NOISE TRANSMISSION THROUGH PAYLOAD FAIRINGS: AN INVESTIGATION OF THE SENSITIVITY TO POINT MASS LOADING

N.S. Ferguson,[1] J.N. Pinder,[2] F.J. Fahy,[1] P. Gardonio[1]

[1]Institute of Sound and Vibration Research,
[2]Institute of Sound and Vibration Research Consultancy Services,
University of Southampton, Southampton, SO17 1BJ. UK.
Tel: +44 (0)1703 593225/593756 Fax: +44 (0)1703 592728

ABSTRACT

Achieving satisfactory noise levels within a launcher fairing has led to the consideration of a wide range of factors, including appropriate noise reduction techniques, which require adequate analytical tools at the early stages of their assessment. This paper provides an example of a possible noise reduction technique and the analytical means for its theoretical assessment.

An efficient method for prediction of the noise transmission through an idealised model of a spacecraft launcher fairing has been developed using a spatial decomposition of the external noise field, a modal description of the fairing modelled as a honeycomb sandwich cylindrical shell and a modal description of the interior acoustic cavity. Also incorporated are the dissipation and radiation damping of the shell. This allows for a rapid assessment of the preferred properties of the structure to give the lowest interior levels without the requirement to solve a large number of numerical (FEM or BEM) models.

The method is more appropriate than Statistical Energy Analysis (SEA), since it accommodates non-diffuse external acoustic fields and the investigation of the coupling in the low-frequency regions of most concern where the modal overlap factors are low.

To investigate the sensitivity to non-uniformity present in real structures and also variations that might be caused by the addition of passive noise control treatments, an extended model has been developed to introduce a finite number of discrete mass additions to the shell. This is shown to perturb the natural frequencies of the shell, introduce structural modes with higher order circumferential and axial wavenumber components and to be able to reduce the possibility of strong coupling between the shell and the external and internal acoustic fields. Parametric studies have revealed the effects of incidence angle, internal acoustic medium and structural properties. However, the latter were restricted to uniform orthotropic structures that exhibit ideal coupling conditions at resonance.

The possibility also exists to enhance the coupling between some modes due to the influence of the masses, and so the effects on coupling of individual mode pairs (structural and acoustic with the same circumferential order) should not be considered in isolation from the complete model. A method for optimisation of applied mass and location could be a future development.

1. INTRODUCTION

Several passive methods have been assessed to control the internal sound field within spacecraft launchers to avoid damage to sensitive payloads, including structural influences and internal absorption [1,2]. An added benefit is that reduced flight noise levels and corresponding qualifications levels could lead to cost reduction in the design of the payloads. Some approaches such as increasing the absorption within the launcher or generally adding mass to the structure in order to increase the transmission loss through the structure introduce a significant mass penalty. An analysis briefly presented in this paper allows for a rapid prediction of the approximate effect of various changes that could be considered as alternatives to typical passive noise control measures. Parameters that have been investigated have included the effect of the angle of incidence of the acoustic field, the structural stiffness of the fairing (assumed cylindrical), the properties of the internal fluid (i.e. air or helium) and non-uniformity of the fairing by the modelling and introduction of discrete additional point masses.

2. THEORETICAL MODELLING

To produce a design tool for rapid low cost parametric assessment of transmission loss influences, a Modal Interaction Analysis (MIA) was developed [2,3] in preference to a fully coupled numerical solution. This analysis includes the external field which excites the structure, that is modelled as coupled to the internal acoustic cavity (payload bay) by applying MIA [4]. The procedure calculates the spatial description of the external acoustic excitation field in terms of axial and circumferential components that excite the modes of a simplified structure (cylindrical). The structural modes then couple and excite the acoustic cavity modes [3]. Due to the orthogonality properties of the modes in the circumferential direction, there is a simple spatial selection of modes that can couple and hence an external acoustic circumferential component of order n will only contribute to the internal acoustic field of the same circumferential order. Thus one can consider the

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transmission loss between pairs of structural and acoustic cavity modes of the same circumferential order, or consider the reduction possible within a frequency band by examining those order pairs, if any, which exist within the bandwidth. Apart from the spatial or wavenumber coupling, there are also frequency terms [3] that indicate whether the structural and/or the internal acoustic modes are close to resonance within the frequency band of interest, and even if the modes are off-resonance there may be a significant contribution from a large number of these terms. Both the spatial and frequency proximity of structural and acoustic modes thus significantly influence the modal coupling, the PC program implementing the analysis being named PROXMODE [2].

