Noise of sliding rough contact

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Abstract. This article is a discussion about the origin of friction noise produced when rubbing solids having rough surfaces. We show that noise emerges from numerous impacts into the contact between antagonist asperities of surfaces. Prediction of sound sources reduces to a statistical problem of contact mechanics. On the other hand, contact is also responsible of dissipation of vibration. This leads to the paradoxical result that the noise may not be proportional to the number of sources.

1. Introduction

Friction of solids having rough surfaces produces a typical wideband noise with generally a low sound level. This is for example the case when rubbing hands or pushing a small object on a table. Other examples are the sounds produced by a piece of sandpaper or a scouring sponge on a saucepan. In all these examples, the contact is weak and mechanical interaction is confined to the interface region at the top of surface asperities. All these sounds may be referred to as roughness noise [1].

This study focuses on the question of the mechanical origin of roughness noise. We present simple experiments and numerical simulations which aim to clarify what happens into the contact and leads to the noise emission. Discussion is conducted in the case of flat metal samples whose surface have been unpolished.

2. Acoustic characterization of roughness noise

A simple way to produce a roughness noise is to rub against each other two nominally flat pieces of metal (figure 1), such as proposed in Ref. [2]. Surfaces have been prepared by electric-discharge machining which leads to a random topography. Figure 2 shows the probability density function of heights of asperities of the samples. The values of quadratic roughness (standard deviation), Skewness and Kurtosis are specified. The Kurtosis is close to 3, the theoretical value for a Gaussian distribution. The distribution is clearly Gaussian, homogeneous and isotropic.

Several remarks must be pointed out concerning noise emission.

The sound level increases with sliding velocity as it can easily be checked with the metal samples (try it rubbing your hands!). The evolution law is of exponential type [3]

$$\Delta L_{\mathrm{p}} = 20 \log_{10}(V_2/V_1)^{\beta} \tag{1}$$

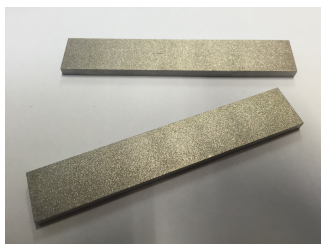where ΔL_{p} is the difference of sound pressure level (dB) between speeds V_1 and V_2.

Figure 1. Simple experiment of friction sound with two steel sample with surfaces prepared by electric-discharge machining.

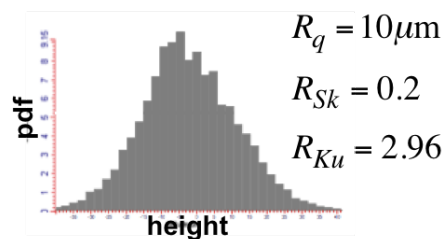$$R_q = 10\mu\text{m}$$
$$R_{Sk} = 0.2$$
$$R_{Ku} = 2.96$$

Figure 2. Distribution of asperity heights.

The sound level also increases with roughness. An accurate measurement of the dependence of sound with roughness leads again to an exponential law [4, 3]

$$\Delta L_\text{p} = 20 \log_{10}(R_2/R_1)^{\alpha} \qquad (2)$$

where ΔL_p is the difference of sound pressure level between roughness R_1 and R_2.

Roughness noise is characterized by a relatively flat power spectrum. In figure 3 is shown the power spectral density of friction sound recorded by a microphone for two different sliding velocities. The spectrum is very wide and the entire audio frequency band is covered. This wide band nature of sound is the main difference of roughness noise with other types of friction sounds such as squeal noise or other narrow band sounds (a fundamental frequency and few harmonics).

In figure 3, we can also observe some emerging frequencies. These peaks exactly match with the natural frequencies of solids without contact. The mechanical contact between the solids does not modify the natural frequencies (a strong contact stiffness would result in a shift towards high frequencies). The mechanical coupling between samples is therefore weak. This is an important characteristics of roughness noise. This condition is generally established under a light contact pressure.
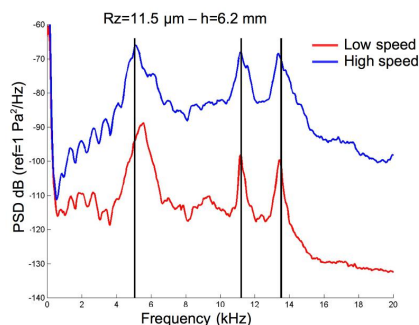

Figure 3. Power spectrum of roughness noise produced by sliding two parallelepipedic metal pieces. Red, low sliding speed; blue, high sliding speed. The vertical black lines indicates the natural frequencies of solids.

