VIBRATION INDUCED DUE TO ACOUSTIC EXCITATION IN DIFFUSE FIELD CONDITIONS

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The paper presents an experimental approach to quantify the vibrations induced due to acoustic excitation in diffuse field conditions. An empirical formulation correlating the varying sound field and vibration level generated in floors and walls in diffuse field conditions has been developed. A lower limiting frequency of 125 Hz for good diffusion is observed due to random wide band acoustic excitation in diffuse field conditions, below which lower vibration levels are registered due to discrete room modes.

INTRODUCTION

Structural vibrations results from both air-borne and ground-borne excitation. The induced vibration in buildings results from various external sources, for example, traffic, blasts, construction activities, sonic boom, low frequency noise from aircrafts flyover, impulsive impacts or human activities. The acoustic waves exert fluctuating forces over the building elements, causing them to vibrate which may be enhanced by resonances in case the frequency of sound waves interacting lie within the domains of natural frequency of structure. The acoustic-elastic coupling may be pronounced at lower frequencies, that is, at natural frequencies of a building, room, or wall vibration. Hubbard [1] correlated the measured accelerations for a number of different types of noise inputs on the basis of peak noise level and found that measured acceleration levels increase linearly as the input level increases. Walls, floors, ceilings and large windows respond mainly in the "oil canning" modes at frequencies below 100 Hz and their motions are controlled largely by the beam elements. The response of windows was observed to be 0.015 g/Pa, while the wall acceleration levels were observed to be 10 dB lower than the window levels. Hodgdon et al. [2] demonstrated a threshold of rattle to be 97 dB. The investigations revealed that the *A*-weighted sound levels correlate poorly with acceleration levels, while the unweighted Sound Exposure level L_E and the maximum sound pressure level correlate well. Santos Lopes et al. [3] worked on the determination of a noise level limit to be imposed on any music sound equipment operating inside the sensitive building in order to avoid damaging vibrations in the building facades. A sound level limit of 105 dB(A) was proposed corresponding to the maxima velocity value of 0.7 mm/s for the root mean square velocity measured in a direction normal to the wall, and 3.5 mm/s for peak normal velocity. A frequency limitation of 63 Hz corresponding to sound pressure of 80 dB(A) was also proposed as limit to be imposed by an electronic device, connected to a microphone.

The difference between a mechanical excitation and acoustic excitation of a given structure is actual coupling between the structural modes and applied excitation. The coupling efficiency depends upon how well the sound waves interact with structural modes in case of acoustic excitation [4]. The problem of acoustic fatigue is also very critical for design of aircraft structures subjected to high acoustic loads due to which light weight structures are tested in reverberation chambers to simulate the launch conditions. Statistical Energy Analysis (SEA) has been widely employed by many researchers to predict vibroacoustic problems for interconnected mechanical systems [5,6]. Chang and Nicholas [7] used Green's functions to study the frequency response of structural–acoustic systems. The sensitivity of the structure to diffuse acoustic field has been modelled explicitly by Cremer and Heckl [8] using the principle of acoustic reciprocity, wherein the sound power radiated by the structure is analyzed numerically when a mechanical force *F* acts upon the structure.

$$\frac{|v'|^2}{|p'|^2} = \frac{8\pi}{k^2\rho^2c^2}\frac{P_{rad}}{|F|^2} \quad (1)$$

where k is the wave number, ρ is the density of the medium and c is the speed of sound. $|v'|^2$ is the structural velocity response squared at a certain point A of the structure due to a diffuse acoustic sound field with a (spatially averaged) sound pressure level $|p'|^2$ (the sound incidence case), whereas P_{rad} is the acoustic sound power which is radiated by a force F acting on the same point A.

Rozen et al. [9] discussed a numerical procedure to predict the disturbances due to acoustic excitation of machinery. The sensitivity of a simple structure consisting of a cantilever beam and a base plate to diffuse acoustic field excitation typical for the sound fields in clean rooms was predicted and measured. It was observed that simulations agree reasonably well within the measurements in a reverberant room. A recent study by Løvholt et al. [10] reveals that low frequency sound interaction with the fundamental frequencies of the building components combined with air leaks in the building envelope are the main factors that govern transmission of sound into the building. There are very few such studies that discuss the low-frequency coupling of the acoustic pressure field to the building dynamics using a 2D finite element model.

