On Tangential Friction Induced Vibrations in Brake Systems

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Abstract The basis for the analysis of friction in brake systems is the brake pad's tribological interface. An investigation of this interface reveals friction intensive surface structures. These so-called "patches" are extremely hard and carry the main part of the friction power. By complex interaction processes of wear and heat these patches are generated permanently but leave the system after a certain period of time. So there is an equilibrium of flow of contact patches in the brake pad interface, with the outcome being a dynamic macroscopic friction coefficient, whose "inertia" can be well described by differential equations in the form of special balance equations. Systematic expansions of these balance equations even allow, for the first time, a simulation of different test cycles of the AK-Master test for friction materials with high accuracy. These friction force variations are generated by the dynamics of the local surface geometry and can explain physically effects of measurements, which were up to now described by control theoretic approximations [7, 8].

Beside these effects the dynamics of friction is influenced by lateral vibrational dynamics of these patches on a very fast timescale. This timescale is so fast that processes of patch growth and destruction are negligible. Beyond that, the vibration frequencies of the patches, as well as the actual local friction power on each of these surface structures, vary over a wide range of values, which is the result of a great variety of patch sizes and heights in the interface. Generally, one would expect a smoothing of these local and stochastically distributed vibration effects. It can however be shown, that the oscillations of the patches are subject to synchronization processes, with the result being in-phase patch vibrations on macroscopic areas of the brake pad of significant size. Thereby, self-excited vibrations of the patches can lead to lateral oscillations of the pad's friction force on a macroscopic scale. These are able to excite the whole system of brake pad and disk.

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1 Introduction

Nearly everybody has used once sandpaper to modify wooden surfaces by hand. And surely everybody has observed agglutinations on the sandpaper at the grinding process. At least this is to be observed when you are grinding the finish (Fig. 1).

During grinding the swarf is moving between wood and sandpaper. At some parts of the sandpaper thin and hard patches are growing up, formed by heat and wood powder. With your fingers you feel hot spots through the sandpaper at exactly these patches. At these patches the friction power produces heat but nearly no wear any more. When there are to many patches the sandpaper gets unusable.

In technical applications for high wear rates the production of such patches has to be avoided. This can be done by the modification of the form of abrasive grain -this controls the density and the form of wear particles- and by introducing elastic belts, which controls the stability of the patch growing process. This will be described elsewhere.

In technical applications for high friction forces but low wear rates these surface patch structures are essential. A typical technical system with these properties are brake systems [1, 2]. Here one can observe these hard surface structures, the so called patches [3, 4]. These patches carry the main part of friction power of the system and their dynamics influences the dynamics of global friction [5, 6].

These patches control and determine in a fundamental way the friction and wear properties of the brake system. The basic structure of a brake pad consists of a comparative supple matrix, with embedded small and hard structures or particles. An example for an ancient brake material with such properties is wood (e.g. used to slow down windmills at forthcoming storms). Modern brake materials in automotive engineering consist of a soft polymeric matrix with embedded SiO_2-particles, iron swarfs or similar materials.

By pressing the brake pad material onto the brake disk, wear debris is produced. At the starting point the main part of the debris is generated from the polymeric

