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Wear and friction in sliding point contact tests

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Tribology Trust Silver Medal 2017

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For those who have known me for a long time, the subject of this talk will come as no surprise; my views on the subject of tests that involve sliding a ball on a flat counter-surface, are well known. Some may well recall me describing sliding hertzian point contact tests as random result generators! Sure, ball on flat tests have their place, as basic screening tests, but I very much doubt that they should be used as the basis for serious tribological research. Sure, you can rub a ball against a piece of rubber, or a coated surface, but will your test deliver a meaningful answer?

Of course, as someone who makes a living from designing and manufacturing tribometers, I would be happy to sell you an instrument that rubs a ball on a ball or a ball on a flat, but in due course, I might try to persuade you to consider other test geometries.

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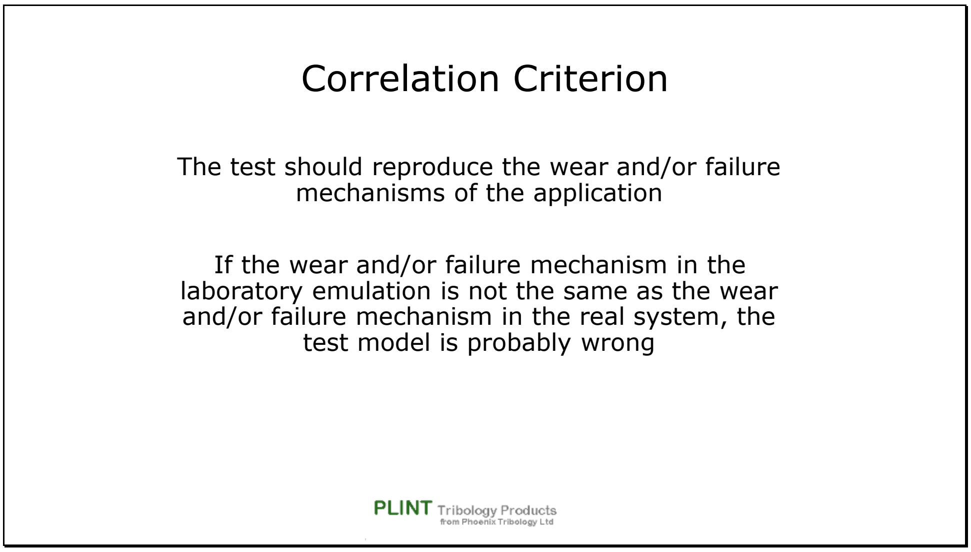

The most important criterion for correlation between model test and actual application is that the test should reproduce the wear and/or failure mechanisms of the application. We can be confident that if the wear and/or failure mechanism in the laboratory emulation is not the same as the wear and/or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

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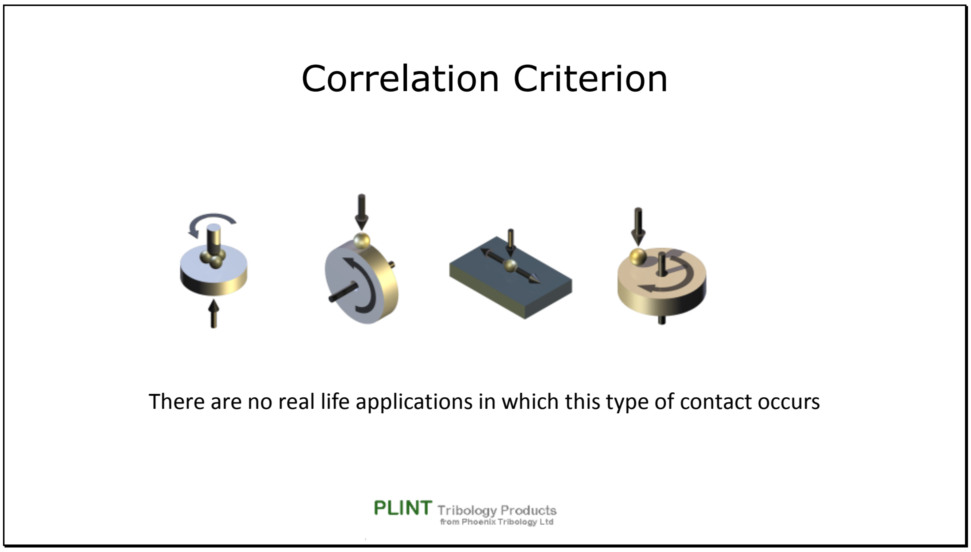

We must start from the position that, in the real world:

1. There are no macro-scale engineering applications involving sliding hertzian point contacts. The only real system hertzian point contacts involve sliding and rolling and pure rolling, 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 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 point contacts; it should be obvious that these tests do not model anything in the real world. It therefore begs the question as to why we use so many sliding point contact tests, for lubricant and material evaluation.

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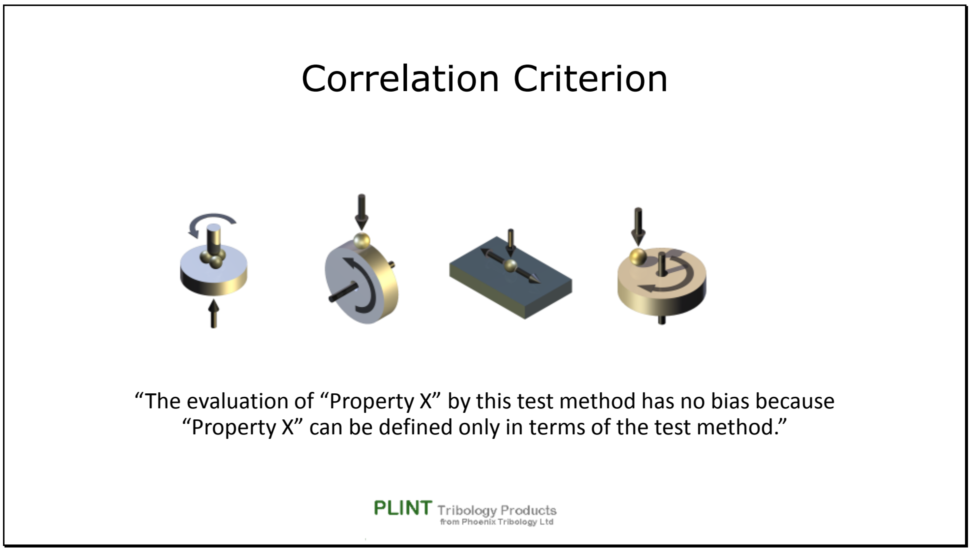

Many of these point contact tests were invented before we understood both surface chemistry and contact mechanics. Many were developed as lubricated scuffing or seizure tests, for evaluating extreme pressure properties of lubricant additives.