The present work aims in determining the amplitude of vibration levels induced due to sound fields in diffuse field to investigate the probability of damage in the buildings due to intense sound fields. An empirical formulation correlating the noise levels and vibration of walls and floors is developed. The magnitude of vibration levels generated is analyzed in frequency domain to understand the behaviour especially at low frequencies. However, the coupling efficiency gets accentuated in mid frequency region especially at coincidence zone. A source within the room will excite multiple resonances and thus the sound field is composed of the addition of the many standing waves that the room supports, whilst at the high frequencies, the wavelength is small compared to the room dimensions and also the acoustic energy levels are attenuated. Thus, in the high frequency region, sound waves are unable to excite the bending modes in the structure.

LABORATORY INVESTIGATIONS

An investigation was carried out in a diffuse field conditions in laboratory to measure the vibrations levels generated due to diffuse sound field excitation. The measurements were conducted in Reverberation chambers at the National Physical Laboratory, New Delhi. The dimensions of reverberation chamber are 6 m × 6.5 m × 7 m. The reverberation chamber is a room within another room, both rooms being reinforced concrete [11]. The outer room has a floor slab 300 mm thick supported on folded RCC plates and wall and ceilings are 125 mm thick. The inner room is floated on a 150 mm thick bed of coarse dry sand washed free of mud and silt. The sand bed is initially covered with 50 mm thick fiberglass and 25 mm thick particle board. The walls of inner chamber are 125 mm thick RCC resting on the floated floor made of highly polished terrazzo concrete. Imparting high polish to the surfaces, the viscous drag and thereby energy loss is minimized. The measured value of reverberation time for empty room with random uncertainty less than ± 0.1 s is shown in Figure 1.

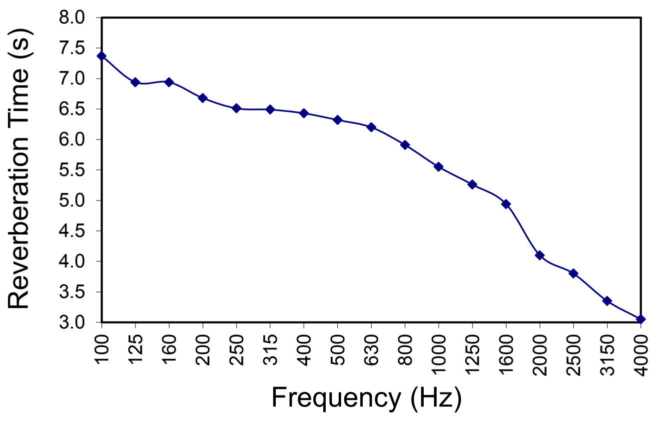

Figure 1. Reverberation time measurements for empty room

The walls, floor and ceiling are non parallel, the average inclination between the walls being 6° and between floor and ceiling 2° to 3°. To prevent the resonance modes, the 125 mm cavity between the inner and outer walls is partially filled with mineral wool blanket to cover 30% of the area. The ceiling of inner chamber is made of polished stone slabs 50 mm thick resting on steel girders and the plenum between the inner ceiling and outer room roof slab is partially filled with mineral wool to damp out resonance modes in this space. The double entry doors as shown in Figure 2 made of sandwich construction consisting of two sheets of 16 gauge mild steel on the outside and one sheet of 1.6 mm thick lead in the middle, with 25 mm air gap on the either side of lead sheet filled with fibre glass. The door panels fit into a rebated 14 gauge sheet steel frame filled with concrete after fixing in position, with rebates lined with soft rubber so as to give a good seal when door is tightly closed with wedged latch [11].