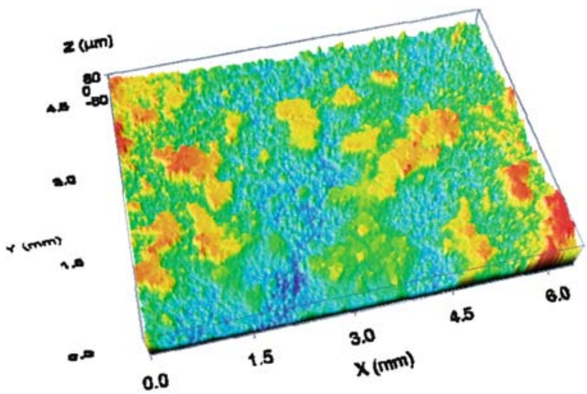

Fig. 1 Confocal microscope picture of grinding paper

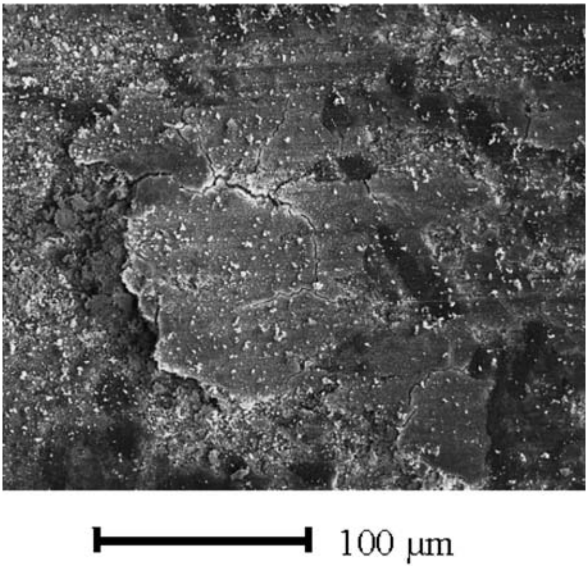

Fig. 2 Patch structures on a brake pad (REM – picture)

matrix. This debris is similar to the wear of friction produced by using a rubber on a sheet of paper (Fig. 2).

The wear particles move through the contact zone. When reaching the rim of the contact zone, a fraction of the wear particles sticks at the brake disk and moves again into the contact zone after one rotation of the disk. Another part is emitted to the environment.

Two important things will happen when, caused by the wear of the polymeric matrix, a SiO_2-particle or an iron swarf reaches the surface of the brake pad, see Fig. 3. First, due to the higher wear rate of the polymeric matrix compared with the hard particle, the particle is getting pressed into the polymeric matrix. Secondly

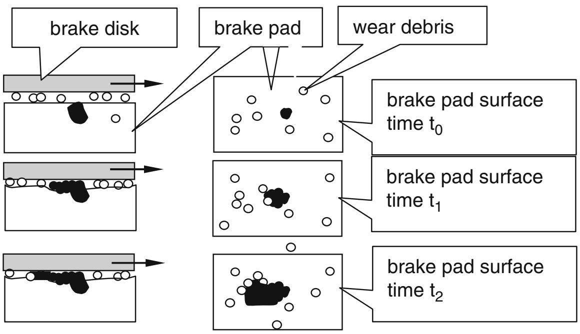

Fig. 3 Section of the brake system at different moments $t_0 < t_1 < t_2$, growth of a patch. On the *left* a side view, on the *right* a *top* view on the pad surface section

the wear particle flow in the boundary layer is disturbed. As a result of the increasing local normal and tangential stress in the surrounding area of the particle, the local temperature will also increase. For technical brake systems the stress and the temperature may increase until alloying processes between the hard particle and the wear particle flow will occur (some analysis indicates even the possibility of creating ceramic structures). This leads to an accretion of hard and thin contact patches on the braking material contrary to the wear particle flow.

Hereby the hardness of the patches is in the range of those for the inhomogeneities.

With this generation of contact patches, the brake material surface can be divided into two different contact zones, see Fig. 4. The contact zone of type I is defined by the contact of the polymeric matrix and the contact zone of type II is defined by the hard patches. The main energetic conversion is produced at the contact zones of type II. So the major part of the breaking power is dissipated in this contact zone of type II. That leads to very high temperatures of the contact patches that even the integral temperature of the brake pad surface will reach about 300°C.

The wear around the contact patches, the heat generation and mechanical stress are leading to crack growth in the hard contact patches. Furthermore the asperity structure of the break disk enforces local movements/vibrations of the contact patches on the soft polymeric structure and leads also to a crack propagation. These effects results in a destruction of contact patches after a certain time (see Figs. 5 and 2)

The boundary layer on the brake material is characterized by the growth and the destruction of contact zones of type II on the surface of the soft polymeric matrix. The main part of the incoming energy is dissipated by these contact patches [5, 6]. Therefore the friction is defined by a dynamic equilibrium of processes that generate and destruct these patches. The current friction is not a stationary function of e.g. normal force or velocity but is determined by the history of the load and the wear. Whether it is well known, that friction, heat and wear are intimately connected, usually friction laws are formulated without heat and wear. An argument for this is

