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RESPONSE OF NUCLEAR REACTOR GAS CIRCUIT STRUCTURAL COMPONENTS TO CIRCULATOR GENERATED NOISE

1. Introduction

The gas circulators used in gas-cooled nuclear reactors to circulate the high pressure primary coolant can generate a considerable amount of energy at acoustic frequencies which is transmitted round the gas circuit and can cause vibration of the structural components.

To give structural integrity over the full reactor life, it is necessary to ensure that the acoustically induced vibration levels are not sufficiently high to cause fatigue failure. Estimates must therefore be made of noise levels within the gas circuit and of the resulting response of structural components.

The interaction between the noise field and the structure is extremely complex and not amenable to analysis by classical methods. However, the Statistical Energy Method which describes the interaction in terms of the statistical properties of the noise field and the structure is available and may be used, within its limitation, to provide estimates of structural response and to identify problem areas where acoustic testing may be required.

2. Definition of Noise Levels

In an advanced gas-cooled reactor, the coolant is typically CO_2 at a pressure of 30 bars and noise levels of over 160 dB may occur close to the circulator. The spectrum associated with the noise depends upon the type of circulator used. All circulators generate broad-band noise caused by changes in lift generated by the impeller blades in response to turbulence in the inlet flow. The particular circulators used at the Dungeness 'B' Nuclear Power Station also generate considerable amounts of energy at blade passing frequency and its harmonics, this being predominantly due to rotor-stator interaction. As the circulator is a variable speed machine driven via a fluid coupling, the blade passing frequency can vary over a range from 135Hz to 450 Hz.

The noise output of the machine was quantified from tests on an 0.3 scale model running at three times normal speed to give a representative tip Mach number. The results were extrapolated to full size, reactor conditions using the following expression based upon the theoretical expression for noise generated by a blade moving in a turbulent flow:

$$W \propto \frac{V^n d^2 \rho}{c^3} \qquad (1)$$

It is seen that the variation of radiation efficiency is basically a function of the ratio of the resonant frequency of each mode to the critical frequency, i.e. the frequency at which the bending wave length equals the acoustic wave length (585 Hz for the current example). Below the critical frequency $(f/f_c)^2$ gives a good fit whereas well above the critical frequency 'a' approaches unity. The biggest variation occurs around the critical frequency.

The second example is that of a thermal insulation cover plate. These plates which are used to retain the thermal insulation lining the reactor pressure vessel are typically 2 ft square ¼in thick mild steel plates mounted on a single ¾in diameter stud welded to the pressure vessel liner. The radiation loss factors for the first few modes have been calculated using the aforementioned computer programme and the results are shown for the cases of central and offset stud locations in Figure 2. Here the radiation loss factor is seen to vary considerably from mode to mode within the same frequency range and to depend strongly upon geometric details such as stud location.

These two examples illustrate the restrictions in the use of the Statistical Energy Approach. For the large plate, the value of η_{rad} is a well-defined function of frequency and use of Equation (2) integrated over a wide frequency range including many modes will lead to an acoustic calculation of response. There is no need to evaluate the response of each mode individually, an alternative form of Equation (2) which includes the modal density of the structure may be used, i.e.

$$S_v = \frac{2\pi^2 n_S C}{\omega^2 M\rho} \frac{\eta_{rad}}{(\eta_{rad} + \eta_M)} \qquad (5)$$

The modal density n_S may be calculated from the area and thickness of the plate and is not strongly dependent upon the actual geometry of the plate. Detailed calculations of modal frequencies for irregularly shaped plates are not therefore necessary.

Even for discrete frequency excitation, the response of any mode within the blade passing frequency range can be evaluated using a value of η_{rad} which takes into account the relatively small spread of this parameter around its basic variation with frequency. It is, of course, always assumed that a resonance of the structure coincides with the excitation frequency.