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 materials have mutual solubility, hence poor tribological compatibility.

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.

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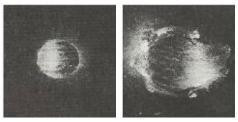
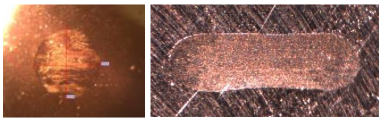
Correlation Criterion



Sliding Four Ball



ISO Fuel Lubricity

Does anything real look like this?

Examples of typical Quality Control Test!

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

Surface topography obliterated!

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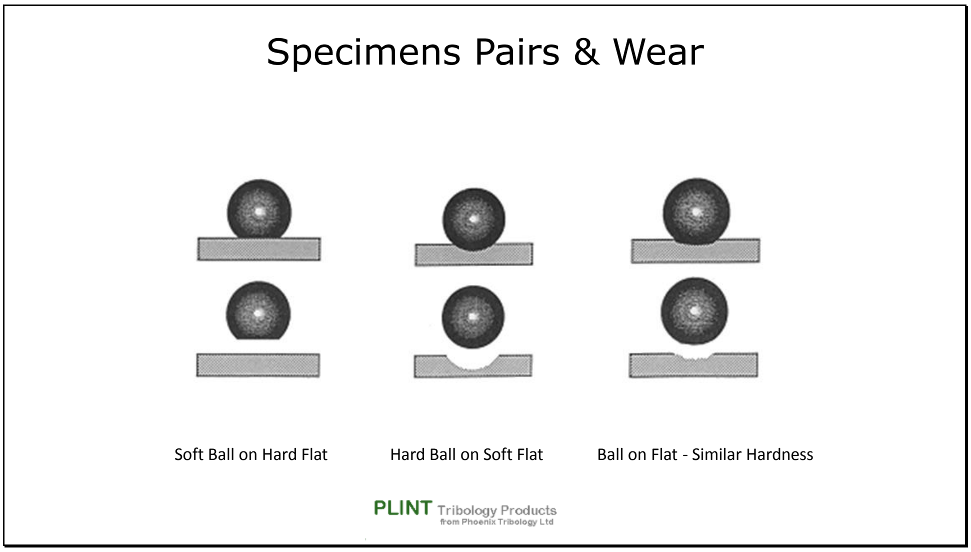

Many wear tests involve running a soft pin or ball on a hard disc or plate. Under these conditions, the wear occurs on the softer material, sometimes accompanied by the generation of a transfer film on the harder material. Measurement of material lost from the softer pin or ball is relatively easy.

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 takes 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.

Normalizing wear volume by sliding distance (mm^3/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.

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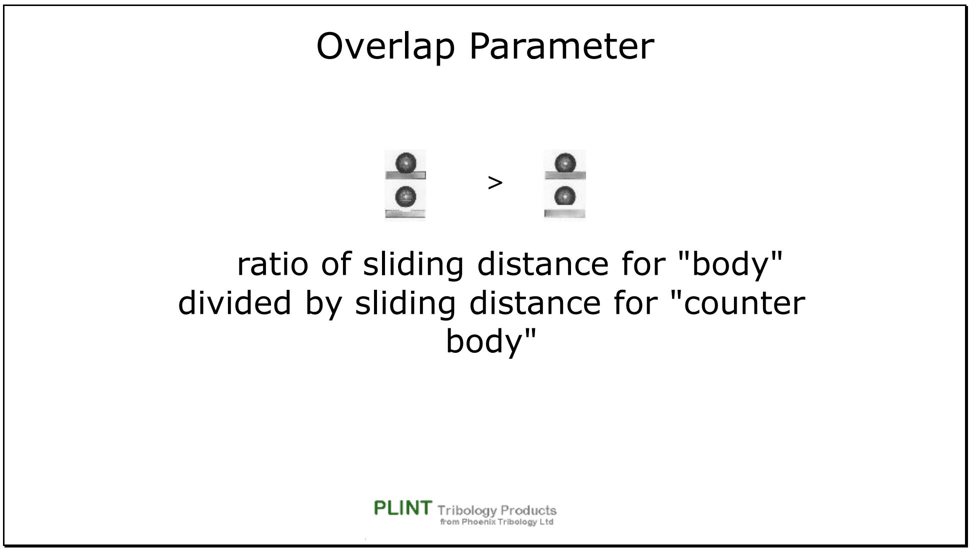

We can of course adjust the way that the wear is shared between the two surfaces, by altering the “overlap parameter”.

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.

