Proceedings of the Institute of Acoustics

NOISE GENERATED BY INTERMITTENT STRUCTURE BORNE VIBRATIONS

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INTRODUCTION

The noise level due to structural vibration can be predicted with some confidence under steady state conditions. The source of disturbance can be a machine housed somewhere inside the building and vibration can be passed into the structure through the resilient mountings, through pipe work and by sound pressures impinging on the structure. The predominant frequencies for each of these paths are likely to be different. How much of these locally induced vibrations are transmitted to another part of the building is difficult to estimate. Wherever possible, a transfer function should be determined by a direct measurement on site. The prediction of the regenerated noise level in the receiving area can then be done with some confidence, as has been demonstrated recently by Walker et al (1) for a studio in the BBC. The measured transfer function will include any effects of structural resonances.

An intermittent disturbance inside a building or from outside can also set up structural vibration, but whether a state of resonant response is set up will depend upon the duration of the disturbance and upon the vibration characteristics of the sound radiating parts of the structure. Similarly, whether a standing wave can be set up will depend upon the room acoustic properties and upon the duration of the vibration in the structure, which can last longer than the disturbance. A source of intermittent disturbance is a railway, either surface or underground. The transmission paths for surface trains are air borne sound striking the facade of a building and by ground borne vibration. From each path some regenerated noise can arise in a part of the building remote from the source when the vibration and room acoustic conditions are favourable. The disturbance from an underground railway is almost entirely by ground borne vibration but the low frequency noise level created in a room may be unacceptable even though the vibration cannot be felt.

The literature on measured levels of disturbance from railways is extensive. For data on the wayside vibration from BR main line trains see Dawn (2) and more information on experience in Europe is to be found in the Proceedings of the ORE Conference (3) held in Rotterdam in 1989. Some examples are given in

BS 8233 - 1987 (4) of the noise level beside a railway line with mixed traffic and for different types of rolling stock. The references quoted in the Standard should be consulted further.

A LINK BETWEEN SOUND AND VIBRATION

A simple approach is to consider the near field pressure level generated by a single vibrating surface. The sound power, W, produced by a radiating surface which is vibrating with the same velocity all over can be shown to be:-

$$W = \rho c S_x V^2 \quad \text{Watts} \qquad \ldots (1)$$

where ρc = impedance of air in Rayls

S_x = radiating area in m^2

V = velocity in r ms m/s

Expressed in d B s relative to $W_{ref} = 10^{-12}$ Watts

$$L_W = 10 \log \frac{W}{W_{ref}} \quad \text{d B} \qquad \ldots (2)$$

A semi empirical expression has been given by Beranek (5) for the near field S P L in a room generated by a vibrating surface:-

$$L_p = L_W + 10 \log \left[\frac{1}{S_x} + \frac{4}{R}\right] + 0.5 \quad \text{d B} \qquad \ldots (3)$$

where $R = \dfrac{S \bar{\alpha}}{1 - \bar{\alpha}}$

S = total surface area in m^2

$\bar{\alpha}$ = average absorption coefficient for the room

It has been assumed that a steady state condition has been established. The S P L calculated applies only to the frequency for which the velocity, V, has been estimated or measured. Complications can arise, as discussed by Allaway (6), when the radiating surface vibrates at more than one frequency and when there are several radiating surfaces vibrating out of phase at different frequencies.

The radiating surface is unlikely to vibrate with a uniform velocity all over and it is difficult to estimate the value to allocate for the area, S_x. The average absorption coefficient can be calculated in the normal manner from the specific coefficients for the different surface finishes for the frequency of the vibration. Normal atmospheric conditions are assumed and air absorption has been neglected, i.e. low frequencies.

As an example, consider a room 3 x 5 x 3 high (m) total surface area, S = 78 m^2 and let the floor be the radiating surface, maximum area S_x = 15 m^2. The velocity is at the threshold for human sensitivity at say 0.1 mm/s for low frequencies. The value for $V = 10^{-4}/\sqrt{2}$ r m s m/s. The influence on the estimated S P L by taking only a part of the radiating surface area, S_x, is given in the Table for two very different values of the average absorption coefficient; values rounded off to whole d B s.

	$\bar{\alpha}$ = 0.01			$\bar{\alpha}$ = 0.1		
radiating area S_x m^2	15	5	1.5	15	5	1.5
S P L L_p d B	82	78	73	73	69	66

At very low frequencies there would be hardly any disturbance but at frequencies above say 30 Hz the noise will be intrusive even though the floor vibration cannot be felt.

