Notes on Rubber Friction

Laws of Friction:

In dry sliding between a given pair of materials under steady conditions, the coefficient of friction may be almost constant. This is the basis for two EMPIRICAL Laws of Sliding Friction, which are often known as Amontons Laws and date from 1699. They are in fact not original but a re-discovery of work by Leonardo Da Vinci dating from some 200 years earlier.

Amontons Laws of Friction can be stated as follows:

> Friction is proportional to normal load.
>
> The friction is independent of the apparent area of contact.

A third Law of Friction was added by Coulomb (1785):

> The friction is independent of sliding velocity.

These three Laws are collectively known as the Amontons-Coulomb Laws. They are based on EMPIRICAL OBSERVATIONS only and there is NO PHYSICAL BASIS for these Laws. If a tribological contact does not appear to behave in agreement with these Laws, it does not mean that there is something suspect about this behaviour. These Laws are not FUNDAMENTAL in the same way the Newton's Laws are fundamental.

Most metals and many other materials in dry sliding conditions behave in a way that broadly agrees with the First Law. Contacts between metals and ceramics and metals and polymers rarely agree with the First Law.

Most materials agree with the Second Law, with the exception of polymers.

Most materials agree with the Third Law, but only over a moderate range of sliding velocities. The transition from rest to sliding at low velocities does not agree with the Third Law and at high sliding velocities, in particular in metals, the dynamic friction coefficient falls with increasing velocity. Further Laws have subsequently been added, until we end with:

> Friction is proportional to normal load.
> Friction is independent of the apparent area of contact.
> Friction is independent of sliding velocity.
> Friction is independent of temperature.
> Friction is independent of surface roughness.

These are, in sum, the classical laws of friction. Ceramics and polymers usually do not conform to these laws.

Modern understanding of friction stems from the work of Philip Bowden and David Tabor (mostly at Cambridge) between the 1930s and the 1970s and is based on careful analysis of contact mechanics. Their model for sliding friction assumes firstly that all frictional effects take place at the level of micro (or asperity) contacts and that the total friction force has two components: an adhesion force and a deformation or ploughing force. The former is associated with the real area of contact at an asperity level, the latter with the force needed for the asperities of the harder surface to plough through the softer surface. These assumptions are sufficient to explain why many material contacts do not behave in accordance with the classical Laws of Friction.

In a metal-metal contact, the deformation at an asperity level is mostly plastic. This means that the real area of contact is proportional to load. Increasing load leads to an increase in the number of asperity contacts rather than an increase in the average asperity contact surface area; more asperities are brought into action to support the increased load. Because of this, there is minimal increase in penetration depth of the asperities. As the ploughing component of friction depends on penetration depth, it is thus not highly dependent on load. The adhesion component however is proportional to the real area of contact, hence the load. Hence, the total friction in this type of contact is effectively proportional to load. It is of course important to note that even this agreement with the classical Laws breaks down once oxide and other surface films are present or once work hardening at an asperity level takes place.

By comparison with the metal-metal contact, metal-ceramic and metal-polymer contacts tend to give rise to elastic deformation at an asperity level. In ceramics, this is because of very high hardness. In polymers this is because the ratio between Young's modulus and hardness is low. This means that, except in the case of contact between a polymer and a very rough surface, the contact is almost completely elastic.

A further consideration in respect of contacts involving polymers is the strong time dependence of their mechanical properties; most polymers are visco-elastic.

In those contacts where the deformation at asperities level is elastic (as opposed to plastic) the real area of contact for a single asperity will be proportional to the load raised to the power 2/3. The real area of contact thus increases by less than proportional to load. Because of this, the friction coefficient tends to decrease with increasing load, but this is only true with a relatively smooth metal counter face, where adhesion friction predominates.

Whereas surface roughness does not have much impact on the friction in a metal-metal contact other than during running-in processes, this is not the case with the metal-polymer contact. Minimum friction is achieved with a metal surface roughness of around 0.2 Ra. With higher surface roughness, the ploughing contribution to friction increases sharply with increased penetration of the polymer surface, whereas with very smooth surfaces the adhesion component of friction increases dramatically. Of course, these frictional responses will be modified by the presence of either transfer films or entrained debris.

Before leaving the issue of surface roughness, it is worth noting that in addition to the bulk effect of surface roughness, asperity orientation and shape also have an effect on friction. With a metal surface ground in one direction, the frictional response of a polymer sliding across the surface may depend on the orientation of the motion relative to the direction of sliding. This can prove a particular problem in running a polymer pin on the surface of a metallic disc in a pin on disc configuration.

Now, whereas in the metal-metal contact, over a limited speed range, we can ignore the effects of sliding velocity, we cannot do the same for the metal-polymer contact. This is because of the visco-elastic properties of the polymer: the higher the deformation velocity, the higher the effective Young’s modulus of the polymer. This results in lower surface penetration at higher speeds and hence lower ploughing friction and a lower real area of contact and hence lower adhesive friction.

