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EXCITATION MECHANISMS FOR STRUCTURE-BORNE SOUND

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1. INTRODUCTION

Noise in the Nineties is something we shall have to live with. Our only hope - and as acousticians our goal - is to reduce this noise so that there will be a Quieter Britain and a quieter rest of the world.
In order to achieve this goal it will certainly be necessary to apply all the available and conceivable strategies; such as

- education; i.e. to make all people aware that noise is a nuisance and possibly a health hazard (especially for the others) and therefore all unnecessary noise sources should be avoided and whenever there is a choice, the quieter alternative should be used;
- legislation; i.e. to apply some pressure so that noise is reduced even if no immediate financial benefit can be seen;
- economy; i.e. to compare the cost of noise control with the long-term social costs that are caused by noise;
- planning; i.e. to make sure that noise aspects are taken into account at an early planning stage so that noise sources are concentrated and separated from potential quiet zones, etc. etc.;
- technology; i.e. to apply the existing knowledge in machine design to build machines and vehicles that produce only a minimum of noise, and to use all kinds of noise attenuation by enclosures, absorbers, mufflers, shields, dampers, elastic mounts, isolating walls, active control, etc. to reduce the noise that cannot be prevented at the source.

In this article I shall discuss only the technology aspects of noise control and even there I shall restrict myself to structure-borne sound.
The main reason for this restriction is that air-borne sound sources - especially those that generate noise by unsteady or turbulent flows - have already been investigated quite thoroughly by a great number of researchers, whereas the many different mechanisms that are responsible for structure-borne sound excitation have received much less attention.

2. IMPACT NOISE

Noise caused by the impact of two bodies is a very common sound source. It can be found in forge hammers, typewriters, rattling of loose parts, operation of saws, footfall noise, rain on the roof and many other occasions. Basically it consists of two parts [1,2].
There is the so-called acceleration noise which is due to the fact that even if all elastic deformations of the impacting bodies were ruled out there always would remain the sudden deceleration of the impacting body [3,4]. This deceleration, or velocity jump can be Fourier - decomposed giving a wide frequency spectrum (Fig. 1). If for each frequency the radiation properties of the decelerated body are known, the total radiated sound can be calculated. If numbers are put into the equations in Fig. 1 one finds that for example for a forge hammer, this acceleration noise in the worst case can give sound power levels close to 120 dB. This is certainly less than the total radiated sound power from a forge hammer, but it is nevertheless a surprisingly

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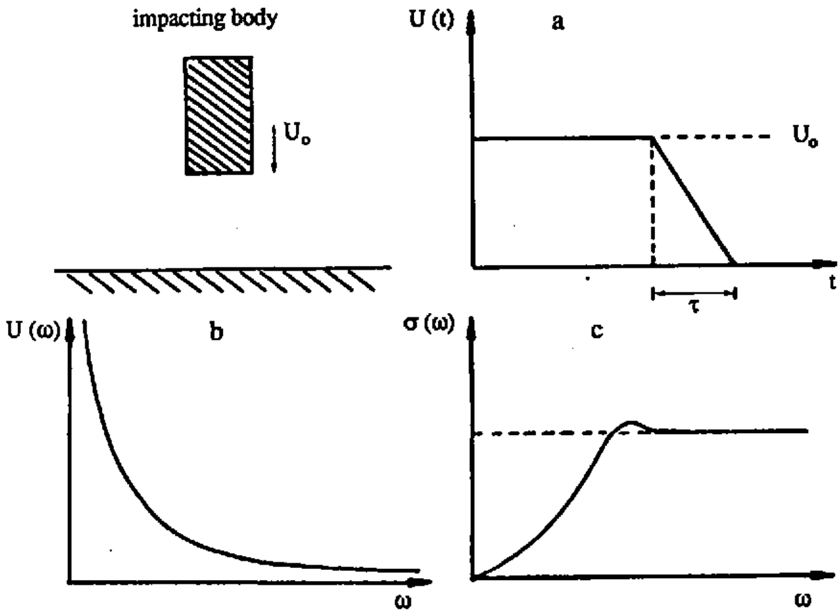

Radiated sound energy

$$E = \frac{1}{2}\,\rho\, V_v\, U_o^2 \left[1 - e^{-\frac{c\tau}{a}}\left(\cos\frac{c\tau}{a} + \sin\frac{c\tau}{a}\right)\right] \left(\frac{a}{c\tau}\right)^2$$

Spectrum shape

$$\sim \frac{1 - \cos\omega\tau}{4 + \frac{\omega^4 a^4}{c^4}}$$

Fig. 1 Acceleration noise
a. time history of velocity
b. spectrum of velocity
c. spectrum of radiation efficiency
ρ,c = density and speed of sound in air, V_v = virtual mass
a = radius of equivalent sphere, σ = radiation efficiency

high value. According to Sir Geoffry Taylor the sound energy of the acceleration noise is equal to the kinetic energy of the "virtual", hydrodynamic mass that continues to move when the solid body is suddenly stopped and therefore leads to a compression of air.

The second part of impact noise is the so-called ringing noise. It is caused by the vibrations of the bodies that are set into motion by the impact. The amplitudes of the vibrations are determined by

- the acting forces
- the input impedances and
- the propagating properties of the structure-borne sound in the excited structures.

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At low frequencies the spectrum of an acting force is solely determined by the transmitted momentum; the high frequencies are given by the pulse shape. The borderline between the two frequency ranges is given by the impact duration (see Fig. 2).

