Guidance - Rubber Friction Testing

Laws of Friction

The classical laws of friction are as follows:

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.

The laws do not apply to polymers. Contacts between 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 Fourth and Fifth Laws do not apply to polymers.

Metal-polymer contacts tend to give rise to elastic deformation at an asperity level. This is because, with a polymer, 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.

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 adhesive friction predominates.

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

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.

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 surface topography 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 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 polymers, 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!

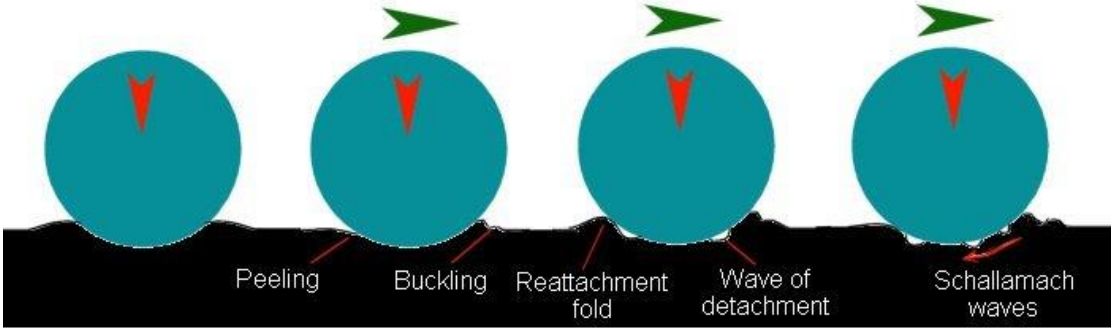
Rubber Friction Testing

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, for 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.



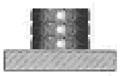
Under relative motion, “waves of detachment” form at the leading edge of the contact and flow across the contact area away from the leading edge. Rather than gross sliding over the complete contact area, the surface displacements move in folds or buckles. Before the rubber can buckle, it must first be 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.

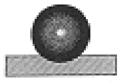
The friction force, as with all contacts, depends on the real area of contact between the rubber and the counter-face, the interfacial shear strength and the deformation properties of the rubber. The real area of contact is a function of the hardness and surface roughness, the applied load and the relative 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.

Challenges

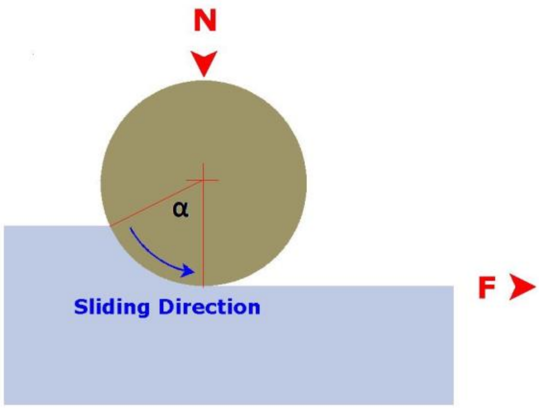
In the early 1990s we were involved in a collaborative venture with Dr Alan Roberts at the Malaysian Rubber Producers Research Association with the aim of producing a more reliable method of measuring friction in rubber, using a specially designed reciprocating tribometer. Our starting point was to choose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



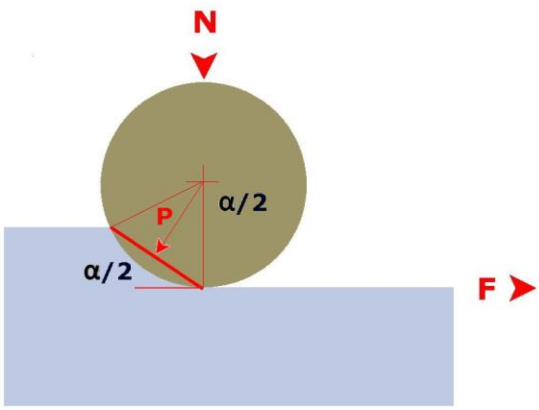
Flat-on-Flat geometries were eliminated at an early stage. Achieving and holding a flat on flat contact geometry represents a significant challenge and, even if achieved, the arrangement suffers from many significant weaknesses, all of which give rise to poor repeatability. Principle among these is the 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, as the following analysis demonstrates.



With a hard ball on rubber flat geometry, common to various test standards, we have the same indentation issues as with polymers in general.



