine at CSIRO, Division of Building Construction and Engineering in Melbourne, Dr. J. Pan at the University of Western Australia and Dr. L. Wood At R.M.I.T. in Melbourne.

In December, 1991 a rare opportunity will arise for engineers, acousticians and researchers interested in active control of noise and vibration to learn more about this exciting field. A four day intensive course involving six well known speakers from the U.S.A. and the U.K. who are all leaders in their field and four speakers from Australia will be offered at the University of Adelaide.

In this and the next issue of Acoustics Australia, a brief overview is given of the current state of the art of active noise control. Due to the limited space available, it was not possible to treat active vibration control, but this may be the subject of a future issue.

Impulse Acoustics

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ABSTRACT: The ability to time - isolate desired components and obtain information simultaneously over a wide frequency range are two aspects of impulsive sounds which can be utilized in a variety of measurement situations. However, impulse measurements place special demands on the instrumentation. Consideration is given to a range of impulse sources and to the requirements of the receiver and recording system. Then a number of applications of impulse acoustics are reviewed with the advantages and limitations being outlined.

1. INTRODUCTION

A review of the acoustics literature indicated that there have been relatively few applications of impulsive sound as a measurement tool. Yet modern digital technology has made recording and analysis simple and its use offers several advantages over continuous wave techniques [1]. This article seeks to outline the potential of impulse acoustics and to describe the necessary instrumentation.

An impulsive sound can be defined as a sudden pressure change in the medium which, after reaching a high pressure level for a short time interval, returns to the ambient pressure. Impulses have three important characteristics:

(i) a large pressure for a short duration, often with a well defined waveform,

(ii) a relatively broad frequency band, and

(iii) an inherent phase relationship between the frequency components forming the impulse.

Before considering the instrumentation an example of how impulses can be used advantageously will be considered.

2. AN EXAMPLE OF IMPULSE MEASUREMENT

Consider the house and garden sketched in Figure 1(a). Sound reaching a receiver in the garden from a source located outside the fence can arrive there by a variety of different paths. If a sinusoidal sound source is used, then the resultant signal will also be a sinusoid with a level depending on the various path lengths and on the phase and amplitude changes which occur on reflection or diffraction of the sound. It would be difficult to separate the various components experimentally.

Alternatively, if a short duration impulsive sound is used, then the various contributions will arrive at different times and can be identified by relating the delay to the path length [2], as indicated in Figure 1(b). The amplitude of the individual pulses gives an indication of the importance of that contribution, while changes to the pulse shape can be related to the mechanism (e.g. reflection or diffraction) involved in the sound reaching the microphone.

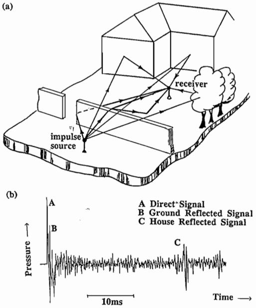

Fig. 1. (a) Some possible ray paths and (b) a time trace obtained across a residential fence by using an impulse source.

To take advantage of this concept several conditions should be met. The source duration must be short so that the pulses are sufficiently separated in time. Preferably, the pulse waveform should be relatively simple and reproducible to permit shape changes to be easily identified. Finally, the level must be sufficient for the impulse to dominate any background sounds. While these principles are not new [3], modern instrumentation makes their application easier. A suitable source and receiver are also needed.

3. SOURCES

Nine techniques for producing impulse noise for hearing tests have been listed [4]. More generally, impulse sources may include explosives, a sonic boom simulator [5] and even watershock [6] for the production of high pressure pulses in liquids.

Tone bursts are generated electronically and converted into an acoustic pulse through a loud speaker. Inevitably, the response of a loud speaker distorts the waveform, thereby altering the frequency content. One way of producing an impulse is to determine the transfer function of the loud speaker (that is the ratio of the output to input signal at each frequency) and then use this known function in conjunction with a waveform synthesizer to generate an apparently distorted input signal which, when emitted by the loudspeaker, produces the desired waveform [3,7,8]. Such impulse generators have the advantage that the waveshape, risetime and duration can be controlled and hence the spectral content adjusted, within limits. Also the impulse is reproducible and can be repeated frequently. Limitations include the low output level generated by the speaker and residual ripple preceding and following the main impulse.

More direct ways of creating an impulsive sound including hitting a metal plate with a hammer, bursting a balloon [9] or discharging a gun. Figure 2 shows waveforms obtained simultaneously by placing a microphone on either side of, and equi-distant from sources. The hammered plate tends to ring, producing a relatively long lasting sound which, like the popping balloon, is quite different at the two microphones.

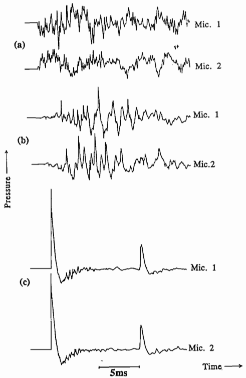

Fig. 2. Waveforms obtained at two microphones located the same distance on either side of (a) a hammer blow on a metal plate (b) a bursting balloon and (c) a discharging gun.

An asymmetric sound field is inappropriate for many measurement situations as it is difficult to correct satisfactorily for changes of waveshape with angle. Perhaps even more important, both a hammer blow and the bursting balloon tend to vary between successive events, making it difficult to obtain reliable data. A hammer blow and similar sounds caused by the collision of two more objects are often classified as impact noise [10] and will not be considered further in this article.

For measurement purposes, a more useful impulse source is a gunshot, such as that shown in Figure 2 (c), which is of shorter duration and tends to be reproducible on a shot-to-shot basis. Live bullets are not recommended! The bullet can be lethal and can produce a shock wave which exhibits non-linear effects. An effective source is formed by discharging a shot-shell primer held in a suitable solid metal holder, by hitting a pointed firing pin into the primer with a hammer [11]. If this mechanism discharges into a tube, say 50cm or more in length, then the hot gas emerging from the tube forms an effective point sound source a few millimetres in front of the tube. Measurements indicate that the sound follows the inverse square law and has conical symmetry about the barrel of the gun. This means that the waveform and intensity does depend on the angle from the gun axis but the waveform is invariant anywhere on a cone formed around this axis. These features make such a gun a suitable source for many types of impulse measurements.

Other impulse sources which have been used include electrical sparks [12], where voltages up to 30kV are discharged between electrodes. A lower voltage system has been produced for model studies [13]. The electrical discharge is more nearly omni-directional than a gunshot, although significant shape variations occur between sparks. The acoustical effect of explosives used in quarry blasts or by the military have been studied at long distances [14-17] and at short range [18]. These large scale explosives have decidedly dangerous aspects and are best left to experts.

Most explosive sounds have a similar waveshape [19] initially a short duration high pressure compression with a fast risetime followed by a longer, and often more randomly varying, rarefaction. The frequency content of the pulse depends on the duration and nature of the source. For a gunshot the range is typically from 100Hz to 15kHz, with most of the energy around 1kHz, as indicated Figure 3. A spark discharge is typically one tenth the duration so the frequency range is correspondingly ten times greater.

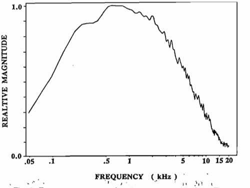

Fig. 3. Frequency content of an isolated gunshot from Fig. 2(c).

A spark discharge or blank being detonated can produce levels ranging from 120dB to 160dB or more. Assuming a background level of 55dB, the dynamic range typically exceeds 65dB which imposes constraints on the recording system. However, it does mean that impulse measurements can be performed satisfactorily in the presence of relatively high background levels.

In Figure 2(c), the delayed reflection of the impulse from the ground is quite apparent. Similar reflections must occur in the other waveforms shown in the figure as they were taken using a similar geometry; the reflections have merged with the direct component. This is a major advantage of using short duration impulses as the unwanted components can often be identified by their separation in time. Small secondary reflections from people, tripod legs or parts of the source structure can fall in the tail of the main pulse and must be avoided.

Non-linear shock conditions occur close to very high pressure level sources, so it is necessary to avoid taking measurements closer than some pre-determined distance. For example, when detonating blanks resulting in levels about 160dB at 2m from the muzzle, it was determined that beyond 2m the sound behaved linearly, following the inverse square law. Inside this distance the measurements were unreliable, partly due to shock conditions and partly due to debris from the discharge causing mechanical noise on the microphone. Many quieter sources display linearity to within a few centimetres of the source.

4. RECEIVERS

The important requirement is that the measuring microphone faithfully captures the impulse waveform, which implies that it should not saturate because of the high level

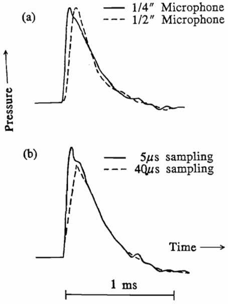

Fig. 4. (a) Comparison of normalized gunshot waveforms recorded with Bruel & Kjaer microphones type 4135 and 4165, (b) effect of altering the sampling time, both waveforms recorded using the same 1/4" microphones.

and that is has an adequate frequency response. If the gunshot of Figure 2(c) is recorded with a 1/2" microphone with an upper frequency response rolling off at 20kHz, a longer risetime will be apparent compared to that obtained using a 1/4" microphone responding up to 100kHz, as shown in Figure 4(a). When the same gunshot is recorded using a 1/8" microphone, responding up to 140kHz, the waveform is unchanged, indicating that the 1/4" microphone had an adequate response. Also, the lower sensitivity 1/4" microphone is less likely to saturate.

5. THE RECORDING SYSTEM

A number of instruments have been designed to determine the peak value [20, 21] and duration [22] of impulses. These instruments are particularly useful in studies of hearing damage due to impulse noise. The latter topic has been reviewed recently [23] and is the subject of ongoing investigations [24, 25]. In this section the emphasis is placed on faithfully recording the impulse waveform.

A tape recorder can be used to capture impulse sounds, but analogue systems have a number of limitations. Typically the dynamic range required by impulses far exceeds the 50dB available from most magnetic tapes while there must be no phase distortion over the full frequency band [27].

More reliable are the modern digital acquisition systems, where the performance depends largely on the behaviour of the analogue to digital converter (ADC). The dynamic range depends on the number of bits produced by the ADC, theoretically being 60, 72 and 96dB for a ten, twelve and sixteen bit system respectively. However, the larger bit size requires longer to convert the sampled analogue signal into a digital word, which restricts the sampling frequency. Many instruments utilize a 10 bit ADC with a conversion time of 10μs, corresponding to a maximum sampling frequency of 100kHz, which is appropriate to cover the audio range. A similar 16 bit ADC might require 600μs per conversion, with an upper frequency of only 1.7kHz. Faster systems can be purchased, e.g. 16 bit with a 1MHz upper limit; these are technically more sophisticated and therefore more expensive.

To avoid aliasing [28], the input signal must be bandwidth limited to less than half the sampling frequency. Often this is achieved by switching in an anti-aliasing filter linked to the selected sampling frequency. As filtering may significantly change the recorded waveform it is essential that a suffficiently fast sampling rate is chosen. Consider the gunshot of figure 2(c), which has the frequency content shown in figure 3. The majority of the signal lies below 15kHz although there is some energy above this frequency. The solid curve in figure 4(b) was produced using a 5μs sampling time, i.e. an upper limit of 200 kHz which is well above the expected frequency content of the pulse. When the sampling time is increased to 40μs a significant change in the waveform occurs, broken curve in figure 4(b), even though the upper limit of 25kHz would appear to be beyond the main frequency band of the impulse.

Sufficient memory to store the complete pulse is another important factor. A signal lasting 0.1s requires at least 2×10^4 words if sampled at 200kHz while 1024 words would permit only 5ms of the pulse to be stored.

Often the digitised waveform has to be analysed into its frequency components by using the Fast Fourier Transform

(FFT) [28] which requires an N point data set where N is 2^n with n integer. If the number of actual data points does not meet this criteria additional points, set to the ambient value, can be added at the start and/or end of the time trace without changing the frequency spectrum, although the frequency resolution depends on augmented time, not the actual recording interval.

The transform effectively analyses a cyclic pattern formed by repeating the N point sequence. If the tail of the impulse has not returned to the ambient condition after the N points, then the pattern has a step where the next set joins on which will distort the true frequency spectrum. Often this leakage is reduced in frequency analysers by using "window functions", such as Hanning, Hamming, etc., which deemphasise data near the start and tail of the time window. However, these window functions may also modify the impulse itself, particularly if it occurs near the start of the trace. For impulses is it preferable to use a rectangular window, which gives equal weighting to all data points, and bring the tail gradually down to the ambient value before the end of the time trace by modifying the data in a computer. Note that any curtailing or modification to the tail will alter the lower frequency components, but with care the effect can be minimal. If the impulse waveform is contained within, say, a 1024 point trace it is sometimes useful to extend the data to 4096 points by adding additional words at the ambient level on either end of the time trace prior to applying the FFT. This has the advantage of decreasing the frequency resolution, by a factor of 4 in this example.

6. COMMERCIAL INSTRUMENTATION

The range of commercially available equipment for the capture and manipulation of impulse data is rapidly expanding in its versatility. To permit the simultaneous capture and comparison of waveforms, two or more inputs are essential. Because the impulses may arrive at different times, either long records or the ability to delay the triggering of one trace relative to the other is necessary. Another asset is pretrigger, which enables small pulses prior to the triggering impulse to be captured.

The simplest pulse capture system is a digital oscilloscope. A few years ago an advanced system provided dual 1024 word channels and a 1μs sampling time. Currently four channel systems, each with 16K record lengths and 0.1μs sampling times are available. Multichannel data acquisition systems which use a PC for mass storage of data are another option. While two or four input channels are normal, systems with 24 channels are available. In some instruments the sampling rate is independent of the number of channels while in others the sampling interval increases as more channels are involved. Both the above systems depend on the host PC for data manipulation. Sophisticated computing packages are available to perform Fourier analysis and other mathematical processes. One claim for such PC based systems is that as technology improves it is only a matter of updating the program compared to replacing dedicated push-button equipment.

Waveform analysers (or transient recorders) and spectrum analysers are evolving into similar instruments as their computing options become more extensive. Some years ago, waveform analysers typically provided longer time traces and more scope for manipulating the data in the time domain than spectrum analysers, which captured a fixed record length and converted this directly into the frequency domain. Now both types of instruments can be obtained which capture traces up to 10^5 words and permit a desired portion to be selected for frequency analysis. Another useful feature is the ability to ensemble average successive waveforms to reduce random effects due to source variations or turbulence in the medium. It is essential that the system has provision to exactly align waveforms before addition otherwise distortion of the waveshape will occur. Interfacing either type of instrument to a PC gives complete flexibility.