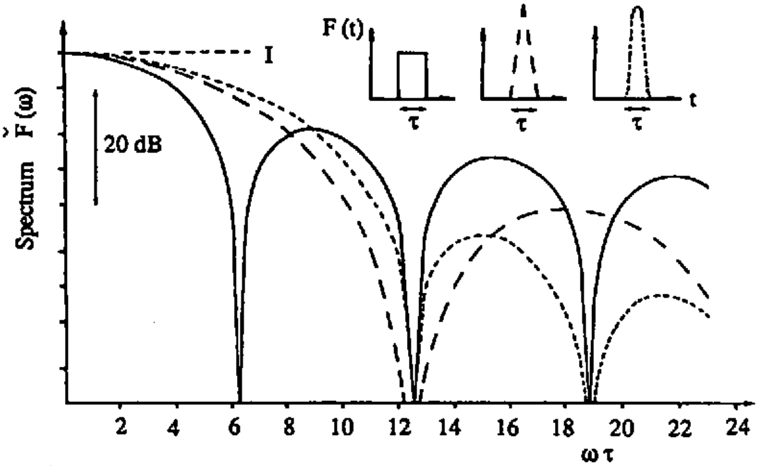

Fig. 2 Spectra of pulses of equal momentum and equal duration but different shape

$$I = \int F(t)\, dt = \text{momentum}$$

As can also be seen in the figure; the smoother the pulse the lower the high frequency content in the spectrum. This is the reason why - whenever possible - elastic layers or other methods (example see Fig. 3) are used to make impacts smoother.There are several other ways of pulse shaping but in all cases the aim is to smoothen it. Very often changes within a few milliseconds can give noticable improvements because they affect the high frequencies ($f > 1/\tau$) which are the most annoying ones for the human ear.

The input impedance of a structure which is excited by an impact, can influence the structure-borne sound in two ways. If the impedance contains a stiffness component (a negative imaginary part when the exp (jωt) notation is used), it changes the pulse shape of the impact just as any resilient layer would do. The magnitude of this local elasticity depends very strongly on the contact area and the shear stiffness of the material. As a good estimate one can assume that the contact stiffness is

$$s_C = \frac{\pi}{2}\, GD/(1-\mu)\ . \qquad (1)$$

Here G = shear modulus, D = average diameter of contact area, μ = Poisson's ratio. This formula holds when the thickness of the excited structure is greater than twice the diameter D. When the impacting body has a mass m, the force spectrum in the range $f > (1/2\,\pi)\,(s_c/m)^{1/2}$ is reduced.

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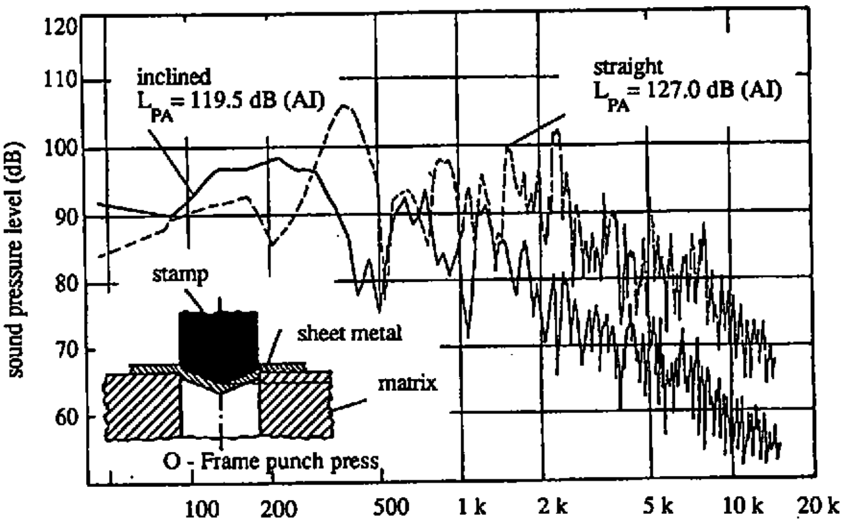

Fig. 3 Noise reduction of a punch press by pulse shaping (after [5])
a. stamp with straight surface
b. stamp with inclined surface
L_{PA} = A - weigthed sound power level

Quite often the contact area is a function of the acting force; in this case one either works with the maximum contact area or has to apply Hertzian contact theory (see e.g. [6]).
The mechanical power P(ω) which is transmitted by a force into a structure is given by

$$P(\omega) = \frac{1}{2} |F(\omega)|^2 \, \mathrm{Re}\,\{1/Z\} \ . \qquad (2)$$

Here Z is the impedance and F(ω) the spectrum of the acting force (after the influence of local elasticity has been taken into account). Since impacts contain a wide frequency band, it is usually sufficient to use for Z the value of "the corresponding infinite system"; i.e. to assume that for the power transfer only waves travelling away from the excitation point are important. The underlying assumption here is the same as in the corresponding air-borne sound problem: in the frequency average the sound powers transmitted by the same source into a reverberant room and into an anechoic room are identical.
There are lists of input impedances in the literature [7,8]; they show that a good estimate for the value of Z to be introduced into Eq. (2) is

$$\mathrm{Re}\,\{1/Z\} \approx 1/\omega\, m_\lambda \ . \qquad (3)$$

Here ω is the angular frequency, and m_λ the mass of the structure that lies within a sphere of radius λ/3 around the excitation point. λ is the wavelength in the structure. For a plate of thickness h and density ρ one would find $m_\lambda = \rho\, h\, \pi\, (\lambda/3)^2$, with λ being the bending wavelength. It