3. Mechanical origin of roughness noise

The mechanical process responsible of roughness noise must be discovered at the scale surface asperities. Surface asperities have a typical size of order of dozen of micrometres. They represent obstacles against motion. Percussion at the top of antagonist asperities act on solids as short and light impacts but at a very high rate. Since the surface is random by nature, these impacts are disordered events (Fig. 4).
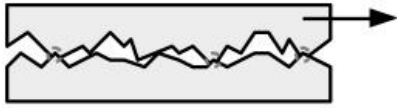
Figure 4. Mechanical origin of friction noise. Percussion of asperities produces vibration responsible of noise emission.

These micro-impacts behave like small hammer strokes which mechanically excite the structure. Since the Fourier transform of a pulse is constant, we now better understand why noise spectrum is so wide. More accurately, we may assess the impact duration for instance with a model of an elastic sphere impacting a rigid plane. For a steel-steel contact, the impact duration is of order of 0.1 ms. The frequency bandwidth of sound is therefore larger than 10 kHz (we measured sound up to 50 kHz with a 1/4" condenser microphone).

The rate of impacts may also be assessed by as follows. The length of correlation in the contact may be defined as the smallest distance of motion for which the spot population is entirely re-newed (Fig. 5). This definition was given by Rabinowicz to explain the transition between static and kinetic friction [5]. Rabinowicz measured this distance D_0 by an indirect technique and obtained that it ranges from 1 to 10 μm for different metallic contacts.

In static condition, the number of contact spots is generally low say, of order of $n_0 \sim 10$ which corresponds to a bearing rate of few percents. But in sliding condition with a speed V, under the assumption of quasi-static movement, the population is entirely re-newed after a time D_0/V. The impact rate is therefore

$$n = n_0 \frac{V}{D_0} \sim 10^5 \quad \text{impacts/s} \tag{3}$$

for a typical sliding speed $V \sim 10$ cm/s and $D_0 = 10$ μm.

A so large impact rate $n \sim 10^5$ combined with a short duration $\tau \sim 10^{-4}$ s leads to a large overlap of impacts (say dozens of impacts at the same time). It is therefore impossible to ear (or measure) separately the different impacts. The fact that roughness noise is perceived as a steady-state noise by the human ear stems from microscopic transient events but so numerous that they appear as a permanent source.
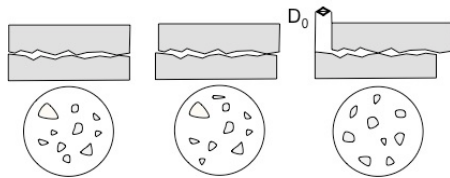

Figure 5. Correlation length of an interface. Under a sliding of length $< D_0$, the population of spots is not significantly modified. When the sliding is $> D_0$, the contact is entirely affected and new spots are created.

4. Numerical simulation of roughness noise

The numerical simulation of friction noise is possible in principle with the only laws of contact mechanics. This is a problem of elasto-dynamics in the presence of contact.

Since the impact duration is of order of 0.1 ms, the time step in numerical simulation must be much lower. In practice we used a time step of 10^{-8} s with classical time-integration schemes. Contact detection between the top of asperities requires a very fine mesh. We applied classical contact algorithms (penalty or Lagrange multipliers methods) with spatial step of 4 μm.

However, the wavelengths of sound and elastic waves in solids are usually very large (30 cm for sound at 1 kHz and several centimetres for elastic waves). So, a finite element model of the elasto-dynamics problem does not require a so fine mesh.

This contradiction between a fine mesh for contact detection and a gross mesh for vibrations highlights that roughness sound simulation is basically a multi-scale problem numerically ill-posed.