7. FURTHER APPLICATIONS

When an impulse sound occurs in an auditorium, the multitude of reflected pulses reaching a receiver can be used to estimate the acoustic qualities of the room [29]. Simultaneous reflections will overlap, exposing intense echoes, while the reverberation time and frequency response can be derived from the impulse response. Impulses have also been applied to measurements of the acoustic transmission of ducts [30].

Using an impulse with a well defined waveform, the change between the incident pulse shape and that reflected from some flat surface can be used to determine the magnitude and phase of the reflection coefficient and hence the characteristic impedance of the surface. This has been achieved on wall sections [31] and fibrous surfaces [32] using a spark discharge and on soils using sine packets [33] and gunshots [34]. Because the required signals can be distinguished from reflections from the walls, impedance measurements can be performed without the use of an anechoic room - a major advantage over continuous waves. Another advantage is that impedance measurements can be determined rapidly over a wide frequency range - typically 500Hz to 10kHz for a gunshot. This is particularly useful if dynamic effects, such as the impedance variation as moisture is added to the soil [35], are to be studied. It is difficult to adapt such techniques to low frequencies. If an impulse source has dominant frequencies down to, say 20Hz, then the resulting waveform has a much longer duration than a gunshot and so it is difficult to achieve suitable time-isolation of the direct and reflected pulses, unless greatly increased distances are employed.

The above techniques can also be used to determine the attenuation of acoustic barriers by replacing the reflected pulse by one diffracted over the barrier [2, 36]. Secondary pulses may prove a useful source of additional information. For example, if a barrier has a crack in it then sound leaking through will reach the microphone located behind the barrier before the main diffracted impulse. The size of the earlier pulse relative to the main diffraction gives a measure of the leakage [37, 38].

If an impulse is propagated through air above a grassy plane then close to the source the direct and ground reflected pulse will be time-isolated. At larger distances the path difference becomes negligibly small and these components merge. This behaviour for different impedance surfaces has been investigated using both tone bursts[39] and gunshots [19]. In practice, wind and temperature gradients cause bending of the sound rays giving rise to regions of either sound focusing or a shadow zone. The changes in the pulse shape as it propagates through the medium can be used to study these processes [40, 41]. While the above studies were made in the atmosphere, similar principles can be applied to pulses in the oceans [42]. When impulses are used, the results are obtained with a "snapshot" of the meteorological conditions rather than an average over many seconds as is commonly encountered in continuous wave measurements.

8. CONCLUSION

While there are situations where impulse noise can be annoying, for example quarry blasts, rifle-range gunfire and sonic booms, the emphasis in this paper has been on the use of impulses to solve acoustic problems. Because the required components can often be time-isolated, impulse measurements can be taken indoors without an anechoic room and their wide frequency range means that data is gained simultaneously over most of the audio range.

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Global Control of Sound Transmission into Enclosed Spaces - How Does It Work?

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ABSTRACT: Active noise control appears poised for wide-spread use in the control of low frequency sound transmission into enclosed spaces. This paper provides a brief overview of how such systems physically achieve sound attenuation

1. INTRODUCTION

Advances in engineering materials have led to a decrease in the strength to weight ratio in modern passenger carrying vehicles, in particular aircraft and automobiles. It has also led to the development of more fuel-efficient, yet louder (in the case of modern turbo-prop engines), propulsion systems. It is expected by passengers, however, that modern advances should not result in any decrease in creature comfort; on the contrary, we expect to be pampered with ever-increasing comfort. This has led to a problem for structural acousticians, as their old ally, mass, is being eliminated before their very eyes. How can low frequency noise be controlled without substantially increasing the mass of these sleek, modern carriers? Active noise control may be one possible solution. But how does such a system work?

The basic components of an active noise control system, as shown in Figure 1, can be divided into two broad categories; the physical control system, and the electronic control system. The physical control system consists of the control sources (speakers, shakers, etc.), which provide a controlling, or "cancelling", disturbance, and error sensors (microphones, accelerometers, etc.), which provide a measure of the residual disturbance. An optional third component is a reference sensor, which will provide a measure of the impending primary disturbance. (While this component is "optional", its inclusion greatly enhances the effectiveness and stability of the control system [1]. Where the primary noise disturbance is periodic, such as generated by rotating machinery, acquisition of this "reference signal" is not a difficult task.) The electronic control system uses the error and reference signals to drive the control sources to achieve the desired sound attenuation.

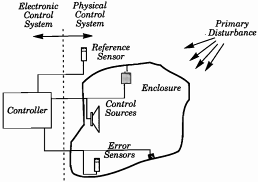

Figure 1. Components of an active noise control system.

The vast majority of active noise control systems currently under investigation for implementation in sound transmission problems utilise a reference signal to construct a feedforward control system, shown in block form in Figure 2. (Although it may be advantageous to combine feedforward and feedback in one system [2], pure feedback control systems are rare in active sound transmission control systems, and will therefore not be considered here.) This type of control system changes the characteristic impedances of the structural/acoustic system to the impending primary source disturbance, or, in control terminology, it modifies the zeroes of the system. To be practically implemented, this arrangement must be adaptive, using the signal from the error sensor (which is to be nulled) to modify the control system generated signal. This is necessary to accommodate the changing response characteristics of the structural/acoustic system (plane, automobile, etc.) which accompany changing external parameters such as temperature, air pressure, and age.

Parallels between advances in microprocessor technology and advances in active noise control are often drawn. While

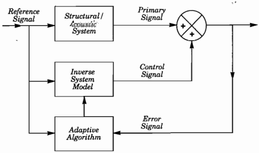

Figure 2. Adaptive feedforward control system.

it is true that the availability of fast, inexpensive microprocessors has enabled practical implementation of active noise control systems, the arrangement of the structural/acoustic, or physical, part of the system is of equal importance in achieving satisfactory levels of sound attenuation [3,4]. At this stage, however, there is no direct analytical methodology for the design of the physical part of active systems for controlling sound transmission into enclosed spaces; rather, each case must be considered individually. In fact, it is only recently that such systems have begun to be moved from the "pure research" basket to the "research and development" basket.

2. PHYSICAL CONTROL MECHANISMS

Before examining the physical means by which sound transmission into enclosed spaces is controlled, a brief review of the concept of structural/acoustic modal coupling is in order. When sound is transmitted from outside an enclosed space to inside, such as propeller noise into an aircraft fuselage, the outside disturbance first sets the enclosing structure into motion. The structural vibration modes then couple with the interior acoustic modes, resulting in an energy transfer from the structure into the acoustic space. For structures which are at least of "moderate" size, which constitute the vast majority of enclosed spaces targeted for active noise control, and where the acoustic medium is not particularly dense, such as air, the response of the structural/acoustic system can be considered in terms of the structural *in vacuo* mode shapes, the acoustic cavity rigid-walled mode shapes, and the modal coupling between the two. Not all structural modes will excite all acoustic modes; in fact, quite the opposite. For modal coupling to occur, the product of the structural and acoustic mode shape functions at the structural/acoustic boundary (the wall), integrated over the entire contacting area, must be a non-zero number. For this type of coupled system, the total response can be considered in two regimes; structure-controlled, where the majority of the total system energy is in the shell, and cavity-controlled, where the majority of the total system energy is in the acoustic space.

Research directed towards the global active control of enclosed sound fields can roughly be considered in two categories, divided by the type of control source used; acoustic control sources located in the enclosure, and vibration control sources attached directly to the structure through which the sound is being transmitted. Acoustic control source work has received the majority of this division of labor, and so will be considered first.

Acoustic Control Sources

When employing a global active noise control strategy to the problem of sound transmission into an enclosed space, the aim is to reduce the spatially-averaged levels of squared sound pressure, or acoustic potential energy, defined as

$$E_p = [1/(2\rho_o c_o^2)] \int_V <p^2(\vec{x})> d\vec{x}$$

where $p(\vec{x})$ is the complex acoustic pressure at some point in the enclosed space, ρ_o and c_o are the density of, and speed of sound in, the acoustic media, respectively, and V denotes the volume of the enclosed space. The acoustic pressure at any location in the enclosed space is the sum of contributions from a (theoretically infinite) set of acoustic modes,

$$p(\vec{x},t) = \sum_{i=1}^{\infty} a_i(t)\phi_i(\vec{x})$$

where ϕ_i is the i^{th} acoustic mode shape function, and a_i is its complex amplitude. When acoustic sources are used in an active system controlling sound transmission, it is easy to view the physical control mechanism as one of "cancellation", where the goal of the active control system is to excite the acoustic modes in the enclosure with equal amplitude and opposite phase to that of the primary source. However, simple interference of two sound fields would result in large noise reductions at some interior locations at the expense of increased sound pressure in other. Implying this as the physical mechanism responsible for global sound attenuation leads to the (in)famous catch-cry of active noise control researchers, "where does the energy go?" [5]. To answer this question, a closer look at the sound transmission problem is required.

The energy transfer from the structural to the acoustic modes is dependent upon the input impedance of the acoustic modes at the structural/acoustic interface, which is proportional to the acoustic modal pressure at this boundary. In exciting the acoustic modes out of phase with the primary excitation, the control source causes a reduction in the modal pressure at the interface, which in turn acts to decrease this input impedance. Thus, the amount of energy accepted by the acoustic modes is reduced. Further, by reciprocity the impedance seen by the control sources looking into each acoustic mode is similarly reduced. This mutual unloading of the primary and control noise sources is the impedance mechanism utilised in the active control of sound transmission using acoustic sources.

Vibration Control Sources

As mentioned, the alternative to acoustic control sources in the enclosed space is vibration control sources attached directly to the structure. Vibration control sources achieve global sound control by altering the velocity distribution of the structure. This can have two different effects, corresponding to two different physical mechanisms. The first of these, which is the most obvious, is to reduce the levels of vibration which cause the noise [6]. For a coupled enclosure this does not necessarily mean reducing the total structural vibration, but rather reducing the vibration levels of the principle noise-producing (coupled) structural modes. This effect, termed modal control, is most prevalent when the response of the system is structure-controlled, and is due to an increase in the structural input impedance of these modes to the external sound pressure excitation field [7].

The second effect which vibration control sources can have upon the velocity distribution of a structure, often predominant for a cavity-controlled response, is to alter the relative amplitudes and phasing of the structural modes (termed modal rearrangement) [7,8]. This can have the effect of reducing the total modal energy transfer into an individual acoustic mode from the set of structural modes coupled to it.

For example, consider the control of sound transmission into the rectangular enclosure of Figure 3. In this case, the responses of the dominant structural and acoustic modes are shown for a frequency near the (0,0,1) acoustic modal resonance (cavity-controlled response) for normally incident plane wave primary excitation and a single vibration source in the center of the panel. Even though the amplitudes of the structural modes have not decreased, the rms amplitudes of the acoustic modes have been reduced by approximately 20 dB, leading to a reduction in acoustic potential energy of approximately 40 dB. How is this possible?

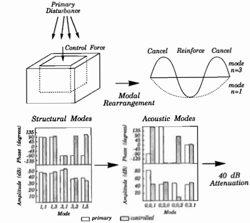

Figure 3. Active control of sound transmission through a rectangular panel into a cavity using a vibration control source where the principle mechanism is modal rearrangment of both the structural and acoustic modes.

Modal rearrangement is the mechanism at work. The change in relative amplitude and phasing of the structural modes has led to a reduction in the overall levels of energy transferral into a given acoustic mode from the structural modes with which it is coupled. Referring to Figure 3, if the amplitudes and phases of the structural modes are adjusted correctly, their individual excitations of the given acoustic mode will tend to "cancel" each other out; that is, the structure becomes "self-unloading".

There is an interesting sideline arising from this dual-mechanism nature of vibration source active noise control. Initial research directed towards using vibration control sources on aircraft modelled the aircraft as plain cylinders. The modal coupling characteristics of a plain cylinder are such that essentially, at low frequencies, each acoustic mode is driven by a single structural mode. It was found that when using a limited number of control sources, sound attenuation by modal control near the resonance of either the acoustic or structural mode in a coupled pair was significant. However, off-resonance sound attenuation was poor. Often, modal rearrangement will work off-resonance, but this was not a viable prospect in a plain cylinder, as it required at least two structural modes to be coupled to a single acoustic mode to work. The addition of a floor-like longitudinal partition into the model, however, alters the modal coupling characteristics so as to "turn-on" the modal rearrangement mechanism, improving off-resonance performance [8]. This phenomena is very unusual, where the complexity of the model improves the result!

3. CONTROL SOURCE/ERROR SENSOR ARRANGEMENT OPTIMISATION

As mentioned earlier, there is no direct analytical method for design of the physical part of an active sound transmission control system. There are, however, several concepts that are commonly employed when analytically assessing the maximum performance of such a system. It is intuitive that for an active noise control system to be effective, it must be able to both excite the modes (structural and/or acoustic) excited by the primary noise source (controllability), and also measure the response of these modes (observability). Ideally, this would lead to the use of one control source and error sensor per mode [9], an ideal not practically realisable. It is more desirable to design active control systems using a relatively few, judiciously placed, transducers. For a simple structure, such as a rectangular enclosure, "good" acoustic control source and error sensor placement positions in the corners (where the acoustic modes have antinodes) are obvious. For more complex structures, such as an aircraft fuselage, the optimum arrangement is not so clear.

One reason that is has thus far proved, in general, impossible to determine directly analytically the optimum physical arrangement of the control sources and error sensors is because sound power attentuation is not a linear function of control source location [10], and because the optimum error sensor locations are coupled to the control source locations [11,12]. For acoustic and vibration control sources, however, control source volume velocity and force, respectively, are linear functions of sound pressure. Therefore, acoustic potential energy can be expressed as a quadratic function of control source volume velocity or force, and the problem solved to determine the optimum volume velocity or force for a given control arrangement [7,13,14]. This process can be implemented in a numerical search routine to optimally locate the control sources. The problem is, however, that six numerical integrations are required at each location! This process can, however, be simplified and sped up by re-expressing it as a linear regression problem and using commercially available software to perform the required calculations (this has the added advantage of simultaneously placing both the control sources and error sensors) [12].