In order to simplify the analysis, we can approximate the arc with a chord and then make the assumption that the friction coefficient is constant along that chord, which is, perhaps, quite a big assumption. We then have:

Tribometer Applied Load = N

Tribometer Measured Friction $=$ F

Resolving:

$$F = \mu P \cos(\alpha/2) + P \sin(\alpha/2)$$

$$N = -\mu P \sin(\alpha/2) + P \cos(\alpha/2)$$

Hence:

$$\mu = (F - N \tan(\alpha/2) / (N + F \tan(\alpha/2)$$

Not:

$$\mu = F / N$$

To be precise, the “apparent” coefficient of friction is:

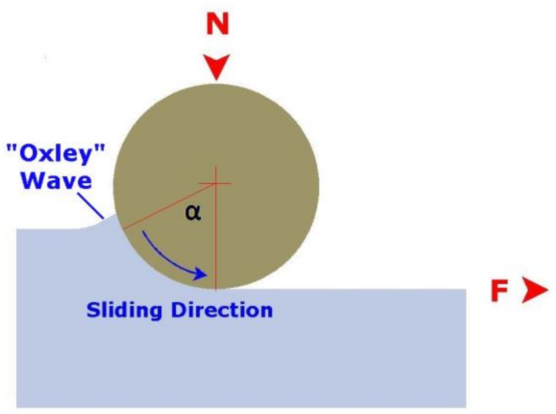
$$F / N = [\mu + \tan(\alpha/2)] / [1 - \mu \tan(\alpha/2)]$$

If μ and α are small:

$$F / N \approx \mu + \tan(\alpha/2)$$

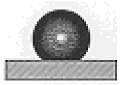
In other words, an “adhesion” term plus a “deformation” term.

However, it must be noted that this cannot be a steady state solution; discounting elastic compliance, which will of course be rate dependent, it is apparent that the ball must move up, to bring the contact point to the original surface level, in the process generating an “Oxley” wave.

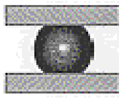


An explanation of the different regimes of friction and wear using asperity deformation models.
J M Challen and P L B Oxley - Wear 53 (1979)

Switching the material pairs around to give rubber ball or hemisphere on hard flat, in other words a self-locating geometry, eliminates indentation thus avoiding load/friction force interaction; the contact area is flat.

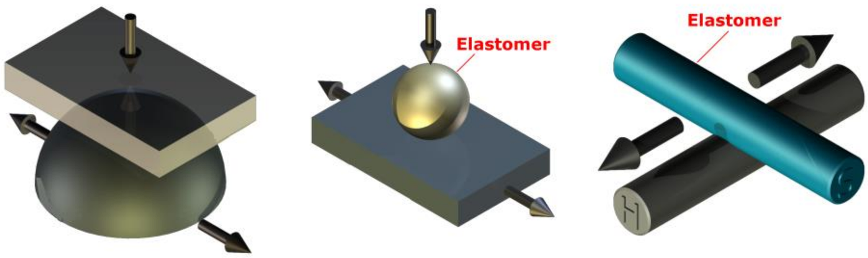


Our preferred solution was to use the self-locating contact geometry of Rubber Ball or Hemisphere on Hard Flat. The avoidance of indentation eliminates the problem of load/friction force interaction, associated with the hard ball on rubber flat arrangement, and the self-locating nature of the sample eliminates the alignment issues associated with cantilevering of 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.



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

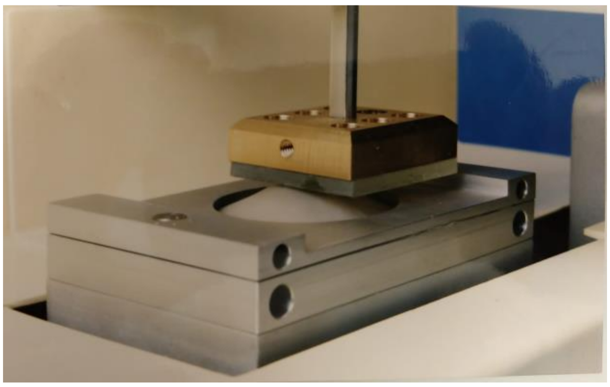
Applying the lubricant before applying the load

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.