Figure 1 shows the simplified idealised model considered with an external incident plane wave at an angle α to the cylinder axis. The blocked pressure field is calculated on the cylinder surface using the diffraction of the incident field by a 'rigid' cylinder. The n^{th} component of the incident field is termed the 'scattering coefficient' for circumferential mode n and is a function of frequency. The generalised force that the external field exerts on the structural modes is then evaluated. The generalised force F_{mn} is a single force that when moved through the virtual displacement δq does the same work as all the external forces, in this case blocked pressures, when the system is moved through the virtual displacement $\delta q \phi_{mn}$, where ϕ_{mn} is the $(m,n)^{th}$ mode of the cylinder. The contributions can be described in acoustic fatigue terminology as Joint Acceptance Functions [3], in which the response of the structural modes is then given as the product of the generalised force in the mode with the 'modal' receptance. The measure of spatial coupling is indicated by the magnitude of the generalised force, whilst the proximity of the excitation frequency to the response frequency of the mode is indicated by the amplitude of the 'modal' receptance term.

Coupling of the structural and internal acoustic cavity modes is again calculated using a classical response analysis [5] with one term representing the spatial matching of the modes and another frequency dependent term which allows for the situation of one excitation frequency component in the external field matching the resonance frequency of the internal acoustic modes. Hence maximum transmission of energy from the external field occurs in the case of spatial external coincidence (external field with the structural mode) combined with internal coincidence (structural mode with the internal acoustic mode) occurring at or near a frequency corresponding to a resonance in the corresponding structural and internal acoustic modes. Because the MIA approach does not include energy balance consideration across the shell, maximum internal energy is limited to the average modal energy levels being set equal to those of the shell's structure.

Alternative methods to control the transmission are either to change the acoustic characteristics of the internal cavity in some manner [1] or change the structural modal characteristics of the fairing in such a way that either one can shift the resonance frequencies or change the spatial coupling. One effect that can be demonstrated is the variation of stiffness of the fairing, simply by reversing the axial and circumferential stiffnesses typically used for an Ariane fairing (see Figure 2).

An alternative approach that was developed for this study has been to consider the effect of non-uniformity of mass of the fairing [6]. The reason for this latter investigation is two-fold in that practical fairings are non-uniform with added trim, connecting mechanisms etc, and it is desirable to understand the likely sensitivity to non-uniformities. Also it may be possible by altering the structural characteristics slightly that the strong coincidence effects corresponding to maximum transmission can be avoided by employing additional mass less than the mass of passive treatments. The theory is presented elsewhere [6] and is based on an expansion of the modes of the non-uniform fairing in terms of the modes of the uniform cylinder in a method analogous to a Fourier series expansion of a periodic waveform. The modes of the uniform cylinder, assumed simply supported, can be shown to be of the form:

$$w(r,\theta,t) = A_{mn} \sin \frac{m\pi x}{\ell} \sin n(\theta-\alpha)\, e^{j\omega_{mn}t} \qquad (1)$$

- w = flexural displacement normal to shell mid-plane
- x = axial coordinate
- θ = circumferential coordinate ($\theta = 0$ corresponds to the plane containing the direction of propagation of an incident acoustic plane wave)
- ℓ = length of cylinder
- m = number of half wavelengths in the axial direction
- n = number of full wavelengths in the circumferential direction
- ω_{mn} = natural frequency of the $(m,n)^{th}$ mode
- α = orientation of the mode (arbitrary)
- A_{mn} = modeshape amplitude
- u,v = corresponding axial and tangential displacement respectively.

The modes of the non-uniform cylinder are expanded in terms of the modes of the uniform cylinder and are of the form:

$$w(r,\theta,t) = \left(\sum_{m,n} A'_{mn} \sin\frac{m\pi x}{\ell} \sin n\theta + B'_{mn} \sin\frac{m\pi x}{\ell} \cos n\theta \right) e^{j\omega t} \qquad (2)$$

where A'_{mn} and B'_{mn} are coefficients that represent the contributions of the modes of order (m,n) that are odd and even in θ respectively. The amplitude and phase of the coefficients are evaluated using an energy formulation based on the Rayleigh-Ritz formulation [6]. The modeshapes are now more complex and examples are given in Figure 3.