4. WHAT DOES THE FUTURE HOLD?

As their mechanisms of operation are known, and there is some form of design methodology available (albeit inefficient), will active noise control systems begin appearing on planes, trains, and automobiles anytime soon? The answer is yes, probably not this year or next, but possibly on a commercial scale by the end of the decade. Several flight tests of experimental systems have already been undertaken [15,16,17]. There are still implementation problems to be overcome, such as practically viable control sources (although new-generation piezo-electric ceramics look extremely promising), and certain electronic control system necessities such as system transfer function modelling. However, as testament to the field's bright future, at least six companies "specialising" in active control are operating in the United States and United Kingdom, although as yet the only truly commercial system is for the relatively simple problem of plane wave sound propagation in air handling ducts.

5. SUMMARY

Active systems for controlling sound transmission into enclosures produce sound attenuation by altering the characteristic impedances of the structural/acoustic system. Acoustic control sources reduce the transfer of energy between the coupled structural and acoustic modes by reducing the acoustic pressure on the surface of the structure.

Vibration control sources, however, have dual-mechanism characteristics. Firstly, they increase the structural input impedance of the primary offending structural modes to the external sound field. Secondly, they reduce the modal energy transfer by altering the relative amplitudes and phasing of the coupled structural modes.

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Acoustics of The Chinese Qing

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ABSTRACT: The Qing is a bowl-shaped musical instrument, commonly used in Buddhist religious ceremonies in China. The principal modes of vibration, which result from the propagation of bending waves around the circumference, are quite similar to those of a bell. The pitch of the Qing is determined almost entirely by the fundamental (2,0) partial, although the (4,0) partial is clearly heard as an overtone.

The qing (also known as shun or ching) is a bowl-shaped musical instrument, commonly used in Buddhist religious ceremonies, where it is often paired with a muyu or wooden fish of about the same size (as shown in Figure 1). Qing generally range from 10 to 40 cm in diameter and 8 to 35 cm in height, although one large qing from the Han dynasty (206 B.C. - 210 A.D.) measures 75 cm in diameter [1]. When used in religious ceremonies, the qing generally rests on a silk pillow and is struck at the rim with a wooden stick. Figure 2 shows four bronze qing from 10 to 18 cm in diameter. In ancient times, the qing was often engraved with the text of the Buddhist Sutra, whose wonders would be conveyed by the sound of the qing.

The principal modes of vibration result from the propagation of bending waves around the circumference. Viewed in the axial direction, these modes resemble those of a bell, with the (m,0) mode having 2m nodes around the mouth [2]. The (m,1) and higher families of bell modes are not observed, however, Figure 3 shows holographic interferograms of some of the more prominent vibrational modes of a 18 cm diameter qing (the largest one in Figure 2). Modes (2,0) through (9,0) are identifiable in the top two rows, but it is difficult to assign mode numbers at the high frequencies.

Mode frequencies are shown in Figure 4 as a function of m, the number of nodal diameters, for the four qing in Figure 2. Sound spectra from the 18 cm qing, freely suspended from rubber bands and resting on the silk cushion, are shown in Figure 5. The upper spectrum in each case is recorded when struck, and the lower spectrum 0.5s later. Note that the decay rates are comparable in the two cases, indicating relatively little damping from the cushion. In both cases, the partial radiated by the (4,0) mode has the largest amplitude. Frequencies of the main partials in a 18 cm and a 15 cm qing are given in Table 1, along with their ratios to the fundamental in each case. Note that no harmonic relationship exists among the partials. The pitch of the qing is determined almost entirely by the fundamental (2,0) partial, although the strong (4,0) partial is clearly heard as an overtone.

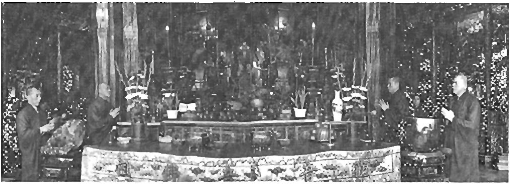

Figure 1. Scene from a Buddhist temple, showing a large qing (right) and a muyu (left).

Table 1. Mode frequencies and ratios in two qing

Mode	18cm qing f_{mn} (Hz)	18cm qing f_{mn}/f_{20}	15cm qing f_{mn} (Hz)	15cm qing f_{mn}/f_{20}
2,0	346	1.00	434	1.00
3,0	953	2.75	1180	2.72
4,0	1751	5.06	2130	4.91
5,0	2691	7.78	3267	7.53
6,0	3748	10.83	4496	10.36
7,0	4644	13.42	6182	14.24
8,0	6255	18.08		
9,0	7363	21.28		

The authors thank Prof. Kuo-Huang Han in the Northern Illinois University School of Music for his enlightening discussions and especially for loaning us the qing used in these studies.

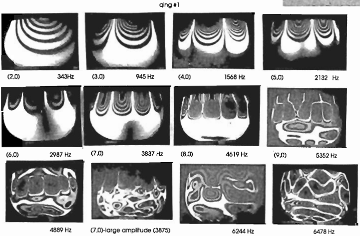

Figure 3. (left) Holographic interferograms of a 18 cm diameter bronze qing, showing modal shapes of the (m,0) modes (top two rows). Modes in the bottom row are not identified, except for the second one, which is the (7,0) mode at a higher amplitude than in the photograph immediately above it.

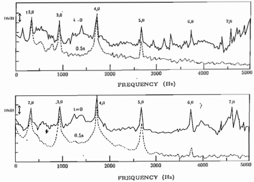

Figure 5. (above) Sound spectra of 18 cm qing freely suspended on rubber bands (upper) and resting on a silk cushion (lower). The upper spectrum, in each case, is recorded at the time of striking, and the lower spectrum 0.5s later.

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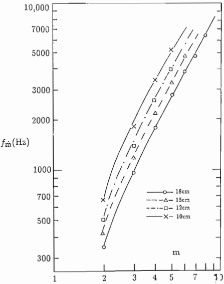
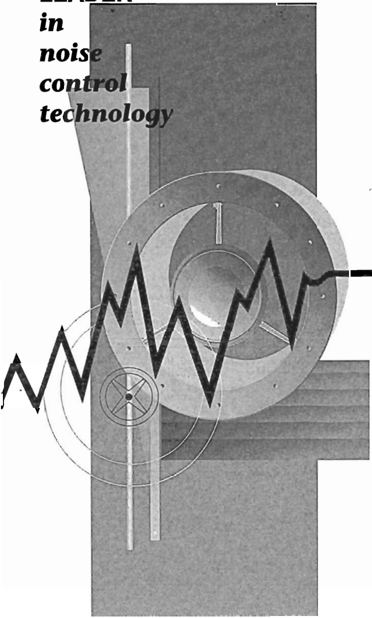
Figure 2. (above) Four qing with diamteters of 18, 15, 12, and 10 cm.

Figure 4. (below) Mode frequencies as a function of m, the number of nodal diameters for qing with diameters of 18, 15, 12, and 10 cm.





Active Control of Plane Wave Noise in Ducts

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ABSTRACT: Active methods of treating plane wave noise in ducts are investigated to identify practical systems. Active attenuators can generally be categorised as suitable for either random noise or periodic noise service. Both categories function as either a sound absorber or sound reflector; this factor requiring greater consideration when dealing with the random noise type. Ways in which some applications affect attenuator performance are discussed.

1. INTRODUCTION

Attenuation of noise using electro-acoustic components dates back to Lueg [1] in 1936. The method utilised a secondary sound to destructively interfere with the primary noise rather than applying a background sound to mask the original. Lueg's electro-acoustic arrangement was extremely simple and lacked application because of significant limitations.

In 1953, Olson and May [2] proposed another approach which reappears now and then in different guises, having aliases like 'virtual-earth', 'near-field' and 'tight-coupled' systems; all exhibiting limited bandwidth, though occasionally accomplishing useful noise reduction. It was not until the 1970–80s that viable active attenuators were proposed; several are described in references [3,4,5,6] and discussed in [7]. These were suited to plane wave sound in a duct, however, with additional complications a few could handle multi-mode noise propagation.

Figure 1 is a simplified block diagram of a basic random noise attenuator for treating plane wave noise in ducts. The principle involves sampling the upstream noise and reproducing it downstream in opposite phase to the original. Microphones and electronics are arranged so that the system only responds to sound which travels in the direction from primary source towards the loudspeaker. Electronics include circuits to make the microphone arrangement directional, and a filter to correct irregular frequency response. The spacing between the microphones and loudspeaker is dictated by the filter's signal propagation delay time.

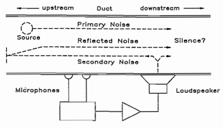

Figure 1. Basic random noise attenuator operating as a sound reflector.

In recent times much work has been focussed on control algorithms that maintain the integrity of the secondary sound. To achieve 20 dB noise reduction, for example, the secondary sound is maintained within 1 dB in amplitude and five degrees in phase over the required bandwidth. Nowadays, accurate control is achieved without great expense, at least for ducts associated with low velocity flows and at noise frequencies beneath the first higher order mode.

Depending on the application, attenuator performance at low frequencies can exceed that obtained with conventional passive silencers design. Active attenuators can be made more compact than a passive attenuator, but some designs only function well in long ducts. Components of active systems can be mounted in the duct walls to avoid interfering with airflow, whereas passive attenuators often have sound-absorbing elements across the duct which introduce undesirable pressure losses.

2. ATTENUATOR TYPES

With respect to functional importance, active attenuators can first be categorised as better suited to either random or periodic noise control. By its nature, the random type can also treat periodic noise, but for this purpose may be the more expensive and less effective of the two types. On the other hand, the periodic noise controller cannot be employed successfully in random noise applications.

The two types can be further grouped in either the sound absorber or sound reflector classes. The former absorbs the primary noise and utilises two or more loudspeakers to produce the secondary sound. Signals applied to the loudspeakers are related in a special way, which necessitates additional electronics. An advantage of the absorber class is that the noise level upstream of the attenuator is not enhanced. Unfortunately it suffers poor power efficiency at frequencies either side of mid-band [8] so may be restricted to low and medium power applications. Olsen classified his near-field system as a sound absorber, however it behaves like a reflector when located inside a duct.

The second group reflects noise back towards the source: in fact the system depicted in Figure1 belongs to this class. Reflectors are less expensive since they only require one loudspeaker. However, where duct acoustic reflections upstream of the attenuator are large, noise attenuation is less than that attainable with the absorbers [9].

Upstream reflections affect a reflecting random noise attenuator in the following manner: referring to Figure 1, half the secondary noise power fed into the duct travels downstream and attenuates the primary noise, while the other half propagates back upstream towards the source. This second half is reflected back by, say, a duct bend or the duct inlet, to augment the original primary noise. Unless the attenuator can entirely cancel all noise propagating downstream, the residual noise rises above that when the upstream reflection is zero.

By design, the absorber type transmits much less power back towards the source, therefore upstream reflections hardly influence attenuator performance.

Of the random noise attenuators, those based on the Swinbanks [4] and Olson [2] concepts receive most attention. Of the periodic noise class, the Chaplin system [6] (sometimes known as the Essex system) is probably best known.

3. PERIODIC NOISE ATTENUATORS

Secondary sound produced by the Chaplin attenuator (Figure 2A) is synthesised using as input a synchronous pulse and the residual noise sensed by a remote microphone. A periodic secondary sound is gradually developed by a microprocessor or digital signal processor to minimise the sound pressure at the microphone. The pulse is often supplied by a transducer which senses shaft revolution. The microphone can be removed from the duct so that it does not sense turbulent pressure fluctuations in the flow. Consequently the attenuator produces a good replica of primary tonal noise and achieves significant noise reduction outside the duct. If necessary, the microphone can be located in the duct to maximise noise attenuation there, but with high flow velocities, turbulence can add substantial irrelevant noise to the microphone signal.

Normal configurations of the Chaplin system place it in the reflector class, although it can operate as an absorber using extra electronics and loudspeaker. Conversion to the absorber class is unnecessary unless it is essential to limit the upstream noise level. However, when operated as a reflector, upstream noise should not increase by more than 3 dB [5].

Another technique creates the secondary noise using specific harmonics of a synchronous pulse. Harmonics are individually adjusted in amplitude and phase, summed, then input to the amplifier. Optionally, a control microphone measures the residual sound to effect harmonic adjustment. The block diagram for the attenuator is similar to that of Chaplin's. This system, less the control microphone, has been applied by Neise and Koopman [10] to reduce tonal noise generated by centrifugal fans. The secondary sound was introduced through perforations in the fan cut-off, using two loudspeakers arranged as a dipole. The arrangement restricted noise within a small region which reduced the number of noise transmission paths.

4. RANDOM NOISE ATTENUATORS

These produce the secondary sound from a sample of the upstream noise. To prevent acoustic coupling between the loudspeaker and microphone, one or the other (or both) is made unidirectional. Figure 2B depicts a Swinbanks attenuator where multiple transducers are connected electronically in a special manner to achieve unidirectionality. The frequency response of the unidirectional configuration is not flat and requires substantial compensation. A variation of the attenuator employs a novel microphone and/or loudspeaker arrangement, described by La Fontaine et al. [11], which avoids such response compensation.

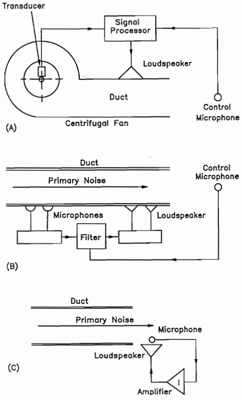

Figure 2. Various active attenuators: (A) periodic type (B) random type; (C) near-field type.

There are several factors contributing to frequency response errors, making signal filtering unavoidable. Nowadays, filtering is performed using a digital signal processor. The processor can self-adaptively adjust the filter coefficients to accommodate variations in component performance and duct conditions such as airflow velocity changes. The latter influences phase alignment of the primary and secondary sounds (these must be displaced 180 degrees). Phase misalignment increases with mean flow velocity, noise frequency and spacing between the primary noise sensing microphone and loudspeaker. Early attenuator filter designs necessitated a large microphone-to-loudspeaker spacing because of long signal processing times, so making the attenuators particularly sensitive to air temperature and flow velocity variations.