For the smaller structure, the large variations of η_{rad} as shown in Figure 2 mean that accurate estimates are more difficult to carry out. The modes are too far apart for expressions based upon modal density to be used and therefore detailed calculations of mode shape and radiation resistance are needed for each mode. This may be carried out for relatively straightforward geometries such as the cover plate example of Figure 2. However, even in this case, the situation is complicated by the fact, that in practice the cover plate has an insulation backing which affects the dynamics to a considerable extent. As a general rule, therefore, it must be assumed that for structures with dimensions smaller than the acoustic wave length, vicinity of the structure and conversely, by the Principle of Reciprocity, governs the response of a structure to an incident pressure wave, i.e. it is a measure of the degree of coupling between the structure and the acoustic field.

(Note that $\eta_{rad} = R_{rad}/\omega M$ where R_{rad} is the real part of the radiation impedance of gas as seen by structure.)

The problem of estimating structural response therefore is basically that of defining η_M and η_{rad}. There is no theoretical method for estimating η_M for a structure. The damping depends upon a number of factors such as numbers of bolted joints, transmission paths for energy into adjacent structures, and recourse must be made to experience or to damping tests on erected components. This, of course, is not possible in the early stages of design and at this stage acoustic values of η_M are unlikely to be available.

4. Estimation of Radiation Loss Factor

A number of approximate formulas for estimating radiation resistance on regular structures under a limited range of conditions have been published in the literature. However, with the advent of readily available computing facilities more general calculations of radiation resistance can be made.

If the mode shape of the structure is known for a given frequency, the acoustic pressure at any point on an imaginary sphere of a radius, which is large compared with the plate dimensions due to the plate vibration, may be calculated. By integrating over the complete sphere surface, the total energy radiated and hence R_{rad} may be calculated. The converse is that the structure will respond to a certain degree due to an acoustic wave passing through any point on the imaginary sphere and impinging on the plate with the appropriate angle of incidence. The total response is given by integrating the effect for waves approaching from all angles, i.e. over the whole of the sphere. A computer programme has been written to perform the integration using numerical integration techniques.

The results of this approach for two different structures are illustrated. The first shown in Figure 1 is for a 10 ft x 6 ft 1in thick steel flat plate, simply supported at its edges. The gas is assumed to be CO_2 at 30 bars and 280°C. The radiation resistance for all resonant modes with wave lengths down to 1 foot have been calculated. Note that Figure 1 gives the radiation efficiency 'a' which is related to radiation loss factor by

$$\eta_{rad} = \frac{\rho C a}{\omega M} \qquad (4)$$

The Statistical Energy Method equation can therefore serve a useful function in preventing excessively pessimistic conclusions being drawn from atmospheric testing. However, once again there are limitations. The reduction in response will only apply if the structural modal frequencies are not sufficiently affected by the increase of pressure to be moved to a frequency range of higher noise level or more efficient coupling between structure and acoustic field.

6. Effects of Edge Cancellation

The theoretical methods used to obtain the radiation loss factors illustrated in Figures 1 and 2 imply that the structural surface receiving the acoustic energy is baffled, i.e. no leakage to the back of the structure is possible. For the case of the symmetric single stud insulation cover plate (Figure 2), the following conclusions may be drawn from the results. For the first rocking mode, the radiation is mainly dipole giving zero volume displacement thus accounting for the low η_{rad} ($< 10^{-5}$) and the low response. For the first umbrella mode, the radiation is monopole with a high volume displacement giving a high η_{rad} and high response. However, the symmetry of the plate gives no net bending moment and therefore low and acceptable stud stresses as illustrated in Figure 5.

If the cover plate is offset on the stud, there is a monopole contribution to the first rocking mode and a stud bending from the first umbrella mode, both modes having high η_{rad}. Initial calculations using the Statistical Energy Method showed that the resulting responses were high and that offset studs should be avoided. However, practical considerations meant that this was difficult to achieve in practice especially as non-uniformities in the insulation backing could lead to asymmetry in the cover plate response.