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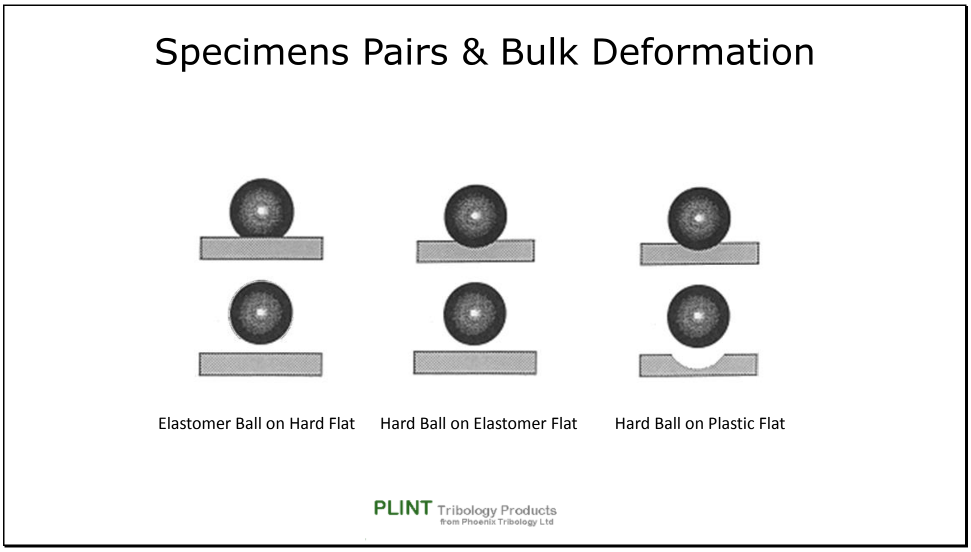

Of course, the ball on flat contact geometry is not reserved exclusively for relatively hard material pairs, but has also been used, with varying degrees of success, for contacts involving polymers.

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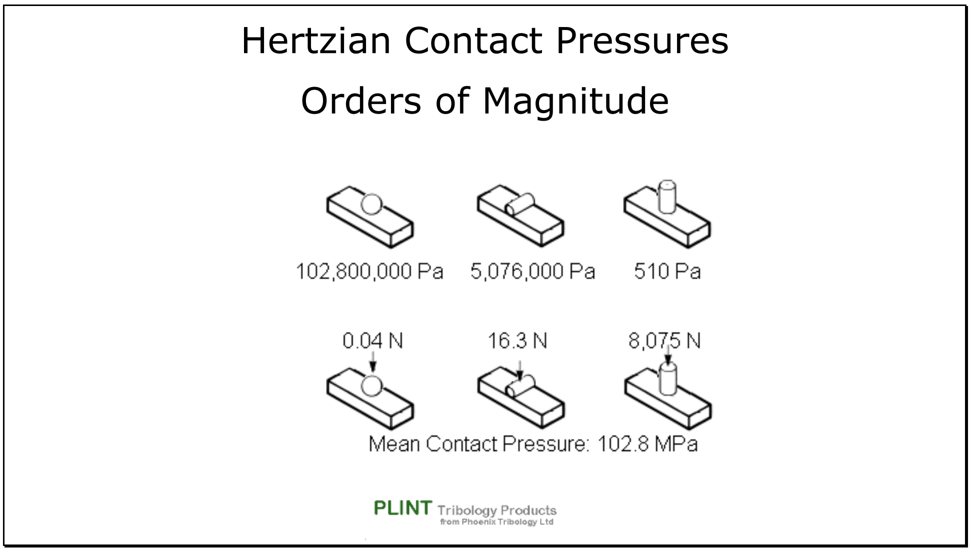

This is a simple illustration of the influence of contact geometry on mean contact pressure, in which we compare a 10 mm steel ball on a flat with a 10 mm diameter cylinder, of the same mass, in line or area contact. This means a cylinder with length 6.7 mm.

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.

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Contact Pressure

Orders of Magnitude

- Point Contact Assumptions:
  - No plastic deformation
  - Contact zone flat
  - No shear stresses in contact
  - Contact radius << ball radius
- For Hard Ball on Soft Flat:
  - Elasto-plastic finite element model
  - Increasing load increases plastic zone

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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.

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.

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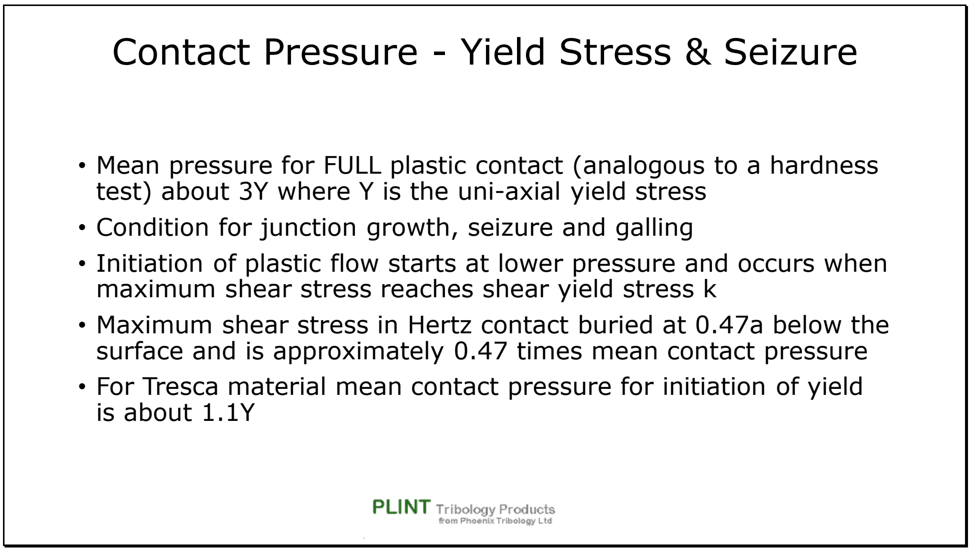

But what happens when we have a contact that is no longer elastic, in other words, a contact where the nominal contact pressure exceeds the yield stress of one or other of the contacting material? Analysis of such a contact leads to the perhaps slightly unexpected conclusion that increasing the load on our ball simply increases the size of the plastic zone. If the ball is the harder surface, we simply have a Brinell hardness test.

The mean pressure for FULL plastic contact (analogous to a hardness test) is about 3Y where Y is the uni-axial yield stress. This is likely to be the relevant condition for junction growth, seizure and galling.

Initiation of plastic flow starts at a lower pressure and occurs when the maximum shear stress reaches the shear yield stress k for the material. The maximum shear stress in a Hertz contact is buried at 0.47a below the surface and is approximately 0.47 times the mean contact pressure. This all means that for a Tresca material, the mean contact pressure for initiation of yield is about 1.1Y. But note that the surface material is still elastic - there is a miniscule plastic enclave under the surface.