should be noted that the value of Z as given by Eq. (3) may differ considerably in its frequency response from the value one would measure by a direct impedance measurement. From the noise control point of view a consequence of Eqs. (2) and (3) is to make a structure as stiff as possible in the regions where the impacts are acting.
The structure-borne sound powers that are transmitted by pulses appear to be low. If for example a concrete floor of thickness h [m] is excited by the standard tapping machine (hammers of 0.5 kg falling from a height of 4 cm in intervalls of 0.1 s), the transmitted power per octave (for $f < 2000$ Hz) is only $6.7 \cdot 10^{-6}\ (0.15/h)^2\ f$ [Watt], which is orders of magnitudes less than the power that is necessary to drive the machine. (f = mid frequency in Hz).
A detailed discussion of the third aspect, the propagating properties of sound in structures, is beyond the scope of this paper, which is restricted to excitation mechanism. But it is worth while to compare the structure-borne sound power P that is transmitted into a structure with the radiated air-borne sound power P_{air} . The ratio of these two quantities is given by the statistical energy analysis [9]. It is

$$P_{air}/P \approx \frac{\eta_R}{\eta_R + \eta} \quad . \tag{4}$$

η is the loss factor of the structure in the vicinity of the excitation region and η_R is the radiation loss factor which for plate-like structures is approximately given by

$$\eta_R \approx \rho_0\, c_0\, \sigma / \omega\, \rho\, h\,. \tag{5}$$

ρ_0, c_0 = density and speed of sound in air, σ = radiation efficiency, ρh = mass per unit area of the structure. For the above mentioned concrete floor one can assume $\sigma = 1$, $\eta \approx 0.01$, this would give

$$P_{air}/P \approx 0.18\ \left(\frac{100}{f}\right)\left(\frac{0.15}{h}\right)$$

(f in Hz, h in m). It is seen that under standard conditions a surprisingly large fraction of the structure-borne sound power is radiated as air borne sound. For highly reverberant structures - and for practically all structures under water - a reasonably realistic upper limit is given by assuming that the structure-borne sound power and the radiated sound power are equal.

3. SUDDEN RELEASE OF STORED POTENTIAL ENERGY

When a spring is under tension it contains a certain amount of potential energy. If the spring is suddenly released, the stored potential energy is converted into kinetic energy, back to potential energy, etc., etc. until it is dissipated by damping. This way the spring is set into rather vehement motion, which, when the frequency of resonance is high enough, radiates sound.
An example in this respect is a punch press, where apart from the impact of the stamp (see previous section) there is a second, usually stronger excitation mechanism. It works the following way: when the stamp is cutting through the work piece, a rather strong force is built up (see Fig. 4); this force also acts on the frame of the press and tends to elongate or to open it (in case of a so-called C-frame); thus the whole frame becomes a spring under tension; when the cutting process is completed; i.e. when the stamp breaks through the sheet metal the frame is suddenly released and vibrates in its resonances. This way sound is radiated (see Fig. 4).

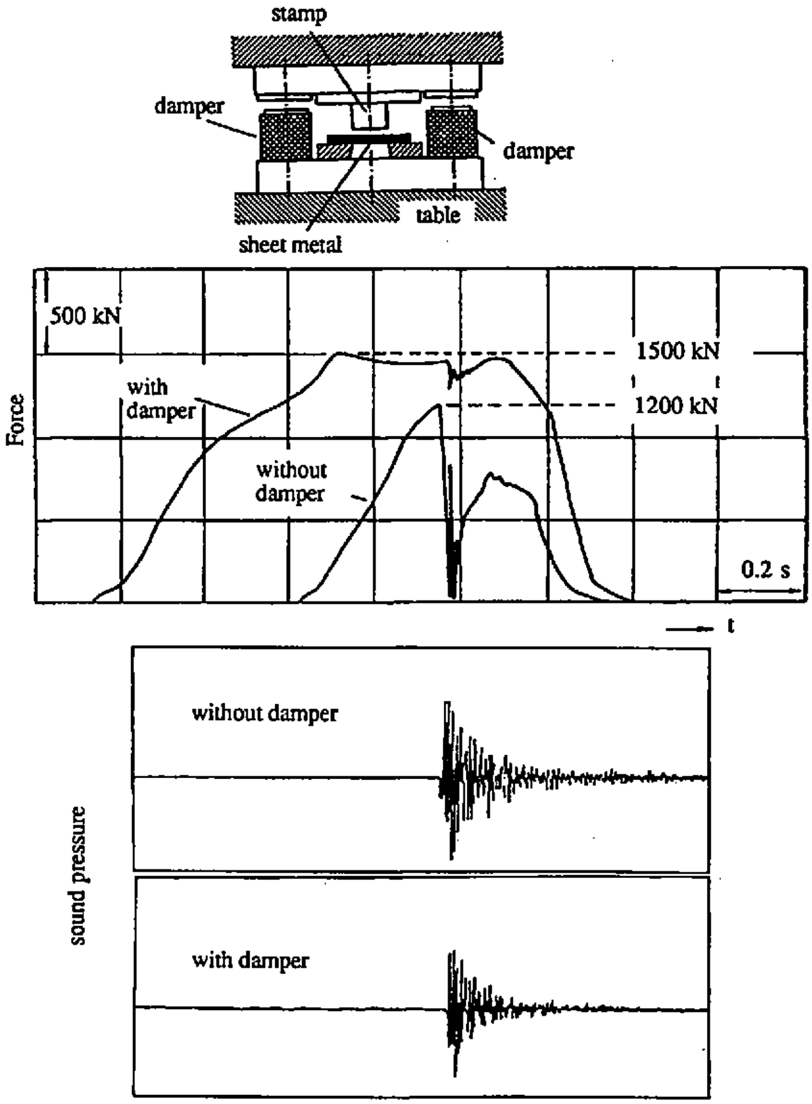

Fig. 4 Forces and sound pressures of a punch press after [5]. The time scale in the pictures is different.