A special test rig was therefore constructed in which high noise levels at low frequency could be obtained. A cover plate with a high degree of stud offset was tested and the response in the first rocking mode was found to be a factor of 12 down on that predicted by theory and for the first umbrella mode, the reduction was even higher being nearly a factor of 30. It was therefore apparent that the radiation parameters calculated were extremely pessimistic and that cancellation due to acoustic leakage round the edge of the cover plate was suppressing the response of the plate to a considerable extent. It should be noted that the plate in question was highly damped due to the insulation and that $\eta_M > \eta_{rad}$ giving rise to a situation where the response would be strongly dependent upon the changes in coupling represented by changes in the radiation loss factor, the overall damping of the system being affected to only a minor extent.

An alternative situation could be envisaged where very low mechanical damping resulted in a system where for discrete frequency excitation the response was effectively proportional to $(\eta_{rad})^{-\frac{1}{2}}$. The reduction in η_{rad} due to the effects of cancellation would then by virtue of Equation (3) lead to an increase in response. As this would not appear to be physically reasonable, some redefinition of the parameters used in the equations seems necessary under these circumstances and imposes accurate estimates of response using the Statistical Energy Method are not possible.

However, the method can still fulfil a useful function by identifying cases of potential high response. All parameters except η_M and η_{rad} are known and therefore it is useful to consider the effect of those parameters on the response. For the broad-band case, response is proportional to $(\mu_1)^{\frac{1}{2}}$ where $\mu_1 = \eta_{rad} / (\eta_{rad} + \eta_M)$ and for the discrete frequency case to $(\mu_2)^{\frac{1}{2}}$ where $\mu_2 = \eta_{rad} / (\eta_{rad} + \eta_M)^2$. The variations of μ_1 and μ_2 with η_{rad} and η_M are shown in Figure 3. The maximum value of μ_1 tends to unity for increasing radiation loss factor whereas μ_2 reaches a peak of $\eta_{rad} = \eta_M$ and further increases in radiation loss factor actually produce a decrease in response.

A maximum possible level of response may therefore be calculated in all cases for broad-band excitation and for discrete frequency excitation in those cases where damping levels are either known or can be estimated from experience. If these maximum levels are within acceptable limits, the structure in question may be eliminated as a potential acoustically-induced vibration problem.

If the structure cannot be eliminated in this way, the designer has the option of either refining the analysis to provide better methods of estimating η_{rad} or of building a representative specimen to be tested in an acoustic chamber. The latter course has the advantage that the mechanical damping would also be represented on the specimen and would not need to be estimated or measured separately. If site construction is sufficiently far advanced for damping measurements to be made on actual erected structures, then the former option may be the most attractive.

5. Gas Damping Effects

For large test specimens and in particular those instances where relatively low frequency response is of interest, it is necessary to test in an atmospheric reverberation chamber. Fortunately, the speed of sound of atmospheric air is close to that of CO_2 at the conditions prevailing in the gas circulator area of most reactors and therefore the relationship of structural and acoustic wave lengths is correct. However, the gas density may be a factor of 25 - 30 down on that occurring in the reactor situation. The radiation loss factor is directly dependent upon gas density which also appears in the denominations of Equations (2) and (3). The effect of a density ratio of 25 has been calculated using Equations (2) and (3) for a range of η_{rad} and for $\eta_M = 10^{-2}$ and 10^{-3} which encompasses the range of damping most commonly encountered in reactor structures. It is seen from Figure 4 that the lower the mechanical damping, the more effect gas damping has in reducing structural response. For $\eta_M = 10^{-3}$ under the broad-band conditions, the response may be reduced by a factor of 5, whereas for discrete frequency excitation the factor may be in excess of 20 which in many cases means the virtual elimination of the resonant response.