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Contact Pressure - Yield Stress & Seizure

- To avoid plastic flow and unwanted strain hardening of surface, do not run tests with contact pressures in excess of 1.1Y
- If contacts in real application do not involve pressures close to or in excess of 3*Y, do not use conditions that produce contact pressure in excess of 3*Y in test
- If you choose to use test conditions that give contact pressures in excess of 3*Y, do not bother running a matrix of tests at different loads
- Note that with a sliding hertzian point contact, centre of contact will be subjected to far greater strain hardening than edges of contact

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If we now add mechanical shear, because of sliding action, we would expect a further decrease in either the applied load or temperature at which yield occurs.

The upshot of all this is confirmation of the general pointlessness, pun intended, of running a matrix of tests with a ball on flat specimen configuration starting at different loads. We learn nothing if our test simply produces the same result, regardless of test conditions. This is equivalent to trying to run a tensile test in which we try to control the load at levels in excess of the ultimate tensile strength of the sample; it does not matter what load we attempt to apply, we always get the same answer.

- To avoid plastic flow and unwanted strain hardening of the surface, do not run tests with contact pressures in excess of 1.1Y.
- If the contacts in your real application do not involve pressures close to or in excess of the 3*Y, do not use conditions that produce contact pressure in excess of 3*Y in your test system.
- If you choose to use test conditions that give contact pressures in excess of 3*Y, do not bother running a matrix of tests at different loads.
- Note that with a sliding hertzian point contact, the centre of the contact will be subjected to far greater strain hardening than the edges of the contact.

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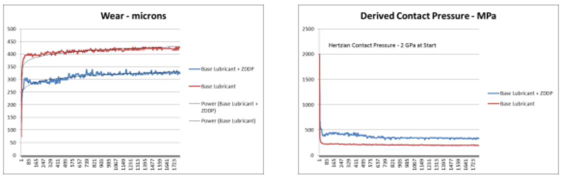
Contact Pressure and Wear



Comparative wear displacement, back-calibrated from post-test scar measurement

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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.

In this example, for comparison of subsequent wear 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!

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Contact Pressure and Wear

- The difference in wear scar width is 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. 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 fluids!
- 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 of the test.

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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.

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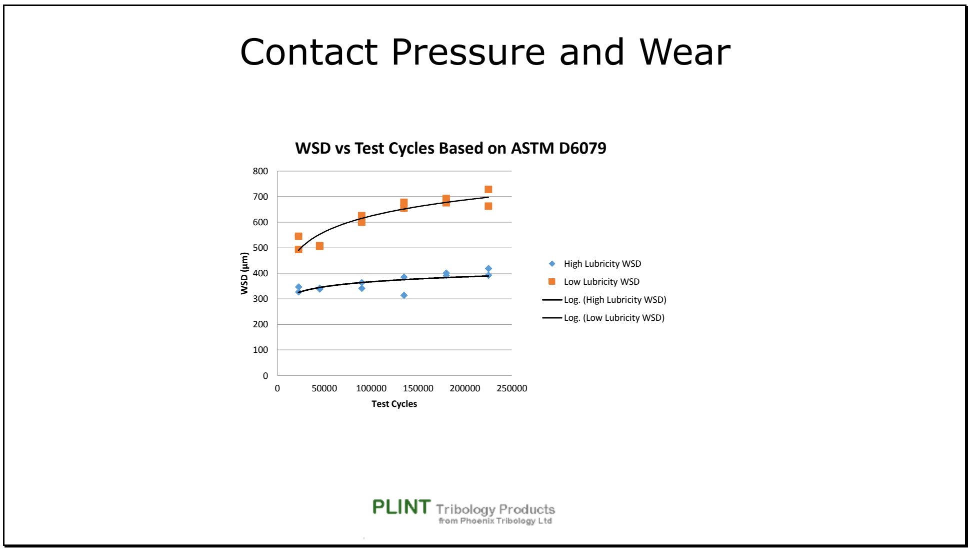

With the wear, or plastic deformation, occurring at the very beginning of the test, once the difference in wear between candidate samples has been established, the number of cycles over which the test is run is somewhat arbitrary.

In this example, the ASTM D6079 diesel fuel lubricity test procedure was run for the specified time, but at different reciprocating frequencies, resulting in the same test duration, but different numbers of cycles.

Tests were run for 75 minutes each at 5, 10, 20, 30, 40 and 50 Hz, the latter being the standard test frequency. The number of cycles per test was therefore 22500, 45000, 90000, 135000, 180000 and 225000 cycles respectively. Two repeat tests were performed at each frequency.

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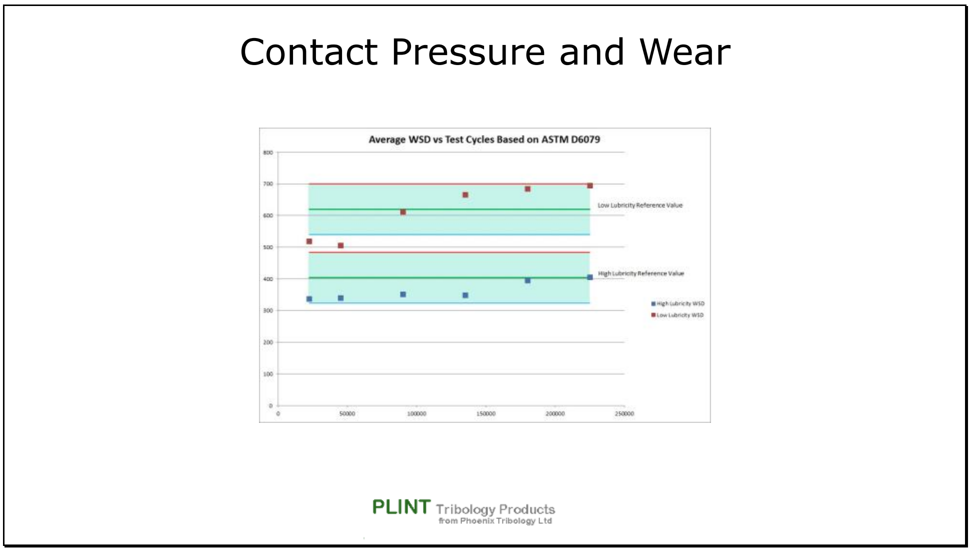

It is apparent that within approximately 50000 cycles the difference in wear between the high and low reference fluids has been established and that not much changes after that.