The consequence is that whereas for the uniform cylinder there are a pair of modes at the same frequency which are assumed to excite one acoustic mode, non-uniformity causes the pair of structural modes to have different frequencies. The pair of modes of the uniform cylinder are identical in the axial (length) direction and are of the same order circumferentially but are odd and even functions respectively (i.e. sin $n\theta$ and cos $n\theta$). Adding mass at certain locations will either affect one or both of a pair of these modes, i.e. adding mass at a node will have no effect, mass as an anti-node may have a significant effect and may also lead to coupling with other structural modes of the uniform cylindrical shell. The modes of the non-uniform shell will in general have circumferential components of many orders and will lead to coupling with many orders of acoustic modes.

NB: If a single point mass is applied, due to the possibility that the uniform cylinder modes can be arbitrarily orientated, it is possible for the modes to be unaltered with the mass located at a node. There will exist other modes in this case that do not correspond to a node at the applied mass location [6].

The effect of non-uniformity on the transmission loss and other considerations are given in the following section. The results presented have not been subject to energy equipartition limitation, which would reduce the effect of added mass under conditions close to coincidence.

Another important factor is that for a given incident acoustic plane wave the blocked pressures on the surface of the cylinder will be symmetrical about the plane corresponding to $\theta=0$, the reference plane. Thus the odd circumferential modal contributions, i.e. sinθ, sin2θ, ... etc will not be excited directly by a pressure with a cosinusoidal variation in θ. A complication is that the odd contributions in the expansion of the structural modes are related to even circumferential components, i.e. a mode could be of the form:

$$\sin\frac{m\pi x}{\ell}\,(\cos\theta - 0.1\sin\theta\ + ...)\ \text{etc.}$$

The corresponding generalised force would depend upon the cosθ, cos2θ, ... etc terms and there would be response in the mode which also implies contributions in the sinθ, sin2θ, ... etc terms. It would then be appropriate to optimise the magnitude and phase of the various even and odd circumferential orders so as to minimise the generalised force and hence excitation of the structural modes. Within the constraints of limited added mass it would seem logical to position it in a non-symmetric (with respect to $\theta=0$) manner on the shell.

3. RESULTS

Figure 4 shows that the noise reduction for an empty sandwich wall shell of overall dimension similar to the Ariane 5 fairing decreases as α, the angle of incidence of a plane acoustic field at a specified frequency, increases. As the angle of incidence of the related noise from the bottom of the launcher is limited, a value of $\alpha = 45°$ was used in subsequent simulations as this corresponded to the least acoustic protection for payloads. However, at launch, the predominant incidence is expected to be at smaller angles for which higher noise reduction values will occur.

The effect of added mass on a fairing producing two modes with different circumferential variations can be seen in Figure 5 for a single point mass. The axial distributions are not shown as the sensitivity of the coupling to the internal field is primarily being changed by the circumferential variation. An alternative approach to visualise the contributions that exist in these modes is to plot the relative amplitudes of the various circumferential orders. Figures 6 and 7 illustrate the effect of applying 4 point masses on the original (4,3) modes (4 = axial order, 3 = number of circumferential wavelengths). A mode can still exhibit a strong sinusoidal component, sin3θ, due to the locations of the masses being in the plane $\theta=0$. The (4,3) modes originally with a cos3θ distribution are significantly altered with other order circumferential contributions. Figure 7 shows the amplitudes of the various circumferential (n) and axial (m) components that contribute for some cases, but note it does not show whether the components are sinusoidal or cosinusoidal.

The resulting change in the noise reduction for a fairing of such dimension has been predicted for the case of a single mass of various values (Figure 8) and distribution of a fixed amount of added mass, 80 kg, in various ways (Figure 9). Figure 8 shows that generally increasing mass is of benefit but in some frequency bands the best noise reduction does not necessarily correspond to the largest applied mass. Above the 50 Hz centre frequency band there is increasing benefit in redistributing the mass to more locations, the locations chosen being deliberately non-equally spaced and located. More importantly than the frequency band results is the effect that can be obtained if there is a dominant structural and acoustic mode coupling that can be targeted. An example considered was the addition of 80 kg (about 11% added mass for the fairing) as four discrete masses located on a diameter in line with the component of the incident travelling wave perpendicular to the cylinder axis at $\theta=0$, $x=0.387\ \ell$ and $x=0.651\ \ell$ and $\theta=\pi$, $x=0.387\ \ell$ and $x=0.651\ \ell$, ℓ = length of fairing (=10.6m). Table 1 below describes the effect on the noise reduction in the dominant mode and the negative effect due to improved coupling between other mode pairs.