Figure 2. (a) Pictorial view of diffusers installed in reverberation chambers at NPL, (b) View of dodecahedral loudspeaker in reverberation chamber with double entry doors

The sound source installed in the room consists of twelve 100 mm × 150 mm elliptical speaker units mounted in a dodecahedral enclosure fed through a power amplifier delivering up to 20 W (rms) output as shown in Figure 2. The omni-directional microphones measuring the sound field are suspended from the ceiling at different heights and different locations so as to cover spatial zones in the chamber. A pink noise generated the acoustic excitation through a dodecahedral loudspeaker system coupled with an amplifier. Sound pressure levels inside the chamber were measured by a Norwegian Electronics 830 dual channel real time analyzer (RTA 830) in linear weighting. The vibration measurements were conducted using a seismic accelerometer B&K 8318 calibrated on primary vibration calibration system by a laser interferometer in frequency range 0.1 Hz to 1 kHz connected to a B&K measuring amplifier Type 2525 and the frequency spectrum of the induced vibration was obtained using an Agilent Audio Analyzer, Model U8903A. The sound field was generated in a varying range from 50 dB to 120 dB and the vibration levels (1 Hz to 1 kHz) on floor and walls of the chambers were measured as shown in the Figure 3 [12]. The magnitude of vibration level was measured at seven different points on the floor and walls of reverberation chamber. The standard deviation of magnitude was observed to be ± 2.88 mm/s^2 for walls and ± 3.67 mm/s^2 for floor vibration. The linear relationship of vibration levels induced due to acoustic excitation plotted in Figure 3 is consistent with Hubbard investigations [1] whereby the acceleration response increase generally as the sound pressure levels increase and follows a straight line relationship based on the assumption of linear behaviour of the structure.

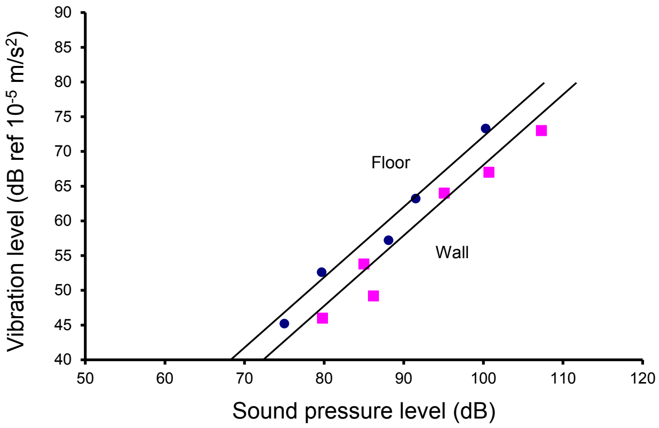

Figure 3. Induced floor and wall vibration due to wideband random acoustic excitation

The induced vibration of the floor ($L_{a(floor)\mathrm{rms}}$) in the frequency domain shown in Figure 4 generated due to the random acoustic excitation is also correlated by a simple regression fit as

$$L_{a(floor)} = L_p \text{ -10log}(f)\text{-10 (dB)} \qquad f \geq LLF\text{ , } r^2 = 0.74 \qquad (2)$$

where $L_a = 20\log(a/a_{ref})$ and $a_{ref} = 10^{-5}$ m/s^2. Thus the above empirical relation can be used to predict the floor vibration induced to acoustic excitation in a diffuse field at different frequencies. The reverberation chamber has a lower limiting frequency (*LLF*) for good diffusion of 125 Hz. The relation between random and the lower limiting frequency is observed to be

$$L_{a(floor)}\,(random) = L_{a(floor)}\,(LLF) + 3 \text{ (dB)} \qquad (3)$$

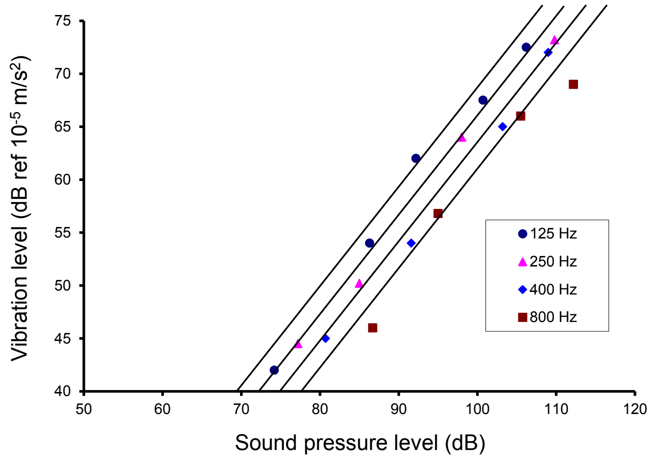

Figure 4. Induced floor vibration due to acoustic excitation of a filtered band at different centre frequencies