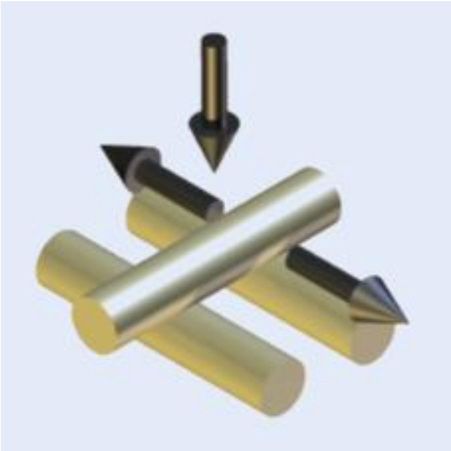
Self-locating Elastomer Sphere on Flat



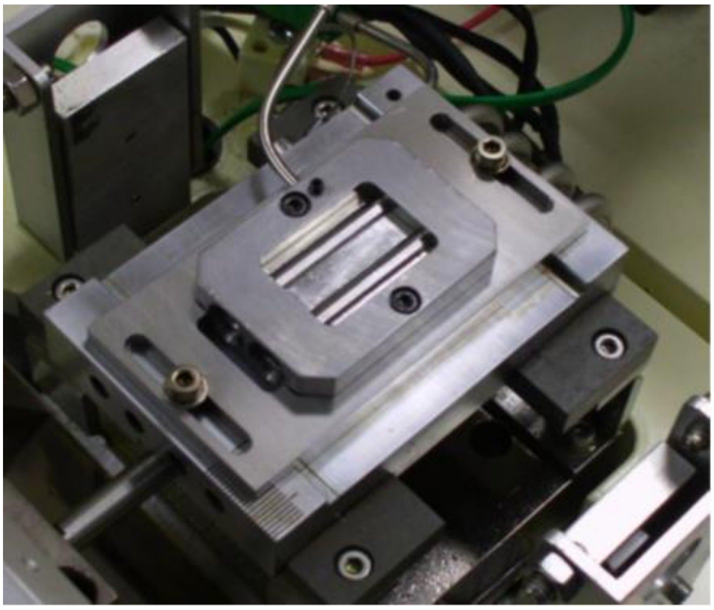
With this geometry, a flat counter-face surface is loaded against an elastomer sphere or hemisphere. This ensures a flat friction contact. In the example shown above, a thin sheet of elastomer (from a surgical gloves) has been stretched and clamped over a hemispherical elastomer hemisphere. In addition to modelling contacts between an elastomer and a hard surface, this geometry has also been used in skin friction simulations.

Self-locating Crossed Cylinder

With the crossed cylinder contact, of elastomer sliding against a steel rod, the friction surface remains parallel to the line of motion; the other way up, with a hard rod sliding along an elastomer surface, a "bow wave" is formed in the rubber, which entirely messes up the friction measurement.



For practical purposes, it is easier to use a "pin on twin" geometry, with the "pin" the elastomer. This works if the pin is a thin sheet of rubber glued to a metal cylinder sample.



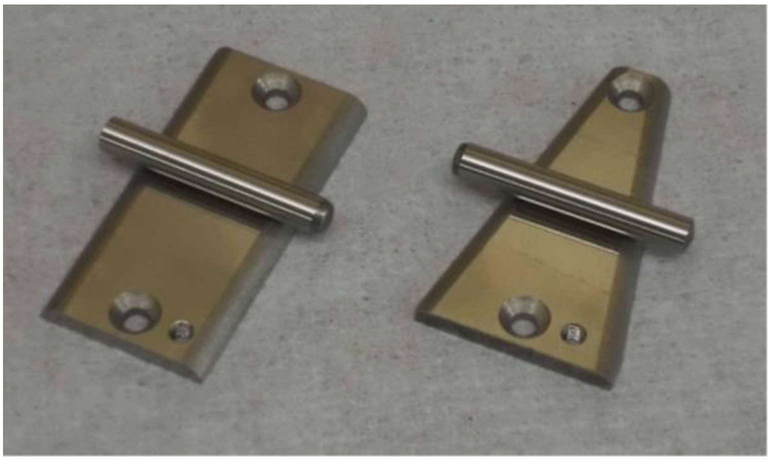
Self-locating Pin on Curved Edge Plate

With the pin on twin geometry, lubricant can flow freely to both sides of the contact, which may not be a particularly good model of a seal. A similar self-locating contact can be formed by using an elastomer pin sliding against a curved edge specimen. The curved edge has two beneficial effects:

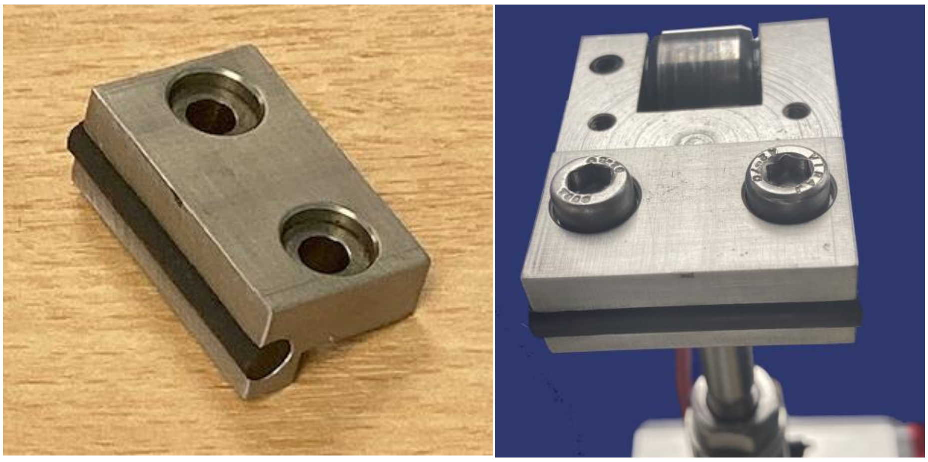
1. With a sharp-edged plate specimen, the edges will cut into the elastomer, increasing the machining or ploughing component of friction. With a pin that does not overlap the edge of the plate, the contact conditions at either end of the pin are indeterminate.
2. By making the pin overlap the edge of the plate, one side of the contact can be lubricated and the other side dry, mimicking the conditions in a seal.

Self-locating Pin on Wedge Plate

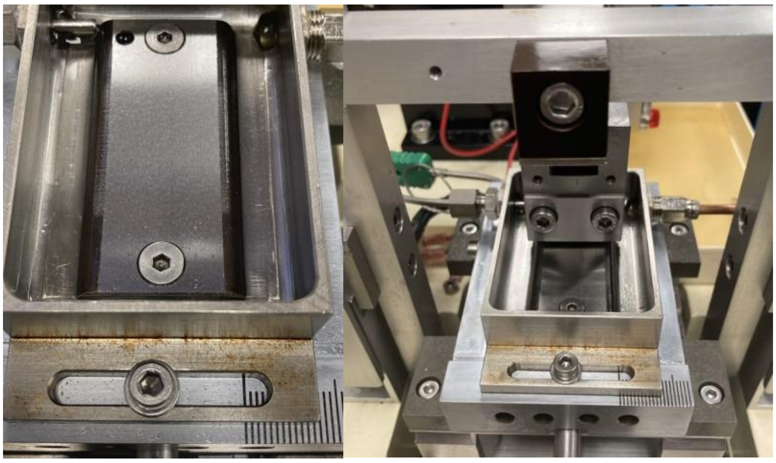
A further advantage of a curved edge specimen is that it allows wedge shaped specimens to be used, such that for a given applied load, the resulting contact pressure varies with stroke.