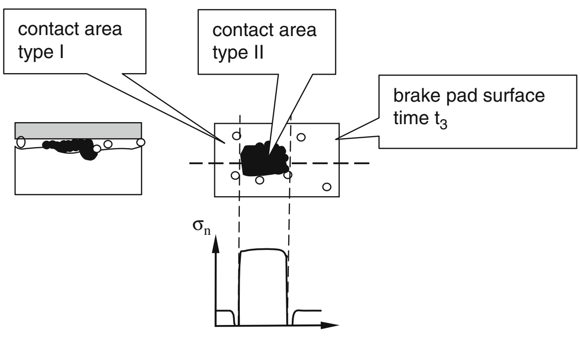

Fig. 4 Section of the brake system, different contact zones (type I and type II) and in principle the normal load distribution along the *dashed line*. On the *left* a side view, on the *right* a *top* view on the pad surface section

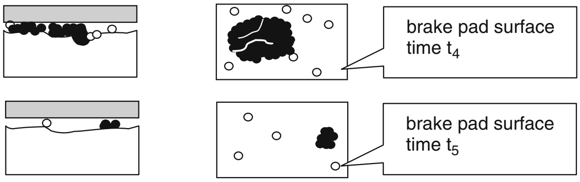

Fig. 5 Section of the brake system at different moments $t_3 < t_4 < t_5$, destruction of a patch. On the *left* a side view, on the *right* a *top* view on the pad surface section

the assumption of different timescales, the phenomena are living on. Heat is often taken into account only via the temperature dependent material datas only.

2 Dynamics of Friction Caused by Heat and Wear

The frictional resistance is significantly determined by the contact areas of type II. These contact patches are depending from wear and heat generation, which depend from frictional resistance by themselves. Figure 6 illustrates the interconnectedness

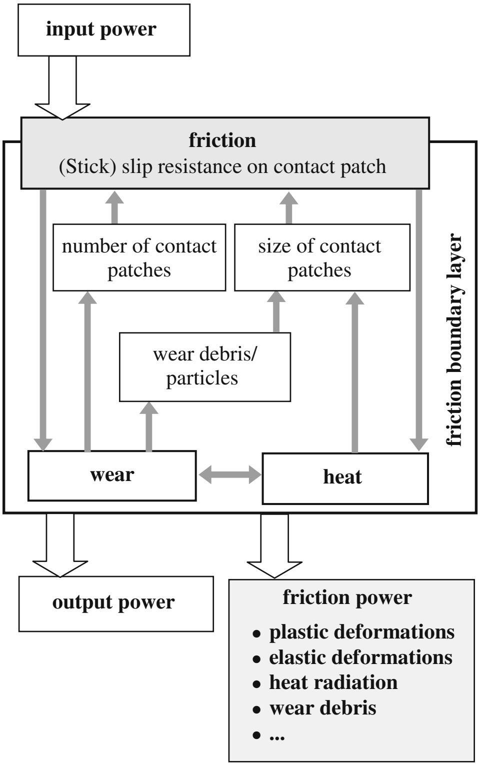

Fig. 6 Friction in technical brake systems

of the above described effects. The illustration therefore shows a closed loop dependency of friction, heat and wear [6]. This shows that friction depends on the wear history. Wear has to be taken into account in a friction law, too, when friction has to be described for shot time intervals.

The dependence on temperature here is given by the growth of patches. The process generates heat, which has to be described by a friction law. This temperature has to be taken into account, when temperature dependent elastic moduli or other temperature dependencies of material data are essential. This last effect has to be included when concrete material behaviour is taken into account, for instance the dynamic behaviour of the patches on the elastic pad. This will be done in the next chapter.

The dynamic equilibrium of accretion and destruction of contact patches is a substantial property of contact area A_{II} and a fundamental consequence of wear.

With respect to Coulomb's ideas the friction coefficient μ is described by the magnitude of the friction force R divided by the magnitude of the normal force N:

$$\mu = \frac{R}{N}$$

Expanding this fraction by the norm v of the tangential velocity vector between the two contacting bodies, one receives

$$\mu = \frac{Rv}{Nv}, \qquad Nv = \left\| \overrightarrow{N} \right\| \left\| \overrightarrow{v} \right\|$$

This formulation gives another interpretation of the friction coefficient μ. It is now a dimensionless measure of the total friction power Rv in units of a characteristic system power, given by the product of the magnitudes of the vectors of normal force and tangential velocity. This characteristic system power is actually an idle power (because the vectors of normal force and tangential velocity are perpendicular to each other) but very handy for the following description of the dynamic friction law. It is a fundamental input parameter for the tribological processes in the brake.

