Sliding Hertzian Point Contact Tests

What wear mechanisms do we actually produce with a sliding hertzian point contact?

Why is a sliding point contact less sensitive than a sliding line contact?

Introduction

Correlation Criterion for Modelling Real Systems
The most important criterion for correlation between model test and actual application is that the test should reproduce the wear and/or failure mechanisms of the application. We can be confident that if the wear and/or failure mechanism in the laboratory emulation is not the same as the wear and/or failure mechanism in the real system, the test model is probably wrong.

Sliding Hertzian Point Contact
We can start from the position that, in the real world:

1. There are no macro-scale engineering applications involving sliding hertzian point contacts. The only real system hertzian point contacts involve sliding and rolling (toroidal transmission) and pure rolling (ball bearings), where the point of contact moves on both surfaces.
2. There are very few engineering applications where the same material is used on both sides of a pure sliding tribological contact. This is to avoid poor tribological compatibility (Rabinowicz).
3. Most practical engineering surfaces, for sliding or sliding/rolling lubricated applications, are much rougher than most standard, sliding point contact test surfaces, where many test standards call for highly polished specimen surfaces.
4. Most practical engineering surfaces are designed to wear, not fail. Many sliding hertzian point contact tests start by failing the surface and running on a failed surface, or, to be more precise, sub-surface.

However, there are many tribological tests using sliding hertzian point contacts, but it should be apparent that these tests do not model anything in the real world. It therefore begs the question as to why we use so many sliding hertzian point contact tests, for lubricant and material evaluation. There are no real life applications in which this type of contact occurs:

In essence, many of these tests were invented before we understood either surface chemistry or contact mechanics. Most were developed as scuffing and seizure tests for evaluating extreme pressure properties of lubricant additives. Of course, there is some logic in using like material sliding on like material if you want to produce surfaces with a tendency to scuff or seize.
The lack of correlation between test and application is of course recognised in most of the relevant standards. For example, ASTM standard test procedures typically include a “Bias” statement of the following form:

“The evaluation of “Property X” by this test method has no bias because “Property X” can be defined only in terms of the test method.”

In other words, it does not correlate with anything else! In many cases, the reasons for this should be obvious.

Conversation held with researchers at one illustrious organisation:

“We could not get our lubricant tests to correlate with our engine tests”

“What tests did you do?”

“We ran tests in accordance with ASTM D5706”

“But that’s a test for EP properties of greases, with a bearing steel ball reciprocating on a bearing steel flat, at room temperature. That was designed to model what happens in wheel bearings on cars, when they are transported on rail flats. What has that got to do with anything in an engine?”

“Well, it’s an ASTM standard and we followed the procedure very carefully”

**Sliding Four Ball Geometry**

Do we ever see anything like this in a real engine or gear-box?

Surface topography obliterated!

Comment made by presenter at Wear of Materials 2011:

“At the end of the project, we ran four ball tests on the fresh and used oils, but could not get any correlation with the engine tests”

Response:

“You have just wasted fifteen minutes of my life!”

**Ball on Flat Geometry**

Do we ever see anything like this in a real engine or gear-box?
ISO Fuel Lubricity Test: Surface topography obliterated!

The first issue to address in designing a test is which way round, in terms of relative hardness, to have the specimen pair. Traditionally, many wear tests have involved running a soft pin or ball on a hard disc or plate. Under these conditions, the wear occurs on the softer material, sometimes accompanied by the generation of a transfer film on the harder material. Measurement of material lost from the softer pin or ball is relatively easy.

If the specimen pairs are reversed, with a harder pin or ball running on a softer disc or plate, we generate a different mechanism or more precisely, mechanisms, depending on the relative hardness, the contact pressure and contact shape. What happens to the disc or plate specimen depends on the nature of the material.

With metallic specimens, plastic deformation of the surface and work hardening take place thus changing the nature of the material. With coated surfaces, repeated passes by a hardened pin or ball may give rise to adhesion-de-lamination and subsequent failure of the coating.

If we define wear exclusively as the removal of material, it will be apparent that if the scar generated on the disc or plate specimen involves plastic deformation (material is redistributed but not removed), then it cannot be considered in the true sense as a “wear” scar. With this contact configuration, the processes involved may be more analogous to forming or machining processes. In the case of forming, we would anticipate plastic deformation, and in the case of machining, removal of material by cutting or ploughing action.