When the radiating surface is a floor slab on the ground beside a railway or over an underground tunnel the vibration level can be assumed to be fairly uniform, unless the slab is very large or loaded unevenly. For floors higher up a building the mode shape has to be taken into account and the effective area of radiation will be reduced but the velocity of vibration can be greater than for the ground slab because of resonances. The lower order mode shapes for a beam have been sketched in Figure 1 and for a panel the mode shapes will be in two directions with modal lines. The interaction of the sound pressures generated by the different areas vibrating in opposite phase is rather complicated, e.g. Allaway (6). The prediction of the S P L by equation (3) can apply under these conditions only to regions close to the radiating surface. For a method to estimate the S P L further into the room recourse should be made to Chapter 10 in Beranek (5).

STRUCTURAL RESONANCE

With modern construction the structural damping tends to be small and a structural resonance can be built up when the exciting frequency is close to a resonant frequency. The amplification at resonance, the Q factor, depends upon the damping, which can be expressed in different ways, such as:-

$$Q = \frac{1}{\eta} = \frac{1}{2\xi} = \frac{\pi}{\delta} \qquad \ldots (4)$$

where η = loss factor for structural damping

ξ = viscous damping ratio

δ = logarithmic decrement for free vibration

A magnification - Q - of 10 to 15 is common for a concrete structure and can be greater for a steel framed building.

It is sometimes stated that a state of resonance will not be built up from an intermittent excitation, such as a passing train. The time required to build up a state of resonance when starting from rest can be calculated for a simple mass-spring-damper system, the excitation being at the undamped natural frequency. The results are given in Figure 2. The time taken is long for a system with very light damping, a high Q value, but then the ultimate amplitude will be very large. For a typical structure with a Q = 15 and a resonant frequency of say 20 Hz, the build up time is slightly longer than 1 sec. Many a train will take much longer to pass by or under a building and resonant vibrations can be built up.

STANDING WAVES

The Beranek equation (3) has been developed for a room with irregularities and large enough to have a diffuse reverberant sound field. This will be the more usual situation but where this is not the case, one or more standing waves can be established. The same question arises as with structural resonance; what will be the build up time? The average absorption and the standing wave frequency will be the determining factors. It will be useful to determine a link between the average absorption coefficient, $\bar{\alpha}$, and the Q factor. The build up time can then be determined from Figure 2.

For a system with damping proportional to the velocity, the free decay is exponential and the decay per cycle is the logarithmic decrement, δ, :-

$$\delta = \log_e \left(\frac{x_1}{x_2} \right)$$

where x_1 and x_2 are the amplitudes of any pair of successive cycles.

A decay rate, D R, in d B s per unit time, 1 s, can be introduced by multiplying with a frequency and by converting $\log_e$ into $\log_{10}$ ($\log_e = 2.30 \log_{10}$).

Hence

$$D\,R = \frac{20}{2.3} f_n \log_e \left(\frac{x_1}{x_2} \right)$$

$$= 8.696\, f_n\, \delta \quad \text{d B/s}$$

where f_n = natural frequency in Hz

From eq. (4) for small damping ($Q > 5$)

$$Q = \frac{\pi}{\delta}$$

Hence

$$D\,R = 8.696\, \pi \frac{f_n}{Q} = 27.3 \frac{f_n}{Q} \quad \text{d B/s} \qquad \ldots (5)$$

In order to link this equation to the room acoustics, we shall use Sabine's formula for the reverberation time, R T :-

$$R\,T = \frac{0.16 \text{ vol}}{- S \log_e (1 - \bar{\alpha})} \quad \text{s}$$

where vol = volume of the room in m^3

S = total surface area in m^2

$\bar{\alpha}$ = average absorption coefficient

For low frequencies (< 500 Hz) and a low absorption ($\bar{\alpha} < 0.2$) the $\log_e (1 - \bar{\alpha})$ can be expanded and only the first term = $-\bar{\alpha}$ need be taken.

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Hence $RT = \frac{0.16 \text{ vol}}{S \bar{\alpha}}$ s

which is the time for the SPL to decay by 60 dB.