We circumvented this difficulty by projecting the equations of motion on a modal basis. If $u(x,t)$ denotes the vibration field and ω_k, $\psi_k(x)$, $k = 1, 2, \ldots$ the natural frequencies and mode shapes, then the modal projection reads,

$$u(x,t) = \sum_{k \geq 1} U_k(t)\psi_k(x) \qquad (4)$$

where U_k is the modal amplitude to determine. The equations of motion on modal amplitudes are

$$m_k \left(\ddot{U}_k(t) + 2\zeta\omega_k\dot{U}_k(t) + \omega_k^2 U_k(t) \right) = F_k(t) \qquad (5)$$

where m_k is the modal mass, ζ the modal damping factor and F_k the modal force induced by contact. The number of equations of motion is very low since relatively few modes are involved in this problem. But computing modal contact forces F_k requires to detect all contact points at each time step. This is achieved by computing the surface deformation on a fine mesh at each time step. This is of course the most time consuming step. The gain compared with classical finite element method is therefore in the time-integration scheme limited to few equations of motion. Details of the method may be found in Ref. [6].

An example of result is shown in figures 6, 7. The studied system is composed of a small cube of length 2 cm made of stainless steel sliding on a large elastic plate of thickness 2 mm. The surfaces have been modelled by numerically generating a gaussian surface of various roughnesses. We can observe in figure 6 that the vibrational level (and therefore the Sound Pressure Level) follows a log evolution with sliding velocity and roughness. This is in agreement with empirical laws of Eqs. (1) and (2). Furthermore, in figure 7, we also observe that the rate of impacts is about 10^5 impacts/s for $V = 10$ cm/s in a fine agreement with Eq. (3). However, it also appears that the impact rate decreases with sliding speed. This may be interpreted as a dynamical effect. Impacts are more strong when the movement is more rapid. This may result in a loss of contact and an increase of flight duration. This phenomenon was not taken into account in Eq. (3) based on quasi-static assumption.

5. Dissipation of vibration
It is well-known in engineering that contacts are responsible of strong dissipation in vibrations. A metallic structure, such as a shell or a frame, is generally highly reverberant and may easily transmit sound and vibration in the overall system. However, the same structure attached by bolts, welding points, or rivets presents a much lower reverberation time. The presence of contact in attachments induces a strong dissipation of vibration by micro-slips into the contact. This mechanism is sometimes used to design low reverberant components in built-up structures. A particularly impressive example is car oil sump made by stamping together two metal sheets instead of a single one. This gives a highly absorbant element which does not contribute to radiate sound.

The following experiment illustrates this phenomenon [7]. We measured the sound pressure level with a sonometre, induced by the sliding of sugar lumps on a table surface (large wood table). Not surprisingly, we observed that the sound pressure level increases with the number of sugar lumps. Sugar lumps are sound sources that may be considered as uncorrelated. Their powers may be summed and therefore the theoretical law is an increase by 10 dB of Sound
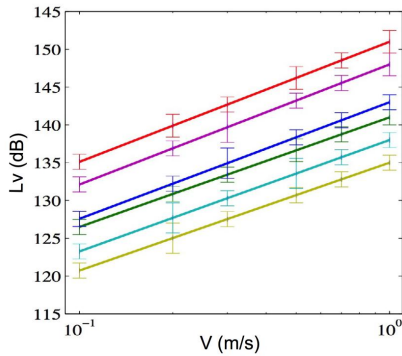
Figure 6. Vibrational level versus sliding speed for various roughnesses (from bottom to top Ra=3, 5, 8, 10, 20, 30 μm).
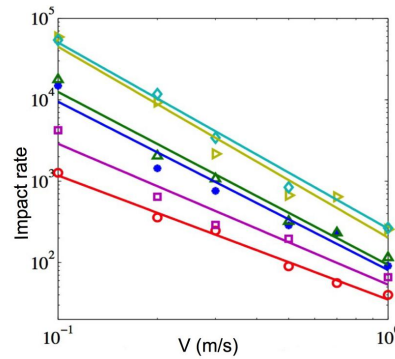
Figure 7. Impact rate versus sliding speed for various roughnesses (from top to bottom Ra=3, 5, 8, 10, 20, 30 μm).

Pressure Level by decade of sugar lumps. The overall increase of 18 dB for a number of sugar lumps ranges from 1 to 100 indicates therefore an experimental slope of 9 dB per decade which is very close to this theoretical value.