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A way to reduce this type of excitation is to install a damper which comes into action just at the moment when the stamp breaks through the sheet metal. This way the total force may increase (see Fig. 4 center part) but the force variation (i.e. the time derivative of the force) becomes smaller and this leads to less sound excitation. The design and operation of such dampers is a rather delicate problem especially when they are actively controlled; but at least for low speed punch presses they gave reductions of up to 15 dB(A).
Noise generation by the sudden release of potential energy is also found in other occasions. One example is the more or less random cutting force that acts at the interface between tool and workpiece in lathes, milling machines, drills, grinding machines, etc.. In all these cases the cutting force undergoes small fluctuations because the material to be cut and removed is not perfectly homogeneous. Thus at one instant the cutting force increases a little, bringing higher tension and more potential energy into the system and at the next instant, when a minute piece of material breaks away, the force is reduced and potential energy is released. Fig. 5 shows narrow band spectra measured on a lathe [10]. It can be seen that a wide frequency band is excited because there is a large number of minute "release pulses" at random intervalls.
A very similar phenomenon occurs when paper or textiles are torn apart. In these cases the existence of fibres and threads makes it quite obvious that there is a rapid sequence of tension increases and releases that acts on the membranes which are formed by the material.

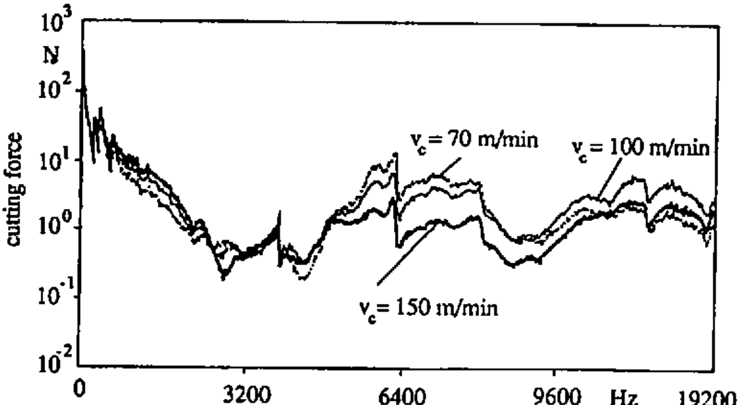


Fig. 5 Spectrum of the cutting force for different speeds measured on a lathe [10]

4. PARAMETRIC EXCITATION

In mathematical terms the situations described in the two previous sections could formally be represented by the matrix equation

$$m\ddot{\xi} + r\dot{\xi} + s\xi = F(t) \,. \tag{6}$$

Here m, r, s are matrices representing masses, dampers, and springs, $\ddot{\xi}$, $\dot{\xi}$, ξ are the acceleration, velocity, and displacement vectors of the vibrating structures, and F(t) is the vector of the outside forces. In Eq. (6) it is assumed that continuous structure are discretised. An important condition underlying Eq. (6) is that the parameters m,r,s do not change in time.

This condition is not always fullfilled as is well known from the many investigations on pendula with time varying lengths and of similar phenomena. The most simple mathematical representation of systems with time varying parameters is

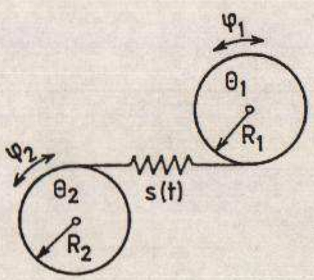
$$m(t)\ddot{\xi} + r(t)\dot{\xi} + s(t)\xi = F_{const}\,. \tag{7}$$

Now there is no more a fluctuating force acting from outside (F_{const} is a constant), instead the mass, damper, and stiffness matrix depend on time.
To get an idea of one important aspect of Eq.(7) all quantities are expressed by their constant and fluctuating parts, i.e. $m(t) = \overline{m} + \tilde{m}$; $r(t) = \overline{r} + \tilde{r}$; $s(t) = \overline{s} + \tilde{s}$; $\xi(t) = \overline{\xi} + \tilde{\xi}$. This way one finds

$$\overline{m}\ddot{\tilde{\xi}} + \overline{r}\dot{\tilde{\xi}} + \overline{s}\tilde{\xi} = (F_{cost} - \overline{s}\overline{\xi}) - \tilde{s}\overline{\xi} - (\tilde{m}\ddot{\tilde{\xi}} + \tilde{r}\dot{\tilde{\xi}})\,. \tag{8}$$

This expression has much in common with Eq. (6) because the left hand side now has only time invariant parameters. Furthermore the first term in brackets on the right hand side usually vanishes, the last term in brackets is of second order. This leaves the term $-\tilde{s}\,\overline{\xi}$. Thus one can say that in the case of small amplitudes and small parameter variations a system with time varying parameters behaves almost like an invariant one with an outside force that is given by the variation in stiffness and the static deflection. In other words: parameter variation or parameter excitation has - for small amplitudes - much in common with excitation by outside forces. For larger amplitudes additional phenomena, such as frequencies of order 1/2, 1/4, etc., and self excitation, can occur.
The discussion following Eq. (8) makes it clear that parametric excitation is most likely to occur when there are strong stiffness fluctuations and when the static deflection is large; i.e. when the system is heavily loaded. Practical examples where this is known to happen are heavily loaded gears and motion over periodic supports.
In gears the time varying parameter is the stiffness of the teeth. In a first approximation each tooth can be considered as a short cantilever beam. The stiffness of this beam varies, because the point of contact changes as two teeth roll on each other. Fig. 6 shows a very simple gear model with the moments of inertia Θ_1 and Θ_2 representing the gear wheels and the stiffness s (t) representing the variable tooth stiffness.The figure also gives the stiffness variations when in the average 2.2, 2.56, 2.91, 3.0 teeth are in contact. The total deflection of a teeth under load is of the order $20\cdot10^{-6}$; this number, however, does not say very much because it is the variation of this deflection that causes vibration and sound. Sound generation of gears by parameter excitation is especially important for gears under heavy loading. It is, however, not the only noise source. There is another mechanism which is due to the force fluctuations that are caused by geometric irregularities of the teethed gears (roughness, unequal spacing, etc.). This second mechanism is the main sound source when the loading is light and therefore the tooth deflection is small. Both mechanisms have been investigated very thoroughly. Much information about this topic and about means to reduce gear noise by design changes can be found in [11-16] and many other papers.