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NOMENCLATURE

C	speed of sound in gas
d	impeller diameter
m	mass per unit area of plate
M	generalised mass of structure
n_s	structural modal density
Sp	sound pressure spectral density (p^2 per rad/sec)
S_v	structural velocity spectral density
v	structural response velocity
V	tip velocity
W	sound power
ρ	gas density
ω	frequency (rad/sec)
η_M	mechanical loss factor
η_{rad}	radiation loss factor

a further limitation on the use of the Statistical Energy Method as represented by equations (2) and (3). However, it must be emphasised that this situation is unlikely to occur in practice when considering nuclear reactor structures.

7. Experience From Commissioning Tests

It is beyond the scope of this paper to discuss the recent commissioning tests on Dungeness in any detail, and it is hoped that these will be covered by a future paper. However, the following observations can be made:

For large platework structures represented by the gas ducts close to the circulators, maximum structural responses at coincidence of a resonance with blade passing frequency were in good agreement, with Statistical Energy Method predictions.

Gas damping effects of 3 - 4 were noted on these large platework structures when comparing results of initial atmospheric circulator tests with results of pressurised tests under gas conditions representative of those occurring during normal reactor operation.

This is consistent with that predicted by Figure 4 taking into account the relative values of radiation and mechanical loss factors expected on these structures at the frequencies concerned.

Small plates where edge cancellation could occur showed a much lower response than similar plates where effective edge baffling prevented cancellation.

CONCLUSIONS

The paper has discussed the situations in which the Statistical Energy Method may be used to estimate the response of nuclear reactor components to the high noise levels prevailing in the coolant gas. It has been shown that for large multi-modal platework structures, the method may be used to estimate acoustically induced response, but for structures which are small compared with the acoustic wavelength, the method may only be used to provide upper-limit estimates.

However, this may still be useful in that potential areas of high response can be identified for further investigation and areas, which are acceptable even at the upper limit of response calculated, can be eliminated from an acoustic test programme. In addition, estimates can be made of the pessimisms associated with acoustic chamber testing under atmospheric conditions.

In general, the Statistical Energy Method provides a useful tool for the estimation of structural response provided its limitations are taken into account. Initial experience from commissioning tests has indicated that, within these limitations, the predictions obtained using the method have been confirmed by measurements under operational conditions.