ASTM D6079 gives a reproducibility figure of 80 microns as the (approximate) 95% confidence level. It transpires that, with the exception of a couple of outliers, tests of 90000 cycles and more, fall within the reproducibility limit.

Plotting the average of each pair of measurements indicates that, perhaps not surprisingly, all the high lubricity measurements fall within the reproducibility limit, indicating that once the initial wear has taken place, further cycles result in very limited additional wear.

With the low lubricity sample, the average measurements for 90000, 135000, 180000 and 225000 all fall within the reproducibility limit.

It would appear that an acceptable result can be achieved within the limits of the standard, running tests at any frequency from 20 to 50 Hz!

It would appear that the choice of frequency and number of cycles is pretty much arbitrary and that comparative results could be achieved with much shorter tests.

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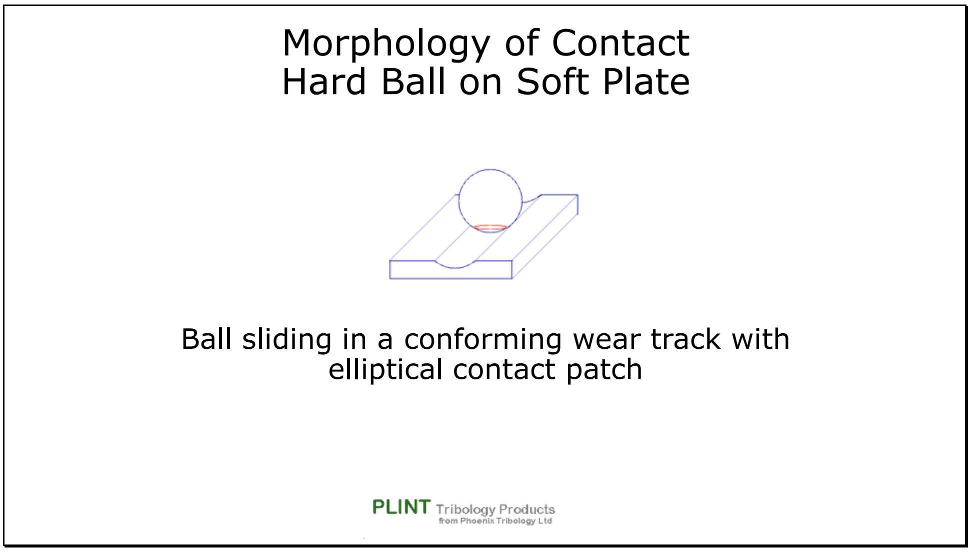

Few serious attempts have been made to explain exactly what is happening with wear scar generation in the sliding point contact tests.

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

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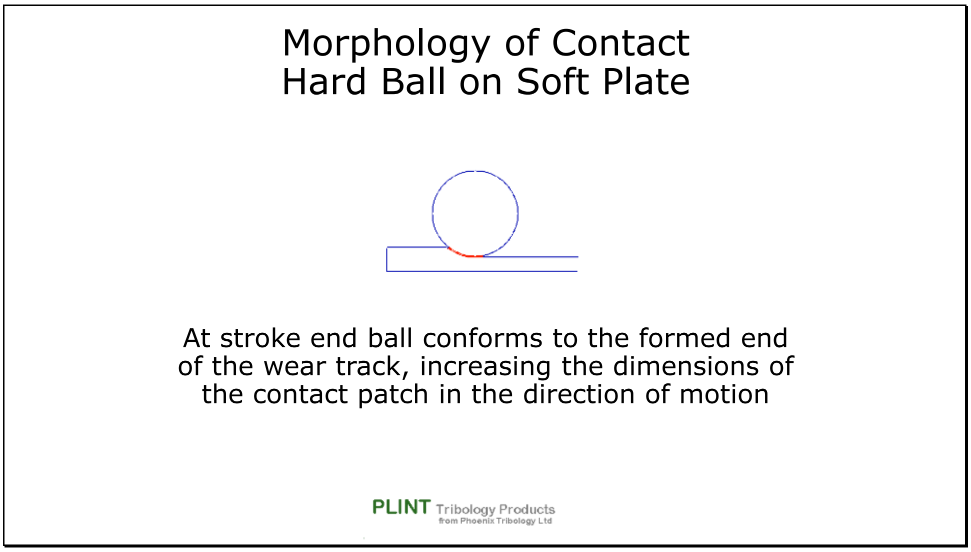

However, with reciprocating motion, at stroke end we would expect the ball to conform to the formed end of the wear track, increasing the dimensions of the contact patch, in the direction of motion.

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”.