In the original case the (4,3) structural mode was the most important and by adding mass its coupling to the (3,3,0) acoustic mode has been reduced by 6.3 dB. In contrast the new modes that have (4,3) order components become significant and whilst the coupling to the (1,3,0) order acoustic mode was negligible before (-20 dB compared to maximum coupling case) these pairs couple better and are 25 dB higher than previously, resulting in a 4.7 dB increase in noise transmission compared to the reference case. The reason for the negative effect is that in the 100 Hz frequency band the best coupling mode is only approximately 2 dB better then the second best coupling pair of structural and acoustic modes. By applying additional mass the natural frequencies of some of the structural modes that are above particular acoustic modes will decrease and approach, for the same main circumferential orders, the acoustic natural frequencies. Increasing the mass loading further will eventually reduce the structural natural frequencies to values below the corresponding acoustic modes so separating the frequencies as before. The noise reduction is both sensitive to frequency separation for particular modal orders and spatial description and this example illustrates the difficulty in optimising in frequency bands with high modal densities. Conversely if the modal density of the structure is low and dominant coupling is due to a single pair of modes, e.g. the (3,1) mode of cylinder in the 50 Hz centre frequency band is 8 dB more dominant than the next highest coupling, then addition of mass at a set of locations will produce beneficial noise reduction.

4. CONCLUSIONS

A Modal Interaction Analysis procedure has been developed and extended to predict the noise reduction given by a launcher fairing. The model allows for non-uniformity of the fairing mass in a specific manner and examples have been shown of the spatial description of the modes compared to a uniform cylindrical shell. Considering a fairing of similar size to Ariane 5, it has been shown to be possible using a simple mass configuration (not presented here) to reduce the coupling strength by 6 dB, averaged over a one-third octave band. Potential exists to optimise such reduction by variation of mass and location for maximum benefit. The analysis would also benefit from development of energy equipartition constraints.

Mass loading has been observed to be particularly suited to low frequency attenuation, where structural modal density is low. At higher frequencies, shifting the structural natural frequencies to improved proximity with the corresponding same order acoustic modes can occur. Individual modal couplings can then be enhanced by added mass. Initial results (Figures 8 and 9) have indicated though the possible benefit of increasing the mass and redistributing the mass for third octave band noise reduction.

No effect of the payload on the internal acoustic modes has been taken into account and the circumferential orientation of the acoustic modes are assumed to have a random phase relationship with respect to the structural modes. A factor of 0.5, to account for this random phase, is introduced within the spatial coupling factor between the acoustic and structural modes. If the orientation of the acoustic modes is known precisely then adding mass could also be used to mismatch the phase in the circumferential direction in addition to shifting the natural frequencies and altering the spatial composition of the structural modes. The possible effect would then be to vary the coupling of both the external and internal acoustic fields with the structure in a way that would lead to a minimum in the noise transmission.

5. ACKNOWLEDGEMENTS

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Table 1 : Relative coupling levels and corresponding noise reduction due to added mass in the 100 Hz ⅓ octave band

Order (mm,n,p)	Interior acoustic modes nat.freq. (Hz)	Shell modes (unloaded) (Reference case) — Dominant coupling modes (to interior): Order (m,n)	Nat.freq. Hz	Δf_{nat} Hz	Cos Amplitude	Relative coupling level, dB	Overall NR in one-third octave band (dB)
(3,3,0)	98	(4,3)	107	9	1	0	0
(1,3,0)	86	(4,3)	107	21	1	-20	
		Mass loaded shell modes					
(3,3,0)	98	(4,3) (4,3)	90 116	8 18	.78 .49	) -6.3)	-4.7
(1,3,0)	86	(4,3) (4,3) (4,3)	116 81 88	30 5 2	.55 .23 .7	)) +5)	