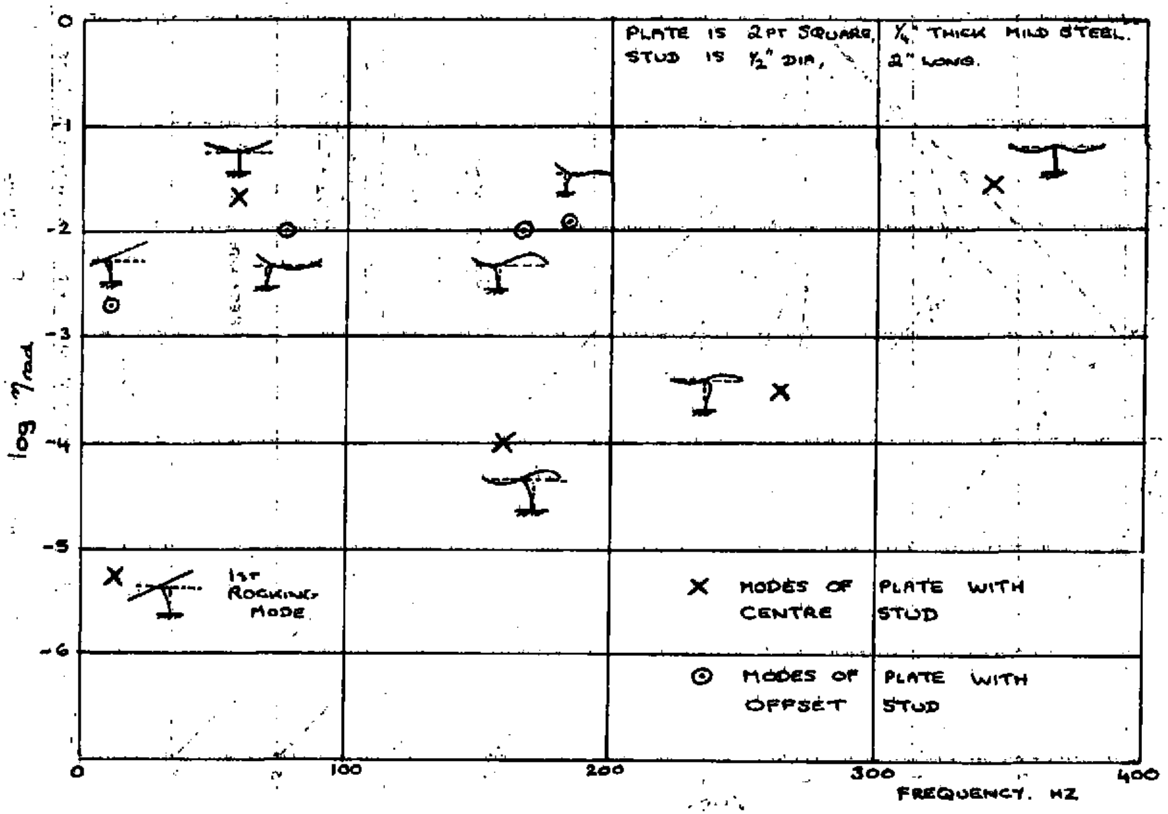

FIG 2. RADIATION LOSS FACTOR OF PLATE MOUNTED ON SINGLE STUD

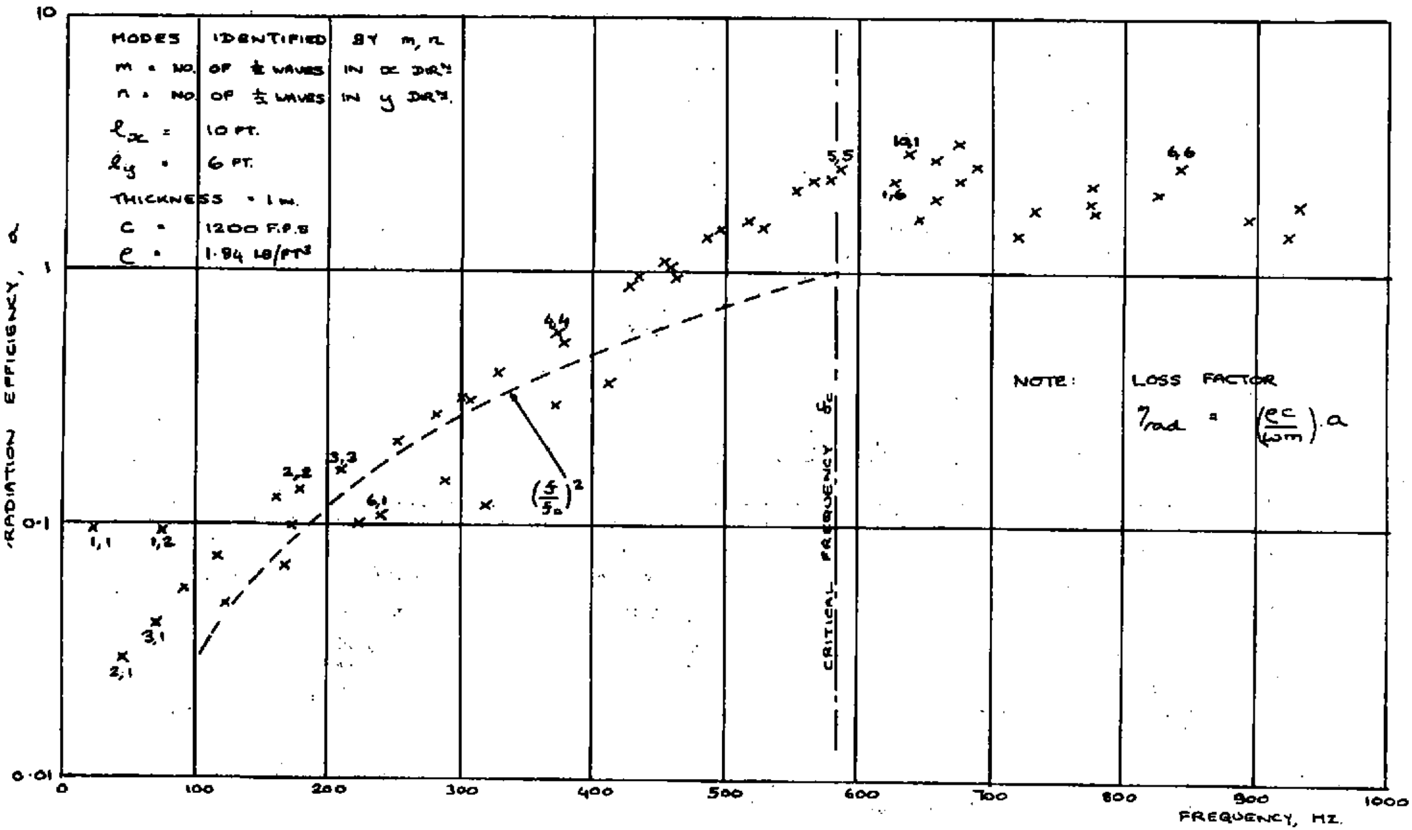

