Noise Pollution
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CHAPTER 14

Solid-borne Noise Control in Buildings and Machinery

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14.1 THE NATURE OF 'NOISE' IN SOLIDS

The noise excited in solids may be of various types:

(a) Single impacts which are registered as separated events by the ear; examples are footsteps or door-slamming in buildings, pile driving, forge-hammer blows, punch presses, textile machines, etc.
(b) Periodic impacts that are interpreted by the ear as tones with harmonic content; examples are the piston slap in internal combustion engines or other periodic machinery, the many small impacts generated in the cutting process of a circular saw, etc.
(c) Almost harmonic as when fluctuating electromagnetic forces act in the body of a transformer or electric motor (cf. household appliances, elevators); other examples are in gear noise due to manufacturing inaccuracies (hence stressing the need for good *quality control*), or to gear-tooth resilience, or in the sometimes rather startling phenomena when feedback and/or resonance effects are involved (e.g. wheel squeal).
(d) More or less random, as in hydraulic systems such as in flushing cisterns, taps, pipes (cf. water-hammer) etc. arising from turbulence, cavitation or presence of 'air-locks'; in devices involving drilling, grinding or polishing; in movement of bodies over irregular surfaces (cf. trains passing over 'wavy-worn' tracks), etc.

The above examples indicate the many and varied sources of noise in solid structures and the sound energy involved is propagated by the different forms of wave-motion (see Chapter 1) which can exist in solids. On reaching the boundaries of the solid with other media, the continued transmission or reflection will depend upon the angle of wave incidence and the relative acoustic impedances of the boundary media. Any internal waves reaching an air-solid boundary will, in general, produce an outward or inward movement of the solid surface and this give rise to sound in the surrounding air.

This list, although incomplete, shows that solid-borne sound *excitation* is almost omnipresent and consequently also the *propagation* of solid-borne sound — or structure-borne sound as it is often called — plays an important role. Examples here are transmission of sound along flanking walls in buildings, propagation of bending waves or other wave types along structural beams, pipes etc., and the transmission of vibrations from machines via their mounting into neighbouring regions.

In the following, an attempt is made to describe (briefly) the important phenomena in solid-borne sound generation and propagation and to give the main hints on structure-borne sound reduction.

14.2 RESUMÉ OF DIFFERENT WAVE-TYPES IN SOLIDS

In gaseous or pure (i.e. non-viscous) liquid, media sound is propagated by compressional waves in the material. Solids, however, have not only a compressional stiffness but also a shear stiffness and consequently sound energy in solids may be transmitted via two types of wave that have different speeds and therefore give rise to some special effects that are unknown in gases or normal liquids. The two basic wave-types are the compressional waves and the shear waves, but rarely is one present alone and they combine with each other to give rise to more complicated wave-fields. Some of these combined waves have special names. The particle motion and the formulas for the speed of wave propagation motion for the various wave types are shown in Figure 14.1(a) and (b). It should be noticed that in some cases the wave speed depends on frequency thus giving rise to dispersion effects.

14.3 MEASURING TECHNIQUES

The quantity that usually is measured in the field of structure-borne sound is the motion in a direction perpendicular to the surface of the structure (only in a few cases is the motion parallel to the surface also of importance). A typical measuring set-up consists of a piezoelectric material backed by a little mass (see Figure 14.2). Such a device, when mounted rigidly to a vibrating surface, produces fluctuating electric charges that are proportional to the acceleration of the surface. Since the measuring devices are very small and light (typically 1–10 grams) — to avoid any mechanical loading of the vibrating object — the electrical signals are rather small, but with modern electronics the signals can easily be brought to a level which allows the use of all the data processing and