Hence the decay rate, DR, in dBs/s:

$$DR = \frac{60}{RT} = \frac{60 \, S \bar{\alpha}}{0.16 \text{ vol}} \qquad \ldots (6)$$

Combining equations (5) and (6)

$$Q = \frac{27.3 \times 0.16 \text{ vol } f_n}{60 \; S \; \bar{\alpha}} = \frac{0.073 \text{ vol } f_n}{S \bar{\alpha}} \qquad \ldots (7)$$

Considering the example again, which is a room 3 x 5 x 3 m, the volume = 45 m^3 and S = 78 m^2. Take a value for $\bar{\alpha}$ = 0.1 and the lowest one dimensional standing wave frequency = $\frac{c}{2L} = \frac{341}{2 \times 5}$ = 34 Hz

Hence $Q = \frac{0.073 \times 45 \times 34}{78 \times 0.1} = 14.3$

or by treating the room in three dimensions, the lowest standing wave frequency will be

$$f_n = \frac{341}{2} \sqrt{\frac{1}{25} + \frac{1}{9} + \frac{1}{9}} = 87 \text{ Hz}$$

and $Q = 36.3$

The time for the growth of a standing wave pattern is short, in each case less than 1 sec, see Figure 2.

The effect of air absorption has again been neglected. A much more involved analysis has been given in the book by Kinsler et al (7) but without going as far as the analyses leading to an equivalent Q factor and Figure 2.

TWO CASE STUDIES

In a few flats in a building in the Barbican in London over the railway tracks into Moorgate Station excessive noise and vibration are experienced. The building is over the section where there are no floating track slabs. Vibration and S P L measurements have been taken in a room above the District Line tracks, the room dimensions were 3.35 x 8.6 x 3 m. Vibration measurements at several positions on the floor showed similar peak levels from successive trains, and it has been assumed the floor slab was vibrating uniformly. Measurements of S P L and vibration could not be taken simultaneously, but the spread in levels from different traains was not excessive. A comparison is shown in Figure 3 between the measured S P L frequency spectrum for three trains and a calculated curve for the mean vibration levels. The agreement at low frequencies is close but the predicted S P L values at higher frequencies are too high, probably because the area taken for the radiating surface was too great. There is some evidence of the influence of a standing wave pattern at about 80 Hz.

We had an opportunity some years ago to take measurements in the Royal Festival Hall, for which the piles extend downwards close to the underground tunnels. Vibration measurements were taken on the slate slab in front of the staging and the S P L was recorded simultaneously in the middle of the Hall. There was no audience. Equations (1) to (3) can be worked reversely and the velocity of vibration corresponding to the measured S P L values was calculated. It was difficult to estimate the area of the radiating surface and the two sets of data were linked arbitrarily at 25 Hz. The comparison is shown in Figure 4 and the agreement is satisfactory. The instrumentation at that time possessed insufficient sensitivity to record velocities lower than 10^{-3} mm/s.

CONCLUSION

The Beranek equation (3) can be used to predict S P L due to intermittent as well as steady state vibrations. The build up times for structural resonances and for standing waves are usually much shorter than the passing time for a train.

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REFERENCES

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(6) P H ALLAWAY, 'Noise generation by the vibration of structures' Refrigeration and Air Conditioning, Jan 1970.

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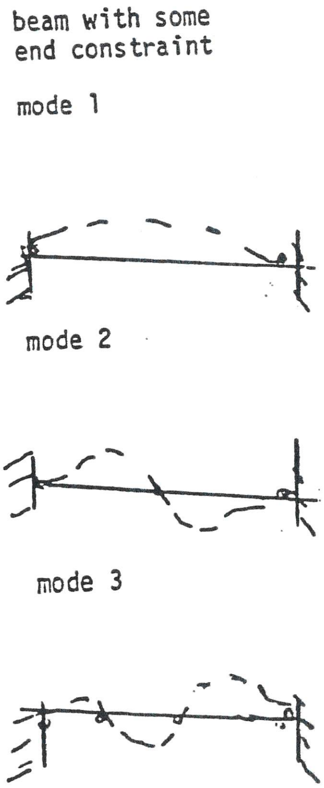