FIG 1 RADIATION EFFICIENCY FOR LARGE FLAT PLATE

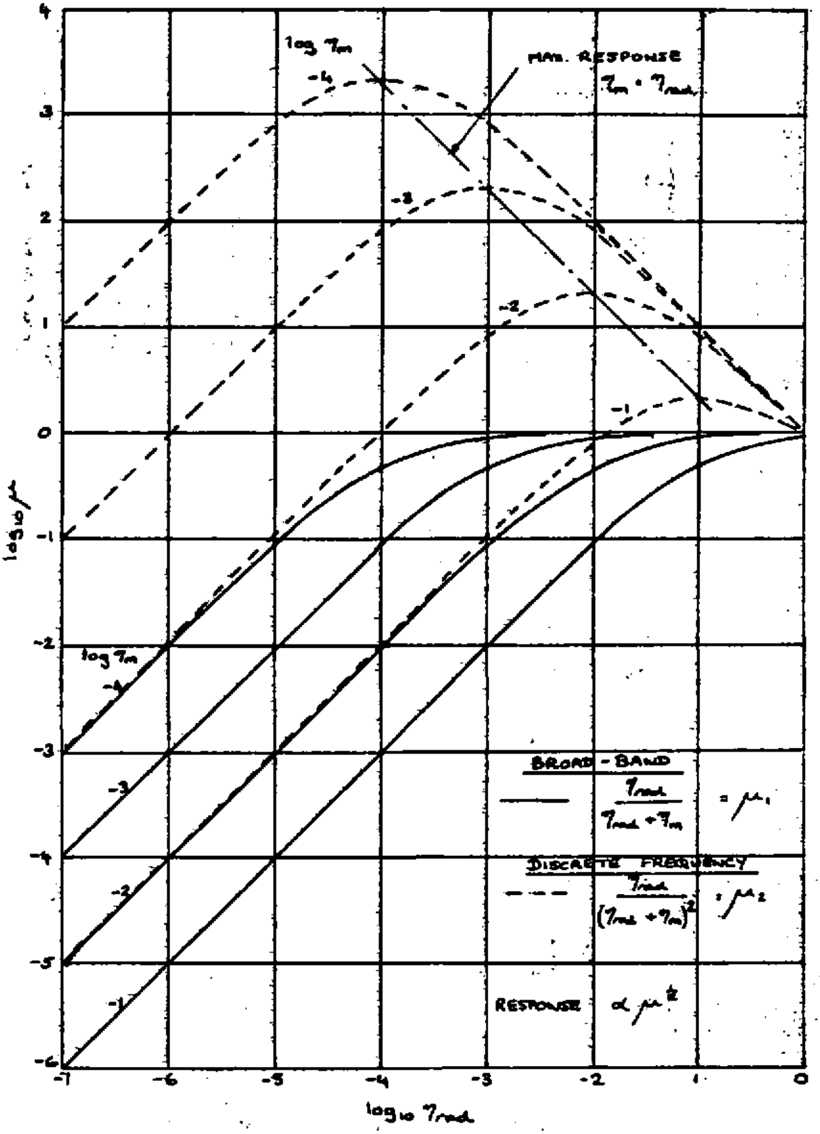

FIG. 3 EFFECT OF RADIATION