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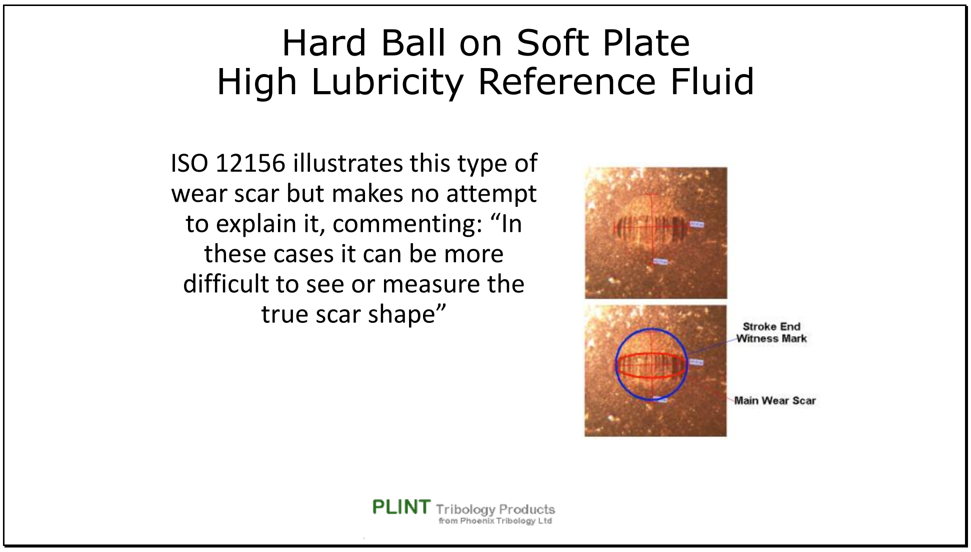

It is apparent that the ball wear scar for the high lubricity reference fluid involves two different wear mechanisms: an elliptically shaped central area subject to severe adhesive wear and end of stroke witness marks with the appearance of three body abrasive wear, perhaps caused by the accumulation of wear debris at the end of the stroke.

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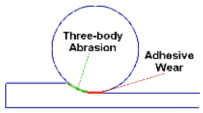
Hard Ball on Soft Plate
High Lubricity Reference Fluid



Ball wear scar for the high lubricity (good) 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.

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Hard Ball on Soft Plate
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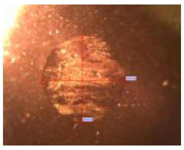

With the low lubricity (bad) reference fuel, the main wear scar and the stroke end witness marks merge into one larger wear scar, with much less obvious directionality. The scar has the appearance of seizure or galling. This would appear to be an example of “junction growth”, with the actual area of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

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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.

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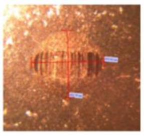
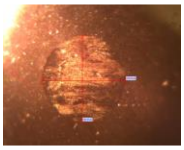
Hard Ball on Soft Plate
Comparison of Wear Scars





The difference between the wear scars is primarily a difference in 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?

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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?

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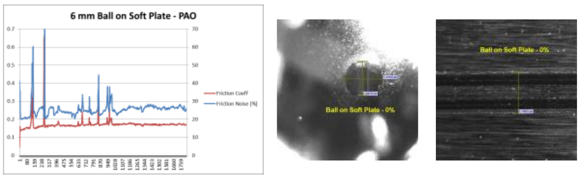
Hard Ball on Soft Plate



6 mm diameter 52100 steel ball sliding against a NSOH BO1 tool steel plate, stroke of 25 mm, load of 28 N and frequency of 5 Hz, lubricated with PAO

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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.

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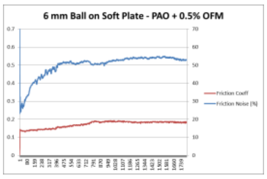
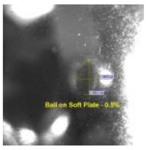
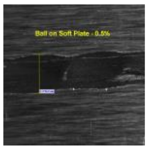
Hard Ball on Soft Plate







6 mm diameter 52100 steel ball sliding against a NSOH BO1 tool steel plate, stroke of 25 mm, load of 28 N and frequency of 5 Hz, lubricated with PAO plus 0.5% OFM

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With the friction modifier, the ball 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.

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Hard Ball on Soft Plate

- With the PAO on its own, 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: junction growth and seizure. There are 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.

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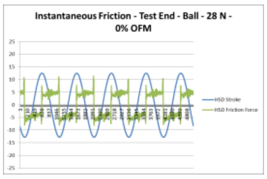
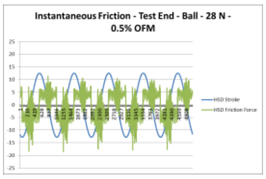
Hard Ball on Soft Plate





- Effect of friction modifier has been to limit adhesion to the point where plastic flow and ratchetting 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.

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The effect of the friction modifier has thus been to limit adhesion to the point where plastic flow and ratchetting 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.

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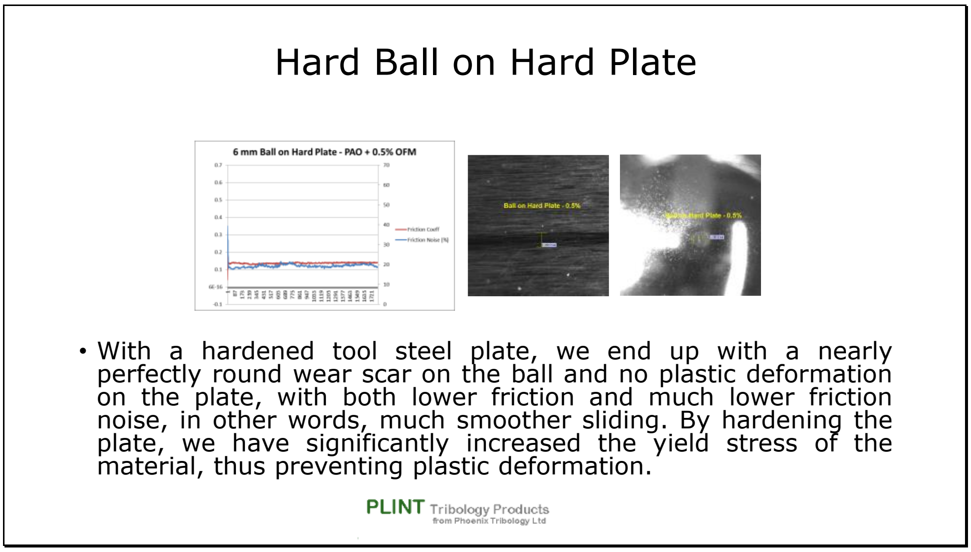

By way of comparison, if we run on a hardened tool steel plate, we end up with a nearly perfectly round and flat wear scar on the ball and no plastic deformation on the plate, with both lower friction and much lower friction noise, in other words, much smoother sliding. Of course, by hardening the plate, we have significantly increased the yield stress of the material, thus preventing plastic deformation.