A control microphone is usually situated in the duct beyond the loudspeaker to inform the digital signal processor of residual noise. The processor adjusts the filter coefficients for minimum residual noise – usually requiring several iterations before an optimum is reached. Complete noise attenuation is not possible due to influences which confuse the control algorithm and distort the primary noise sample. Spurious microphone signals introduced by turbulence pressure fluctuations in the airflow is one of these influences.

For plane wave attenuators, the upper frequency range is restricted beneath the cut-on of the first higher order duct mode. The lower frequency limit is not so clearly defined, but often depends on turbulence pressure fluctuations detected by the sampling microphone because this develops and becomes more difficult to treat with diminishing frequency. In the Swinbanks system, the lowest residual noise attainable is about the same as the turbulence noise heard by the sampling microphone [12]. Turbulence screens must therefore be fitted to the microphones when the flow velocity exceeds a few metres per second, however, below 100 Hz standard screens rapidly lose effectiveness. The 4.5 octave wide-band Swinbanks arrangement described by La Fontaine and Shepherd [9], when using a unidirectional microphone and loudspeaker, achieved 16 dB insertion loss at 50 Hz without flow through the duct. Later tests using 20 m/s airflow reduced this figure to 10 dB with turbulence screens installed.

If either the microphone or loudspeaker is omnidirectional, duct reflections can adversely affect the performance of random noise active attenuators [9]. Typically, performance can be reduced by about 2–6 dB which may seem tolerable. However, consider the attenuator described above, where in the first instance 16 dB insertion loss was achieved at 50 Hz. Combining the influences of turbulence pressure fluctuations and duct reflections, the same attenuator using an omnidirectional acoustic coupler might only provide 4–8 dB insertion loss at that frequency.

Although the conditions cited above are severe, they are not unusual. It becomes obvious that the viability of an active system, particularly the random noise type, depends very much on individual applications. Factors which determine viability and require measurement include: the duct plane wave acoustic power, the contribution plane wave noise makes to the sound at locations beyond the duct, duct reflections either side of the attenuator, and the turbulence pressure level. Shepherd et al. [13] examine how these factors are taken into account when assessing prospective applications.

5. NEAR-FIELD ATTENUATORS

Near-field attenuators can be categorised as random noise systems, however they operate in a distinctly different fashion from other types. Figure 2C shows a configuration proposed by Olson and May for attenuating random noise emission at a duct outlet. The obvious advantage is that the loudspeaker and microphone are more likely to survive destructive flue gases. Another suggested configuration had the loudspeaker and microphone mounted just inside the outlet.

The Olson 'electronic sound absorber' was initially devised as a zone silencer, so it is understandable that the arrangement, when located externally to the duct, only performs well in the near-field of the outlet, and not in the far-field, apart from a few remote positions at some frequencies. The reason is readily explained using mathematics, however, in essence the negative feedback loop – which involves microphone, acoustic path, amplifier and loudspeaker – tries to achieve zero acoustic pressure at the microphone, but not necessarily elsewhere. Therefore, the effectiveness of this configuration diminishes as the observer moves away from the microphone. La Fontaine et al. [14] explain methods of improving the performance of such systems.

Eghtesadi et al. [5] moved the near-field system well inside the duct while investigating monopole attenuators and called it the 'tight-coupled monopole active attenuator'. Duct loading proved beneficial in that a reasonable bandwidth was achieved and 5–20 dB insertion loss was attained across two octaves with 5 m/s airflow. Noise reduction measured inside and outside the duct should agree at frequencies below the duct cut-on frequency. Though Olson described the arrangement of microphone and loudspeaker as an absorber, it clearly operates as a reflector inside the duct.

These systems are marginally stable. Good noise attenuation can only be obtained when the amplifier gain is set almost to the point where the attenuator bursts into oscillation. There is considerable difficulty maintaining accurate gain since duct acoustics influence the feedback loop.

6. CONCLUSION

Since the 1930s several methods of active noise attenuation have been proposed to treat plane wave noise inside a duct. Many can provide useful levels of noise reduction. Some claim superiority over passive systems in special applications, particularly where low frequency noise is concerned.

Active systems incur less pressure drop in air-ducts and, as a result, smaller fans can be employed. Beneficially, smaller fans produce less noise.

Nowadays the technology to manufacture active attenuators is readily available. Nevertheless, few are available off-the-shelf.

Random noise attenuator performance is more difficult to predict than for passive systems. Major reasons are that the former is particularly sensitive to the mode of sound propagation inside the duct and to duct flow conditions, whereas the latter is not. On the other hand, passive systems cannot provide the level of attenuation attainable with active systems at low frequencies, nor are they as compact. Some active systems can only function in long ducts.

The performance of periodic noise attenuators is more easily determined with the control microphone located outside the duct. Possibly, the periodic type will find more application in the immediate future than the random noise type.

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Active Control of Sound Radiation from Vibrating Structures

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ABSTRACT: Active control of sound radiated by vibrating surfaces is becoming more and more feasible with the development of fast electronic signal processors and novel vibration transducers. In this paper, progress to date is reviewed, the physical mechanisms responsible for control are described and directions for future research are outlined.

1. INTRODUCTION

The design of a system for the active control of sound radiated by a vibrating structure can be divided into two distinct parts: the design of the control source and error sensor system, and the design of the electronic control system. To begin, the required control source number and arrangements for a specified noise reduction over a specified frequency range must be determined. Next the error sensor number and arrangement which will allow the calculated reduction in noise level to be achieved must be found. Finally, the electronic control system must be designed to suit the number of control sources and error sensors. The required reduction in vibration level or sound pressure level at the error sensors will determine the accuracy required of the signal processing hardware (that is, is 16 bits accuracy needed or is it possible to achieve the same results with a much less expensive 10 or 12 bit system?). It is of no use designing the electronic controller before the physical system is properly understood, as it is a fallacy to assume that the electronic controller by itself determines the extent of the noise reduction achieved. In general, it is the number and layout of the control sources and error sensors which are more important, as this determines the maximum achievable noise reduction, which is a benchmark against which the electronic controller performance can be measured. As structures respond to a single frequency or band of frequencies in a way which can be described in terms of resonant and non-resonant normal modes of vibration, it is important that the control sources can control all vibration modes which contribute to the sound radiation in the frequency range of interest. It is also important that the error sensors can observe all of these modes.

When controlling sound radiation from a vibrating structure into free space, it is generally desired to reduce the total radiated sound power rather than the sound pressure at one or two locations. Thus the error sensors have to be arranged so that they can observe the radiated sound power. If microphones are used, this will generally entail the use of more than one, especially if the excitation covers a range of frequencies. A trade off will almost invariably have to be made between the practical number of error microphones and the need to accurately measure the radiated sound power so that the maximum reduction theoretically achievable with the control source arrangement can be approached.

To control the sound radiated by a vibrating structure, either vibration sources to the structure or acoustic sources located in the acoustic medium surrounding the structure, or a combination of both may be used. Similarly, error sensors may be either structural vibration sensors which sense only vibration modes contributing most to the radiated sound or there may be one or more microphones placed strategically in the acoustic medium surrounding the structure.

In this brief paper, the work done in this field up to the present time will be summarised, physical mechanisms responsible for control and techniques and equipment used for control will be described, and directions for future research will be outlined.

2. PROGRESS SO FAR

The first attempts to control sound radiated by vibrating surfaces were concerned with tonal noise radiated by large electrical transformers [1-3]. This work involved the use of loudspeakers placed near the transformer tank to control the noise radiated to one or more community locations. It was found that if global control was to be achieved in all directions, then it was necessary to use a large array of speakers, almost as large as the transformer tank itself. Use of one or two speakers resulted in reduced noise levels in some directions at the expense of increased noise levels in others. Also the angular spread of the directions of reduced levels was generally quite narrow. Although not stated by the authors, the physical reason for the above mentioned behaviour is that to achieve global noise reduction using acoustic control sources, it is necessary for the sources to change the radiation impedance "seen" by the transformer tank. This can only be done by using a large array of control sources. If only one or two small sources are used, areas of reduced noise level are achieved solely by local destructive interference effects at the expense of other areas of increased level where constructive interference takes place. Thus in this latter case, the radiation impedance "seen" by the transformer (and hence its radiated sound power) is barely changed, and as a result the overall radiated sound power of the transformer plus control sources is generally larger than the sound power radiated by the transformer itself when only one or two control sources are used, even though there will be some locations (particularly error sensor locations) where the sound level will be reduced.

It is only relatively recently that there has been a concentrated effort to develop practical control systems to control the sound power radiated by vibrating surfaces. Knyazev and Tartakvskii [5] were the first to investigate the control of sound radiation by using control forces on the vibrating structure. in 1988, Deffayet and Nelson [4] analysed the active control of sound radiated by a finite rectangular panel using acoustic control sources. In 1989 Hansen et al [6,7] compared the relative effectiveness of control forces and acoustic sources for controlling sound radiated by a rectangular panel. Analytical modelling of the active control of sound radiated by a rectangular panels has been undertaken by Walker [8], Fuller [9] and Pan et al [10].

More recently research efforts have focussed upon the control of orthotropic panels [11], the use of piezo ceramic crystals to provide the control forces [12-13] and shaped PVDF (poly vinyl difluoride) sensors instead of microphones to provide the required controller error signal [14-16].

Investigation of the physical mechanisms [6,17-19] involved in controlling sound radiation from a simple vibrating surface has provided an understanding of the complexity of the problem and has also resulted in the determination of the influence of geometric and structural/acoustic variables on the maximum achievable reduction in sound power. This work has led to the formulation of strategies for the optimum design of multi-channel systems for the simultaneous control of a number of sources and error sensors [20-23].

As the use of control forces on heavy structures such as transformer tanks poses problems of generating control forces of sufficient magnitude, work has begun [24] on the use of a thin enclosure, which may be perforated or solid steel, placed around the noise radiating structure. Such an enclosure can be adequately excited using piezo ceramic crystals which are relatively inexpensive ($10-20 each).

3. PHYSICAL CONTROL MECHANISMS

It is of considerable interest to be able to identify the physical mechanisms underlying the active control of sound radiation from a vibrating surface. Only if these are properly understood will it be possible to determine the limitations on the amount of noise reduction which would be achievable with an ideal electronic controller. In two recent studies [18,19], the effect of various parameters such as control source location, error sensor location, control source type (acoustic or force) panel size, structural damping, excitation frequency and panel response type (resonance or forced) has been evaluated theoretically for a simply supported, baffled rectangular panel vibrating at a single frequency and radiating into free space. In one study [18], the primary excitation force was a single point force at the antinodal location of a (3,1) mode (2 vertical nodes and no horizontal nodes). In the second study [19], the primary excitation consisted of four in-phase forces located near each of the four corners of the panel. The general conclusions of each of the studies are similar, so here we will concentrate on the detailed results obtained using only a single point primary excitation force. The panel was excited off-resonance at a single frequency between the (2,2) and (3,1) modal resonances. Figure 1 shows the maximum achievable reduction in radiated sound power as a function of location of a single control force, assuming an ideal electronic controller and assuming that the error sensor(s) can accurately measure the radiated sound power. Even under these ideal conditions it can be seen from the figure that for locations of the control source other than on top of the primary source (which is a trivial case and represents the control source at the centre of the concentrated set contours shown in the figure) the maximum achievable reduction in radiated sound power is 22.4 dB and this will only occur for one location of the control force. Improper location of the control force can result in achievable sound power level reductions as low as 4 dB. The size of the panel depicted in Figure 1 was such that the acoustic wavelength corresponding to the excitation frequency was three times the largest panel dimension.

For this test case, two fundamental physical control mechanisms were identified by calculating modal vibration levels on the panel before and after control. With the single control force located at the right hand maximum of Figure 1, the control mechanism was modal amplitude reduction; that is, the vibration amplitudes of the modes contributing most to the sound radiation were significantly reduced. It was also found that for the simple case considered here, a single error microphone properly located in the far field of the panel was able to provide an error signal which allowed the sound power to be reduced by an amount very close to the maximum possible.

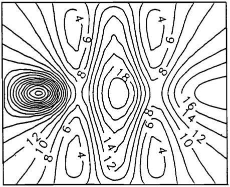

Figure 1. Maximum calculated achievable levels of sound power attenuation (dB) as a function of the vibration source location on the panel.

With the single control force located in the centre of the panel, the dominant control mechanism was found to be that of modal phase rearrangement. That is, the controller rearranged the temporal phases of the radiating modes in such a way as to effectively reduce the overall panel radiation efficiency. This can be better understood if one notes that the total sound power radiated by a vibrating surface is **not** the sum of the powers radiated by each mode. Rather, the modes combine together to provide a particular panel velocity distribution which will be characterised by a particular radiation efficiency. With this control mechanism, it is likely that the r.m.s. vibration levels on the structure will, in some cases, increase under control, even though the radiated sound power will be reduced.

Increasing the panel size so that it was one rather than one third of a wavelength across had a dramatic effect on the results. The maximum reduction in sound power theoretically achievable with a single control source was reduced from 22.4 to 16.9 dB. Also, the modal rearrangement control mechanism was no longer operative, due to the increased panel size.

Increasing the panel loss factor from $\eta = 0.04$ to $\eta = 0.2$ also had a dramatic effect on the maximum achievable reduction in sound power. Apart from the trivial case of the control source on top of the primary source, the maximum achievable sound power reduction was reduced to 8 dB. Also the modal amplitude control mechanism was no longer effective; control was only achieved by a rearrangement of the relative modal temporal phases.

With acoustic control sources, the mechanism responsible for a reduction in radiated sound power was found to be a change in radiation impedance "seen" by the panel as a result of the presence of the acoustic sources. Thus the acoustic sources act to "unload" the panel. Clearly, it is not possible then for a single small acoustic source to provide a significant reduction in the power radiated by a large structure such as an electrical transformer, as such a source could not acoustically unload the transformer. However this does not preclude the control source from providing local areas of sound cancellation at the expense of other areas of increased sound level.

As well as needing to understand the physical mechanisms involved to design an optimum system to control sound radiation from a vibrating surface, it is also necessary to realise that all vibration modes contributing to the radiation must be both controllable by the control forces and observable by the error sensors. Clearly a control force located at a modal node cannot control that mode and equally clearly a vibration sensor located at a modal node cannot provide an error signal for that mode. Also an acoustic sensor located at a minimum point in the radiation field generated by a particular mode may not provide adequate error information for that mode. Thus if a single acoustic error sensor is used to provide a signal proportional to the total sound power radiated by a structure, then it is extremely important that it is located so that it can best measure the required quantity. Measurements made using the simple test arrangement just described indicated that the maximum achievable reduction in sound power, could vary from 11 dB to 22.4 dB (for the optimum location of a single control force) dependent upon the location of the far field acoustic error sensor.