DISCUSSION

The enclosed space in the reverberation room can be considered as a complex resonator possessing many normal modes of vibrations excited by introducing a sound source into the room. The acoustic energy supplied by the source is considered as residing in the standing waves established in the enclosed space. The characteristic frequencies of vibration of the standing waves depend upon the room size and shape whereas the damping of these waves depends mainly on the boundary conditions. The extent of diffusion can be judged by uniformity of reverberation time within the volume of room, linearity of sound decay at different points in the room and uniformity of sound intensity distribution within the room. The experimental investigations carried out for measurement of reverberation time show that the decay curves at all positions in the room and at all frequencies have a smooth, linear drop of at least 40-45 dB from the initial sound cut-off. The distribution within the room of sound level of filtered band of white noise is within ± 0.5 dB at high frequencies and within ± 1 dB at low frequencies. Diffusing plates have also been additionally suspended from ceiling and oriented at random to enhance the state of diffusion in the room as shown in Figure 2(a). The standard deviation of correlation coefficient (sin kr / kr) was measured to be within ± 0.06 [11,13] in the frequency range 100 Hz to 125 Hz.

The expression for modal density that applies approximately to rooms of any shape including cylindrical rooms is given by [14]

$$\frac{dN}{df} = \frac{4\pi f^2 V}{c^3} + \frac{\pi f S}{2c^2} + \frac{L}{8c} \qquad (4)$$

where V is the volume, S is the total surface area and L is the sum of length of all edges of the room. At higher frequencies, there is fairly even modal distribution and spacing between the characteristic frequencies is close, while at low frequencies, there are very few modes. So, the average sound energy density is not the same throughout the enclosed space and thus the sound field is not diffuse. A diffuse field can be established in a rectangular room if there is at least 20-30 modes in the measurement bandwidth [15], and there is at least one mode per Hz. In the present case, the number of modes has the value 21 for f = 100 Hz and Δf = 13 Hz (1/6 octave bandwidth). Since the room is not symmetrical, the eigen-tone frequencies cannot be calculated easily. However, if the room is assumed to be rectangular with dimensions corresponding to the average dimension, the eigen-tones between 110 Hz and 125 Hz would have been spaced 1 Hz apart. The lower limiting frequency for good diffusion is observed to be 125 Hz. Where diffuse conditions exist, Figure 3 shows that the acceleration level increases linearly with acoustic excitation (L_{in}).The diffuse field conditions are however difficult to achieve in a normal build up areas and thus there may exist deviations from the results predicted due to empirical formulation. This may be attributed due to the spatial distribution of sound field, inherent damping of the system and excitation of resonances in case the vibration frequency falls within bounds of natural frequency of structure. Hodgson observations [16] reveal that diffuse field theory can be applied in the case of an empty room with quasi-cubic dimensions, specularly reflecting surfaces and uniform surface absorption. However, it has been experimentally demonstrated that even in small rooms, the uniformity of the sound field can be significantly improved with diffusers [17, 18]. Schroeder [19,20] described a cross-over frequency that denotes approximately the boundary between reverberant room behaviour above and discrete room modes below for airborne sound in reverberant enclosures calculated empirically from equating the half-power bandwidth B ($B = 2.2/T_{60}$) of the resonances with three times the average asymptotic spacing Δf ($\Delta f = c^3 / 4\pi V f^2$) between resonance frequencies giving f_s as [20]

$$f_s = 2000\sqrt{\frac{T_{60}}{V}} \qquad (5)$$

where T_{60} is the reverberation time of the room in seconds and V is the volume of the room in m^3 and the factor 2000 (which contains the velocity of sound) guarantees that on average, at least three resonances fall within the half-power bandwidth of one resonance at frequencies above f. In the normal rooms, the frequency f of Eq. (2) becomes equivalent to f_s of Eq. (5).