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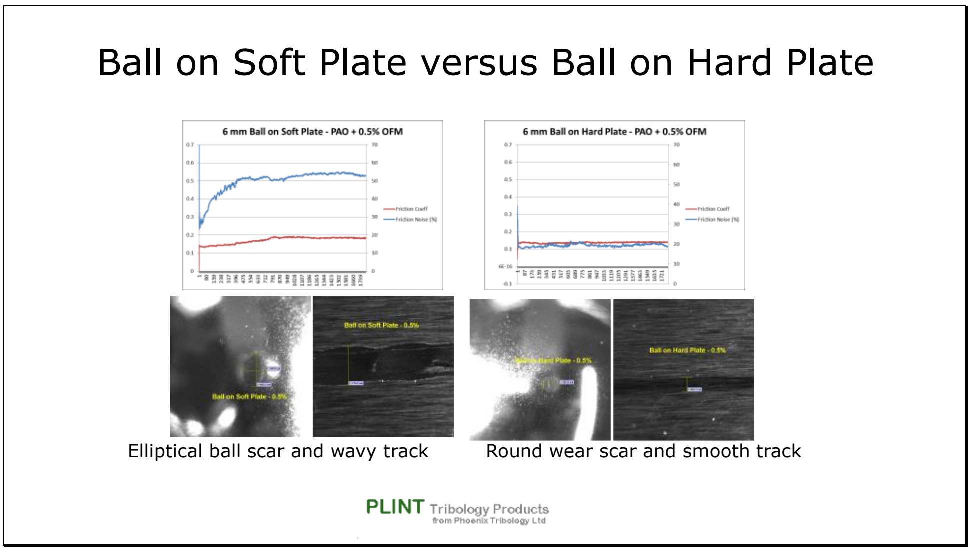

So, in this sliding hertzian point contact test, changing the hardness of the plate sample has a significant influence on both the wear and the associated frictional response.

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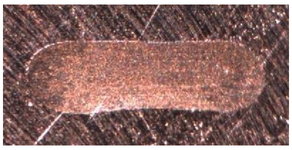
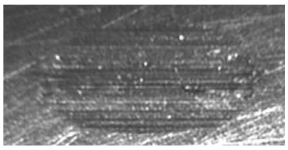
Ball on Soft Plate versus Ball on Hard Plate



ISO Fuel Lubricity – Ball on soft plate
Obliteration of the surface



Optimol – Ball on hard plate
Severe adhesive wear

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

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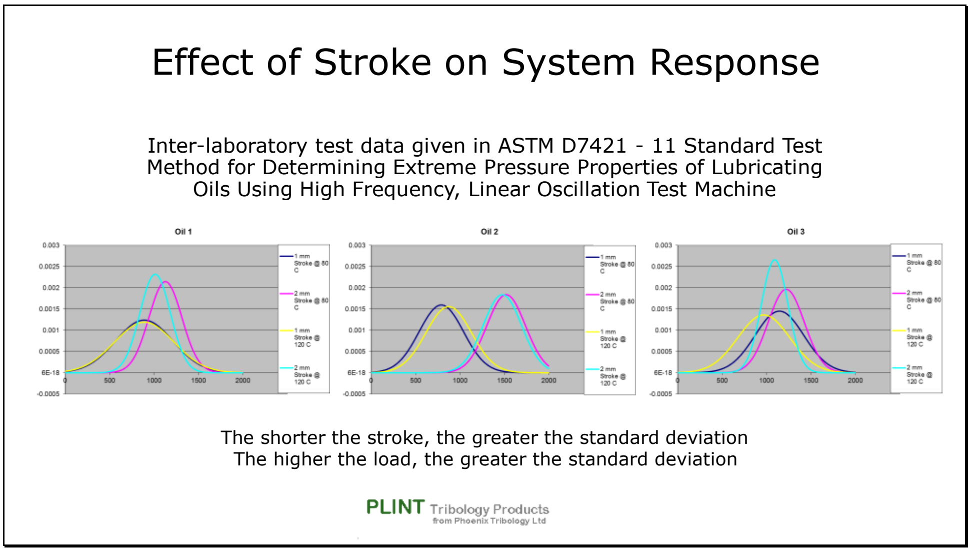


How does the stroke length, in other words, the overlap parameter, affect the response?

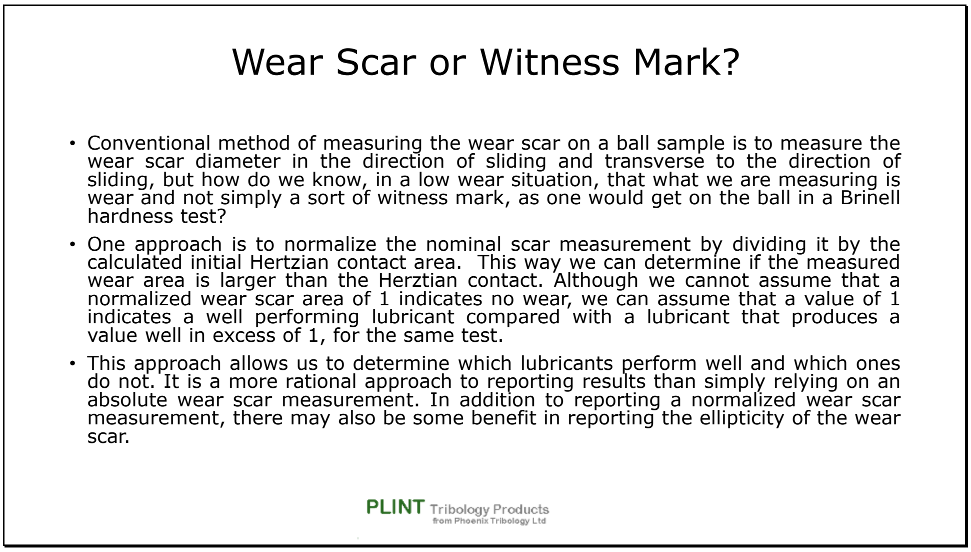
This example is taken from the inter-laboratory test data given in ASTM D7421 - 11 Standard Test Method for Determining Extreme Pressure Properties of Lubricating Oils Using High Frequency, Linear Oscillation Test Machine. Table 1 gives results for tests run on three different oil samples at either 1 mm or 2 mm stroke. Plotting standard deviation gives the results shown.