Another usually low frequency parametric excitation occurs when a heavy body is rolling over a beam on supports. In this case the stiffness "seen" by the rolling body is different at different positions. Fig. 7a shows a model that was used to make numerical simulations of this phenomenon. Fig. 7b gives some results for the rail motion at different times and for the wheel motion as it runs over the rail. At present this mechanism does not seem to be of major importance, but it should be kept in mind that high speed trains pass one hundred or more ties per second and therefore experience rapid stiffness variations.



$$m_{eq}\frac{d^2X(t)}{dt^2}+s(t)X(t)=F$$

$$m_{eq}=\frac{\theta_1\theta_2}{\theta_1R_2^2+\theta_2R_1^2}$$

$$x(t)=R_1\varphi_1(t)-R_2\varphi_2(t)$$

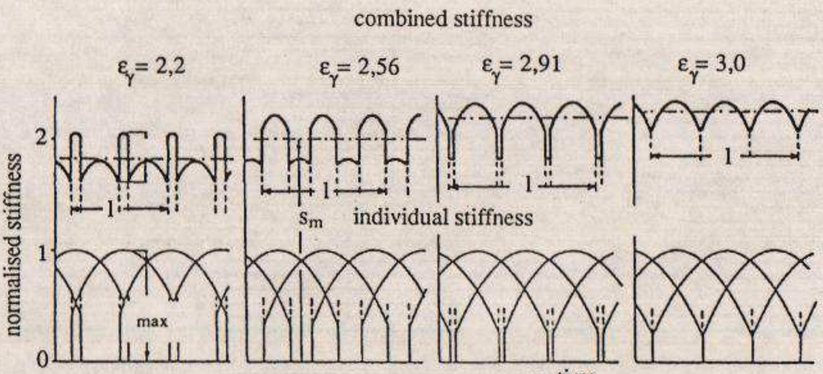


Fig. 6 top: simplified model of gear noise generation by parametric excitation
Θ_1, Θ_2 = moments of inertia, s (t) =variable tooth stiffness
bottom: tooth stiffness as function of time, l = length of contact

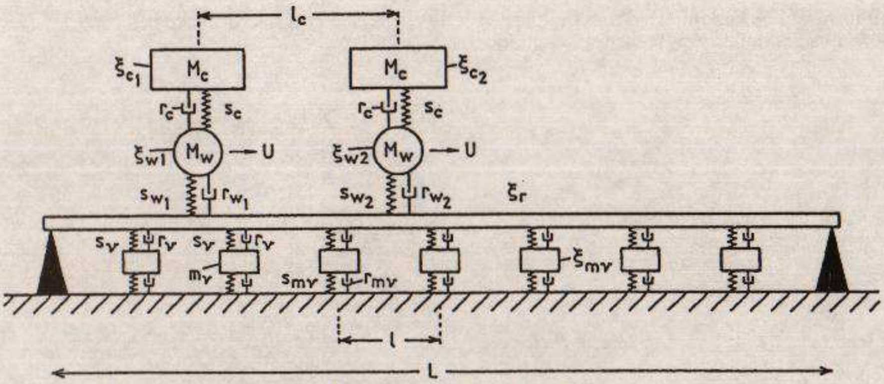
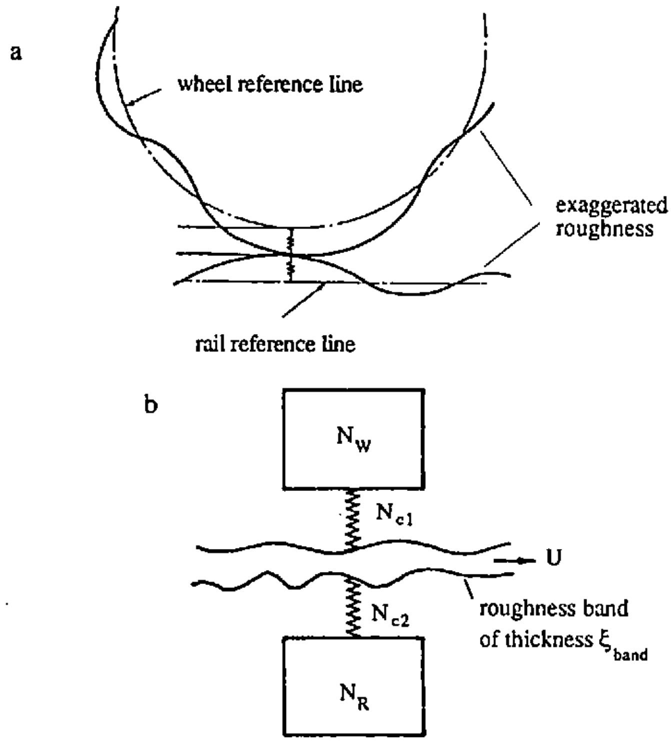


Fig. 7a Model of supported rail section with many supports. Solely parametric excitation by two wheels.

Fig. 7b Time history of velocity "measured" at the wheels and at a different positions of rail. Trainspeed 100 km/h. Support stiffness $s_v = 100000$ kN/m; $r_v = 5.6$ kNs/m. Static rail deflection 0.3 mm.