Normalizing wear volume by sliding distance (mm³/N-m) with hard ball on soft flat is at best an approximation!
In real machines, we frequently find contacting materials of similar hardness, with the result that wear is shared between the two contacting surfaces. The only solution here is to measure the wear on both surfaces, not forgetting that, if the materials are different, the wear rate will still be dependent on which material is used for the pin or ball and which is used for the disc or plate. This is because the energy inputs are different for the two specimens.

![Contacting Surfaces](image)

We can of course adjust the way that the wear is shared between the two surfaces, to convert:

![Conversion Example](image)

We do this by altering the “overlap parameter”.

**Overlap Parameter**
The "overlap parameter" (Czichos) is defined as the ratio of sliding distance for "body" divided by sliding distance for "counter body". For the thrust washer this is 1, for fretting tests it is close to 1, but for pin on disc tests it is variable, but is typically less than 0.05. The overlap parameter also applies for reciprocating tests; the longer the stroke, the less the linear wear on the plate sample for a given amount of wear on the ball.

Bowden and Tabor were right: long stroke, low speed, sliding has significant advantages over short stroke sliding!

**Contact Pressure – Orders of Magnitude**
The following is a simple illustration of the influence of contact geometry on mean contact pressure, let us consider a 10 mm steel ball on a flat and then a 10 mm diameter cylinder of the same mass in line or area contact, which means a cylinder 6.7 mm long.

![Contact Pressure Illustration](image)

102,800,000 Pa  5,076,000 Pa  510 Pa

To get a mean contact pressure of 102.8 MPa on a 10 mm diameter by 6.7 mm long cylinder on edge, a total force of 16.3 N would be required.

To get a mean contact pressure of 102.8 MPa on a 10 mm diameter pin on end, a total force of 8,074.9 N would be required.
Of course, we should remind ourselves that in the case of the point contact that we may not be justified in using the Hertzian contact equations in the first place. For these to be true, it is assumed that the contact is elastic, that the contact zone is flat, that there are no shear stresses in the contact zone and that the contact radius is much smaller than the radius of the ball. This will not be the case if we have a hard ball loaded against a softer flat, where plastic deformation may take place. In this case, we cannot sensibly use the Hertz equations and instead must use an elasto-plastic finite element model to evaluate the contact conditions. Such models lead to the perhaps slightly unexpected conclusion that increasing the load on our ball simply increases the size of the plastic zone. So you logically don’t really need to run repeat hard ball on soft flat tests, at different loads!

**Contact Pressure and Wear - Estimated from On-line Wear Displacement**

Having discussed the nominal contact pressure at the start of a ball on flat test, how does this progress with wear?

The difference in wear scar width is more or less established within the first few minutes. Subsequently, tests continue to run at very much lower contact pressures than at the start, but at different contact pressures depending on the fluid. For comparison of resulting wear or behaviour, should we not be increasing the load on the ZDDP sample, so that it runs at the same contact pressure as the base lubricant?
There would be no such differences in contact pressure in a real machine: the contact pressure in a gear, cam or ring liner contact does not alter just because we decide to test two different additives!

This is a fairly extreme example, starting with a very high hertzian contact pressure, but even with the ISO fuel lubricity test, the contact pressure with the low lubricity reference fluid falls from about 820 MPa to less than 20 MPa in the first fifteen minutes.

**Morphology of the Contact – Hard Ball on Soft Plate**

Few serious attempts have been made to explain exactly what happening with wear scar generation in the sliding point contact tests such as the ISO diesel fuel lubricity test.

For a ball sliding in a conforming groove, we would expect an elliptical contact patch.

![Diagram of hard ball on soft plate](image)

However, with reciprocating motion, at stroke end we would expect the ball to conform to the formed end of the wear track, increasing the dimensions of the contact patch in the direction of motion.