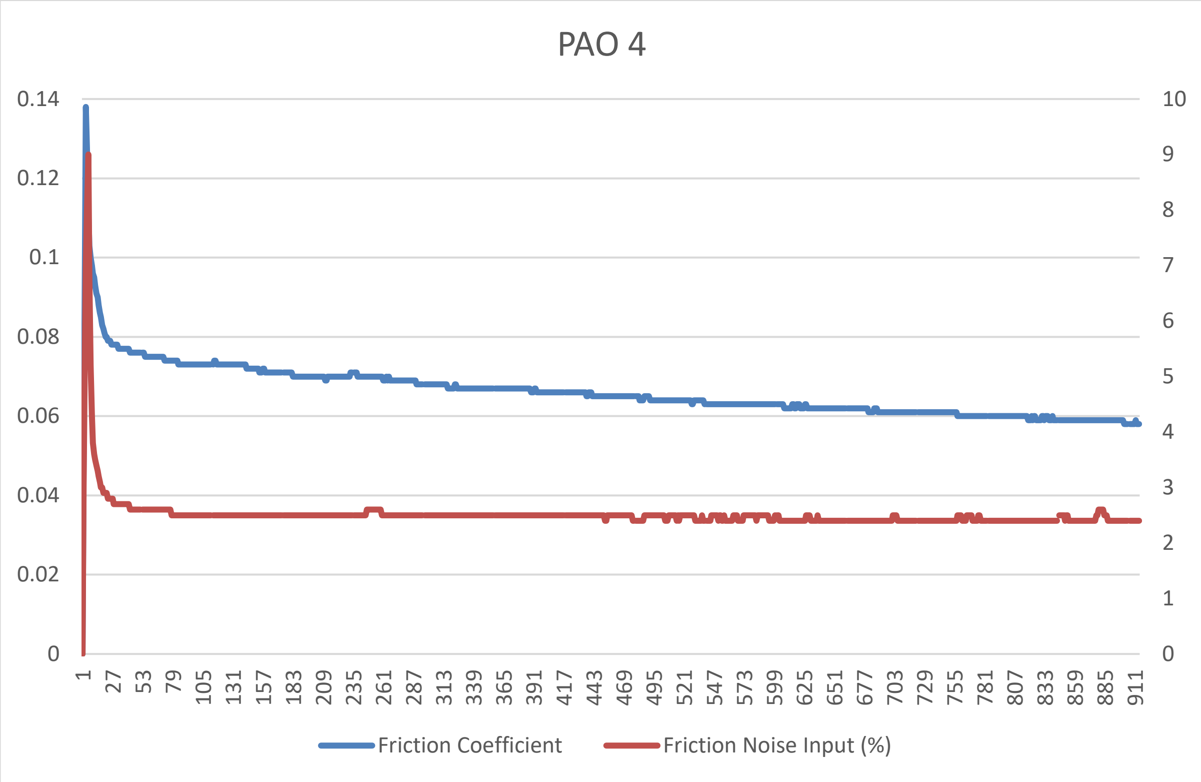
O-ring Section Tooling

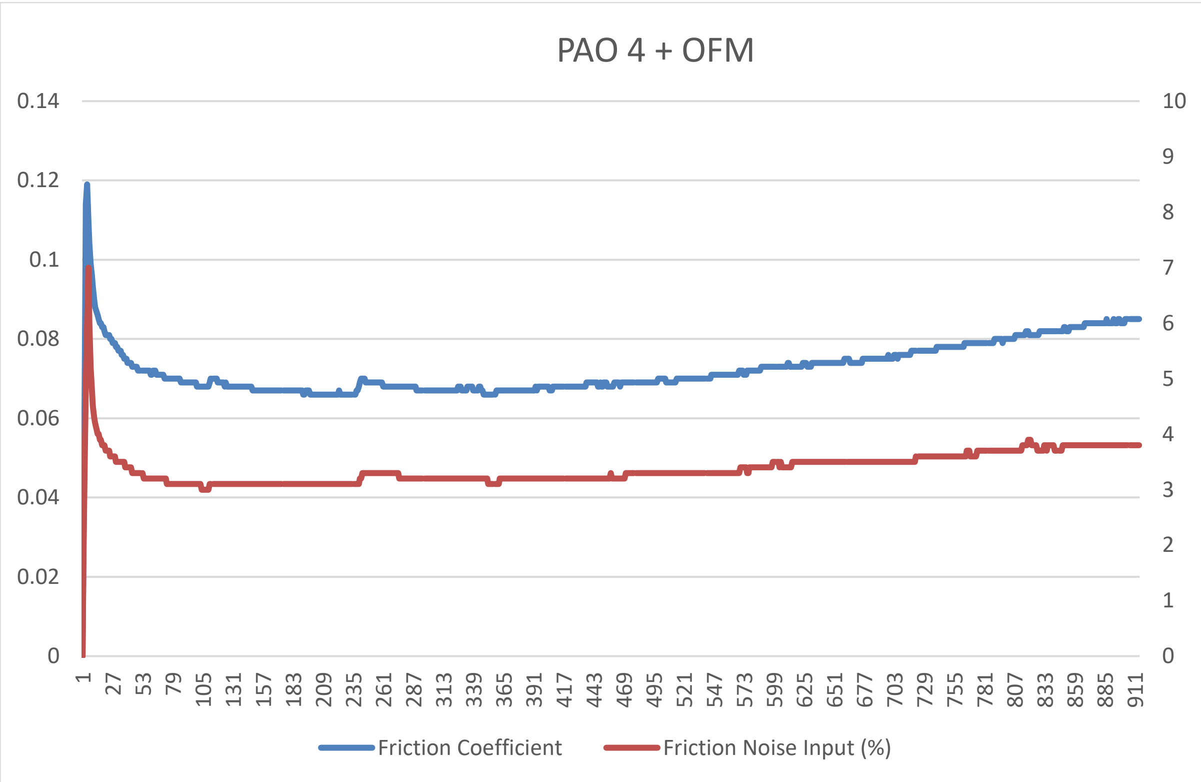


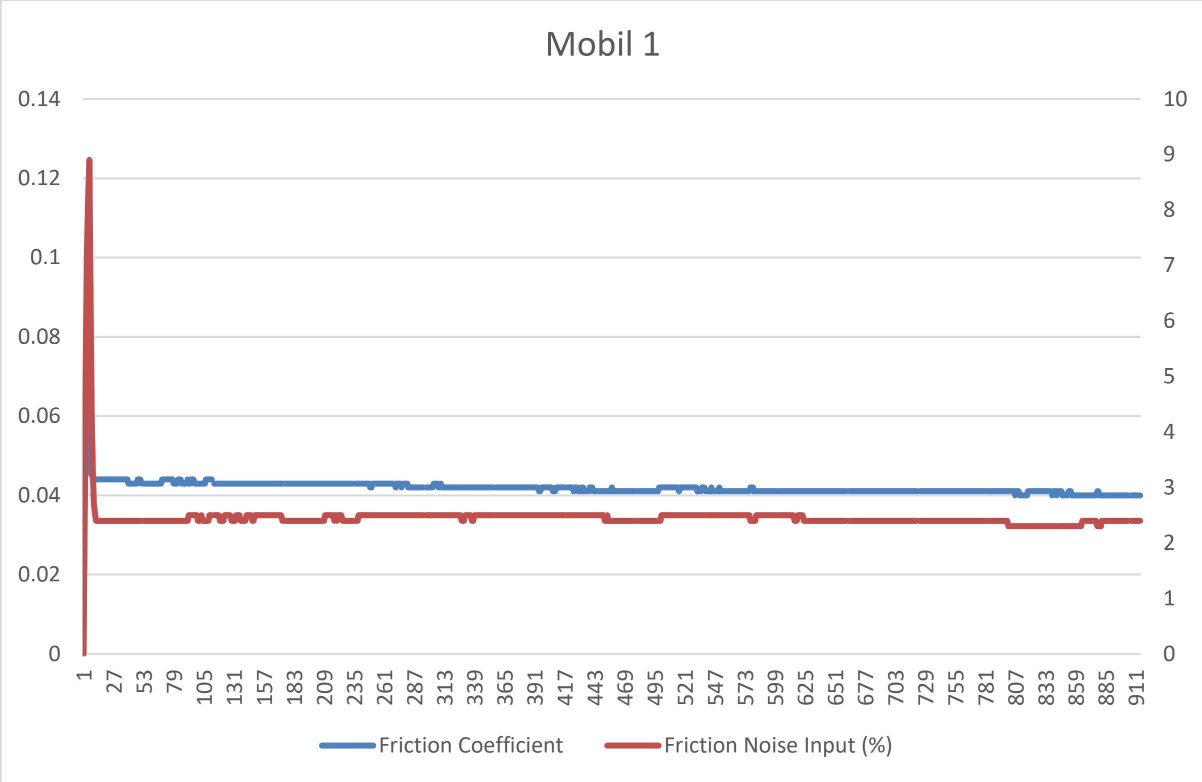
Simple tooling can be produced to mounting a length of O-ring section in a reciprocating tribometer sample holder, as shown above.