5. ROLLING NOISE

Rolling noise is almost ubiquitous because it is responsible for the noise of railways, tyres, and bearings. A good understanding of that part of the railway noise which is caused by surface irregularities of rails and wheels is obtained by the "gramophon theory". This model is based on the assumption that rolling noise is generated by the "playback" of the surface irregularities [18, 19]. Fig. 8 shows the underlying model in its most simple form.



$$\xi_{rail} = \frac{\xi_{band}}{1 + \frac{N_W + N_C}{N_R}}$$

$$\xi_{wheel} = \frac{\xi_{band}}{1 + \frac{N_R + N_C}{N_W}} \quad ; \quad N_C = N_{C1} + N_{C2}$$

Fig. 8 Rolling noise generation by surface irregularities
a. : actual situation
b. : idealized model

It consists of the wheel represented by its resilience N_w, the contact stiffness represented by its resilience $N_c = N_{c1} + N_{c1}$ and the rail represented by N_R. The contact stiffness is divided into two parts and between them a band of variable thickness ξ_{band} (x) is pulled through with speed U. This way a time dependent deflection ξ (Ut) is generated which excites rail and wheel. Since the

forces acting on rail and wheel have to be of equal magnitude but of opposite sign the formulas shown in Fig. 8 can easily be derived. The main conclusion to be drawn out of them is that the component which is softest (i.e. has highest resilience) will vibrate most strongly. Thus it would be advantageous from the noise control point of view to make the contact as soft as possible so that it will "swallow" the surface irregularities and leave rail and wheel almost unaffected. In principle this could be done by using other materials [20] or by properly shaping rail and wheel. Unfortunately practical considerations (especially wear) do not allow major changes in existing rail-wheel systems. As a consequence of that the surface irregularities are almost fully transmitted to the rail or the wheel. Generally at low frequencies the rail is softer than the wheel therefore it vibrates more and (together with the ties between the rails) radiates more. At frequencies above 1000 Hz standard railway wheels have several resonances therefore they are effectively softer (easier to be excited) than the rail and consequently they vibrate and radiate more. In [18, 19] the model has be refined taking into account the fact that the contact area has a finite size (with typical dimensions of 10 - 15 mm) and that motions of all directions are excited.
Apart from the obvious noise control measures such as making the contact area softer and keeping rails and wheels as smooth as possible, one can also get moderate improvements (ca. 5 dB) by applying damping material to wheels and rails and by designing wheels in such a way that the acting forces in radial direction excite only a minimum of axial motion [21]; i.e. wheels should be made as symmetrical as possible.
According to the "Gramophon theory" a rail/wheel system which is geometrically smooth would be perfectly quiet. There is some doubt whether this conclusion is correct. There seem to be other, less important noise generating mechanism in the rolling process. One of them is the already mentioned parameter excitation when a body is moving over a rail of varying stiffness. The stiffness variations may be caused by the supports or by changes in the surface hardness. Another conceivable mechanism might be due to the fact that irregularities of the order of 10^{-5} to 10^{-6} m are sufficient to generate high frequency noise; therefore small vibrations of this amplitude (possibly caused by parametric excitation) can have the same effect as surface irregularities which in turn give rise to vibrations, that are seen as time dependent surface irregularities, etc. etc.. It is not yet known wether this mechanism is of any practical significance. All that can be said is that in the case of gear noise, which has much in common with rolling noise (geometric irregularities, parametric excitation), the corresponding mechanism is important.
For tyre noise generation there is no model as simple as the "Gramophon theory".* The reason is the much greater complexity of tyre noise. The reasons are:

- the speed of structure-borne sound waves around a tyre is comparatively low: 60 - 100 m/s for bending waves, appr. 400 m/s for longitudinal waves [23,24];
- the contact area is large compared with the wavelengths at high frequencies and large compared with the size of tire irregularities such as tread blocks;
- since the contact area is so large, tangential motion in this contact area is of great importance, this brings in the highly non-linear effects of friction;
- a tyre is a complicated structure with double curvature, consisting of several materials and having many inhomogenuities in form of tread blocks.

There are many investigations (see e.g. [24 - 30]) treating the different aspects of structure-borne sound generation in tyres. The most detailed one seems to be [24], where blank tyres on rough roads are considered. The basic ideas in this approach are

- at frequencies below 400 Hz a tyre behave like a ring under tension;
- in the frequency range 400 - 2500 Hz a tyre can be approximated by an orthotropic plate;

* The simple air pumping theory [22] is not treated here because it is valid only in special cases; furthermore it does not involve structure-borne sound.

- for the two frequency ranges the Green's function (or impulse response function) can be found by standard techniques, the Green' function fully represents the linear structure-borne sound properties of the tyre;
- a time domain calculation is used to find the forces acting on each surface element in the contact area, to this end for each time increment the static load, the inertia forces, the surface roughness, and the deflections from previous times and neighboring surface elements are taken into account, the calculations are non-linear because the contact forces can only compress and not expand (adhesion is neglected) the tyre in radial direction;
- a convolution of the forces with the Green's functions gives the tyre surface velocity, this quantity then has to be transformed into a stationary frame to find the radiated sound pressures.