![Diagram of hard ball on soft plate with grooving](image)

Exactly this effect is evident with tests using the high lubricity (good) reference fuel, with wear scars showing an elliptical wear scar with grooving in the direction of motion, plus end of stroke witness marks, which lack directionality. Indeed, ISO 12156 both mentions and illustrates this type of wear scar but makes no attempt to explain it, simply limiting comment to “In these cases it can be more difficult to see or measure the true scar shape”.
It is apparent that the ball wear scar for the high lubricity reference fluid involves two different wear mechanisms: an elliptically shaped central area subject to severe adhesive wear and end of stroke witness marks with the appearance of three body abrasive wear, perhaps caused by the accumulation of wear debris at the end of the stroke. An alternative explanation for the witness marks could be a form of impact fretting; the relative motion of the ball against the end of the plate wear scar must involve surfaces coming into contact and then sliding with very small amplitude motion. Clearly the contact pressure distribution must be varying between mid-stroke and end of stroke.

With the low lubricity (bad) reference fuel, the main wear scar and the stroke end witness marks merge into one larger wear scar, with much less obvious directionality.
The wear scar with the low lubricity fluid has the appearance of seizure or galling. This would appear to be an example of “junction growth”, with the actual area of contact approaching the nominal area of contact.

The difference between the wear scars appears primarily to be a difference of wear mechanism, that being, for the high lubricity fluid, severe adhesive wear, plus something indeterminate on either side of the wear scar, in the direction of motion, and, for the low lubricity fluid, seizure. Why bother measuring the scar dimensions? Why not just report whether the fluid produces seizure or not?

**Ball on Soft versus Ball on Hard Plate**

The following experiments were run with a 6 mm diameter 52100 steel ball sliding against a NSOH BO1 tool steel plate, with a stroke of 25 mm, load of 28 N and frequency of 5 Hz, lubricated with PAO and PAO plus 0.5% OFM.
With the PAO on its own, we have a wear scar with much the same appearance as the wear scar generated with the low lubricity reference fluid in the ISO diesel fuel lubricity test, in other words, junction growth and seizure. Unsurprisingly, we have friction and friction noise spikes.

With the friction modifier, the wear scar looks much the same as the wear scar generated with the high lubricity reference fluid in the ISO test; an elliptical central scar with signs of severe adhesive wear. However, the wear scar on the plate is wavy, which indicates that we have plastic flow.

The effect of the friction modifier has thus been to limit adhesion to the point where plastic flow and ratcheting occurs. The result is that although the friction may be marginally lower and no seizure occurs, the friction noise is much higher, because the resulting surface is bumpier. We can see this effect in more detail if we examine the instantaneous friction signal.
By way of comparison, if we run on a hardened tool steel plate, we end up with a nearly perfectly round wear scar on the ball and no plastic deformation on the plate, with both lower friction and much lower friction noise, in other words, much smoother sliding. Of course, by hardening the plate, we have significantly increased the yield stress of the material, thus preventing plastic deformation.

We can see this kind of behaviour with many different standard hertzian point contact tests:

ISO Fuel Lubricity – Ball on soft plate

Optimol – Ball on hard plate

Obliteration of the surface

Severe adhesive wear
- How do we explain the linear scratching: work hardened wear/oxide debris pulled through the contact?

- What do we have at the right hand end of the scar: adhesive wear?

- What mechanisms are contributing to the resulting friction: ploughing and adhesion, depending on stroke position?

- Do these friction mechanisms have anything to do with friction seen in service, under mild wear regimes?

- What is this test meant to be doing and what does it mean?
EDX confirms oxide at Spectrum 1

The following example is taken from the inter-laboratory test data given in ASTM D7421 - 11 Standard Test Method for Determining Extreme Pressure Properties of Lubricating Oils Using High Frequency, Linear Oscillation Test Machine. Table 1 gives results for tests run on three different oil samples at either 1 mm or 2 mm stroke. Plotting standard deviation gives the following:
With Oil 1 and Oil 3, a better distribution is achieved at 2 mm stroke compared with 1mm stroke. Both Oil 1 and Oil 3 have failure test loads within similar ranges, with the lowest mean value 890 N and the highest mean value 1223 N.

By comparison, Oil 2 has a lowest mean value of 790 N and a highest mean value of 1521 N. The distributions with Oil 2 at 1 mm stroke are broadly similar to the distributions for Oil 1 and Oil 3 at the same stroke. However, at 2 mm stroke, the distribution is now significantly better than at 1 mm stroke and considerably worse than Oil 1 and Oil 3 at the same stroke.