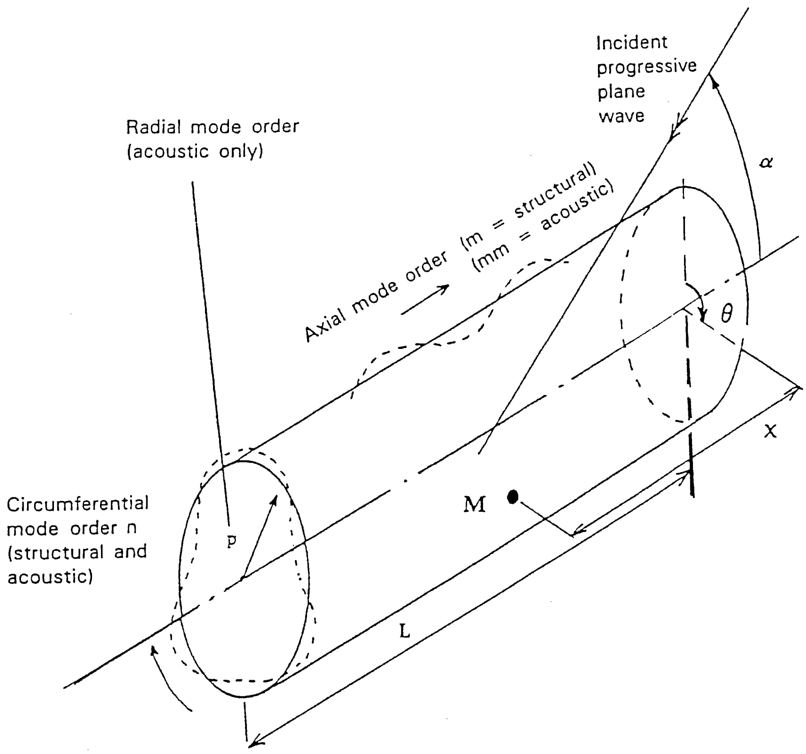

Fig. 1 Cylinder coordinate system and mode order notation. Point mass M at location (x,θ) on cylinder, x = distance from one end and θ = orientation with respect to incident field at angle α to cylinder axis.

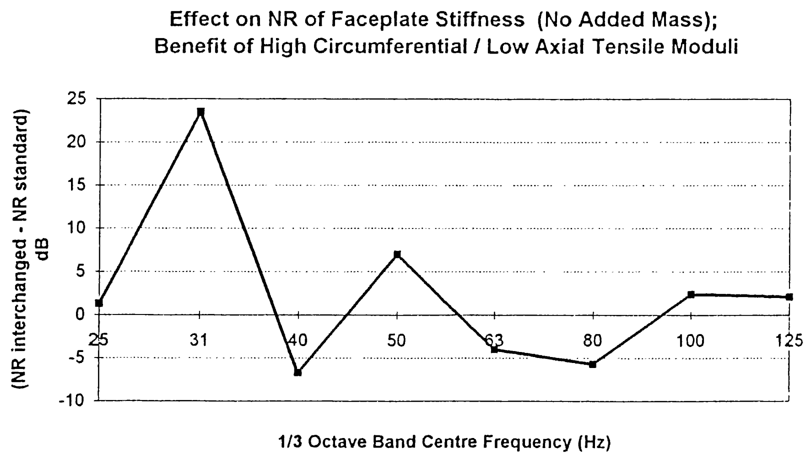

Fig. 2 Effect of interchanging axial modulus of elasticity E_x with circumferential modulus of elasticity E_θ. Noise reduction for standard case corresponds to E_x/E_θ = 3.3, noise reduction for interchanged case corresponds to E_x/E_θ = 1/3.3.