In our final consideration of the classical Laws of Friction, we should perhaps consider temperature. In the metal-metal contact, modest temperatures do not give rise to major changes in the mechanical characteristics of the materials, so it is perhaps safe (over a modest temperature range) to consider that friction is independent of temperature. This is of course no longer the case at elevated temperatures or under conditions at which asperity tip temperatures result in the softening or melting of the material.

In the case of polymers, the Young’s modulus falls sharply with rising temperature leading to an increase in contact area and an increase in adhesive friction. The product of friction and sliding velocity is frictional energy input, giving rise to an increasing contact temperature. This is accompanied by a further softening of the material and increase in friction, which reaches a maximum at the point where the real area of contact approaches the nominal area of contact. Further increase in temperature will cause the polymer to melt or collapse. This is the PV limit of the material.

From the above analysis, it should be clear that with many modern engineering materials, the classical Laws of Friction do not apply. A different set of Laws of should perhaps be postulated as follows:

Friction is NOT proportional to normal load.
Friction is NOT independent of the apparent area of contact.
Friction is NOT independent of sliding velocity.
Friction is NOT independent of temperature.
Friction is NOT independent of surface roughness.

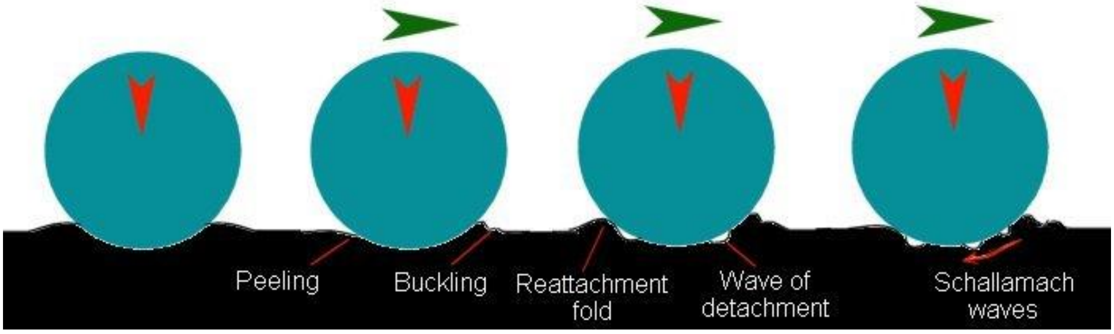
One thing is certain and that is that in the case of the Laws of Friction, the term "coefficient" does not mean a constant multiplicative factor. And we have arrived at this position without once mentioning the name Schallamach!

For dry sliding of un-lubricated contacts for typical engineering materials, we find that wear coefficients vary by four or five orders of magnitude whereas friction coefficients vary much less, in the range approximately 0.2 to 0.8.

In lubricated contacts, we would expect friction coefficients in the range 0.1, boundary or mixed lubrication, to less than 0.01 for hydrodynamic lubrication.

By contrast, under dry conditions, with a rubber or elastomer in sliding contact with a smooth, rigid counter-face, friction coefficients can be very much higher, sometimes in excess of 2. This is as a result of local adhesive forces associated with elastic deformation of the rubber.

A most important feature of this type of contact is that the friction force may, and usually will, vary significantly with both time and displacement. This is as a result of the mechanisms first observed by Schallamach.



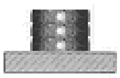
Under relative motion, "waves of detachment" form at the leading edge of the contact and flow across the contact area away from the leading edge. Rather than gross sliding over the complete contact area, the surface displacements move in folds or buckles. Before the rubber can buckle, it must first be peeled from the rigid counter-face and the energy required to do this generates significant frictional resistance.

At the trailing edge, there is a requirement to peel apart the contact. Local recovery and slip can give rise to re-attachment of the rubber at the outermost edge of the contact. The process is cyclic and gives rise to variations in friction.

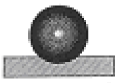
The friction force, as with all contacts, depends on the real area of contact between the rubber and the counter-face, the interfacial shear strength and the deformation properties of the rubber. The real area of contact is a function of the hardness and surface roughness, the applied load and the relative radius of curvature of the contacting bodies. The interfacial shear strength depends on the type of polymer and whether or not the surface is lubricated. These factors in turn depend on other parameters, for example, the time dependent behaviour for the rubber or, in the case of liquid lubricated contacts, lubricant entrainment conditions and squeeze film effects.