The key difference here is that the mean failure test loads at 2 mm stroke with Oil 2 are now much higher, these being 1521 N at 80 C and 1477 N at 120 C. This indicates that the standard deviation increases with increasing load, presumably because of an increase in resisting friction.

So this data demonstrate that:

- The shorter the stroke, the greater the standard deviation.
- The higher the load, the greater the standard deviation.

One might assume that the first effect is to do with the overlap parameter and the second to do with system response.

**Wear Scar or Witness Mark?**

The conventional method of measuring the wear scar on a ball sample is to measure the wear scar diameter in the direction of sliding and transverse to the direction of sliding, but how do we know, in a low wear situation, that what we are measuring is wear and not simply a sort of witness mark, as one would get on the ball in a Brinell hardness test?

One approach is to normalize the nominal scar measurement by dividing it by the calculated initial Hertzian contact area. This way we can determine if the measured wear area is larger than the Hertzian contact. Although we cannot assume that a normalized wear scar area of 1 indicates no wear, we can assume that a value of 1 indicates a well performing lubricant compared with a lubricant that produces a value well in excess of 1, for the same test.
This approach allows us quickly to determine which lubricants perform well and which ones do not. It is a more rational approach to reporting results than simply relying on an absolute wear scar measurement. In addition to reporting a normalized wear scar measurement, there may also be some benefit in reporting the ellipticity of the wear scar.

Considering the wear scars shown above for the 0.5% OFM tests at 28 N, we have the following:

<table>
<thead>
<tr>
<th></th>
<th>Normalized Wear Scar</th>
<th>Ellipticity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hard Ball on Soft Plate:</td>
<td>9.95</td>
<td>0.289</td>
</tr>
<tr>
<td>Hard Ball on Hard Plate:</td>
<td>3.46</td>
<td>0</td>
</tr>
</tbody>
</table>

**Additive Sensitivity**

**Sliding Point Contact Test Sensitivity**

The standard ISO fuel lubricity test is insensitive to increased additive concentration, once a complete, coherent, additive film has been formed. It is perhaps safe to assume that the same applies to lubricants as well as fuels.

Although the ISO test generates a marked difference in wear scar size for nominally high and low lubricity test samples, the test lacks the necessary sensitivity to distinguish between good candidate samples with differing amounts or types of lubricity enhancing additive.

“The downward sweep of disparate sigmoidal curves for the friction reduction effects of different additives becomes close to one downward sigmoidal curve on a molar concentration basis. There are only a finite number of absorption sites for additives to latch on/adhere to, applying Langmuir adsorption theory”. (M F Fox).
Sliding Point Contact versus Sliding Line Contact

By contrast with the sliding point contact, historical data can be found to demonstrate sensitivity to friction modifier concentration using a line contact test configuration.

The key differences between a point contact and a line contact experiment are as follows:

<table>
<thead>
<tr>
<th></th>
<th>Point Contact</th>
<th>Line Contact</th>
</tr>
</thead>
<tbody>
<tr>
<td>Contact Area - Plate Specimen</td>
<td>Varies as specimen wears, with increasing scar width and length</td>
<td>Very nearly constant, with increasing scar length only</td>
</tr>
<tr>
<td>Contact Area - Moving Specimen</td>
<td>Two dimensional increase with wear</td>
<td>One dimensional increase with wear</td>
</tr>
<tr>
<td>Nominal Contact Pressure</td>
<td>Decreases following inverse square law</td>
<td>Decreases following inverse cube law</td>
</tr>
<tr>
<td>Initial Wear Rate</td>
<td>Most likely severe</td>
<td>Most likely mild</td>
</tr>
</tbody>
</table>

Note:

1. For mild wear, we would expect the surfaces to get smoother and for severe wear we would expect the surfaces to get rougher.
2. Obliteration of the original surface topography indicates scuffing or seizure.
3. We would expect a test that starts with a hertzian point contact to produce severe wear and/or plastic deformation during the running-in process, hence a roughening of the surface, followed by smoothing of the surface, as the nominal contact pressure reduces, with an associated transition to mild wear. This would appear to be an example of failing the surface at the start of the test, followed by subsequent running on a failed surface.
4. Most standard sliding point contact tests seem to start with very smooth surfaces and then produce an increase in surface roughness, but most real lubricated systems tend to start with a rougher, engineering, surface with mild wear resulting in smoother surfaces.