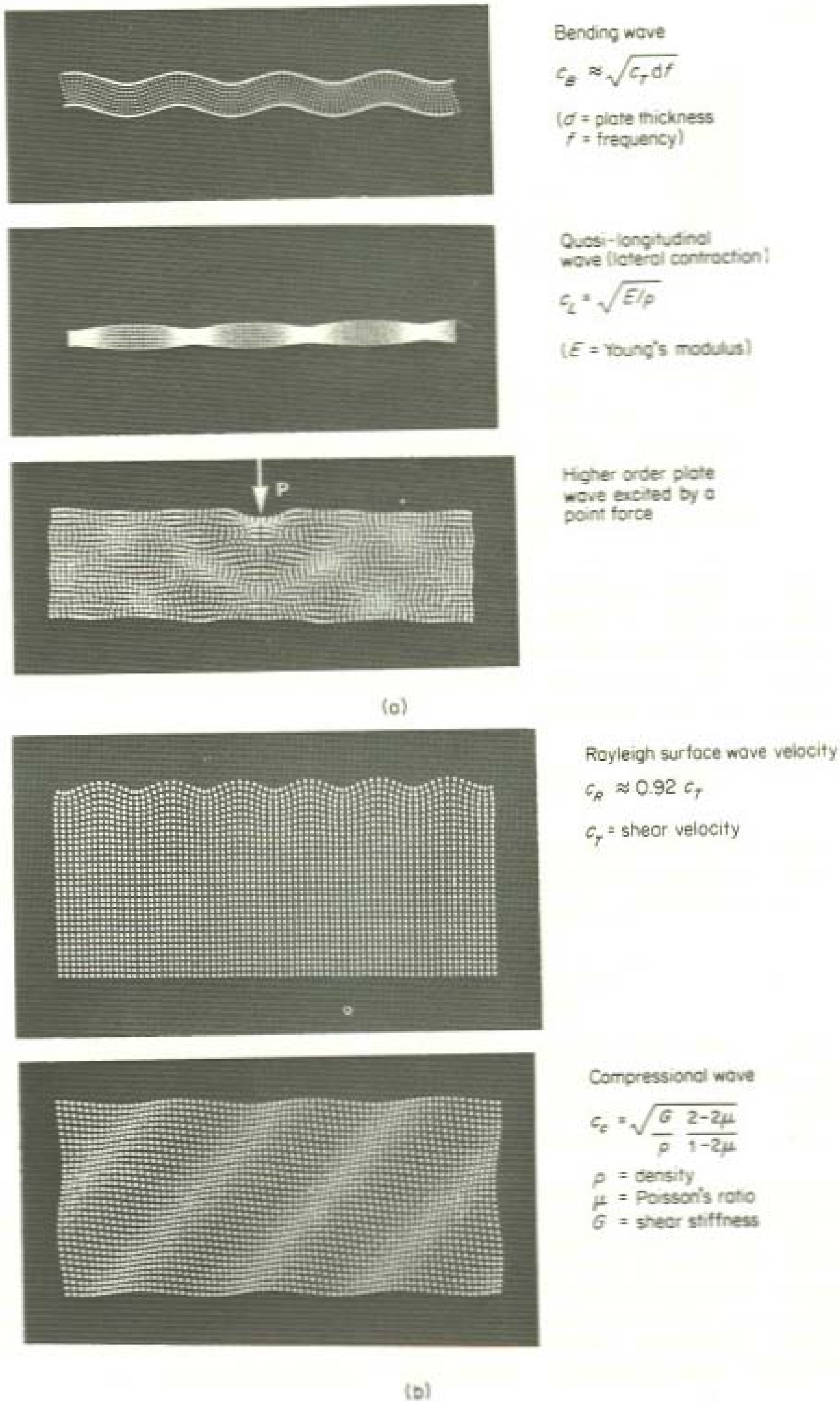


Figure 14.1 Different wave types in solids

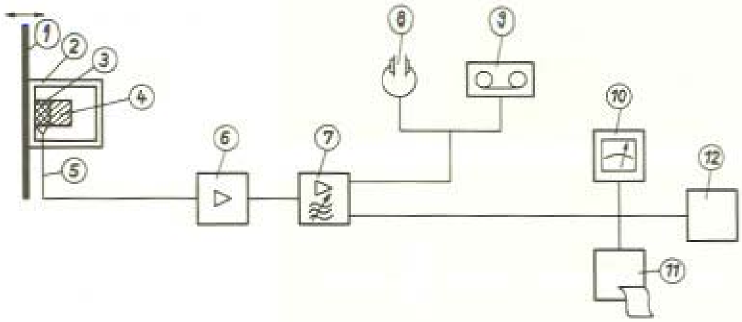


Figure 14.2 (1) vibrating object, (2) casing for accelerometer (not to scale), (3) piezoelectric material, (4) small mass, (5) cable, (6) preamplifier, (7) amplifier and filter, (8) earphones, (9) tape recorder, (10) scale instrument, (11) level recorder, (12) other data reduction (e.g. computer). Items (7)–(12) are the same as those used for normal sound signal processing

data reduction equipment (frequency filters, correlators, level recorders, etc.) that is available for normal sound signals. Figures 14.3(a) and (b) show some examples of measured structure-borne sound spectra. The data are given in terms of the velocity level which is appropriate for comparison with airborne sound data. It is defined as

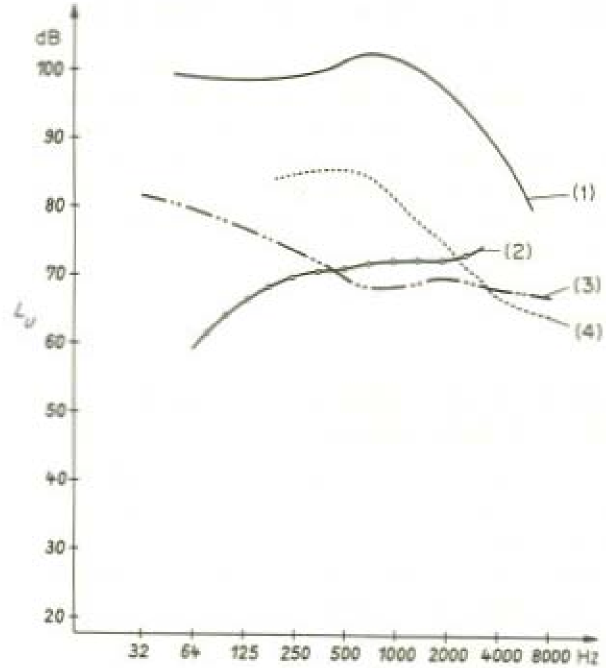
$$\text{Velocity level } L_U = 10 \log \frac{\bar{U}_2}{U_0^2}$$

Where $\bar{U}^2$ is the mean square value of velocity, and U_0 is a reference value. If the levels are given in term of spectra, the velocity level and the so-called acceleration level L_a are related by

$$L_U = L_a + 20 \log \frac{a_0}{2\pi f U_0}$$