For simplicity it is assumed that the friction power is proportional to the total area A_P of contact patches. In this simple analysis the energy flowing in all other contact regions will be neglected. The total patch area A_P on the pad is determined by an equilibrium of flow caused by the growth and destruction of single contact patches. This equilibrium of flow is a fundamental consequence of the closed loop interaction of friction and wear schemed in Fig. 6. Since the friction power is described by μ, the total patch area A_P itself and especially the dynamics of A_P are correlated with μ and its temporal change. So the equilibrium of flow of friction power must be given by a balance equation of μ [6].

This means that the above delineated processes do not affect μ itself but only its derivation with respect to time. So the friction in general becomes a dynamic process. The approach of the equilibrium of flow for μ leads to general balance equations.

Models of the coefficient of kinetic friction usually use the sum of all asperity interactions in the contact area and assume a negligible wear for short time intervals.

The model presented above shows, that because of the dependency of the current size and quantity of contact patches from the history of wear, the wear has to be taken into account even for short time intervals. Therefore the defining equation for μ has to be an balance equation instead of an algebraic equation:

$$\begin{aligned}
\dot{\mu} &= f_1(\mu, a_1, a_2, \ldots) - f_2(\mu, a_1, a_2, \ldots) \\
\dot{a}_1 &= g_1(\mu, a_1, a_2, \ldots) \\
\dot{a}_2 &= g_2(\mu, a_1, a_2, \ldots) \\
&\ldots.
\end{aligned} \tag{1}$$

The accretion of the contact patches is described by the function f_1 and the function f_2 covers the effects that lead to the destruction of the patches. The a_i are further state variables as heat, wear and so on, each of this variables is result of a balance equation too. To fill this law, in a first attempt the variables temperature and wear are taken into account in a rather simple manner: the wear is usually in literature approximately given by the friction power, so wear is given by the product μNv. The heat can be treated in the usual way, where the source is given again by the friction power. These functions can be build up in detail, see [6, 12]. A fourth order equation for μ is given in [9], where even parts of the well known AK-Master test are predictable described by this friction law.

Using a cellular automaton, we reproduced this interaction [10–12] and as a result we obtained precisely the well known surface structures of a break pad, see Fig. 7 showing a section of size 1.5 mm × 1.5 mm. In addition the Automaton computes the topography of the brake pad and the wear particle density.

For very slow variations of the normal force and the velocity and short application times one can use approximately the stationary solution μ_{stat} of the dynamic friction laws. This stationary solution has the well known shape of a Stribeck curve.

The stationary solution μ_{stat} can be written in the following form,

$$\mu_{stat}(N, v) = \frac{\varepsilon}{\varepsilon + x} \Delta\mu + \mu_\infty \tag{2a}$$

Classical measurements of friction acquire always such a stationary solution of the friction coefficient. With these measurement data, one can determine the constants ε, $\Delta\mu$ and μ_∞. The load parameter x is given by

$$x = Nv \tag{2b}$$

When normal load and velocity are changing faster, they affect the topography dynamics described above resulting in non steady state behavior of the friction coefficient itself. The characteristic time scales of these surface dynamics varies from about 0.01 s up to the application time of the brake (Fig. 8).

On very fast timescales, where the effect of geometry changes in the friction surface by the grow and destruction of the patches are negligible the dynamics of friction is influenced by lateral vibrational dynamics of these patches on a very fast