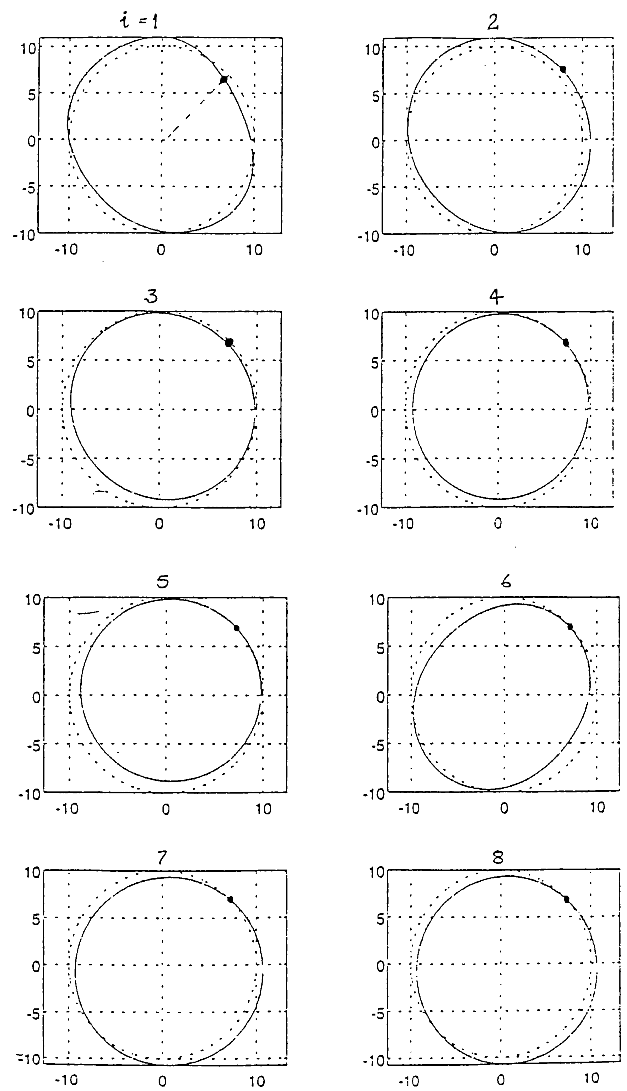

Fig. 3 Circumferential component of the modeshapes of the lowest 8 natural frequencies for a sandwich cylinder. Point mass attached in the midplane, corresponding to x = L/2 (see Fig. 1), of the cylinder. Modeshape normalised with undeformed section also shown (dotted circle radius 10 non-dimensional

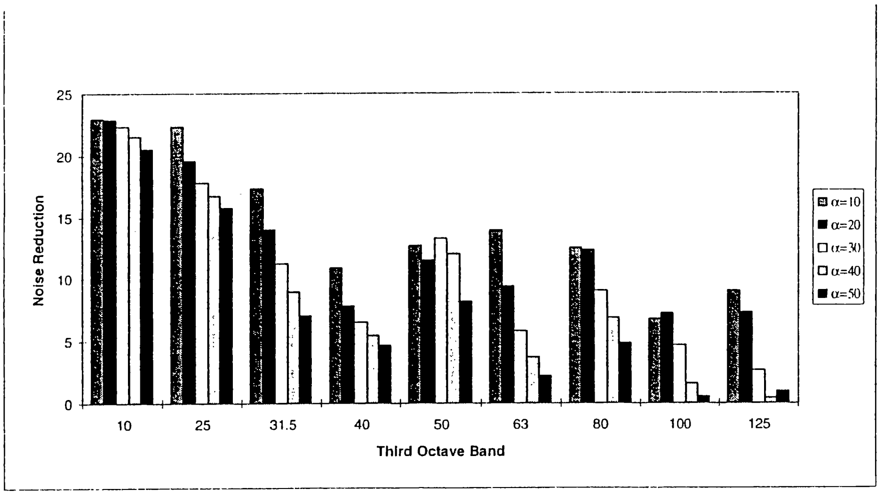

Fig. 4 Noise reduction for a simply-supported uniform honeycomb sandwich cylinder, 2.7m radius and 10.6m length, as a function of angle of incidence of acoustic field.