When the noise source is periodic in nature, it may be advantageous to implement the controller in the frequency domain, or in the time domain using less expensive hardware of limited precision. For these alternatives to be properly evaluated, the effect of phase error on the control result should be evaluated in all cases using an analytical model. For the rectangular plate model considered here, it was found that an error of ±1 degree in phase produced 0.2 dB less sound power attenuation and 15 dB less sound pressure attenuation at the error sensor. Thus if the objective is to reduce the overall radiated sound power, the result appears to be relatively insensitive to phase accuracy. However, note that this was for a case where the maximum achievable reduction in sound power was 22.4 dB. If it were 60 or 70 dB, then the sensitivity to phase accuracy would be much greater.

4. CONTROL TRANSDUCERS AND ERRORS SENSORS

In designing an active control system, decisions must be made regarding the type of control sources and the type of error sensors which will be most appropriate.

Acoustic control sources are generally not as effective as vibration control sources and vibration error sensors are generally more difficult to optimise than acoustic error sensors. If vibration control sources are more appropriate, then the type of vibration source must be chosen. Piezo ceramic crystals are ideal inexpensive alternatives for thin structures such as aircraft. They can be bonded to the structure they need to control using epoxy resin adhesives, and when actuated by a voltage up to 150 volts r.m.s., they expand or contract and induce a bending moment in the structure. Piezo ceramic crystals are typically 38 mm to 50 mm x 0.25 mm thick and are very brittle. However other sizes are available and they can be manufactured to conform to curved surfaces. It is unlikely that piezo ceramic crystals would supply a sufficient control force for heavy structures. On the other hand, piezo ceramic stacks, magnetostrictive actuators, elecytrodynamic shakers and hydraulic shakers can provide relatively large control forces, but they all require a backing mass or some other imaginative means of providing a reaction support.

Piezo ceramic stacks consist of many layers of piezo ceramic crystals stacked together to give a large force output. Magnetostrictive actuators are made using a rod made from an alloy of iron and rare earth elements which extends on application of a magnetic field. For it to act as an actuator, it is necessary to provide either a d.c. bias in the driving signal to the electromagnetic coil surrounding the rod, or surround the rod with a high permeability permanent magnet [25]. The latter is the preferred alternative as the former leads to overheating and amplifier problems. It is also necessary to provide mechanical precompression of the actuator as it is extremely brittle and a magnetic field applied to a non-precompression rod will cause it to end and break. The advantage of magnetostrictive actuators lies in their small size and reasonable force/cost ratio.

Electrodynamic and hydraulic shakers should only be used as a last resort for a number of reasons. It is difficult to properly isolate a hydraulic shaker from the hydraulic system, and if it lies between the servo valve and the actuator, much of the alternating energy can be lost in flexing the hose. Also in many cases the noise made by the hydraulic power pack and hydraulic lines may also be difficult to reduce to acceptable levels. Electrodynamic shakers are undesirable for different reasons. They are generally large and heavy and are often relatively expensive.

If acoustic control sources are chosen, then the choice is simply between speaker and horn drivers, the final choice being dependent upon the frequency range and required power output.

An indirect way of reducing the sound power radiated by a vibrating structure, in particular a composite structure, is to embed shape memory alloy wire in the structure which will change the structural stiffness charactertistics on application of a voltage [26]. This in turn will result in a change in the sound radiation characteristics of the structure.

In deciding upon the type of error sensor to be used, there is a choice of two separate classes - acoustic or vibration. Generally acoustic error sensors (microphones) will provide better results if the objective is to control radiated sound; however, in some cases their use may be impractical. In that case the choice is between the use of accelerometers fixed at appropriate locations to the vibrating surface PVDF film which may be appropriately shaped and distributed so that it only measures the vibration associated with the modes which contribute most of the radiated sound power [15]. Clearly, it is not so easy to achieve this result using accelerometers, and as accelerometers are much more expensive, they are the least preferred option. PVDF film is a flexible form of a piezo ceramic crystal. Both can act as actuators or sensors but the force producing capability of the PVDF film is too small to make it a practical actuator. However, as it is far less expensive than piezo ceramic crystals, it is the preferred choice for a sensor. As a sensor, the material produces a voltage or charge output on being stretched or compressed in-plane. Another option for a vibration error sensor, which also can be arranged to only measure the radiating modes, is a fibre optic sensor [27]. However this is more expensive than the PVDF film option and provides similar results.

5. CONCLUSIONS

When designing an active control system to reduce the sound power radiated by a vibrating structure it is useful to undertake a detailed numerical analysis of the problem to determine the number of control sources and error sensors and their optimum arrangement to achieve the required reduction in sound level over the desired frequency range. The achievable control will also depend upon the size of the structure, the modal density, the excitation frequency and the system damping. Only after the physical system has been properly understood will it be possible to optimise the electronic controller. For example, it would be an unnecessary expense to use a multi-channel controller accurate to 16 bits if the maximum achievable noise reduction were only 10 dB and it was relatively insensitive to the control force amplitudes and phases. Unfortunately, at this stage the analytical modelling has been restricted to single frequency problems. Although it could be extended easily to multiple frequency periodic problems, the extension to include random noise over a wide band is not so straightforward. The analytical modelling so far has also been restricted to simple panels, although optimisation procedures already developed for the placement of control sources and error sensors could be used with finite element and/or boundary element modelling for a more complicated problem. Needless to say, the extent of the computations required for multiple control sources and error sensors on a complex structure is horrendous. At least the analysis and understanding of simple structures will provide some guidelines for the design of control systems for more complex structures.

Thus, future work will be directed at extending the analytical modelling to these more complex cases by developing more efficient analysis techniques, or by more efficient use of existing techniques. Future work will also be directed at the development of PVDF sensor shapes to provide the desired error signal and the development of better magnetostrictive and piezo ceramic actuators to provide larger forces. Also more work needs to be done on the analysis and use of thin, solid or perforated, metal shells, driven by piezo ceramic crystals, surrounding a heavy noise radiating structure as a means of controlling the radiated sound.

Finally, development of more efficient control algorithms and less expensive multi-channel electronic controllers must be undertaken. This work is not discussed here except in the context of the accuracy of the required controller hardware being dependent upon the physical system being controlled. This accuracy requirement can only be determined by detailed analysis of the physical system to be controlled or by extensive trial and error tests.

In conclusion, it may be assumed that there now exists the technology to control periodic sound radiation from vibrating structures. However, a general system does not exist; each system must be designed separately for each structure to be controlled.

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The Use of Waveguides in Acoustic Emission Monitoring Projects

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ABSTRACT: The CSIRO has become involved in three long term acoustic emission monitoring projects. The conditions associated with the structures necessitated the use of waveguides to transfer the transient elastic waves from the surface of the structure to the transducer.

A variety of different waveguide and transducer combinations evaluated for three separate applications are discussed. The various configurations were investigated in both laboratory and field situations so that some idea of the relative attenuations was available together with data on the resultant waveforms.

1. INTRODUCTION

Acoustic emission (AE) refers to the transient elastic waves that are generated in a structure by the growth of defects associated with some form of stressing (mechanical, thermal, chemical etc.). The elastic wave pulses propagate through the structure until a boundary surface is reached, when a surface wave is generated which can be detected by a surface mounted transducer. The use of continuous or quasi-continuous AE monitoring requires that the AE transducers be both in an environment which will not cause excessive deterioration in their performance and be readily accessible for maintenance if required. These restrictions will in many cases only be met if the transducer is attached to the end of a waveguide. The use of a waveguide means that there will be some modification to the nature of the detected pulse, since new factors, including the nature of the coupling of the waveguide to the structure being monitored, the characteristics of the wave propagation through the waveguide, and the coupling of the end of the waveguide to the transducer, will be introduced.

2. STRUCTURES BEING MONITORED

The first structure is a new Platformer pressure vessel operating in a refinery at temperatures up to 580°C; the second is a new large Cryogenic storage tank operating at -40°C; and the third is a fibre reinforced containment vessel operating at 96°C.

The Platformer is a large pressure vessel operating at high temperatures and pressures. The unit reforms the long chain petrol molecule using platinum as a catalyst to produce a fuel with a higher ignition efficiency. The structural integrity of the Platformer was monitored from the manufacturing stage forward so that long term operation with limited maintenance shutdowns will be possible. The Cryogenic tank project will use continuous structural integrity evaluation to provide long periods free of major maintenance and to allow operation of the vessel for extended periods of time (in excess of 50 years) by means of a systematic acoustic emission monitoring program. The fibre reinforced plastic vessel project involves the monitoring of a number of vessels from new for periods in excess of 20 years to provide continuous integrity evaluation and defect location.

3. WAVEGUIDE GEOMETRIES

3.1 Metal Structures

The waveguides used in metal structures are commonly manufactured in mild steel if this material is acceptable; otherwise stainless steel is used, which is superior with regard to resisting corrosion but is inferior acoustically, in that it has a larger attenuation than mild steel.

Waveguides are generally circular and tapered to a blunt point which is then held against the structure being monitored by the use of some hold down device to provide pressure on the blunt point. This form of attachment of the waveguide to the structure is satisfactory for short term investigations or for investigations that are only carried out infrequently and the transducer assemblies removed after each monitoring period.

Permanent monitoring requires more substantial means of attaching waveguides, with the further proviso that the attachment of the waveguide does not itself introduce any lessening of the mechanical integrity of the structure. A change in the geometry of the waveguide so that it flares out at one end for easy attachment by adhesives is one possibility and another is to use some form of welding. Two forms of welding which can be employed and have been investigated thoroughly as part of field monitoring projects are the drawn arc and the electric arc welding techniques.

3.2 Fibre Reinforced Plastic (FRP) Structures

Metal waveguides cannot be readily attached to FRP structures and the alternative is to construct a waveguide using FRP techniques so that such a unit can be readily bonded onto the structure. The waveguide assembly is essentially a bunch of parallel glass fibres which are held together by an appropriate resin and splayed out at both ends to allow for attachment to the vessel and for coupling of a transducer.

4. EVALUATION

4.1 Metallographic

The nature of the junction where a steel waveguide is bonded to a steel vessel was investigated for both the drawn arc and electric welding using standard metallurgical techniques of macro and micro examination and hardness measurements.

4.2 Acoustic Emission

The acoustic properties of the waveguide-sensor assembly were evaluated by observing and comparing the waveforms detected using a standard pencil - lead break for the transducer directly on the surface being monitored and for the same transducer attached to the waveguide. These observations provided data on pulse modification and attenuation associated with the use of a waveguide.

5. RELEVANT THEORY

The nature of elastic wave propagation in a waveguide is fairly complicated, but some idea of the factors affecting the waves and the total response of a waveguide/transducer assembly can be obtained. Many modes of wave propagation are possible and the modes are dispersive (velocity of propagation depends on the frequency) particularly at lower frequencies. However, as the frequency increases, the phase velocity of these complex longitudinal waves asymptotes to the velocity of Rayleigh waves as described by Redwood [1]. The modes exhibit a cut-off phenomenon (phase velocity becomes infinite at a lower limiting frequency) which means that there is a degree of high-pass filtering action associated with the use of waveguides. Plots of phase velocity as a function of the non-dimensional parameter of the radius of the rod divided by the wavelength indicate that for values of this parameter between 1 and 2 many of the modes have values for the velocity of propagation which are approaching the Rayleigh wave velocity. Considering a steel waveguide and taking a velocity of propagation of 4,000 m/sec and a radius of 0.005m (5mm) the value of frequency that make the non-dimensional parameter unity is 800kHz, so that for frequencies less than this value the wave propagation will most certainly exhibit dispersion.

However waveguides are not necessarily of uniform cross-section and also have such items as disks of a larger cross-section attached to one end so that a transducer may be coupled to the assembly. The analysis of these more complicated configurations is easier to handle if electrical transmission line analogies are employed as described by Harris [2] and Skudrzyk [3]. A free cylindrical rod has many resonances and anti-resonances which will produce a complex response but as the frequency rises there is so much overlapping of these resonant responses that the response tends to become almost uniform. If k is the wavenumber (2π divided by the wavelength) and l the length of the waveguide, then the uniform response occurs for values of the non-dimensional parameter $k.l$ between 10 and 100 (the value will depend on the damping in the system). Considering again a steel waveguide with a velocity of propagation of 4,000 m/sec and a length of 0.4m (400mm), then for $k.l = 10$ the value of frequency is approximately 16 kHz. The effect of a plate on the end is to excite higher order modes of propagation in order to satisfy the boundary conditions, and, since these modes are very dissipative, there will be energy loss. The effect on the resonant response of the assembly is to cause a small apparent increase in length so that the frequencies are slightly lowered. Analysis of the situation using transmission line analogies, shows that the other effect of a disk, apart from the generation of higher order modes, is to produce a mass loading, so that if f_0 is the resonant frequency of the entire system, f_1 the resonant frequency of the waveguide without the disk and M_1 its mass, and M_2 the mass of the disk, then the following relationship holds:

$$f_0 = M_1 f_1/(M_1 + M_2) \qquad (1)$$

In summary, the available theory indicates that the effect of a waveguide is to produce a high pass filtering action with the velocities of propagation exhibiting dispersion which is most pronounced at the lower frequencies, but the overall response of the system will not be contaminated by a complicated resonance/anti-resonance pattern at the high frequencies.

6. RESULTS AND DISCUSSION

6.1 Steel Structures

The range of steel structures monitored is wide and varied, and this study is restricted to pressure vessels operating at elevated cryogenic temperatures.

6.1.1 Platformer

Since this vessel operates at high temperatures and pressures, it was decided that the transducer assemblies to be used in investigating the performance of the Platformer would be removed after each evaluation, and so welded or permanent waveguides were not considered. The arrangement used was a steel waveguide with a flat contact area which was pressed on the metal surface through a hole cut in the insulation. The flat end was pushed onto the surface with a pressure of approximately 600 kPa using a spring loading arrangement held onto the surface by magnets. The free end of the waveguide passed into a specially designed assembly so that it was in intimate contact with one side of a piezo-ceramic disk reducing interface attenuation. An evaluation of the waveguide arrangement indicated a 5 to 6 dB attenuation when using the waveguide compared to fixing the transducer directly onto the vessel surface. The high operating temperature of the vessel made the use of waveguides essential. Tests using waveguides ranging from 3mm up to 20mm diameter indicated that the size of the waveguide to vessel contact would cause an attenuation of up to 10dB. A flat contact surface between 6 and 10 mm diameter was found to be the best configuration with an attenuation of 5dB. Some investigations were carried out using sharp pointed waveguides and the results of this work gave an attenuation of 20dB for the same diameter waveguide. The use of the integral waveguide making direct contact with the piezo-ceramic disk reduced the signal transmission loss and allowed a more realistic wave-form analysis technique to be used.

