# Noise of sliding rough contact

**Alain Le Bot**

Ecole centrale de Lyon, CNRS LTDS UMR 5513, 36, av. Guy de Collongue 69134 Ecully, France

E-mail: `alain.le-bot@ec-lyon.fr`

**Abstract.** This article is a discussion on the origin of friction noise produced when rubbing solids with rough surfaces. We show that the 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, the 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 of 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 example, the contact is weak and the 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. The 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 others two nominally flat pieces of metal (figure 1) such as proposed in Ref. [2]. The surfaces of the sample have been prepared by electric-discharge machining which leads to a random surface. In figure 2 is shown the probability density function of heights of asperities obtained for 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 the noise emission.

The sound level is of course an increasing function of the sliding velocity as it can easily be checked with the metal samples (try it by 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 $\Delta L_{\mathrm{p}}$ is the difference of sound pressure level between speeds $V_1$ and $V_2$.

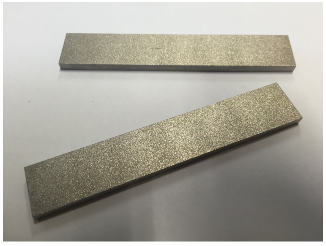

**Figure 1.** Simple experiment of friction sound with two steel sample with surfaces prepared by electric-discharge machining.

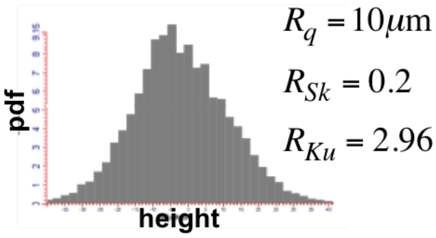


**Figure 2.** Distribution of asperity heights.

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

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

where $\Delta L_{\rm p}$ 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 by sounds. This wide band nature of sound is the main difference of roughness noise with other types of friction sounds such as squeal noise for which the power is confined into the vicinity of a single frequency and few harmonics.

In figure 3, we can also observe some emerging frequencies. These peaks exactly match with the natural frequencies of solids. This coincidence shows that the mechanical contact between both solids does not modify the natural frequencies (a strong contact stiffness would result in a shift towards high frequencies). Consequently, the mechanical coupling between samples is weak. This is an important characteristics of roughness noise. This condition is generally established under a light contact pressure.

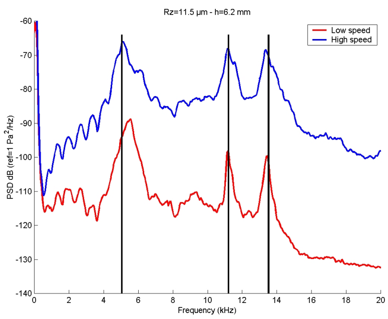


**Figure 3.** Power spectrum of roughness noise produced by sliding two parallelepipedic metal pieces. Red, low sliding speed; blue, high sliding speed. The vertical black lines indicates the natural frequencies of solids.

## 3. Mechanical origin of roughness noise

The mechanical process responsible of roughness noise must be discovered at the scale surface asperities. Surface asperities whose typical size is of order of dozen of micrometres, represent obstacles against motion. Percussion at the top of antagonist asperities act as many light impacts at a very high rate. Since the surface is random by nature, all these impacts are very numerous, short, and disordered events (Fig. 5).

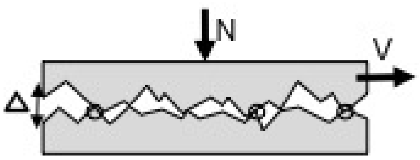


**Figure 4.** Mechanical origin of friction noise. Percussion of antagonist asperities generates a vibration of surfaces responsible of noise emission.

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 the 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 the following reasoning. 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 (a definition 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 tit ranges from 1 to 10 $\mu$m 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 $V$, provided that we assume quasi-static movement, the population is entirely re-rewed 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 $D_0 = 10$ $\mu$m.

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 steady-sate aspect of roughness noise which is perceived as a continuous noise by the human ear, stems from microscopic transient events but so numerous that they appear as a unique one.

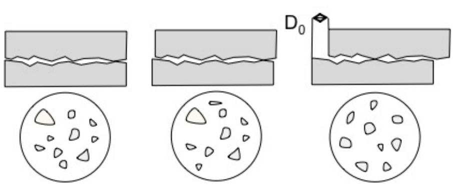


**Figure 5.** Correlation length of an interface. Under a sliding of length $< D_0$, the population of spots is not significantly modified. When the sliding is $> D_0$, the contact is entirely affected and new spots are created.

## 4. Numerical simulation of roughness noise

The numerical simulation of friction noise is in principle possible 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 than that. 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 $\mu$m.

However, the mechanical and acoustical wavelengths involved in this problem are usually very large (30 cm for the acoustical wavelengths at 1 kHz and several centimetres for the mechanical wavelengths). So, a finite element model confined to the only elastic problem would not require a so fine mesh.

This contradiction between a fine mesh for contact detection and a gross mesh for the vibration prediction highlights that roughness sounds is basically a multiscale problem which is 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 $\omega_k$, $\psi_k(x)$, $k = 1, 2, \ldots$ the sequence of natural frequencies and mode shapes, then the modal projection reads,

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

where $U_k$ is the modal amplitude to be determined. The equations of motion on modal amplitudes are

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

where $m_k$ is the modal mass, $\zeta$ the modal damping factor and $F_k$ the modal force induced by contact. The number of equations of motion is very low since we saw that relatively few modes are involved in this problem. But determining 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 a low number of 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 from 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, from 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 the sliding speed. This may be interpreted as a dynamical effect. The more rapid the movement, the more strong the impacts. This may result in a loss of contact and an increase of the flight duration of the slider. This phenomenon is of course not taken into account in Eq. (3) based on the quasi-static assumption.