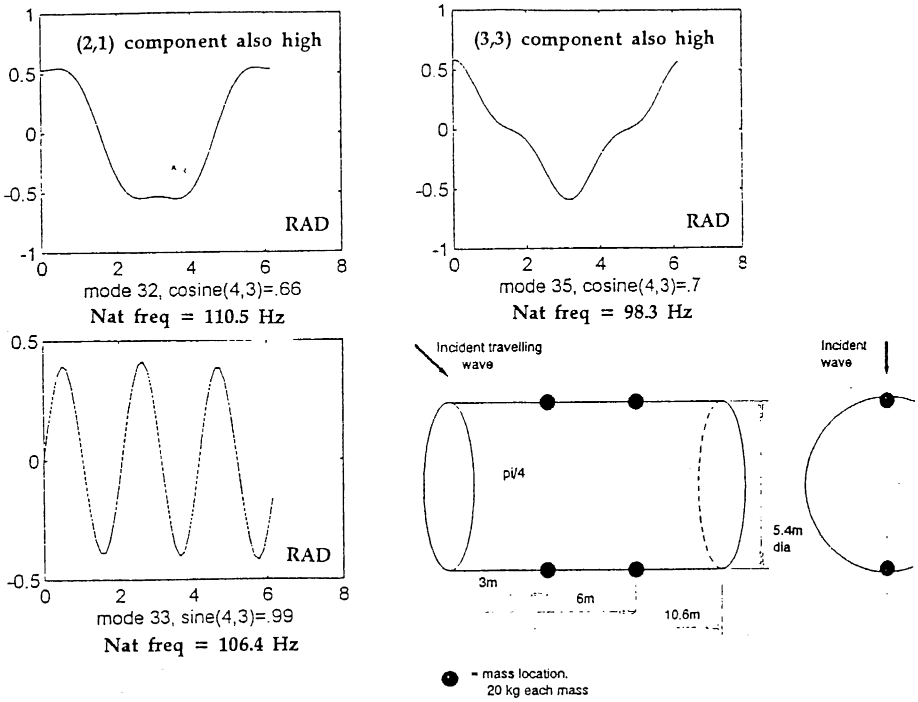

Fig. 6 Variation of circumferential components of modes for a cylinder (r=2.7m, L=10.6m) with four attached masses placed at x=3m and $\theta=0°$ and 180° and at x=6m and $\theta=0°$ and 80° (as shown). Modes with highest (m,n) = (4,3) components shown.

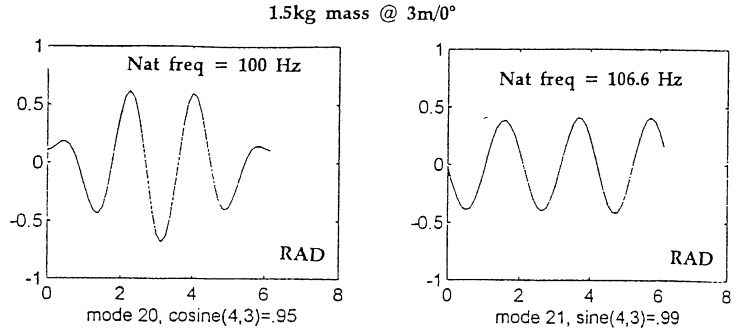

Fig. 5 Circumferential component of the relative amplitudes for two modes of a cylinder (r=2.7m, L=10.6m) with attached mass of 1.5 kg at x=3m and $\theta=0°$. Mode at 100 Hz has predominantly cosine component (relative magnitude 0.95 compared with the sum of the squared component amplitudes) and mode at 106.6 Hz has predominantly a sine component (relative magnitude equal to 0.99).

Fig. 7 Alternative component description of modes with (m,n) = (4,3) components. Relative component amplitudes are shown but phase (i.e. sine or cosine) not shown.

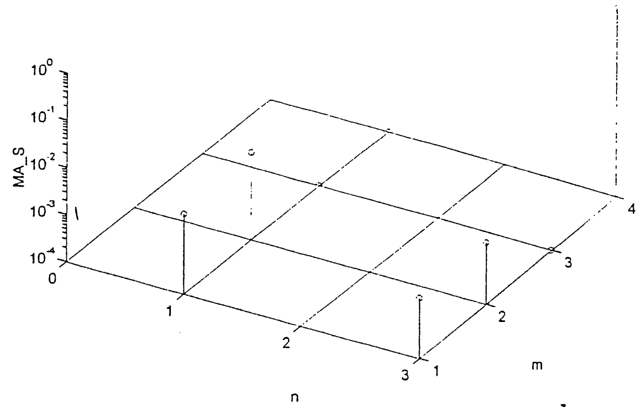

Fig. 7(a) Mode with large (4,3) component but mostly sine (0.99) contribution.