Fig. 9 shows examples of calculated and measured surface velocities and sound pressures [24]

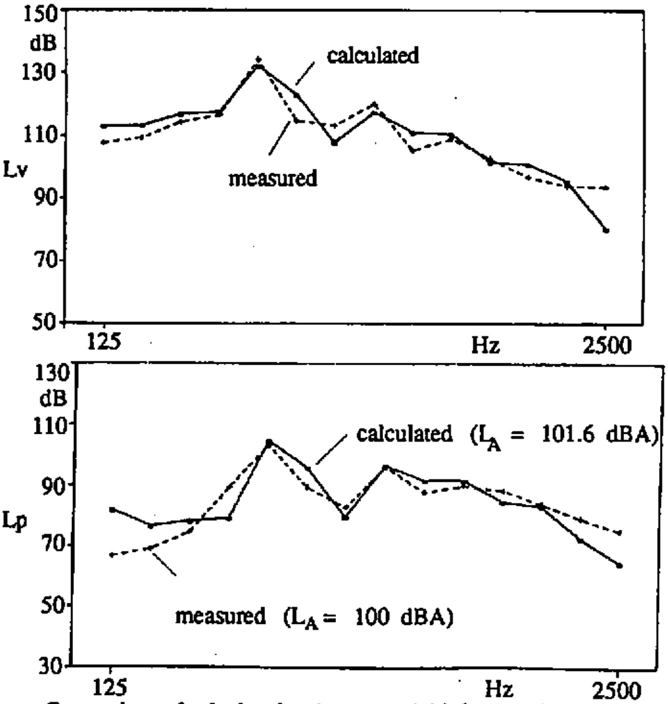


Fig. 9 Comparison of calculated and measured third octave levels of tyre noise of a blank tyre on a drum with rectangular roughness pattern (i.e. cross bars of 1 mm height and a wavelength of 6 cm ± 5 %). After [24].
top : velocity levels (re $5\ 10^{-8}$ m/s) near leading edge, speed : 70 km/h
bottom: sound pressure level in 1 m distance at a speed of 70 km/h

The model was also used to make some parameter studies. They showed that there is not much potential for noise reduction by making small changes in present day tyres. This is not very surprising because tyres have been optimised to a large degree in a long process of trial and error.

Wether radical changes such as other diameters, greatly reduced width, other combinations of hard and soft materials have a realistic chance remains to be seen.
At present the main hope in the field of tyre noise reduction is the use of sound absorbing or even resilient road surfaces [31, 32].

6. STICK-SLIP PHENOMENA

The most common feedback mechanism in structures-borne sound is the self excitation of vibrations by the stick-slip phenomenon. It consists of a rapid sequence of a very short slipping motion followed by a brief period of stick. This mechanism is responsible for the screech of railways in curves, the braking sreech and several other unpleasant sounds that can be produced by chalk on a blackboard, knife on a plate, poorly greased hinges etc.. Stick-slip also is the origin of - the hopefully more pleasant - violin sound. Whether the low frequency rattling of lathes under certain operating conditions is of stick-slip type, is a matter of taste, but it certainly is very closely related to it.
In acoustics the stick-slip phenomenon which is understood best is the violin sound (see e.g. [33-36]). The many investigations made on this topic showed that for a certain time the string moves with the bow (sticking phase) and then rather quickly jumps back (slipping phase). This gives the typical saw-tooth motion (Fig. 10).

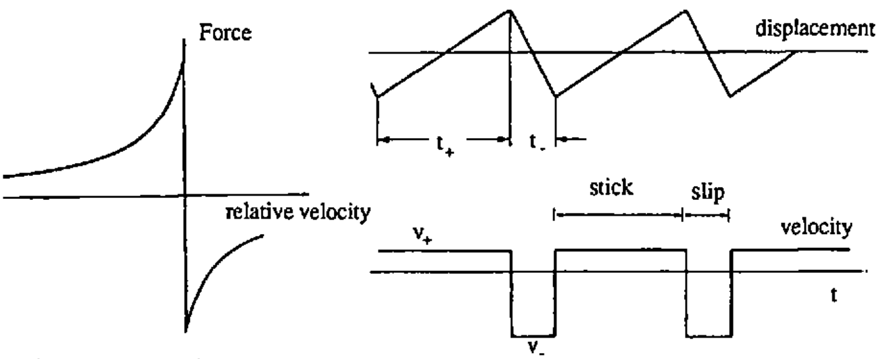


Fig. 10 Force velocity characteristic, displacement and velocity of a violin string excited by a bow

For a mathematical treatment of violins and other stick-slip sounds the most convenient way seems to be the following [35]:
- the string (or other excited structures) is represented by its Green's function;
- for each time increment the forces and velocities in the contact region are calculated using the highly non-linear relation between these two quantities.

This way the characteristic feature of the force/velocity curve can be taken into account. It consists in a decrease of the friction force as the velocity is increased and therefore can lead to instability (i.e. conversion of bow energy into vibratory energy of the string) and finally to a limit cycle. Even the onset of the vibration can be found by this method. Thus the physics of the stick-slip excitation is well understood. But unfortunately it relies on the non-linear force/velocity curve of friction which up to now can only be estimated. Attempts to solve the "inverse problem"; i.e. to

find the force/velocity curve from measured string vibrations have - to the authors knowledge - not yet been successful.
On the practical side the most effective way of controlling stick-slip noise is to change the force/velocity characteristic. Sometimes this can be done by greasing or by bringing liquids into the interface so that the coefficient of friction is changed. Damping is also a possibility provided it absorbs more energy per cycle that is supplied by the stick-slip; in essence this means that damping helps mainly when the mechanism is only slightly unstable.
Sometimes it is possible to avoid stick-slip by making design changes that eliminate the dangerous direction of motion that leads to screech. In railways this can be done [37] by making provisions which allow the two - normally rigidly connected - wheel-sets of a bogie to be steered independently in a curve.