AND MECHANICAL LOSS FACTORS ON ACOUSTIC RESPONSE

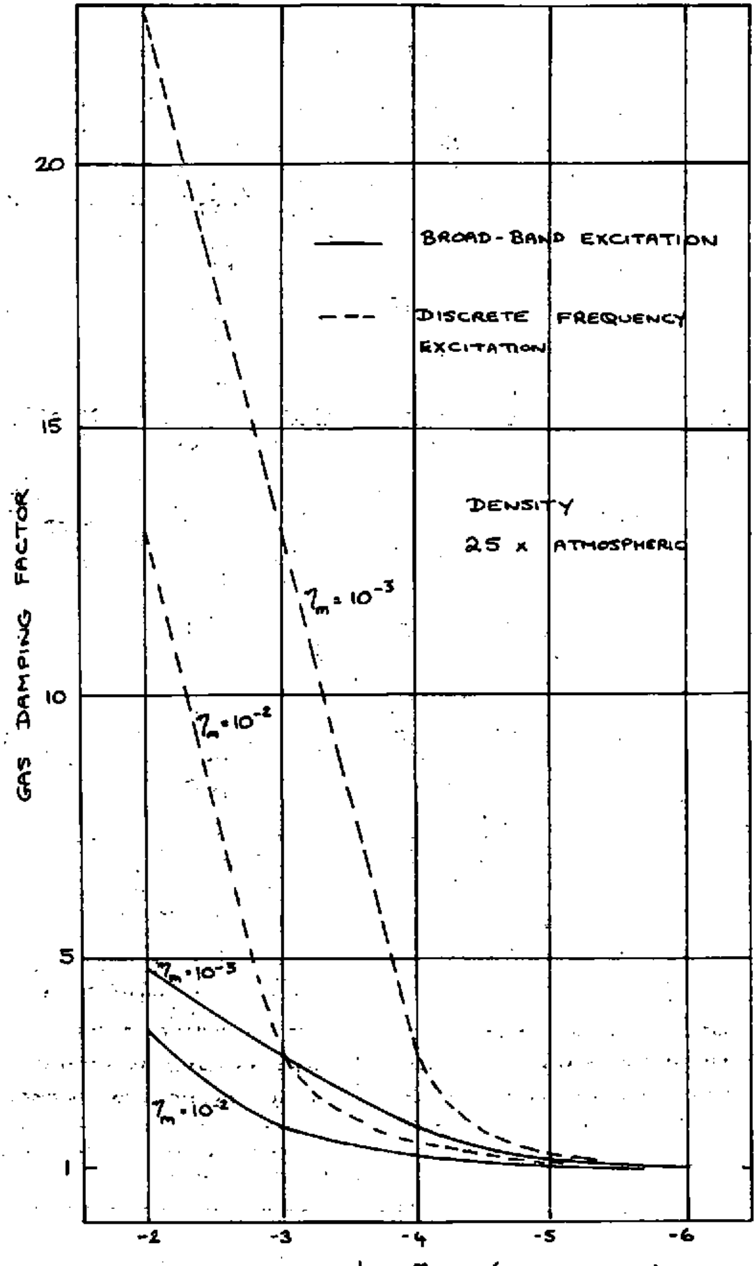

FIG 4 VARIATION OF GAS DAMPING FACTOR WITH