6.1.2 Cryogenic tank

Since it was a requirement of the program that the acoustic emission monitoring be used for 50 to 100 years, the details of the fixing of transducers to the vessel surface were of prime importance. The vessel will have a 100 mm thick sprayed insulation over the total exposed surface, which is one of the reasons why it was necessary to use waveguides. In many applications adhesives, greases, and magnets can be used to hold waveguides onto the surface being monitored, but since in this project very long monitoring times were involved, welding was considered to be the best method to attach the waveguide to the structure. Test welds were made using two methods, drawn arc and electric arc. A major concern when using welded waveguides relates to the introduction of micro cracks in the parent material as a result of the rates of localised heating and cooling during the welding of the waveguides to the parent metal. A series of test welds were made using the proposed waveguide configuration and a number of variations in the welding technique. Initial visual inspection was used to divide the samples into "good" and "poor" classifications. Metallographic evaluation, using both macro and micro examination, indicated adequate fusion and a clean microstructure in the "good" samples, while the "poor" samples exhibited some lack of fusion at the weld interface, a reduced heat-affected zone which did not produce a good weld interface, some inclusions in the weld and a small shrinkage crack in one sample, and a microstructure exhibiting some retained high temperature phase. This indicates the importance of the good welding practices required in attaching waveguides to any structure. Summarising,

* The attenuation and wave propagation studies consistently identified a small decrease (1-2dB) in attenuation when comparing "good" with "poor" weld conditions, though this could be within normal experimental variability.
* The main concern relates to the need for good welding techniques and adequate fusion between waveguide and parent metal to prevent the introduction of cracks within the microstructure.
* The most efficient waveguide diameter was found to be 6mm to 10mm, with attenuation increases up to 5dB being recorded for sizes outside these limits.

While attenuation variations which are considered to be within acceptable experimental limits were recorded in all tests, it was possible to identify significant changes in the detected waveform associated with only very minor variations in the waveguide to plate weld. Some samples were tested in the original form and then sectioned so that up to 50% of the weld and waveguide was removed. While this again resulted in a minimal 0.5-1dB variation in the detected pulse peak amplitude, there was a significant change in the detected waveform indicating changes in the pattern of reflections and the nature of higher order mode generation associated with the propagating waves.

The final waveguide design used a threaded plate which was attached to the outer end of the waveguide with the transducer attached to this plate using a thin layer of adhesive. This configuration was chosen after experimental evaluation of a number of designs, bearing in mind that later tests of this vessel should not be restricted to the use of the present transducers, preventing the application of future advances in transducer and monitoring technology. The extra attenuation associated with the use of this waveguide assembly for a range of configurations was 10dB. The best results were obtained using a waveguide manufactured from solid bar. The waveguide which was welded to the transducer base gave an additional 1dB attenuation, while the threaded connection between waveguide and transducer base gave an additional attenuation of up to 5dB without the use of couplant and consistently 3dB with the use of a couplant.

6.2 Fibre Reinforced Plastic Structures

These structures pose different problems from steel vessels since the waveguides cannot be attached by welding or by pressurised blunt points. The ideal waveguide for this situation would be made from FRP so that such a structure could be readily bonded to the surface using an appropriate resin, ensuring that the differential expansion between waveguide and vessel due to heating/cooling and mechanical stressing is minimised. However, geometrical discontinuities could give rise to significant problems with the structural integrity of the FRP structure and would modify the detected acoustic emission waveform.

A glass-rich waveguide was constructed using many parallel strands of fine glass held by resin which was then cured. The final assembly flared out at the ends, enabling bonding at one end to the vessel and the production of a flat surface at the other end to which the transducer could be attached. Such an assembly gave an extra attenuation of 30 dB on average compared to the transducer fixed directly onto the vessel surface. Since it was a requirement that transducers be left on the hot FRP vessel for operating periods of 3 to 4 weeks, it was not possible to fix the transducers directly onto the vessel surface. A number of tests were made using different commercial and CSIRO transducers, with the transducer being calibrated before and after a period of up to 4 weeks on the hot operating vessel. In all cases a loss in transducer sensitivity was measured within a range of 30 to 50dB, indicating that any realistic monitoring program in any hot and/or hostile environment requires the use of waveguides. Intermediate transducer calibration indicated that some commercial units exhibited significant loss of sensitivity after 5 hours at operating conditions less than 100°C.

7. CONCLUSIONS

The need for long term, permanent and repeated acoustic emission monitoring requires that the data obtained and used in structural integrity evaluation be both valid and reproducible. With the increasing use of acoustic emission as a realistic on-line monitoring technique, it is essential that reproducibility and compatibility of data be assured so that the interpretation of results is an on-going process with a high degree of confidence.

There is a need to ensure that correct techniques are used for the selection and operation of equipment to be used and the interpretation of results, but of equal importance is the validity of the recorded data. This work has shown that it is essential to use waveguides in many industrial monitoring projects, and the attenuation resulting from the use of different transducer configurations must be measured. It is also necessary to achieve a uniform equipment sensitivity over the total structure being monitored, by ensuring that the transducers used are within prescribed sensitivity bands and that there is reproducibility in the coupling techniques used to fix the transducer to the structure.

8. ACKNOWLEDGEMENTS

The work reported has been conducted over some years, and the bulk of the work was supported by the CSIRO, but some significant recent work has been done as part of a number of different monitoring projects supported by the Australian Oil Refineries, TransBos Australia Joint Venture, BHP-Utah Minerals Australia, Erico Australia and the CSIRO. The technical and financial support given by all groups is acknowledged and appreciated.

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Acoustical Activities Around Sydney

Prepared by John Dunlop School of Physics University of N.S.W.

1. INTRODUCTION

Acoustical activities in Australia are centred mainly in Government laboratories, universities, industrial laboratories and in acoustic consultancies. A brief listing of some of these activities and personnel particularly in the Sydney area, is provided for the benefit of delegates attending the 1991 InterNoise and Westprac IV Conferences.

2. NATIONAL ACOUSTIC LABORATORIES

126 Greville St., Chatswood. Dennis Byrne, Research Director

The National Acoustic Laboratories' (NAL) Hearing Services Program provides, through a network of almost 50 Hearing Centres, hearing aids and audiological services to eligible clients who constitute about two thirds of the population requiring such services. The program is supported by the activities of a central Laboratory which also provides broader community services.

Acoustic activities involve research, services, and the provision of extensive special acoustical test facilities.

Research includes hearing aids and their application; speech communication; noise and its effects on humans, particularly the prevention of hearing loss. Related activities include the development and design of acoustic and electro-acoustic devices and measurement systems.

Prevention services include noise surveys, expert advice on hearing conservation, audiometric screening and audiometric calibration.

Special Acoustical Test Facilities include four anechoic rooms (one large, two medium and one small), two reverberation rooms and a large horizontal plane wave tube. As well as being used by NAL, these facilities and associated equipment may be hired for research, development or testing purposes. Acoustical measurement services also available.

3. DEFENCE SCIENCE AND TECHNOLOGY ORGANISATION

DSTO (Sydney), Pier 17-18, Jones Bay Rd., Pyrmont, N.S.W.
Ian Hagan, Chief.

The Sydney laboratories of DSTO undertake extensive research and development in underwater acoustics and signal processing. Present acoustic activities are centred on:

- ocean environmental acoustics • high and low frequency sonar
- side scan sonar • acoustic vision

Publications include: Hall M.V. and Irvine M.A. "Applications of adiabatic mode theory to the calculation of horizontal refraction through a meso scale ocean eddy", J. Acoust. Soc. Am. **86** 1465-1477 (1989)
Cato D.H. "Sound generation in the vicinity of the sea surface: Source mechanisms and the coupling to the received sound field", J. Acoust. Soc. Am. **89** 1076-1112 (1991)

4. C.S.I.R.O.

4.1 C.S.I.R.O. and Australian National University

Canberra, A.C.T. Dr. N.H. Fletcher

Individual projects concentrated on the physics of music and vocal systems in insects, birds and other animals. There is also some work in progress on non-linear vibrations and on ultrasonics.

Publications include: Fletcher, N.H. and Rossing, T.D. "The Physics of Musical Instruments" (New York: Springer-Verlag 1991)
Fletcher, N.H. "Acoustic Systems in Biology" (New York: Oxford University Press, in press)

4.2 Applied Physics Division

National Measurement Laboratory, Bradfield Rd., Lindfield.
Dr. K. Hews-Taylor, Research Program Manager

The Acoustics and Mechanics Program supports a range of research and standards activities in acoustics, vibration and ultrasonics. This Program at present is working in five main areas:

- The development of Australia's national standards and a calibration facility for ultrasonic transducers;
- the design and construction of ultrasonic transducers for medical and engineering diagnostic uses;
- design and fabrication of surface acoustic wave (SAW) devices for use in chemical and biochemical sensors;
- NDT evaluation of adhesively bonded joints;
- ultrasonic propagation and scattering in biological and industrial suspensions.

Publications include: Chan H., Drew P., Chivers R. "Standards for medical ultrasound and transducer calibration at the National Measurement Laboratory", Acoustics Australia 1931-36 (1991)
Collings A.F. "Ultrasonics - a useful tool in biophysical and biomedical research", Acoustics Australia 1937-41 (1991)

4.3 Minerals and Process Engineering Division

Lucas Heights Research Laboratories, New Illawarra Rd., Lucas Heights.
Dr. R.W. Harris, B. R. Wood.

Extensive expertise on acoustic emission and signal processing is available to industry on a consultancy basis.

5. UNIVERSITIES

5.1 University of Sydney

Department of Architecture and Design Science
A/Prof F. R. Fricke, Head

The Department undertakes research on a number of environmental aspects of buildings. In the case of acoustics this interest is liberally interpreted to mean both the external and internal environments of buildings as well as the more fundamental issues of the radiation and propagation of sound.

At present there are three major areas of acoustics research in the Department:

- Environmental noise prediction assessment and control
- Building acoustics • Perceptions of sound

Publications include: Wu Qunli and Fricke F.R. "Determination of Blockage Locations and Cross-sectional Areas by Eigenfrequency Shifts", J. Acoust. Soc. Am. **87** 67-75 (1990)
Wu Qunli and Fricke F.R. "Determination of the size of an Object and its Location in a Rectangular Cavity by Eigenfrequency Shifts", J. Sound Vib. **143** (2) (1990)

5.2 University of New South Wales

School of Physics, Department of Applied Physics
Dr. J. I. Dunlop, Head

The Department of Applied Physics supports a program of research, development and consultancy in the areas of non-destructive testing, mechanical properties of materials and underwater acoustics.

Publications include: Dunlop J.I., "Measurement of acoustic attenuation in marine sediments using an impedance tube", J. Acoust. Soc. Am. **90** 1991 in press
Dunlop J.I., "Non-linear vibration properties of felt pads", J. Acoust. Soc. Am. **88** 911-918 (1990)

School of Mechanical and Production Engineering
A/Prof K. Byrne, R. Randle, R. Overall

Fields of endeavour of interest to the Noise and Vibration Group include vibration and acoustics, sound fields in irregular enclosures, signal processing and analysis.

Publications include: Byrne K.P., and Kelly D.W., "Predicting the reactances of irregularly shaped Helmholtz resonators by the finite element method" Acoustics Australia **15** 21-26 (1987)

Australian Defence Force Academy, Canberra
Dr. Joseph Lai, Head

Six academic members of the Department of Mechanical Engineering and six research staff participate in the activities of the Acoustics and Vibration Centre. Research is concentrated in the following areas: Applications of the Sound Intensity Technique; Signal Processing; Noise Control of Machines; Vibration Analysis Techniques and Control.
Facilities: Anachoic chamber meeting ISO 3745

6. ACOUSTIC CONSULTANTS

There is a large and active group of acoustic consultants practicing in Sydney and surround areas. Thirty three are listed in the Sydney yellow pages telephone directory and many of these belong to the Association of Australian Acoustical Consultants, a national organisation which oversees their practices. Also listed in the yellow pages are many suppliers of acoustic products under "acoustic materials and services".

ACT

July Technical Meeting

In the last decade, the technique of modal testing has been developed providing a very powerful tool for measuring the vibration characteristics of structures. **Dr Hugh Williamson**, Senior Lecturer from the Department of Mechanical Engineering at the Australian Defence Force Academy explained the technique and discussed some practical applications which include investigations of both large and small objects including aircraft, machinery structures, automobiles and musical instruments. This was a joint meeting with the Mechanical Engineering Branch of the Institution of Engineers.

Modal testing and analysis enables the natural frequencies, mode shapes and damping characteristics of the object to be determined. It is used to assess the vibration response characteristics of existing structures and in design development. The detailed information enables the appropriate vibration control measures to be adopted. The technique relies on advanced signal analysis and vibration analysis software. The presentation was followed by a demonstration of the measuring equipment and the analysis procedures using graphic software. This demonstration was held in the facilities of the Acoustics and Vibration Centre located within the Department of Mechanical Engineering.

August Technical Meeting

Aircraft noise and flight path monitoring systems have recently been installed at Sydney and Brisbane airports. **Leigh Kenna**, Chief Engineer, Environmental Monitoring in the Civil Aviation Authority (CAA) described the operation of the two systems and their relationship with the system in Canberra. The systems incorporate 8 and 4 individual monitors respectively, with the noise level data transmitted via telemetry to the control centres at the airport and to CAA headquarters in Canberra.

The system in Sydney is currently being extended by the addition of 3 monitors at Badgery's Creek to monitor environmental noise before, during and after construction of the airport.

Mike Evenett , from CAA Environmental Monitoring, demonstrated how the noise level from each aircraft flyover could be identified and correlated against the airport radar data for identification. He also showed the use of the sophisticated software to analyse and store the data and to prepare the reports.