If we now compare the results generated with 6 mm diameter 52100 ball on soft gauge plate flat at 28 N with a 6 mm diameter x 10 mm long 52100 cylinder on soft gauge plate flat @ 200 N (500 MPa), we get the following:
We get much lower friction coefficient with the line contact specimen, but very little difference in friction between the line contact test with and without the friction modifier. However, when we look at the friction noise, we can see that the line contact test, with the friction modifier, produces much lower friction noise than the line contact test without friction modifier.

Tests were also run using 10 mm diameter balls at lower loads in order to reduce the resulting contact pressure to 1 GPa. The following series shows a selection of results generated using a 10 mm 52100 Ball on Soft Gauge Plate at 10 N load, once again compared with a 6 mm diameter x 10 mm long 52100 Cylinder on Soft Gauge Plate at 200 N.

With the ball on flat configuration, increasing the friction modifier concentration has little effect on the friction coefficient, but a significant effect on friction noise.
With the cylinder on flat configuration, increasing the friction modifier concentration has little effect on the friction noise, and only a small effect on friction coefficient.

But when we come to look at the wear, we get the following:

The issue is this: with no friction modifier, we have seizure with the ball contact and, once again, rather indeterminate wear scars with the different friction modifier concentrations, caused, even at these lower contact pressures, by plastic deformation of the plate sample. By contrast, it would appear that the wear scars generated with the line contact may just be sensitive to additive concentration, up to 0.5% level, but perhaps not much further. The next challenge was then to see if a test could be devised that was sensitive to additive concentrations above and beyond the 0.5% level tested so far.

**Pressure Change Estimated from Nominal Wear Volume**

Perhaps the difference in behaviour could have something to do with how the pressure changes with wear volume for the different contact configurations. Assuming a flat wear scar on the ball or cylinder, we can calculate the change in contact pressure with wear volume based on geometry alone, assuming that we start with a hertzian point contact. The following compares the variation in contact pressure with wear volume for a 6 mm steel ball on steel flat and a 6 mm diameter x 10 mm long steel cylinder on flat, the former with a load of 2 N, the latter with a load of 545 N, both giving an initial, nominal contact pressure of 820 MPa.
The issue is this: with the ball, the contact pressure falls much faster and much further than with the cylinder. Hence we have an initially very severe process followed by very benign conditions. The transition is almost digital! This model, based exclusively on geometry, looks remarkably similar to the derived values based on actual wear data, shown earlier:

**Pressure Change from Detailed Analysis (Professor John Williams)**

Consider a hard ball radius $r$, reciprocating in a straight line on a softer, in other words, wearing surface. The depth of wear is $\Delta$ and the sliding speed is $v$ and the applied load is $W$.

Viewed from the side, normal to the x-z plane, the loading can be considered to be concentrated in a line (rather than a thin ellipse), but viewed normal to the sliding direction there is conformity between ball and track. The semi-included angle is, say, $\beta_0$:

Geometry: \[ \Delta = r (1 - \cos \beta_0) \quad (1) \]
Now, the contact pressure, thought of as the intensity of the line loading, will vary along the arc of contact so that \( p \approx p(\beta) \). But equilibrium must be satisfied so that:

\[
\int_{-\beta_0}^{+\beta_0} p(\beta) \cos \beta \ r \, d\beta = W
\]  
(2)

Now suppose as the process proceeds and the depth of the wear track increases it does so in such a way that the Archard wear law is satisfied at every point. If the centre line wear is \( \delta \Delta \) then at the position \( \beta \) the radial wear must be \( \delta \Delta / \cos \beta \) is the ball is to remain spherical. So applying Archard at these two point and calling the intensity of loading on the centre line \( p_0 \):

\[
\frac{d\Delta}{dr} = K_w p_0 v \quad \text{and} \quad \frac{d\Delta / \cos \beta}{dr} = K_w p(\beta) v
\]  
(3)

So it follows that:

\[
p(\beta) = p_0 \sec \beta
\]  
(4)

Thus applying equilibrium (2):

\[
2 \int_{0}^{+\beta_0} p_0 \sec \beta \cos \beta \ r \, d\beta = W \quad \text{i.e.} \quad p_0 = \frac{W}{2r\beta_0}
\]