RADIATION AND MECHANICAL LOSS FACTORS

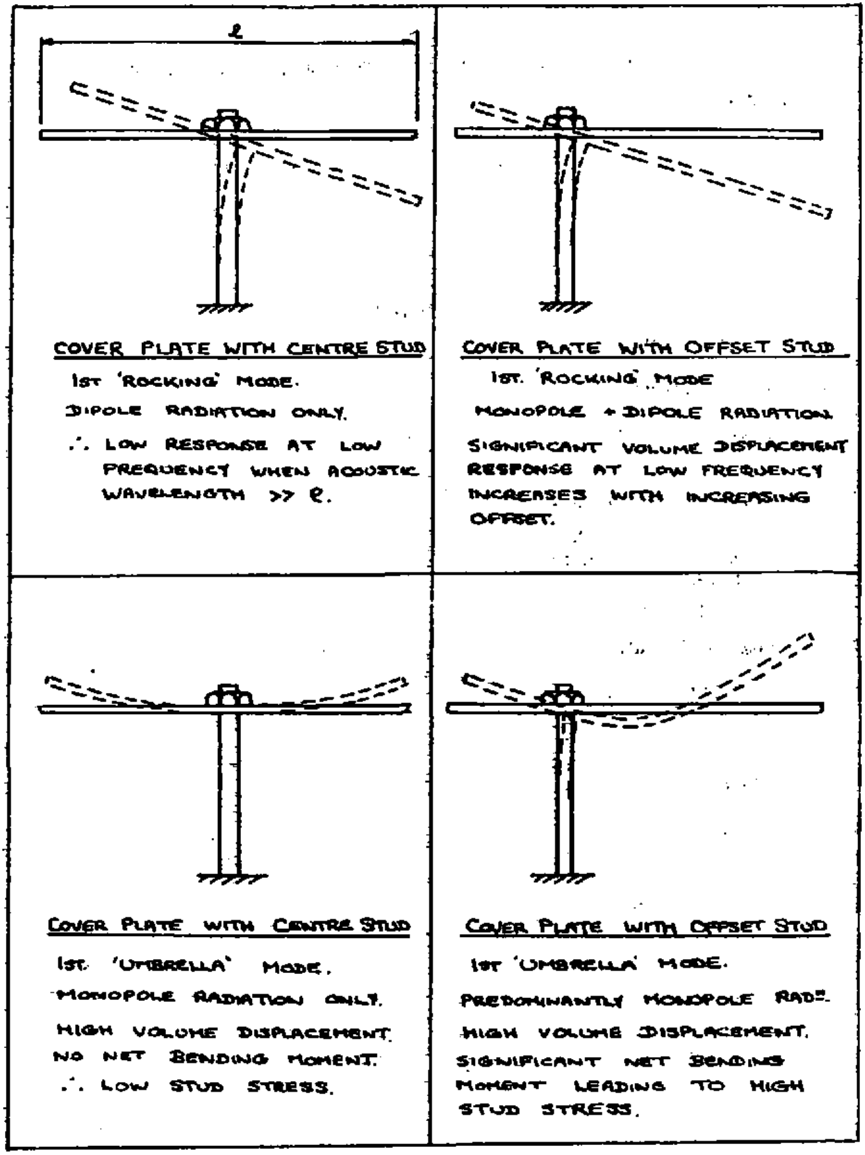

FIG 5 MODES OF VIBRATION OF STUD MOUNTED COVER PLATE