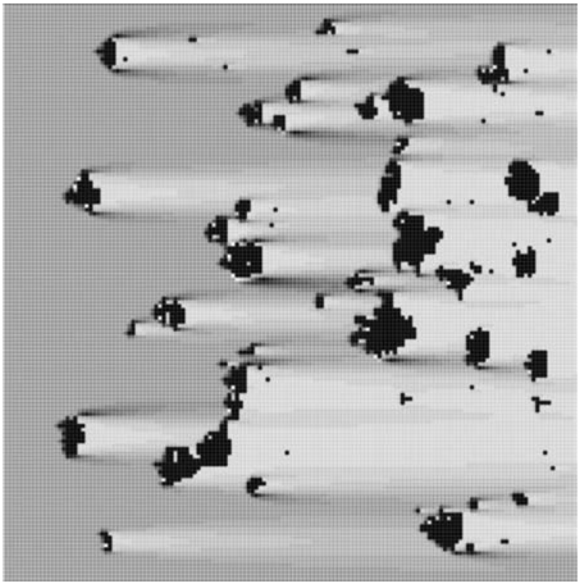

Fig. 7 The friction surface of the pad during braking with patches (*black*) and wear debris (concentration correlated with *gray scale*) computed with a Cellular Automaton

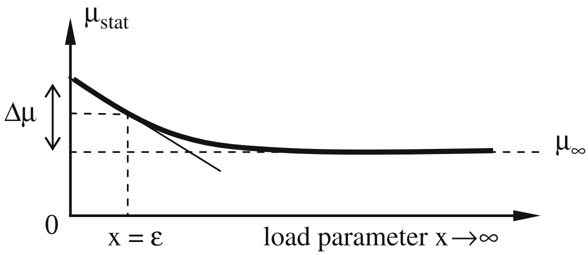

Fig. 8 The stationary behaviour of the friction coefficient depending on the parameters $\Delta\mu$, μ_∞, ε

timescale. The lateral vibration frequency of a single patch can be estimated to cover a range of about 0.5–20 kHz. This timescale is so fast that processes of patch growth and destruction are negligible. On each patch the stationary solution (2) describes the friction force evolution again.

3 On Tangential Vibrations in the Brake System

Due to the fact that the main part of the friction power is carried by the contacts between patch and disk, only these contacts will be considered in the following. The permanent changes in size and number of the contact patches shall be neglected for

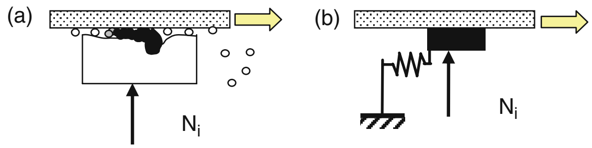

Fig. 9 Lateral patch vibrations (**a**) described by a 1 dof friction oscillator (**b**)

the following examinations, since these changes live on a rather slow time scale with respect to the patch vibrations. Beyond that, only the lateral dynamics are of interest in this chapter. Therefore, the patches appear as classical stick-slip oscillators on a permanently moving surface, which is the brake disk. They are coupled among one another via the polymeric matrix. Each of these stick-slip oscillators is subject to an individual normal force N_i, since patch size and normal force are correlated. Moreover, the patch size controls its elastic connection to the under layer and the vicinity of neighbored patches determines the visco-elastic coupling (Fig. 9).

For the dependency of friction versus sliding velocity, the function (2) is utilized. The equation of motion for this individual patch, decoupled from all other patches, has the simple form:

$$\mathrm{m}\,\ddot{\mathrm{x}} + \mathrm{cx} = \mu(\mathrm{v_{rel}}) \cdot \mathrm{N} - \mathrm{b}\,\dot{\mathrm{x}}, \quad \mathrm{v_{rel}} = \mathrm{v} - \dot{\mathrm{x}} \tag{3}$$

Therein, v is the brake disk's tangential sliding velocity. As is known, such a system is subject to self-induced vibrations. For the case of a sufficiently high friction force, compared to the damping force, a stationary vibration along a given limit cycle is the consequence,

Usually one would expect that due to the wide range of the oscillating masses and the respective stiffnesses (and thereby the frequencies), the oscillation of the patches is not synchronizised and stochastic or chaotic. This type of phasing, which is schemed by the different colours in Fig. 10a, leads to a rather temporally homogeneous load transmission into the disk overloaded by a wideband noise. In case of constant velocity and normal load this mechanism would transport the local friction behavior on each patch to the macroscopic pad.