where f is the centre frequency of the frequency band under consideration and a_0 is the reference value for the acceleration level. It should be noted, that the sound-*pressure* levels in front of a vibrating solid are hardly ever larger than the *velocity* levels as expressed in Figures 14.3(a) and (b).

Figure 14.3 (a) Examples of velocity levels per third octave. Reference value $5 \cdot 10^{-8}$ m/s: (1) Diesel engine on elastic mounts running at full load (2,500 Hp) at 1,700 r.p.m.; (2) Standard tapping machine acting on a 12 cm concrete floor; (3) Electric motor on elastic mounts running at 1,400 r.p.m.; (4) Gasoline engine on elastic mounts running at full load (4 Hp) at 4,000 r.p.m; (b) Examples of velocity levels per third octave. Reference value $5 \cdot 10^{-8}$ m/s. (1) Subway rail when train is passing at 60 km/h; (2) Maximum values of an elastically mounted elevator for six persons; (3) Wall of a house when a street-car is passing with 45 km/h at a distance of 13 m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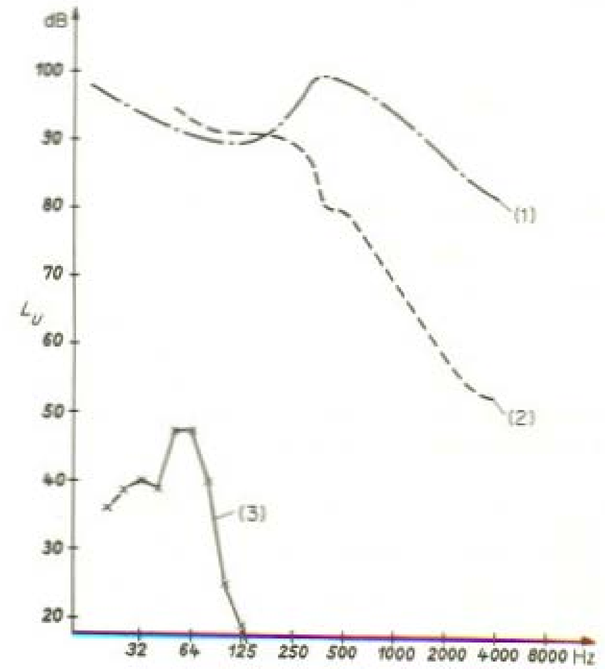
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14.4 REDUCTION OF STRUCTURE-BORNE NOISE AT ITS ORIGIN