The modified equation suggested by Nélisse et al. [15] is given by

$$f_s \approx \sqrt[3]{\frac{\alpha c^3}{4\pi\eta V}} \qquad (6)$$

where α is the model overlap, c is the speed of sound and η is the damping factor. For a damping of $\eta = 5\times10^{-3}$ [15] and model overlap α =3 as proposed by Schroeder, f_s is calculated to be 192 Hz in the present case.

Figure 5 shows the response of floor in g/Pa at different frequency. For sound pressure level of 1 Pa (or 94 dB), Eq. (2) reduces to

$$L_{a(floor)} = 84 - 10\log(f) \quad \text{(dB)} \qquad (7)$$

The overall response of the floor is observed as 0.0002 g/Pa (20.4 mm/s^2/Pa) and for the wall as 0.0015 g/Pa (13.2 mm/s^2/Pa). It can be observed that the g/Pa value diminishes at higher frequencies and also strong coupling of sound waves with the structural modes in dominant in the frequency range from lower limiting frequency (*LLF*) up to 500 Hz.

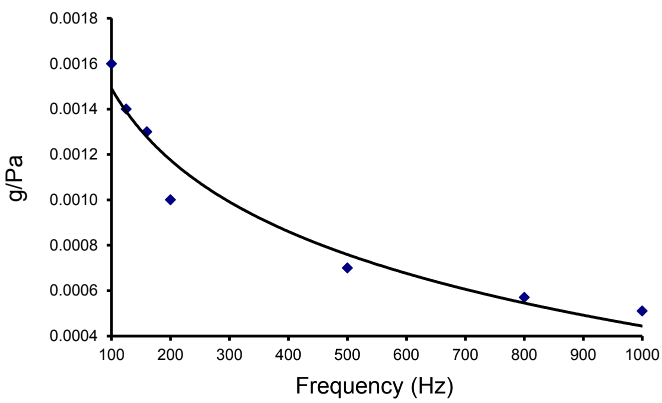

Figure 5. Response of floor to induced acoustic excitation in g/Pa

In most of the practical situations, the acoustic excitation in the free-field conditions and induced vibrations on the facades and low frequency response exhibit a complex behaviour with uncertainties arising from sound structure interaction. Some of the acoustic excitations like sonic boom, cracker bursting and open air detonation of charges can produce high acoustic stimulus of lower frequencies. Since their occurrence is hardly likely and is always confined to outdoors, this affect their coupling to the facade of a structure under free field condition and the induced vibrations never exceed the indoor diffuse criteria as described by Eq. (2). Air borne excitation is mainly low frequency sound waves interacting with building elements like windows, doors etc. causing them to vibrate, while ground borne vibration propagates through building foundation and floor supporting walls. The interaction of sound waves with structure in free-field conditions is quite cumbersome to model as various uncertainties are involved in acoustic-elastic coupling. Thus, a large database for vibration induced due to various noise sources like transportation, aircrafts flyover, blasts etc is required for analyzing the severity and perceived response by the community. For instance, a study conducted to ascertain the magnitude of maximum floor vibration level generated in a historical structure during ceremonial gun firing reveals a vibration level of 6×10^{-3} g (rms) for noise level of 125 dB(A). The floor modes of 25 Hz, 42 Hz, 69 Hz and 100 Hz get amplified during the excitation [21]. Another study conducted for monitoring the transient acceleration induced due to overflying aircrafts landing and take-off over ancient monument reveals a maximum acceleration observed as 3×10^{-3} g (rms) and major resonant modes excited in structure lying in frequency range 10 Hz to 100 Hz [22]. Experience for blasting, explosions and for sonic booms suggest that damage to houses may occur at peak acceleration values between about 0.3 and 3.0 g in the frequency range of 10 to 100 Hz respectively [23]. The widely used German standard, DIN 4150 [24] provides limit values for different types of structures and for different sources of vibration in conjunction with assessment of building damage caused by short-term and long-term vibrations. The generally accepted code of degree of damage to structures is correlated with the ground motion peak velocity and frequency as the strain imposed on the building at foundation level is proportional to peak particle velocity. In present context, for the extreme condition of the random acoustic excitation in diffuse field, Eq. (2) can be considered for predicting the maximum acoustic excitation for structural integrity as prescribed by DIN 4150 and BS 7385 standards [25]. The present study also emphasizes the need for correlating the induced vibration levels with *C*-weighted sound pressure level or sound exposure levels as *A*-weighting devaluates the low frequency noise.