However, when reproducing this simple experiment on a drum membrane, we observed that the Sound Pressure Level is almost constant (we did the experiment from 1 to 80 sugar lumps). This result is rather surprising at first sight. If the lumps are uncorrelated noise sources on a table wood, we may wonder why they are not on a drum membrane.

The key of this phenomenon is not in the additive law of sound power but rather in dissipation process. We may propose the following thermodynamical reasoning. The structure (table plate or drum membrane) is a tank of vibrational energy. Sources (sugar lumps) inject vibrational energy in this tank. The power balance reads $P_{\mathrm{inj}} = P_{\mathrm{diss}}$ where P_{inj} is the power being injected by sources and P_{diss} the dissipated power. The total power supplied by sources is $P_{\mathrm{inj}} = Np$ where N is the number of sources and p the unit power. For dissipation of vibration in materials, a widely-accepted dissipation law [8] is

$$P_{\mathrm{diss}} = \eta \omega W A \tag{6}$$

where η is the damping loss factor, ω the central frequency, W the energy per unit area (assumed to be uniformly distributed), and A the plate area. Substituting P_{inj} and P_{diss} into the energy balance gives $Np = \eta \omega W A$. The energy density is proportional to the number of sources, $W \propto N$. This proportionality of vibrational energy and therefore the radiated sound with the number of sources explains the slope of 10 dB per decade observed on wood table.

But in the case of membrane, natural dissipation of vibrations is almost negligible. In the sugar lumps experiment, dissipation of vibrations mainly stems from the presence of lumps themselves. Micro-slips occurring at the interface between the sugar lumps and membrane are responsible of most of dissipation. It is therefore reasonable to assert that dissipation is proportional to contact area. More specifically, we may infer by analogy with Eq. (6) that

$$P_{\mathrm{diss}} = \eta' \omega W S \tag{7}$$

where η' is an ad-hoc damping loss factor (due to contact) and S the total contact area. Letting $S = Ns$ where s is the contact area of one lump and substituting Eq. (7) into the energy balance give now $Np = \eta' \omega W N s$. We get $W \propto 1$ that is the vibrational energy density is independent of the number of sources. The number of lumps is responsible of an increase of

supplied power, but, in the mean time, it is also responsible of an increase of dissipation exactly in same proportion. This well explains the constant regime of sound observed on drum. When combining both dissipation mechanisms in material and contact, all intermediate regimes exist from constant law of sound to the linear law of 10 dB per decade [7].
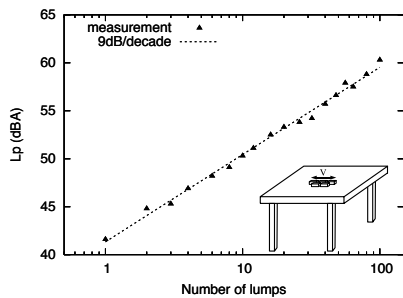

Figure 8. Friction noise generated by sugar lumps sliding on a wood table. The Sound Pressure Level (SPL) increase by 9 dB per decade (linear regime).
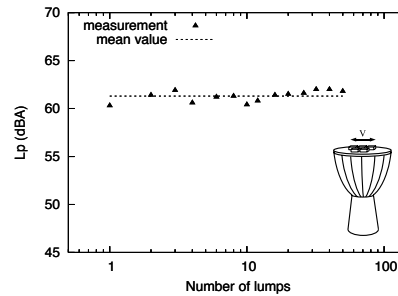
Figure 9. Same experiment on a drum membrane. The Sound Pressure Level (SPL) is almost independent on the number of lumps (constant regime).

6. Conclusion

Roughness noise is a very rich phenomenon. The mechanical problem of contact and vibration is highly multi-scale (scale of asperities much lower than scale of vibration). This is also a problem of statistical mechanics since the surfaces in presence are random by nature. Numerical simulation of contact mechanics is intensively time-consuming but highly optimized methods are possible. They lead to interesting results such as impact rate, impact duration, and strength of impacts.

The duality of contact, source of vibration but also dissipation of vibration, is certainly the most intriguing characteristic of roughness noise. The complexity of local events in contact does not allow a deterministic approach of the problem. But the fact that microscopic events responsible of noise emission are numerous, short and random is a favorable condition to apply methods and concepts of statistical mechanics.

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