The so-called primary noise control — is usually more effective and more economic than noise control which leaves the noise-excitation mechanisms unchanged but reduces the sound propagation by damping materials, enclosures, isolator shields, etc. (the so-called secondary noise control).

There are cases even where primary noise control shows some additional benefits* because there is no law of nature that states that good noise control has to be paid for by some sort of disadvantage.

The most common methods of primary noise control in the field of structure-borne sound are as follows:

14.4.1 Reduction of the Fluctuating Forces and Motions which are Basically Responsible for the Sound Generation.

This can be done by lowering the speed, by using very smooth surfaces when one body rolls over another one, by reducing the mass of those parts that have to undergo high acceleration.

An interesting case in this respect is the development of the typewriter. Originally, the rather heavy platen had to be accelerated and decelerated whenever a letter was typed. As a next step the letters were placed on a light sphere and only the sphere had to be moved. Nowadays the sphere is replaced by a very thin plastic disc which has a weight of less than 1 gram. The next step seems to be that the only moving parts are minute droplets of ink which are sprayed in such a way onto the paper that letters or other symbols are formed. Thus in the development process the mass of the moving parts has been reduced from kilograms to less than milligrams.

14.4.2 Modification of the Time History of a Process

In this case sudden changes, which are responsible for the annoying high frequency sound, are avoided.

Examples are cam drives that are designed in such a way that discontinuities in the motion or in its first derivative cannot occur. Other examples are found in gearboxes where the teeth are inclined to cause a smooth transmission of the force or in many other instances where a suitable mechanical design of a system allows for a smooth transition. All these methods rely on the fact that usually it

* It is known, for example, that sudden impacts cause a lot of noise, they also cause wear, thus if impacts are made smoother, the life time of a machine is likely to increase. Similarly, reduction of unbalanced masses in machinery rotating at high speeds decreases the noise and helps to avoid damage in bearings. As a third example, one may mention the fact that small clearances between the moving parts of machine tools reduce any rattling noise and at the same time improve the precision of the product. In short, quiet machines are better machines.

is not the magnitude of a force or a displacement which is responsible for the sound generation but its rate of change (i.e. the suddeness of an event). Quite often small alterations that are restricted to a few milliseconds of an event can have a remarkable effect on the noise production, provided that discontinuities are smoothed out. Modern flush toilets as compared to those that were built twenty years ago give a good example how noise control at the origin can effectively reduce the sound.