FIG 1 Mode shapes

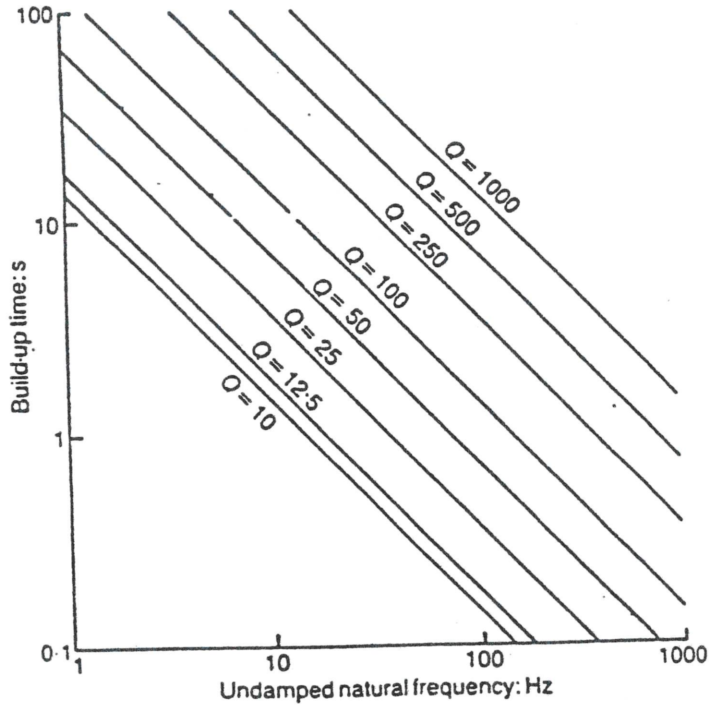

FIG 2 Build-up time for structural vibration

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FIG 3
Comparison between measured and predicted noise levels from trains

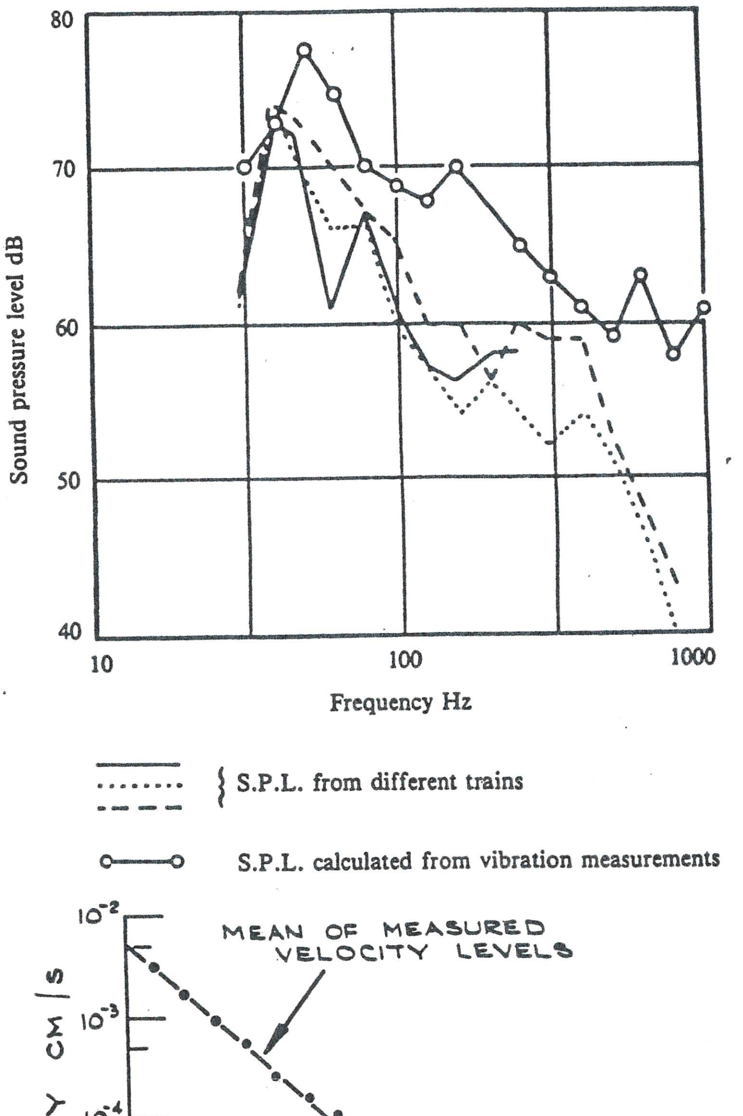

FIG 4
Measured and predicted velocities of vibration from trains in the Royal Festival Hall