Marion Burgess

SOUTH AUSTRALIA

David Bies Prize

In recognition of the contributions of David Bies to the science and practice of, and education in, acoustics the South Australian Division of the Australian Acoustical Society has established a Prize to recognise outstanding contributions in these areas. The prize is available each year and may be awarded to a member of the Society based in SA or a member who has made a meritorious contribution to acoustics in SA. If no nomination is considered to merit award of the prize, it will not be awarded for that year.

Nominations for 1992 close on June 30 1992. Persons may nominate their own work or the work of others. A person considering making a nomination should consult the Chairman of the SA Division, Bob Boyce, tel (08) 207 1823.

* * *

STANDARDS

Standards Australia will host the proceedings of the the **International Organisation for Standardisation** Technical Committee 43 (ISO/TC 43) meetings to be held in Sydney between 5 and 12 December 1991.

The work of ISO/TC 43 is divided between two secretariats which organise Working Groups responsible for preparation of the new standards and revision of the existing. Secretariat 1 is devoted to the broad acoustic application covering such fields as: noise from various vehicles, machinery and equipment, occupational noise hazards, characteristics of noise sources, sound power measurements, and methods of noise measurement and assessment etc. Secretariat 2 concentrates on building acoustics with subjects such as: reverberation in auditoria, absorption coefficients of building materials, sound attenuation by various construction elements, rating of measurement results, flanking transmission etc.

The parallel work to ISO Secretariats and Working Groups is performed at **Standards Australia** through the variety of Acoustics Committees located with the Environment and Safety Group. The Committees are constituted from the multitude of acoustics and vibration experts drawn from governmental and statutory authorities, industry, users and academics. They are organized so as to facilitate efficient preparation of Australian Standards and communication between ISO and Standards Australia. The current committees (with Chairperson) are as follows:

AV/1 -	Acoustics/vibration terms, units and symbols (Mr K. Cook)
AV/2 -	Instrumentation and measurement techniques (Dr K. Hews-Taylor)
AV/2/1 -	Acoustics and vibration data acquisition systems (Mr R. Piesse)
AV/3 -	Human effects (Dr V. Bulteau)
AV/3/1 -	Hearing conservation (Mr L. Challis)
AV/3/2 -	Hearing protection devices
AV/3/3 -	Audiology (Mr R. Piesse)
AV/3/4 -	Noise on ships and offshore platforms (Mr K. Murray)
AV/4 -	Architectural (Prof F. Fricke)
AV/4/1 -	Sound transmission in buildings and of building elements (Mr N. Gabriels)
AV/4/2 -	External envelope attenuation (Prof. A. Lawrence)
AV/5 -	Community noise (Mr W. Davern)
AV/5/2 -	Aircraft and helicopter Noise (Mr L. Kenna)
AV/5/3 -	Road traffic noise (Prof. A. Lawrence)
AV/5/5 -	Noise in harbour and river waterways (Mr K. Murray)
AV/6 -	Machinery noise (Dr L. Koss)
AV/7 -	Noise from office and household equipment (Mr L. Challis)
AV/8 -	Vibration and shock instrumentation (Dr L. Koss)
AV/9 -	Vibration and shock application (Mr W. Neville)
AV/10 -	Vibration and shock human effects (Prof. E. Betz)

Standards Australia committee members, through participation in the work of ISO, are instrumental to Australia's input to the global standardization. This is reflected in the Standards Australia policy on adoption of the international standards from ISO as a preferred way for publication of the national Australian Standards. The policy is in agreement with the **General Agreement on Tariffs and Trade** (GATT) Standards Code.

The aim is to ensure that the standards produced by one nation, or a block of nations, do not act adversely to restrict trade rights and opportunities of other nations. GATT was created through the realization among nations that it was possible for any nation to unfairly restrict trade (in their own benefit) by means of standards. It was further realized that in the world trade it was no longer tenable that products of one nation should not harmonize (receive equal treatment) with those of other nations.

The GATT Standards Code agreement came into force on 1 January 1980 and requires signatory nations to abide by certain practices, which may be identified as follows:

1. National Standards will be based on International Standards;
2. International Standards will be used if such exist and applicable;
3. When nations develop their own standards they will do so openly, and will publicise the fact for other nations awareness;
4. There will be mutual acceptance of other nations test method and certification.

In this high spirit of cooperation and global understanding the ISO meetings in Sydney will be held with the full support of the Australian acoustical fraternity.

Mark P. Potocki

* * *

WORKSAFE AUST.

A phone survey has revealed that Worksafe Australia's **Noise Management at Work Control Guide** (reviewed in Acoustics Australia Vol 19 No 1) has been well received by the 42 organisations which have purchased it. Responses ranged from "outstanding" and "impressive" to "very useful" and "good and concise". The core section of the Guide - the 5 step noise management program - was one of the most popular parts, along with the modules on such topics as staff training and cost benefit which were highly rated.

Respondents' descriptions were positive, describing the guide most frequently as comprehensive (64%), informative (41%) and easy to understand (35%). Nearly two thirds of the buyers had already used, or said they intended the use, the Guide to implement noise management programs. A majority reported they would use it to carry out a wide range of projects, including workplace audits, noise surveys and staff training. One company setting up a program found the Guide useful in pinpointing deficiencies in existing programs and modifying or redesigning them where necessary. Worksafe Scientific Officer, Dick Waugh, said the results confirmed the value of research and consultation on which the Guide was based.

Further information about the Guide, or other aspects of Worksafe's noise management resource package: Dick Waugh, Worksafe Aust, GPO Box 58, Sydney 2001, tel (02) 565 9555

* * *

FASTS

The Society is a member of the **Federation of Australian Scientific Societies (FASTS)** and in a recent newsletter the President of FASTS, Tony Wicken, has stated that: *FASTS' existence has coincided with a stronger realisation in the Government of the importance of Science and Technology. We have seen some changes but it is generally felt that we need to be more pro-active than reactive.* These feelings were endorsed at a meeting of Professional Status on 30 August. It was considered that FASTS had a role to act on behalf of the many societies which did not individually have the resources to participate effectively in the procedures which will be necessary to establish competency standards. Communication with the various bodies set up by the Government would be required. FASTS is organising the 1991 meetings for the National Science and Technology (Budget) Analysis Group (NSTAG). It has provided a submission to the Australian Science and Technology Council (ASTEC) and to all members of Federal Parliament.

INTERNOISE 92

INTERNOISE 92 will be held in Toronto, Canada from 20 to 22 July 1991. This will be the twenty first in the annual series of international conferences on noise control engineering. While technical papers in all areas of noise control engineering will be considered for presentation, the conference theme is *Noise Control and the Public.* The call for papers has been released and abstracts of proposed papers, with the Internoise 92 Abstract Cover Sheet, are required by 10 January 1991. The manuscripts of papers will be required by 15 April 1992.

Further information: INTERNOISE 92 Secretariat, PO Box 2469, Arlington Branch, Poughkeepsie, NY 12603, USA Fax (914) 473 9325

* * *

New Studios for ABC - Claiming to be the "most advanced acoustic space in the southern hemisphere", the new ABC Ultimo Centre in Sydney presented a formidable challenge for engineers, architects, acoustic consultants and builders. The $150 million complex replaces eleven buildings formerly occupied by the ABC. The complex includes a large hall, studios, control rooms and recording booths for Radio National, 2BL and 2JJJ as well as offices and conference rooms. The Eugene Goossens Hall, 31m x 20m x 13m, is designed as the "home" for the Sydney Symphony Orchestra and is the largest recording studio in Australia. Music performed in the hall is expected to be recorded to CD standard.

The need for superior acoustic properties in most of the complex called for innovative wall and ceiling constructions as well as sound absorbing treatments. Extensive use was made of Bradford Insulation Acoustic Grade Rockwool, Gyprock Fyrecheck and Fibreboard. The final acoustic performance checks were conducted by Arup Acoustics (UK) and included hiring a train to run along the track near the complex late at night.

* * *

ARL

Acoustic Research Laboratories Pty Ltd is a new, wholly Australian owned and operated, high technology instrumentation manufacturer. The company designs and manufactures innovative systems-engineered measuring and control products for environmental and industrial noise and vibration monitoring. ARL has been involved in extensive product development for almost two years and the fruits of that effort are only now becoming available as production instruments. In addition to providing a standard product range for noise and vibration measurement, ARL offers its services as a custom design and production facility for special purpose monitoring systems. The directors of the company are **Dean Gillies, Robert Fitzell and Don Craig.**

Further information: Acoustic Research Laboratories, 169A Pacific Highway, Hornsby, NSW 2077, Tel/Fax: (02) 476 4198

* * *

Companies Amalgamate

dB Metal Products Pty Ltd and Noiseal Products Pty Ltd have joined forces to provide a wider range of products and services. The company will trade as dB Metal Products Pty Ltd under the directorship of **John Payton** and **Keith Porter**. The company plans to manufacture Noiseal doors, windows and industrial silencers in addition to the successful dB panel systems for enclosures, air handling units and quiet booths.

Further information: Factory 1, 21 Green St, Doveton, Vic 3177, Tel (03) 793 2340, Fax (03) 794 5193.

PEOPLE

New Members

•Interim Admissions

We have pleasure in welcoming the following who have been admitted to the grade of Subscriber while awaiting grading by the Council Standing Committee on Membership.

New South Wales
Mr B J. Clarke, Mr P G Knott, Mr J Xie

South Australia
Mr K Payens

Victoria
Mr C J D'Rozario, Dr K Legge, Dr E A Lindqvist, Mr P McMullen

•Graded

We welcome the following new members whose gradings have now been approved.

Subscriber
New South Wales
Mr G J Gannon, Mr N A Wilkinson

Member
New South Wales
Dr J C S Lai (ACT), Mr A I Zelnik

Queensland
Mr A C Monkhouse

South Australia
Mr W A Reflinski

Western Australia
Mr P R Baster, Ms P Gabriels, Mr T S Saw

* * *

It is with pleasure that we welcome **Warburton Franki** as a sustaining member of the Society.

* * *

We were sorry to hear of **Paul Dubout's** bad fall in which he sustained facial damage. Paul's retirement has not been the happiest of times. We are grateful to Paul for acting as one of our Assisting Editors, but, under the circumstances he has had to relinquish that position. We wish Paul all the best for the future.

BOOK REVIEWS

HEARING - AN INTRODUCTION TO PSYCHOLOGICAL AND PHYSIOLOGICAL ACOUSTICS

Stanley A Gelfand

Marcel Dekker Inc, New York, 1990, pp 535, Hard Cover ISBN 0 8247 8368 9. Australian Distributor: DA Books, PO Box 163, Mitcham, Vic 3132. Price A$71.25

This book is an ambitious attempt to introduce the reader to the sensory modality of hearing - a sense ranking in importance with vision, in the conduct and enjoyment of life for most people. In this second edition, the book deals with the physical auditory system and the phenomenon of auditory perception, with reference to anatomy, function and the research procedures employed by investigators in the various associated fields. Thus, from the interaction of sound pressure with the pinna and the eardrum to the activity of single nerve cells in the auditory cortex, the author discusses anatomy, neurophysiology and psychophysics.

The presentation of the often complicated and subtle concepts of this discipline is deliberately informal. In general, Gelfand provides explanations in plain language, without recourse to overly-technical jargon or formalism. His approach reflects his acknowledged inspiration derived from his interaction with students and it is tutorial-like. Thus, he discusses the historical development of the contemporary ideas held by research workers, even to the extent of explaining outmoded interpretations of earlier research and how they have come to change. Throughout the book, specific research results are discussed for their role in the development of hypotheses. The reader is given a good sense of the ongoing detective work that is auditory research. Each chapter has a bibliography of the literature quoted and some specific recommendations are given for wider or follow-up reading on some issues.

The text is liberally illustrated as an aid to explanation. Figures predominantly are taken directly from research papers, including line diagrams, graphs and reproductions of micro-anatomical or histological preparations. The quality and usefulness of the Figures, however, is variable. Certain of them have neither sufficient legend nor associated explanatory text to be useful. The quality of half tone reproductions is only mediocre.

In a few cases, the explanation of a concept fails utterly. Some sentences are flawed by missing words and would remain unintelligible to the student reader. In another case (the explanation for the occurrence of N_1 and N_2 peaks in the compound action potential), the undefined term, the "diphasic response" is invoked as explanation, to leave the reader wondering what that phenomenon is.

Despite some shortcomings, as an attempt to provide a broad overview of the topic of hearing, with generally informal and clear discussion of issues, the book is successful. The further aim, of wishing to impart a sense of enthusiasm for the ongoing elucidation of how we hear, makes the book commendable. Provided back-up resources are available to the reader to cover for the relatively few inadequacies of this book, it will provide a good introduction for the student to the subject of hearing.

Ken Hill

Ken Hill is a Fellow in the Research School of Biological Sciences at the Australian National University. His work investigates the neurophysiological mechanisms of sensory coding in the ear.

* * *

THE PHYSICS OF MUSICAL INSTRUMENTS

Neville H. Fletcher
& Thomas D. Rossing

Springer- Verlag, New York, 1991, pp 620. Hard cover ISBN 3-540-96947-0, Australian Distributor: DA Books, PO Box 163 Mitcham Vic 3132. Price A$ 87.25

In their preface the authors state: "The reader we had in mind in compiling this volume is one with a reasonable grasp of physics and who is not frightened by a little mathematics . . . We have not pursued formalism for its own sake. Detailed physical explanation has always been our major objective".

Considering the immensity of the subject, the authors have admirably succeeded in achieving their objective in the course of 620 pages. It is rare to find a book that clearly and concisely explains the basis of mathematical treatments while,within the same covers, includes a comprehensive discussion of the most relevant experimental investigations. The authors have an enviable reputation for doing this in their own series of investigations into the physical and acoustical aspects of a wide variety of musical instruments.

The first 8 chapters (a little over 200 pages) comprise a concise textbook on the basic theory of Vibrating Systems (Part I) and Sound Waves (Part II) which includes discussion of transient behavour as well as several sections devoted to non-linear behaviour. These two parts of the volume would make a good stand-alone textbook. It is becoming clear now that almost all musical instruments are non-linear. With a non-linear aural detector in our heads, there is much food for thought regarding the appropriate methods to use for measurement in support of the newly developing theory.

There is a fair sprinkling of the traditional mathematician's soothing syrup. After some involved or lengthy derivation, while the reader stares at a somewhat fierce equation, a calm voice whispers: 'It is therefore clear that . . . ', or 'clearly it follows that . . . .