And so:

\[
p(\beta) = \frac{W}{2r\beta_0} \sec \beta
\]  
(5)

So the wear equation (3) becomes:

\[
\frac{d\Delta}{dt} = K_w p_0 v = K_w \frac{W}{2r\beta_0} v
\]  
(6)

Differentiating equation (1) we have:
So, equating the right hand sides of (6) and (7), we have:

\[
\frac{d\Delta}{dt} = r \sin \beta_0 \frac{d\beta_0}{dt}
\]

(7)

But call the nominal pressure on the contact \( P = \frac{W}{\pi r^2} \) and remembering that the integral of \( v \) with time must be the total sliding distance \( L \), we can write (8) as:

\[
\int_0^{\beta_0} \beta_0 \sin \beta_0 d\beta_0 = \frac{\pi}{2} K_w L
\]

(9)

Now, \( \int x \sin x \, dx = \sin x - x \cos x \), so we have an expression relating the growth of angle \( \beta_0 \) with applied load \( P \), distance \( L \) and wear constant \( K_w \):

\[
\sin \beta_0 \cos \beta_0 = \frac{\pi}{2} K_w L
\]

(10)

Now, if angle \( \beta_0 \) is relatively small (our data suggests typical values not exceeding 10\(^\circ\)) then we can use the small angle approximations for sine and cosine: \( \sin \theta \approx \theta - \theta^3/6 \) and \( \cos \theta \approx 1 - \theta^2/2 \), giving:

\[
\beta_0 = \left[ \frac{3\pi}{2} K_w L \right]^{1/3}
\]

(11)

We can now substitute this in the expression for the linear wear \( \Delta \) to get:

\[
\frac{\Delta}{r} = \frac{1}{2} \left[ \frac{3\pi}{2} K_w L \right]^{2/3} \rightarrow 1.4 (K_w L)^{2/3}
\]

(12)

Or, in terms of wear volume, since the cross-sectional area of the groove is \( r^3 (\beta_0 - \sin \beta_0 \cos \beta_0) \) and again using the small angle approximations:
In other words, the volume wear is linear with sliding distance and load, the reduction of load intensity being compensated by the increase in the load area. It would be interesting to see if there is any evidence that the linear wear rate goes with sliding distance to the 2/3 and volume with distance.

Experimentally, one would normally measure the width of the track, say 2a, but since $\beta_0$ is small:

$$a \approx \sqrt{\Delta \times 2r} \quad \text{i.e.} \quad a \approx r \sqrt{\frac{2 \Delta}{r}} \quad \text{so from (12) } a \approx 1.67r \left( KwPL \right)^{1/3}$$

(14)

The data for a 6 mm ball on soft plate with PAO suggest $P = 28 \left( \pi \times 0.003^2 \right) = 0.99$ MPa and after 1739 cycles $L = 1739 \times 0.05 = 87$ m, with $a = 380$ microns. Hence, from (14):

$$Kw = 5 \times 10^{-12} \text{ m}^2 \text{N}^{-1}$$

$$= 5 \times 10^{-3} \text{ mm}^3 \text{N}^{-1} \text{m}^{-1}$$

Pin on disc data from Archard and Hurst suggest for mild steel on tool steel a value of $Kw = 7 \times 10^{-3} \text{ mm}^3 \text{N}^{-1} \text{m}^{-1}$; this is surely too close to be a coincidence!

**Sliding Point Contact Repeatability Depends on Lapped Surfaces**

Why do the specimen surfaces have to be so smooth and un-representative of real engineering surfaces?

**SRV: Ball (10 mm dia) and Disc of same hardness**

**Ball:** Rc 60 to 62

Test disc: Vacuum arc re-melted (VAR) AISI 52100 steel with an inclusion rating using Method D, Type A, as severity level number of 0.5 according to Test Methods E45 and Specification A295/A295M or an inclusion sum value K1# 10 according to DIN EN ISO 683-17 and spherodized annealed to obtain globular carbide, Rockwell hardness number of 60-62 on Rockwell C scale (HRC), the surfaces of the disk being lapped and free of lapping raw materials.

$$0.5 \mu m < Rz \quad (\text{DIN}) < 0.650 \mu m$$

$$0.035 \mu m < Ra \quad (\text{C.L.A.}) < 0.050 \mu m$$

$$0.020 \mu m < Rpk < 0.035 \mu m$$

$$0.050 \mu m < Rvk < 0.075 \mu m$$

Running-in process for SRV tests calls for an initial load of 50 N, giving a nominal maximum hertzian contact pressure of 1.718 GPa, with $a = 0.0118$ mm. The yield strength of 52100 hardened to Rc 60 to 62 is
approximately 2 GPa, so the contact should still be elastic, although there cannot be much margin, once frictional shear is introduced.