CONCLUSIONS

An empirical formulation correlating the vibration induced due to acoustic excitation in diffuse field conditions is developed. In practical situations, as the diffuse field conditions don't prevail, the interaction of sound waves with structural modes gets diminished resulting in weak coupling and thus lower vibration levels are registered. The transition frequency also called the Schroeder frequency thus governs the interaction of sound waves with structural modes in practical situations. It may be noted that the vibration of any structure is dependent upon the material properties and boundary conditions in addition to the external forcing function. The paper considers the forcing function (diffuse field conditions) only and can't be generally applied to other structures with a definite level of confidence. However, it quantifies the maximum vibration levels induced in a diffuse field set up for adjudging the severity of vibration levels induced due to high acoustic loads. Further investigations in this regard on vibration induced due to acoustic excitation in free field conditions and correlation of the induced vibration with weighted acoustic excitation (*A*-weighting, *C*-weighting, L_{max} and Sound Exposure Level) in free field conditions shall be helpful in better understanding of sound waves interaction with structural elements in practical situations. A comparison with diffuse field conditions is also to be investigated for characterizing the vibroacoustic behaviour of structures explicitly in different situations.

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ACOUSTICS AUSTRALIA REVIEW SURVEY 2013

In July 2013, the Council of the Australian Acoustical Society established a panel to review the current production of Acoustics Australia, the value of the journal to the membership of the Society and propose changes as necessary.

The editorial board, with reference to the review panel, developed a survey to assess the opinions of the membership regarding the journal itself and on plans for alternative distribution means. The questions in the survey were aimed to provide a balance between ranking type responses and free text comments.

Soon after the distribution of the August 2013 issue of the Acoustics Australia journal, the members of the Australian Acoustical Society (AAS) were asked by email to give their views on Acoustics Australia via an online survey. Of the approximately 500 members, 156 responded to the survey. A summary of the finding is given here and the full report an analysis is available on the webpage at

http://www.acoustics.asn.au/members/forms/Acoustics_Australia_Review_Survey-2013.pdf

This is in the members only area so you will need to log in to access.

While it is clear from the results is that there is a great diversity in the opinion of the journal, the bulk of those that responded do read and find the journal useful at least some of the time. The comments suggesting Acoustics Australia be more like the Acoustics Bulletin, (a non peer review journal) were offset by those saying Acoustics Australia should strive to achieve a higher impact factor. There were a number of comments suggesting preference for more Australian related articles and fewer theoretical papers from overseas. There also needs to be more clarity to the readership of the editorial process and the plans for future issues.

There was a mixed but strong feeling about editorial control for the letters to the editor with particular recommendations to clearly identify any background or vested interest of the letter writer and, where appropriate, to allow the right of reply in the same issue.

In regard to the format for distribution, the results indicate that it is time to go to a full electronic (pdf) distribution for the journal as the bulk of the respondents accepted a move to an electronic version. The distribution of responses to the question about preference for hard copy versus pdf if there was no change in membership fee is shown below. A move away from hard copy distribution will be a major cost saving for the journal production. For those few who are prepared to pay for a hard copy at a cost recovery price, a limited print run using a more cost effective process, could be made available.

Some commented that more articles may be submitted if the website is made clearer, the future planning of special issues is publicised and if members are invited to submit articles. This could be achieved with ad hoc emails from the General Secretary and specific prompts to individual members plus more information in the journal itself about the process and seeking submissions.

Marion Burgess