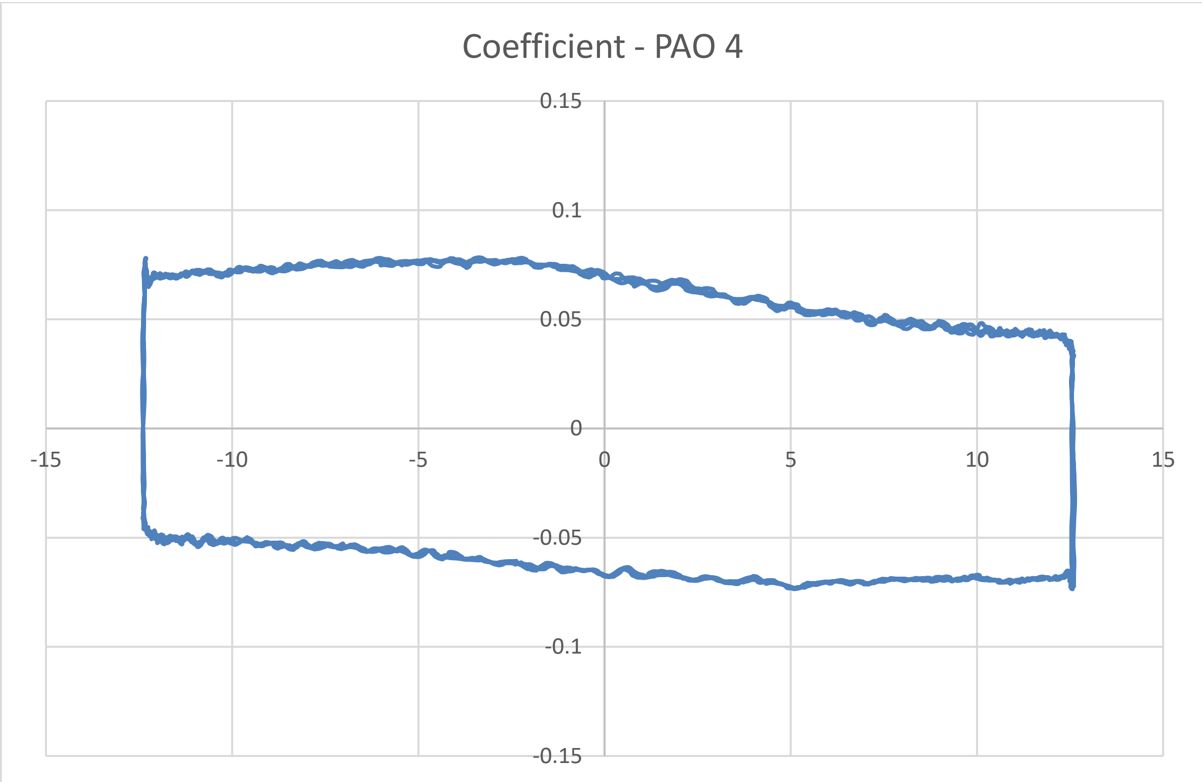


Used in conjunction with a curved edge specimen, edge effects are avoided, and lubricant can be confined to just one side of the contact, with some interesting effects:


PAO 4
0.14
0.12
0.1
0.08
0.06
0.04
0.02
0
10
9
8
7
6
5
4
3
2
1
0
1
27
53
79
105
131
157
183
209
235
261
287
313
339
365
391
417
443
469
495
521
547
573
599
625
651
677
703
729
755
781
807
833
859
885
911
Friction Coefficient
Friction Noise Input (%)