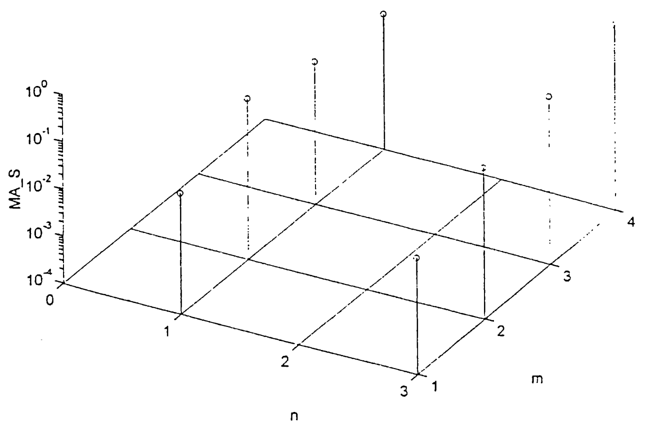

Fig. 7(b) Mode with large (4,3) component but mostly cosine (0.87) contribution.

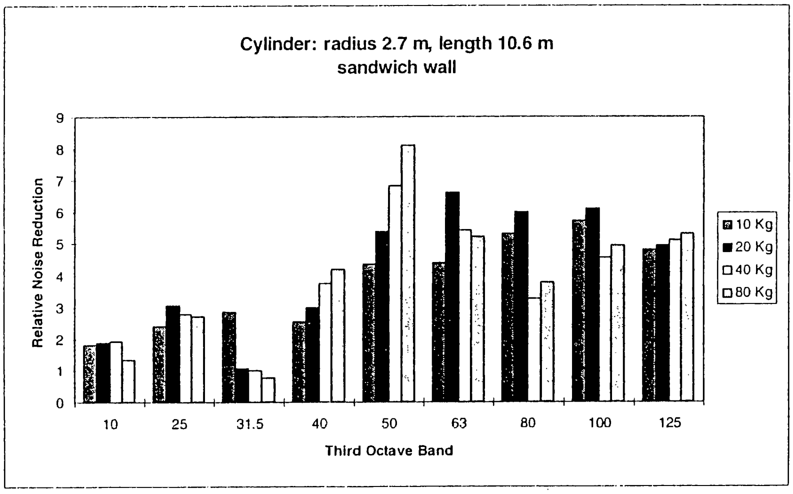

Fig. 8 Relative noise reduction, compared to uniform cylinder (r=2.7m, L=10.6m), for increasing addition of mass at one point corresponding to $x = L/\sqrt{2} = 7.5m$ and $\theta = 2\pi/\sqrt{2} \equiv 254°$ for an incident acoustic field at 45° to cylinder axis.

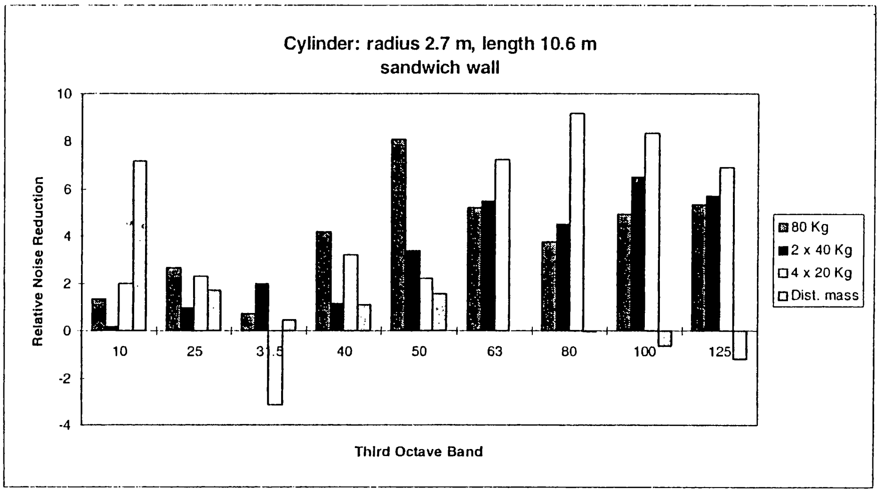

Fig. 9 Relative noise reduction, compared to uniform cylinder (r=2.7m, L=10.6m), for the addition of 80 kg total mass either at one $((x,\theta) = (7.5, 254°))$, two $((x,\theta) = (7.5, 254°)$ and $(6.1, 208°) \equiv (L/\sqrt{3}, 360/\sqrt{3}))$, and four points $((x,\theta) = (7.5, 254°), (6.1, 208°), (L/2 = 5.3, (1 - 1/\sqrt{2})\ 360° = 105°)$ and $((1 - 1/\sqrt{2})\ L = 3.1, 360°/2 = 180°)$ or equally distributed.