Part III comprises 4 chapters on String Instruments; Part IV, 5 chapters on Wind Instruments and Part V, 4 chapters on Percussion Instruments. All of these chapters are distinguished by a skilful balance between a detailed description of the instruments, the physics of their operation, a discussion of the acoustical output, interleaved with copious diagrams taken from the literature. The selection of illustrations has obviously involved a great deal of time and thought. Occasionally, however, the captions are too brief (probably the original captions) and do not always draw attention to the essential features under discussion or to the method of measurement. The input admittance curves for stringed instruments (chapter 10) would be improved with the frequencies of the open strings marked.

The longest chapters are those devoted to Bowed String Instruments, the Piano, Flutes and Flue Organ Pipes, Drums; areas in which considerable amounts of research have been conducted in recent years. The treatment in each case is balanced and comprehensive and would be an excellent starting point for anyone interested in conducting further work. The versatility of the book is well illustrated with the space devoted to all manner of percussion instruments, drums and bells. For each chapter there is a comprehensive, up-to-date set of references.

The whole book forms an impressive introduction at a fairly advanced level to the extensive field of traditional musical instruments with some reference to their electronic counterparts. The book is clearly printed with excellent layout and reproduction of diagrams. It is highly recommended for all those interested in the basic priciples of operation of musical instruments, as a starting point for research in this area or as a comprehensive reference book.

Howard Pollard

Howard Pollard was associate Professor in Physics at UNSW until his retirement. His research interests continue to include musical acoustics. He is Chief Editor for Acoustics Australia.

ADVERTISER INDEX

Readers are asked to mention this publication when replying to advertisements

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FRONTIERS OF NON LINEAR ACOUSTICS 12th ISNA

M.F. Hamilton and D.T. Blackstock (Editors)

Elsevier Applied Science, 1990, pp 642, ISBN 1 85166 537 4. Australian Distributor: DA Books, PO Box 163, Mitcham, Vic 3132. Price A$162.75

This book contains the proceedings of the 12th International Symposium on Nonlinear Acoustics held in Austin, Texas during 27-31 August 1990. The first symposium of this series was held in New London, Connecticut, 23 years ago. As indicated in the preface to the book, the purpose of this symposia series is to serve as benchmarks in the evolution of the field (on nonlinear acoustics). This book contains a total of 92 papers presented at the Symposium. These papers, covering a wide range of topics, have been organised into the following 14 sections:

Invited Papers: 7
Waves Containing Shocks: 6
Atmospheric Propagation, Relaxation and Inhomogeneous Media: 6
Sound Beams: 6
Parametric Arrays: 6
Reflection and Scattering: 8
Waveguides: 7
Acoustic Streaming, Instability and Chaos: 4
Parameter of Nonlinearity: 7
Biomedical Applications: 6
Cavitation: 4
Interaction with Bubbles: 7
Waves in Solids: 12
Special Topics: 6

The recent evolution of the new science - "chaos theory" has highlighted the importance of understanding nonlinear phenomena in virtually all fields of study because the 'real world' is full of nonlinear dynamic processes. This book provides a fairly up-to-date view of the progress made so far in nonlinear acoustics, most of which has practical applications. However, just like most other proceedings, the papers have been written mainly for the specialists in the field, especially researchers. For those working in one of the topic areas listed above, the book may serve as a good reference to keep in touch with the progress of nonlinear acoustics. The quality of reproduction from obviously "camera-ready" manuscripts is good.

Joseph Lai

Joseph Lai is a Senior Lecturer in the Department of Mechanical Engineering at the Australian Defence Force Academy. He has an ongoing interest in the analysis of nonlinear dynamic processes.

NEW PRODUCTS

CIRRUS

Integrating Sound Level Meter

The **CRL 222A** is a new version of the popular integrating sound level meter, CRL 222. It has retained the advantage of the analogue scale, is only 23 mm deep and has the capability to measure L_{eq} and Sound Exposure Level (SEL) as well as conventional sound level. The measuring range of 63 dB and a maximum L_{eq} period of 10 hours allows use over a full working day. The CRL 222AK is a complete package with the acoustic calibrator, windshield etc in an attache case.

Modular Acoustic System

ARIA harnesses the power of modern software with the portability of laptop computers to provide a modular acoustics measurement system. ARIA requires a processor card to receive the acoustic signal and then software modules are added as required to build a customised acoustic analyser. Each software module fulfils a dedicated task and these include: Architectural Acoustics, Building Control Acoustics, Real Time Dual Channel Narrow Band FFT Analysis, Environmental Monitoring and Event Recording, Real Time Sound Intensity Analysis, Acoustic Power Determination and Time Domain Spectrometry. As each module is software based, the upgrading of the system is simply achieved by installing extra software.

Further information: Davidson, 17 Roberna St, Moorabbin, Vic 3189, Tel: (03) 555 7277 Fax: (03) 555 7956

ACOUSTIC RESEARCH LABORATORIES

Environmental Noise /Vibration Loggers

Designed and manufactured in Australia, the Enviro-Log range has been specifically developed to meet the needs of medium to long term, outdoor monitoring applications where rugged, inconspicuous design and low cost are paramount.

The **EL-015**, a type II Lin/A weighted statistical noise logger which captures 16 samples/second with a 90 dB dynamic range capability. 2048 statistical results may be stored on the Eger which stores statistical linear rms, AS 2670 human sensitivity weighted and peak hold levels. It is intended for logging structural vibration in occupied buildings and the frequency range is 1 to 1 kHz and can operate for up to 14 days.

Spatial Sound Field Monitor

The **SFM-16** is a 16 channel 1/3 octave sound level meter for measurement of sound pressure level and spectrum (from 50 to 10kHz) at discrete locations. The parallel sound level meter circuits and filters are designed to comply with the requirements of AS 1259 and AS Z41. Control is via a front panel 'qwerty' style keyboard and results may be displayed in either text or graphics on a liquid crystal display. The system, fully programable, permits RS 232 communication with MSDOS computers for long term data storage and analysis. An internal, rechargable battery permits operation for up to 24 hours when mains power is not available.

Further information: Acoustic Research Laboratories, 169A Pacific Highway, Hornsby, NSW 2077, Tel/Fax: (02) 476 4198

BRUEL & KJAER

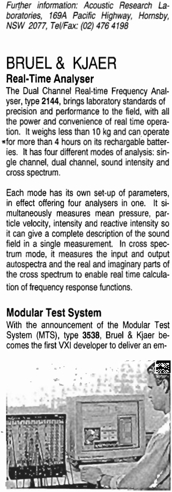
Real-Time Analyser

The Dual Channel Real-time Frequency Analyser, type **2144**, brings laboratory standards of precision and performance to the field, with all the power and convenience of real time operation. It weighs less than 10 kg and can operate for more than 4 hours on its rechargable batteries. It has four different modes of analysis: single channel, dual channel, sound intensity and cross spectrum.

Each mode has its own set-up of parameters, in effect offering four analysers in one. It simultaneously measures mean pressure, particle velocity, intensity and reactive intensity so it can give a complete description of the sound field in a single measurement. In cross spectrum mode, it measures the input and output autospectra and the real and imaginary parts of the cross spectrum to enable real time calculation of frequency response functions.

Modular Test System

With the announcement of the Modular Test System (MTS), type **3538**, Bruel & Kjaer becomes the first VXI developer to deliver an em-



Bruel Kjaer's Modular Test System Type 3538 bedded slot 0 controller that provides full client/server support, a resident multi-user, object oriented operating system, and floating point signal processing. The Felix operating system is based on advanced software bus technology and enables flexibility. Program development takes place on a UNIX workstation connected to the VXI mainframe via the network. The MTS architecture supports a variety of programming languages, including Felixpascal or C for embedded execution.

Further information: B&K, PO Box 177, Terrey Hills NSW 2084 Tel: (02) 450 2066 Fax: (02) 450 2379

DIAGNOSTIC

Real-Time FFT Analyser

The Diagnostics Instruments, type **PL 202**, Real-time FFT Analyser is fully portable with a bandwidth of 20 KHz. A memory card facility is provided for applications and data storage and the instrument is menu driven. A large, high contrast, LCD display allows spectra orbital plots and waterfall displays to be achieved. It is an excellent analysis tool for rotating machinery as well as acoustic, structural and transient investigations.

Bearing Analyser

The **PL 35** is designed to ensure reliable bearing diagnosis, using the Diagnostic Instruments Envelope Signal Processing method. Use is made of a data-base to achieve on-the-spot diagnosis of bearing condition before failure occurs. A tough, portable instrument that can operate with a PC.

Further information: Kenelec Pty Ltd, 1st floor, 38-40 Albany St, Crows Nest, NSW 2065 Tel (02) 439 5500, Fax (02) 438 4313

BRADFORD INSULATION

TEL Product Range

Bradford Insulation have recently launched their latest range of industrial glasswool products. Manufactured with the "**TEL**" process, the products have a higher compressibility, softer feel and easier handling than before. The products are: Flexitel, Supertel, Thermatel, Ductel, Quietel, Ultratel, Multitel, Specitel, Linacoust, Attenuliner and SPI.

Further information: Bradford Insulation in your Capital City or Level 12, 2 Dind St, Milson's Point, NSW 2061

ACOUSTEX

With world wide concern for the environment it is pleasing to see a safe, efficient insulation material based on the use of recycled paper. Spray on **Acoustex** is both a thermal and acoustic insulation product which is sprayed directly onto any surface - usually beneath metal, concrete or plaster ceilings or roofs. The recycled paper is high quality white or off white paper which gives an attractive architectural, monolithic, textured finish and achieves excellent sound absorption as well as insulating against heat transfer and condensation. Acoustex contains no synthetic mineral fibres, is fire resistant and the application is fast.

Further information: Acoustex Pty Ltd, Dissik St, Cheltenham Vic 3192, Tel (03) 555 9941

INC DECIBAR

DECIBAR sound absorption materials are manufactured from flexible polyurethane foam, while DECIBAR sound barrier materials are manufactured from high density loaded PVC. The DECIBAR vibration damping materials may be applied by spray or trowel and the DECIBAR composite materials can combine any of the properties of absorption, barrier or damping.

Recent additions to the range and which are specifically targeted at the building and construction industries include DECIBAR ALAF and DECIBAR AFHR. DECIBAR ALAF is an aluminium foil faced fire retardant acoustic foam. DECIBAR AFHR is a hydrolysis resistant acoustic foam offered with a 20 year lifetime.

Further information: INC Corporation Pty Ltd, 22 Cleeland Rd, South Oakleigh, Vic 3167. Tel (03) 543 2800, Fax (03) 543 8180.

DIARY

• Indicates an Australian Activity

CONFERENCES and SEMINARS

1992

March 4-6, AUBURN
2nd INTERNATIONAL CONFERENCE ON RECENT DEVELOPMENTS IN AIR- & STRUCTURE-BORNE SOUND AND VIBRATION
Details: Congress Secretariat, Dept Mech Eng, 201 Ross Hall, Auburn University, Alabama 36849-3541, USA.

March 8-11, SAN DIEGO
36th ANNUAL CONVENTION OF THE AMERICAN INSTITUTE OF ULTRASOUND IN MEDICINE (AIUM)
Details: AIUM Convention Dept, 11200 Rockville Pike, Suite 205, Rockville, MD 20852-3139 USA.

May 11-15, SALT LAKE CITY
MEETING OF ACOUSTICAL SOCIETY OF AMERICA
Details: Acoustical Society of America, 500 Sunnyside Blvd, Woodbury, NY 11797, USA.

May 25-29, GANSK
5th SPRING SCHOOL ON ACOUSTO-OPTICS AND ITS APPICATIONS
Details: A.Sliwinski, Institute of Experimentl Physics, University of Gdansk, Wita Stwosza 57, 80-952 Gdansk, Poland.

July 20-22, TORONTO
INTERNOISE 92
Details: Congress Secretariat, PO Box 2469, Arlington Branch, Poughkeepsie, NY 12603,USA

Aug 28 - Sept 1, TOKYO
INTERNATIONAL SYMPOSIUM ON MUSICAL ACOUSTICS
Details: ISMA 92 Tokyo Secretariat, c/ Acoustics Laboratory, Ono Sokki Co, 1-16-1 Hakusan Midoiku, Yokohama 226,Japan

September 3-10, BEIJING
14th ICA
Details: 14 th ICA Secretariat, Institute of Acoustics, P.O. Box 2712, Beijing 100080, China

September 12-14, NANJING
INTERNATIONAL SYMPOSIUM ON ACOUSTICAL IMAGING
Details: 14 th ICA Secretariat, Institute of Acoustics, P.O. Box 2712, Beijing 100080, China

September 14-18, LONDON
EURONOISE 92
Details: Institute of Acoustics, PO Box 320, St Albans, Herts, AL1 1PL, England

October 12-16, ALBERTA
1992 INTERNATIONAL CONFERENCE ON SPOKEN LANGUAGE PROCESSING
Details: ICSLP-92, Catering and Conference Services, University of Alberta, 103 Lister Hall, Edmonton, Alberta, Canada T6G 2H6

• November 26-27, BALLARAT
AAS Annual Conference
Details: John Upton, B&K, PO Box 233,Moonee Ponds,Vic 3039 Australia

• December 14-18, HOBART
11th AUSTRALASIAN FLUID MECHANICS CONFERENCE
Details: 11 AFMC Secretariat, Dept Civil & Mech Eng, University of Tasmania, GPO Box 252C, Hobart 7001

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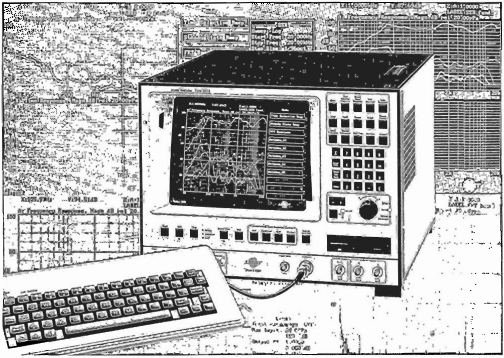

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The 2012 is designed to help you in your work. You can create autosequenced measurement routines, which run your measurements automatically, saving you time and effort during repetitive measurement tasks. And the 2012's multi-tasking design means that you can be performing new measurements while processing previous data, and printing results, all at the same time.

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