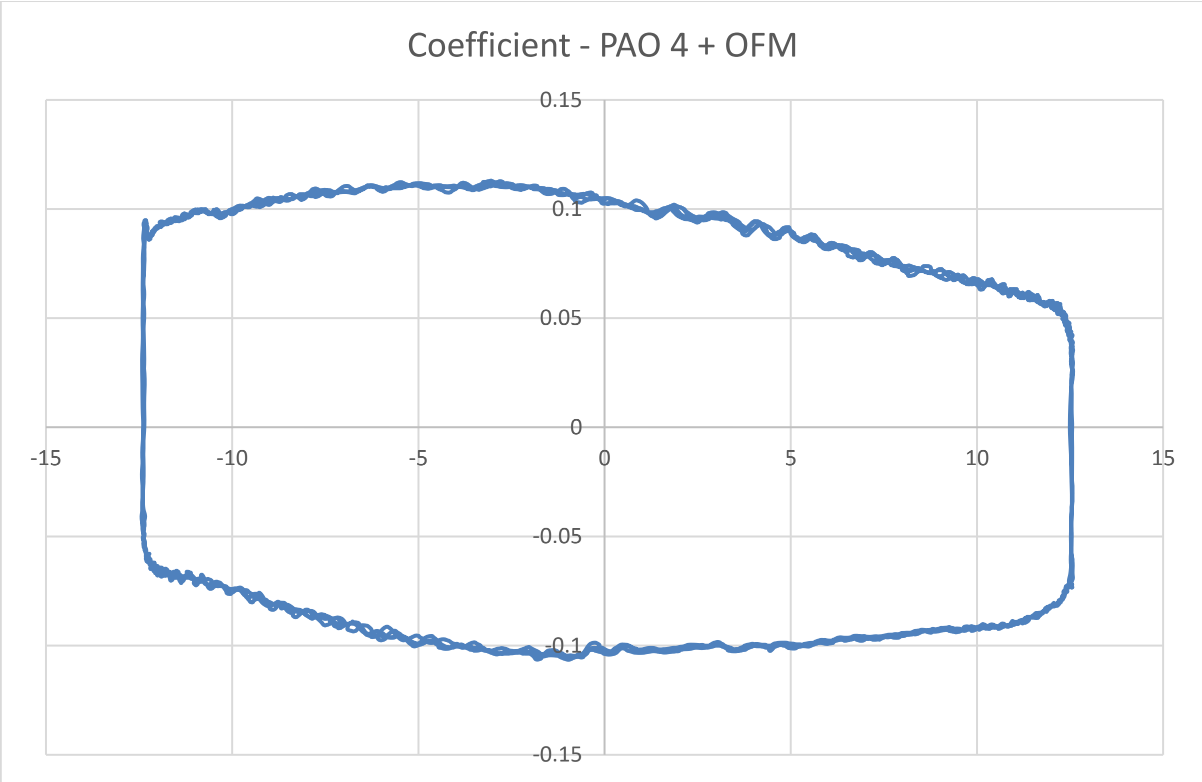

PAO 4 + OFM
0.14
0.12
0.1
0.08
0.06
0.04
0.02
0
10
9
8
7
6
5
4
3
2
1
0
1
27
53
79
105
131
157
183
209
235
261
287
313
339
365
391
417
443
469
495
521
547
573
599
625
651
677
703
729
755
781
807
833
859
885
911
Friction Coefficient
Friction Noise Input (%)



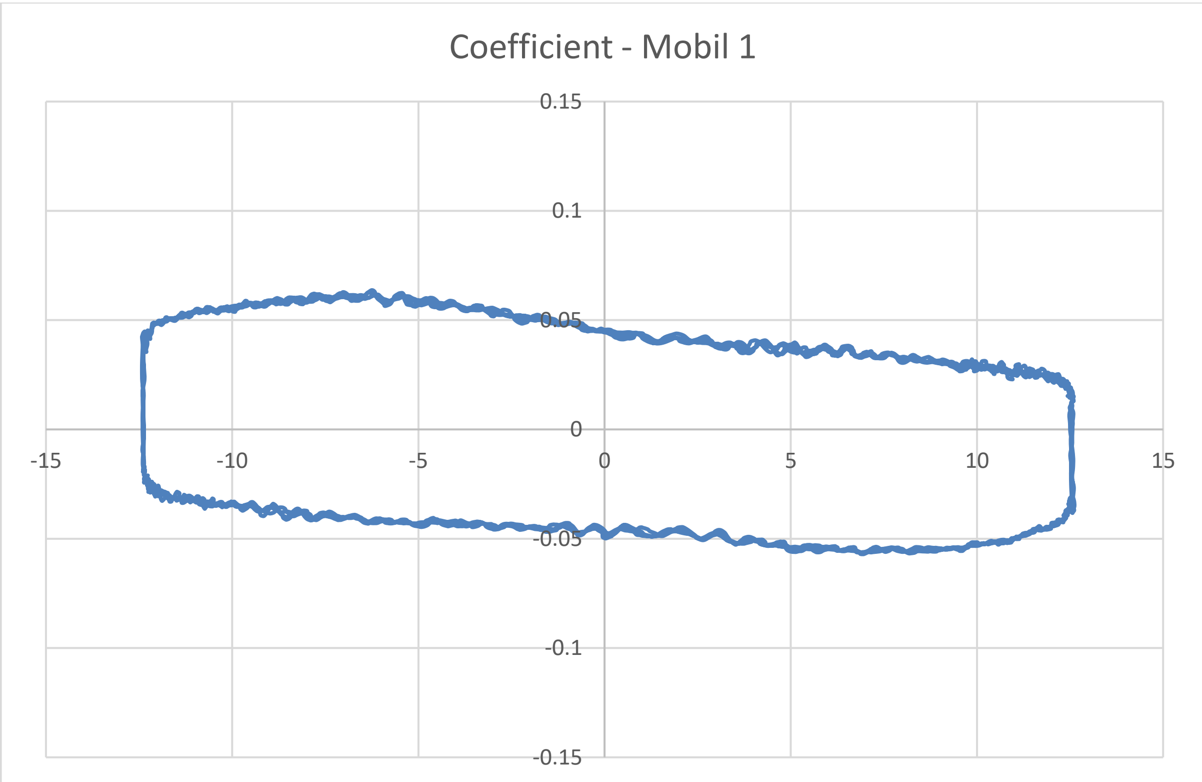


Lubricated Side

Un-lubricated Side



Lubricated Side Un-lubricated Side



Lubricated Side Un-lubricated Side

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!

Rubber friction measurement can be tricky, and that is even before we have introduced a lubricant!