14.4.3 Detuning of Resonances and/or Interruption of Feedback Loops

Sometimes structural noise sources generate an almost harmonic tone with an amplitude that is much higher than the rest of the spectrum. Examples are resonances of electric motors or generators, large pipes, double walls, etc. and stick slip phenomena that generate screech sounds, rattling of lathes, etc. Since, in such cases, resonance effects and feedback mechanisms are involved noise reduction can be achieved by detuning the system or by interrupting the feedback loop. Unfortunately, this is said much more easily than it is done. The reason for this is that — apart from electric machinery where the situation is fairly well understood — an exact prediction of resonances and feedbacks is almost impossible. Therefore, one is compelled to design those structures that are apt to cause difficulties according to the best of the existing knowledge but allow for some changes whenever a prototype is available. A rather novel and effective method to improve resonating constructions is the so-called modal analysis. This method which usually consists of an impact excitation of a structure and many subsequent velocity or acceleration measurements gives the resonance frequencies as well as the nodal patterns of a vibrating structure. Thus it is possible to find out the frequencies that do not give rise to resonances and those points that have the least motion (i.e. nodal points) and which therefore are the most suitable as mounting points. Good results are also obtained when modal analysis is combined with the finite element method or when holographic methods are applied to study the vibration patterns of structures.

14.4.4 Compensation of Fluctuating Forces or Motions by their Opposite

Noise in solids usually can be described by some linear differential equations, i.e. the principle of linear superposition holds. Thus if a force F causes a velocity v, a force $(-F)$ causes a velocity $(-v)$, or in other words if both forces are present simultaneously the velocity vanishes. This principle can be applied in noise control whenever it is possible to make some lever arrangement that replaces fluctuating forces by fluctuating moments. There are even some attempts to add an artificial — usually electrodynamic — force generator that acts as an anti-noise source, which if properly phased reduces the total noise output.

14.5 REDUCTION OF SOLID-BORNE NOISE DURING PROPAGATION

It is a general principle in environmental engineering that a pollutant can be controlled the easier the nearer this is done to the source. It is also known that a single well-defined pollutant can be handled much better than mixture of many — possibly unknown — contributers to a pollution situation.

Applied to structure-borne noise control, this means that one should bring all methods of secondary noise control, such as damping, isolation, etc., as close to the original sound source as possible; it also means that a source which generates just one wave type at one frequency (so to say one noise pollutant) can be reduced much easier than one that produces many wave types in a random manner over the whole frequency range.

14.5.1 Resilient Layers

Resilient layers, such as rubber elements, springs, etc. are the most common way of reducing structure borne sound transmission. Figure 14.4 gives a few examples. They may act in two ways. If they are close to the noise source (e.g. floating floors, carpets or other resilient layers in buildings) they reduce the suddeness of impacts and thereby avoid high frequency noise excitation. If they are at some distance from the source (more than half a wavelength) they reflect the incoming structure-borne sound energy. Resilient engine mounts, the rubber elements separating the wheels and bogies of modern subway coaches, resilient clamps in piping-systems or suspended ceilings (spring damping) are examples. The efficiency of such devices can be quite high (20 dB or more) provided they are applied very close to the original source; and provided they are resilient.

As a rough rule of satisfactory resilience one may take that the resilient mount has to be much softer than those parts of the adjacent structure that lay within a quarter of a wavelength.

14.5.2 Vibration Damping (Thermal)

There are several materials, especially high polymers, that generate heat when they are vibrating. The amount of vibratory energy that is transformed into heat by this way, i.e. the structure-borne sound damping can be quite substantial. Materials are available that transform more than fifty percent of the vibratory energy into heat within one cycle. Unfortunately such materials have no structural strength therefore they are combined with other load carrying structural elements. Some constructions obtained this way are shown in Figure