7. OTHER SOURCE MECHANISMS

7.1 Unbalanced forces in rotating machines

It is well known that unbalanced masses in rotating machines cause forces that have the periodicity of the number of revolutions. Since the amplitude of these forces grows with the square of the speed they are especially dangereous at high RPM and require careful balancing.
Another type of unbalancing occurs in reciprocating engines, when there are pairs of equal pistons in opposite phase. In this case the unbalanced force having the frequency of the revolution is compensated, but higher harmonics occur because the system consisting of crank shaft, connecting rod, and piston does not make a purely sinusoidal motion even if the angular speed of the crank shaft is absolutely constant. It can be shown that the remaining unbalanced force has the frequency of twice the number of revolutions and that its amplitude is $2\,\omega_0^2\,ma^2/l$, ω_0 = angular frequency, m = mass of one piston, a = crank radius, l = length of connecting rod.
The sound caused by this force is very important for cabin noise in automobiles in the frequency range 100 - 300 Hz.

7.2 Electromagnetic forces

In electric motors and generators there are strong electromagnetic fields. The major part of the forces generated this way is stationary or moving with a constant speed. If all structures were continuous such forces could not generate sound (as long as the speeds are subsonic). This, however, is not the case; real life motors are highly discontinuous with segmented strators and rotors. This leads to small force fluctuations which excite the structures and lead to sound radiation (details see e.g. [38 - 40]).
In electric transformers there are in addition to the electromagnetic forces rather strong magnetostrictive forces which lead - because of the square law dependence between electric current and force - to frequencies which are twice the a.c. frequency and harmonics thereof.

7.3 Hydrodynamic forces

Hydrodynamic forces are generated by vortex generation, turbulence and cavitation. They can lead to violent structure - borne sound excitation and radiation as one can hear near water taps, pressure reduction valves, high speed flow in pipes, ship propellers, turbulent boundary layers at aircraft fuselages, etc. Since these questions are closely related to flow noise, they are not treated here (see e.g. [41,42]).

7.4 Nonlinear phenomenon

Apart from stick - slip there are other nonlinear phenomena that lead to structure - borne sound. Since they are not yet very thoroughly investigated, only a short list is given :

EXCITATION MECHANISMS FOR STRUCTURE - BORNE SOUND

- snap - through noise which is known from jumping crackers and from the thin metallic sheets that are used to simulate thunder in theaters; in ordinary life this type of noise is generated when large metallic sheets are transported or when they are straightened. Wether rustling of paper or plastic foils is of this type does not seem to be known;
- buckling is very similar to snap - through noise because is needs a certain initial force and then leads to a sudden unstable motion;
- piston slap in internal combustion engines or other phenomena whereby a body moves between parallel constraints with a small amount of clearance; in this case the the stiffness "seen" by the moving body changes between zero (when it is out of contact) and a very high value (when it hits a constraint); this can lead to an amplification of a small initial deviation to a fairly large sidewise motion;
- in recent years the interest in deterministic chaos has grown tremendously; this has also affected structure - borne sound research; as an example [43] Fig. 11 shows the frequency response of thin foil where the vibration amplitudes were much larger than the thickness.The existence of a chaotic motion can be seen; other examples in this field are investigations on gear rattle [16] and bearing noise [44]

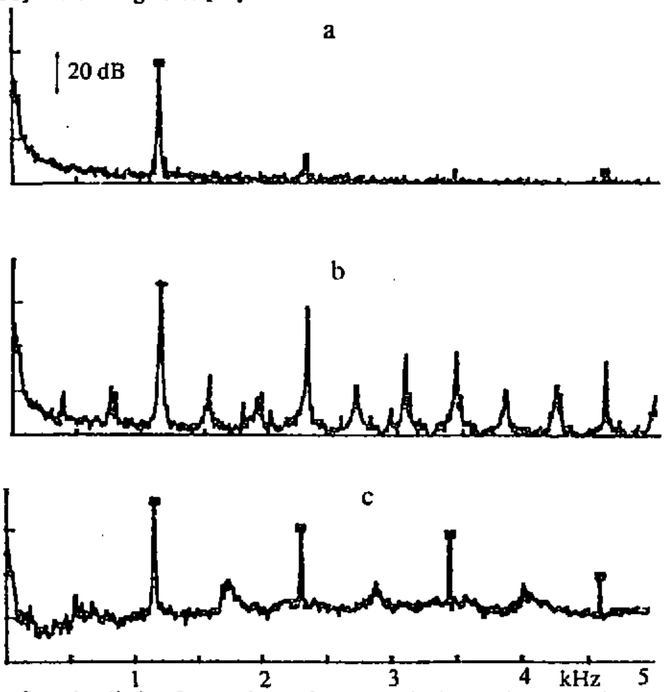


Fig. 11 Sound radiation from a sheet of paper excited by a sinusoidal force at 1150 Hz
a. : low amplitude (5.9 units); single frequency
b. : medium amplitude (26.4 units); harmonics and subharmonics
c. : high amplitude (58.4 units); broad band noise plus few harmonics

8. CONCLUDING REMARKS

Basically research in structure - borne sound is an application of the principles and methods that were found and developed by famous scientists such as d´Alambert, Bernoulli, Langrange, Helmholtz, Kirchhoff, Love, Lamb, and most notabely Lord Rayleigh, whose books on the Theory of Sound, after more than 120 years, are still a "must" for everyone working in this field. The fact, that the principles are old, does, however, not mean that work in this field is outdated. According to my opinion the contrary is true. Research on the generation of structure - borne sound is an intellectually very challenging field.Although the fundamentals are old it involves interesting, modern scientific fields such as machine design, material sciences, contact mechanics, modern developments in dynamical systems, numerical mathematics, advanced measurement techniques etc. This also seems to be seen by a slowly growing - but still too small - number of students who do not want to work in the glamorous and fashionable modern fields of science but instead intent to combine good scientific work with an attempt to make the world a little quieter.

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