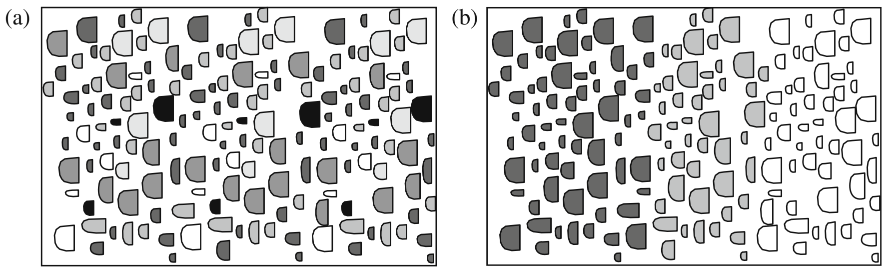

Fig. 10 The phase of the patch oscillators, (**a**) unsynchronized state, (**b**) synchronized state

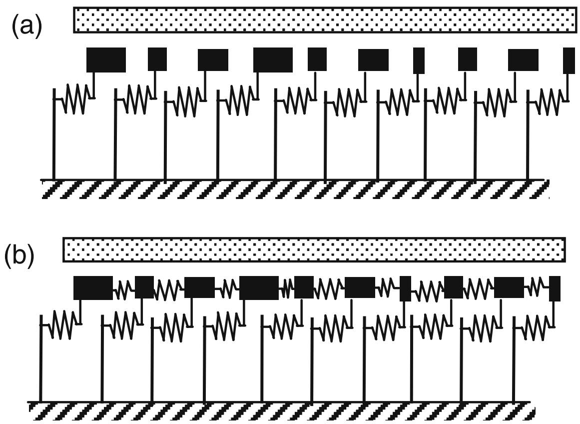

Fig. 11 (**a**) 10 decoupled stick-slip-oscillators (**b**) 10 coupled stick-slip-oscillators

By the vicinity of the patches the oscillators are weakly coupled. The question is whether there can occur a synchronization (see Fig. 10b) of these patch oscillations under special circumstances, so that macroscopically the load transmission has a periodic character. In this case that phenomenon leads to a lateral excitation of the disk-pad-system and can cause noise problems. This fundamental mechanism will be illustrated by the systems in Fig. 11.

As a major consequence of the coupling, all patches run through the friction curve in-phase. The friction curve has in general a falling characteristic in brake systems. On the macroscopic scale, this results in significant oscillations of the global friction force along the whole pad surface rather than in only small stochastic variations of the friction force versus time, as would be the case in an unsynchronized system, Fig. 12 (a). Simple estimations indicate that the vibration frequencies of the patches

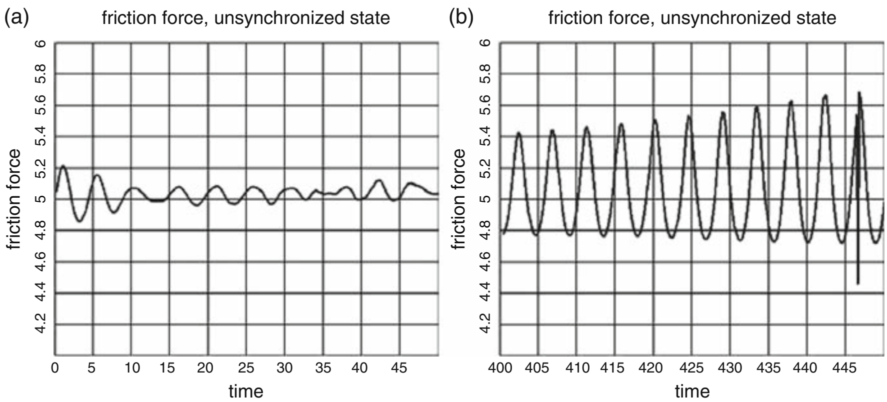

Fig. 12 Resulting friction force of ten stick-slip oscillators in the (**a**) unsynchronized state, (**b**) synchronized state

can be expected in a range between 1 and 10 kHz. The high-frequency stick-slip oscillations can excite vibrations of the whole system of brake pad and disk.

4 Conclusions

The investigations on lateral dynamics in brake systems, discussed in this paper, revealed the existence of macroscopic stick-slip oscillations in the high-frequency range. These can be fundamental for the explanation of generally known phenomena, such as the squealing. The effect of excitation caused by the synchronization can be even more significant using the real number of oscillating patches, which is magnitudes of order larger than in the system with ten patches shown in this paper.

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