Rubber friction testing - the challenges

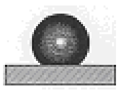
In the early 1990s we were involved in a collaborative venture with Dr Alan Roberts at the Malaysian Rubber Producers Research Association with the aim of producing a more reliable method of measuring friction in rubber, using a specially designed reciprocating tribometer. Our starting point was to choose a well-defined contact geometry.



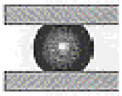
Flat-on-Flat geometries were eliminated at an early stage. Achieving and holding a flat on flat contact geometry represents a significant challenge and even if achieved, the arrangement suffers from many significant weaknesses, all of which give rise to poor repeatability. Principle among these is the tendency, even with a well supported sample, for the rubber to act as a cantilever, with the contact tilting in the direction of motion. This increases the pressure at the leading edge and reduces the pressure at the trailing edge. And for lubricated tests, the entrainment conditions are unsatisfactory and poorly defined.



The Hard Ball-on-Rubber Flat geometry, common to various test standards, was investigated and our views with regard to its suitability confirmed. With indentation, the line of action of the friction force ceases to be horizontal and load/friction force interactions are generated, the effect of which is indeterminate. The arrangement does however provide more analysable entrainment conditions in lubricated tests.

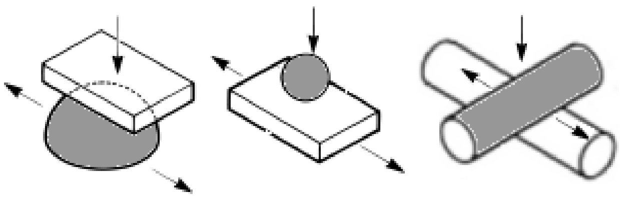


Our preferred solution was to use the self-locating contact geometry of Rubber Ball or Hemisphere on Hard Flat. The avoidance of indentation eliminates the problem of load/friction force interaction associated with the hard ball on rubber flat arrangement and the self-locating nature of the sample eliminates the alignment issues associated with the flat on flat contact. Further to this, the contact geometry and entrainment conditions are well defined and do not vary significantly with tilt.



A further advantage of using a rubber ball as a sample is that it may also be tested in a rolling contact configuration, between two hard flats, to investigate further the contribution of hysteresis and of adhesion to friction. The latter can of course be varied by, for example, spreading talc on the surface or by adding a lubricant.

About the only other fully self-locating contact geometry that one could conceive for a reciprocating test application would be a crossed cylinder arrangement. We did not try this.



In addition to the more conventional test parameters: load, temperature, sliding speed etc, we found that a key variable for our tests was dwell time, both following the initial application of load and at the end of each stroke. Controlling the dwell time precisely had a significant impact on improving test repeatability and the instrument we developed included the means for controlling the application and removal of load.

A typical test sequence would involve applying a load and waiting, typically for 30 seconds, for the rubber sample to relax before motion was commenced. At the end of a stroke, we had the choice of either starting the reverse stroke immediately, pausing for a specified period with the load still applied or removing the load at the end of the stroke, then re-applying the load with a pause before commencing the return stroke. Low sliding speeds were used, in dry tests to minimize frictional heating of the samples and in lubricated tests to prevent hydrodynamic separation of the surfaces.

With lubricated tests we observed phenomena associated with the method of application of the lubricant. The test choices here included:

Applying the load before applying the lubricant

Applying the lubricant before applying the load

With load before lubricant, sliding started with an effectively dry contact. With lubricant before load, a time and viscosity dependent squeeze film effect would occur. A further variation was to apply the lubricant to one side of the contact only.

With dry tests, the influence of relative humidity on frictional behaviour was limited by ensuring that the tests were run in an air conditioned environment. It was not until some years later, working with another party, that we extended the capabilities of our experiments to allow precise control of ambient conditions, which became progressively more important as we reduced sample temperatures to below freezing.

Finally, an alternative to reciprocating sliding, with or without pauses, was to remove the load and separate the specimens at the end of a stroke and perform the reverse stroke with the specimens out of contact. The specimens were then re-loaded into contact and further forward stroke performed. In this way, a series of unidirectional strokes are performed. Depending on the rubber under test, uni-directional sliding could result in high wear rates and pattern abrasion, with macro-scale ridges forming in the surface. By contrast, testing the same material in reciprocating sliding would suppress the formation of pattern abrasion and generate lower wear rates, with a much finer scale surface roughness generated. This is intrinsic abrasion as defined by Schallamach.

Practical Challenge

This problem involves the friction between a rubber hemisphere and a hard surface.

The rubber hemisphere is loaded into contact with the hard surface elastically, producing a given contact area.

A thin disc of the same rubber material is made with exactly the same diameter as the contact area of the hemisphere under load. This is then subjected to the same applied load as the hemisphere. Care has been taken to support the disc in a holder to minimize the cantilever effect.

Both rubber samples are then set in sliding motion. Is the friction force the same in both cases?

No!