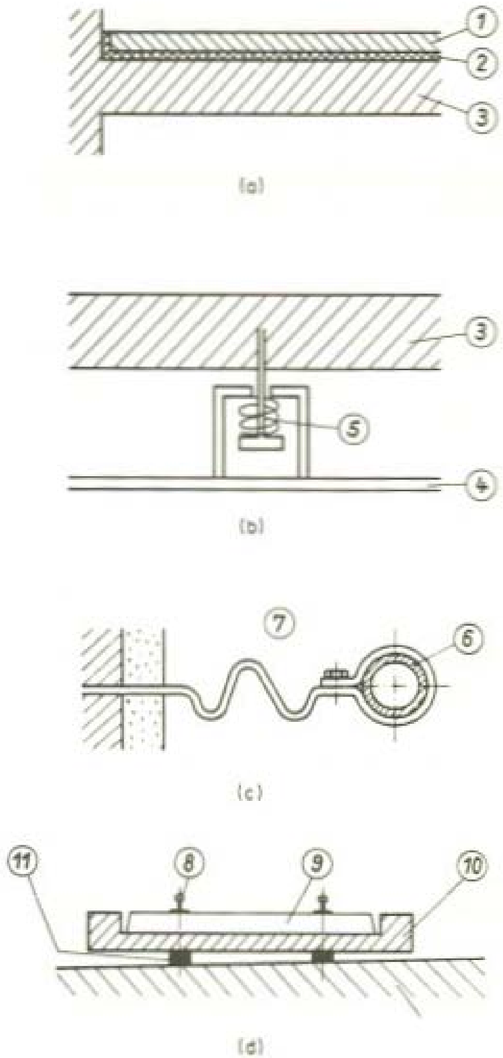
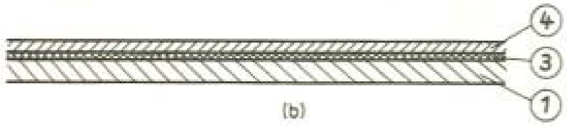


Figure 14.4 Application of resilient layers. (a) Floating floor; (b) Suspended ceiling on springs; (c) Flexible pipe clamp; (d) Resilient track mounting. (1) floating floor, (2) fibrous material appr. 10-mm thick, (3) structural floor, (4) suspended ceiling, (5) spring, (6) pipe, (7) resilient clamp, (8) rail, (9) sleeper, (10) heavy concrete beam, (11) elastomer mounts

(a)



(b)

Figure 14.5 Application of vibration damping. (a) Constrained layer; (b) Sandwich plate. (1) base plate, (2) damping layer consisting of a highly damped material with rather high elasticity, (3) thin shear layer, optimum thickness depends on thickness of base plate and cover plate, (4) cover plate

14.5. Other ways of structure-borne sound damping incorporate the friction in materials like sand (especially in architectural acoustics) or the viscosity of thin layers of air or oil between two thin plates (Figure 14.5). Vibration damping devices give a considerable reduction of resonant amplitudes and very effectively attenuate structure-borne sound over longer distances. When applied to constructions that consist of many different parts, they often are of limited use in the vicinity of the sound source.

14.5.3 Added Masses and Other Discontinuities

Structure-borne sound waves are reflected to a smaller or larger degree whenever they come to a discontinuity. Thus any change in cross section or material, any change in the direction of propagation (sound around corners and bends), as well as added masses give a certain reduction of structure-borne sound propagation. Usually the improvements obtained this way are rather small (e.g. the reduction at wall-joints in building is of the order of 3–10 dB). Only heavy added masses are of practical importance as noise-control devices. They give good results when they are applied to thin structures near the sound source and when the frequencies of interest are fairly high. Noticable improvements are achieved if an added mass is heavier than the surrounding structure within a quarter-wavelength.

14.6 FUTURE TRENDS

It is the opinion of this author, that in the near future research in structure-borne sound will concentrate on getting a better understanding of the different sound-excitation mechanisms. Such a better understanding will certainly help to find new ways of noise control at the origin. In many cases such investigations will incorporate the study of nonlinear effects, a subject that hardly has been touched in structure-borne sound. It may also be that such sophisticated methods as the use of antisound sources will be applied more and more.

With regard to structure-borne sound propagation, there are still many open questions with respect to buildings consisting of many prefabricated, large elements. It seems to be possible to design this type of buildings in such a way that the sound isolation is good; but the design rules are not yet quite clear. Another promising aspect is the availability of resilient materials (elastomers) that can withstand very high loads. With such materials it is possible to isolate very heavy machinery, complete buildings, and long stretches of subway or railway tracks.

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Furthermore, there are numerous papers on the topic in the following scientific journals: *Journal of Sound and Vibration*, *Journal of the Acoustical Society of America*, *Acustica*. There are also survey papers and short contributed papers in the proceedings of Internoise, FASE (Federation of the Acoustical Societies of Europe), DAGA (Deutsche Arbeits-gemeinschaft für Akustik) which appear every year or every second year.