## 5. Dissipation of vibration

It is well-known in engineering that contacts are responsible of a strong dissipation of 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, resulting in a highly absorbant element which does not contribute to radiate sound.

The following experiment illustrates the phenomenon [7]. We measured with a sonometre the sound pressure level induced by the sliding of sugar lumps on a table surface (a large wood table). Not surprisingly, we observed that the sound pressure level increases with the number of sugar lumps. We may add that sugar lumps are sound sources which are uncorrelated. Their

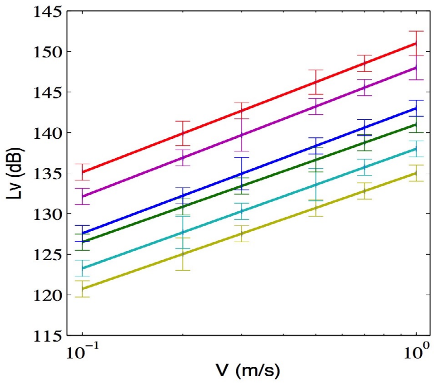


**Figure 6.** Vibrational level versus sliding speed for various roughnesses (from bottom to top Ra=3, 5, 8, 10, 20, 30 $\mu$m).

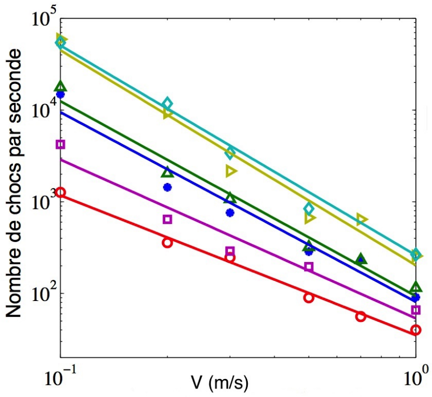


**Figure 7.** Impact rate versus sliding speed for various roughnesses (from top to bottom Ra=3, 5, 8, 10, 20, 30 $\mu$m).

powers add together and therefore the theoretical law is an increase by 10 dB of Sound 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 the theoretical value.

However, when reproducing this simple experiment on a drum membrane, we observe 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 powers but rather in the 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 $P_{\text{inj}} = P_{\text{diss}}$ where $P_{\text{inj}}$ is the power being injected by sources and $P_{\text{diss}}$ the dissipated power. The total power supplied by sources is $P_{\text{inj}} = Np$ where $N$ is the number of sources and $p$ the unit power. When the dissipation of vibration occurs in the material by hysteretic damping, a widely-accepted dissipation law [8] is

$$P_{\text{diss}} = \eta\omega WA \tag{6}$$

where $\eta$ is the damping loss factor, $\omega$ the central frequency, $W$ the energy per unit area (assumed to be uniformly distributed), and $A$ the plate area. Substituting $P_{\text{inj}}$ and $P_{\text{diss}}$ into the energy balance, the energy density is found to be proportional to the number of sources, $W \propto N$. This result explains the additive law and the slope of 10 dB per decade.

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

$$P_{\text{diss}} = \eta'\omega WS \tag{7}$$

where $\eta'$ 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 a single lump, 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 the supplied power, but, in the mean time, it is also responsible of an increase of dissipation exactly in the same proportion. This well explains the constant regime observed on the drum. Other experiments reveal that when combining both dissipation mechanisms, all intermediate regimes exist from constant to linear [7].

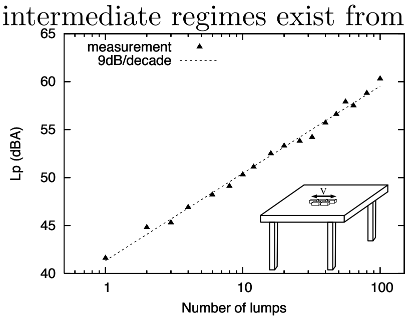


**Figure 8.** Friction noise generated by sugar lumps sliding on a wood table. The Sound Pressure Level (SPL) increase by 9 dB per decade (linear regime).

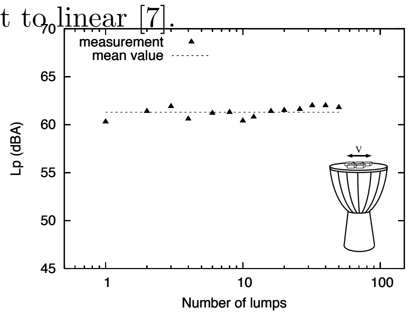


**Figure 9.** Same experiment a drum membrane. The Sound Pressure Level (SPL) is almost independent on the number of lumps (constant regime).

## 6. Conclusion

Roughness noise is a very rich phenomenon. The mechanical problem of contact and vibration is strongly multiscale (scale of asperities versus scale of vibration). This is also a statistical problem of contact mechanics since the surfaces in presence are random by nature. The numerical simulation of the contact mechanics is intensively time-consuming but optimized methods may lead to interesting results such as impact rate, impact duration and so on.

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


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