PCS: **Hard ball (6 mm dia) on soft disc**

Note: Cannot sensibly use Hertz equations but should use elastic-plastic FE model!

**Ball:**  Rc 58 to 66

Test disc: AISI E-52100 steel machined from annealed rod, having a Vickers hardness “HV 30” scale number of 190 to 210 (according to ISO 6507-1), lapped and polished to a surface finish of Ra < 0.02 μm

Vickers 210 correspondence to a tensile strength of 710 MPa. For a Tresca material the mean contact pressure for initiation of yield is about 1.1Y, where Y is the uni-axial yield stress, hence, in the absence of frictional shear, we can assume that plastic flow will occur at a mean pressure of 781 MPa.

ISO fuel lubricity test runs with an applied load of 2 N, giving a (nominal) maximum hertzian contact pressure of 826 MPa, with a = 0.034 mm! The surface may still be elastic, but there will be a plastic enclave below the surface.

If we now add mechanical shear, because of sliding action, we would expect a decrease in either the applied load or temperature at which yield occurs, plus movement of the peak stress closer to the surface.

**Beilby Layers**

Both SRV and PCS disc samples are lapped, but little information is given with regard to the process. Why is it apparently necessary to have an amorphous lapped surface in order to achieve good test repeatability? What effect do the Beilby layers generated have on the morphology of the contact?

We are dealing with very small contact areas. Lapping produces uniform, fine, surfaces, resulting in a large number of asperity contacts, with the load distributed over this large number of contacts. Variations in the number of contacts, sample to sample, are likely to be small, thus resulting in better repeatability, the finer the surface.

Or, putting it another way, as we make the surfaces smoother, we would expect the ploughing component of friction to decrease compared with the adhesive component. For rougher surfaces, sampled by a very small contact area, variation in ploughing component may give rise to nominally poor repeatability, sample to sample.

If we have a hard ball loaded on a soft surface, with a load sufficient to plastically deform the surface, what happens when we start sliding? We have the equivalent of a sliding Brinell hardness test: bulk plastic deformation as opposed to plastic deformation at an asperity level.

Of course, we do not have this problem if we use a line or area contact and this leads us to the conclusion that for reasons of sensitivity, reduced contact pressure, the avoidance of seizure and the use of normal engineering surfaces, sliding line contact tests have advantages over sliding point contact tests. However, even with sliding line contact tests, there is a requirement for test optimisation in order to deliver increased sensitivity.
Conclusions

In this paper we have attempted to explain what it is that is produced in a sliding hertzian point contact test. For the most part, the only justification for running a sliding point contact test is that it allows rapid generation of a wear scar that can be measured, at best, with a degree of uncertainty as to the resulting wear mechanism or mechanisms involved and what exactly to measure. Whether this is a meaningful measurement is open to question, bearing in mind that there are no real engineering applications involving a sliding point contact. Perhaps it is fine for a screening test, but if we want to model real systems, we need to do better than this!

Summary

Sliding point contact tests:

- Essentially pass/fail test
- Repeatability depends on having highly finished sample surfaces in order to maximise the number of asperities supporting the applied load at test start
- Nearly impossible to avoid plastic flow at the start of a test either on or below the surface and obliteration of the surfaces
- Because of small contact scale, number of available active metal sites is very limited, making tests rather insensitive to additive concentration
- Rate and range of fall in contact pressure at the beginning of the test produces an almost “digital” response
- Most standard tests use material pairs that are unrepresentative of material pairs in real sliding contacts
- Does not model anything in the real world

Wasting time trying to:

- Improve sensitivity
- Achieve correlation with real applications
- Need to do something different!

Perhaps it is time we re-named all such tests “Sliding Brinell Hardness Tests”!