The distributions for all three oils at 1 mm stroke are broadly similar. At 2 mm stroke, the distributions for Oil 1 and Oil 3 are significantly better, but for Oil 2, the distributions are not significantly better. The key difference here is that the mean failure test loads for Oil 2, at 2 mm stroke, are now much higher, suggesting that the standard deviation increases with increasing load, which would presumably be accompanied by an increase in friction and an increase in contact area. So this indicates that:

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

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

Slide 31



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 Herztian 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.

Slide 32

Wear Scar or Witness Mark?

Wear scars for the 0.5% OFM tests:

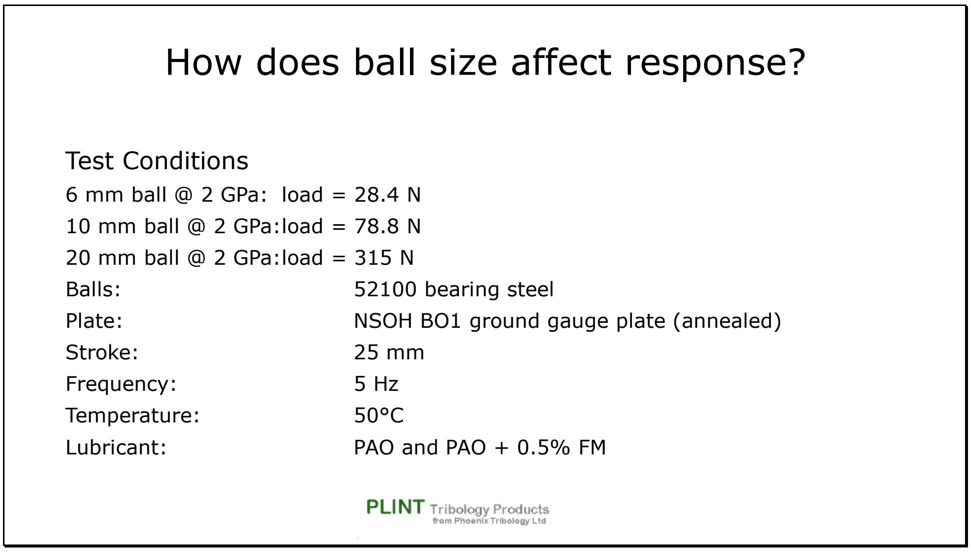
	Normalized Wear Scar	Ellipticity
Hard Ball on Soft Plate:	9.95	0.289
Hard Ball on Hard Plate:	3.46	0.0

In addition to reporting a normalized wear scar measurement, there may also be some benefit in reporting the ellipticity of the wear scar.

Normalized Wear Scar sizes and Ellipticity for the 0.5% OFM tests, shown earlier, are given in the table:

	Normalized Wear Scar	Ellipticity
Hard Ball on Soft Plate:	9.95	0.289
Hard Ball on Hard Plate:	3.46	0.0

Slide 33



A range of different ball sizes are used in a variety of different standard tests. In theory, there should be simple equivalence between tests run with different ball sizes, but the same contact pressures. To investigate this, experiments were run using 6 mm, 10 mm and 20 mm balls, with loads adjusted to give a nominal contact pressure of 2 GPa.

Test Conditions

6 mm ball @ 2 GPa:	load = 28.4 N
10 mm ball @ 2 GPa:	load = 78.8 N
20 mm ball @ 2 GPa:	load = 315 N
Balls:	52100 bearing steel
Plate:	NSOH BO1 ground gauge plate (annealed)
Stroke:	25 mm
Frequency:	5 Hz
Temperature:	50°C
Lubricant:	PAO and PAO + 0.5% FM

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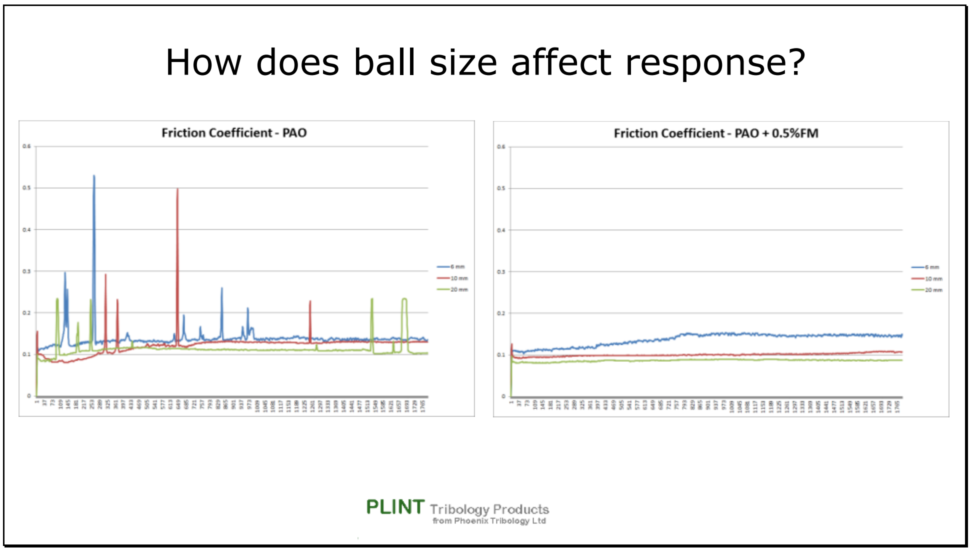
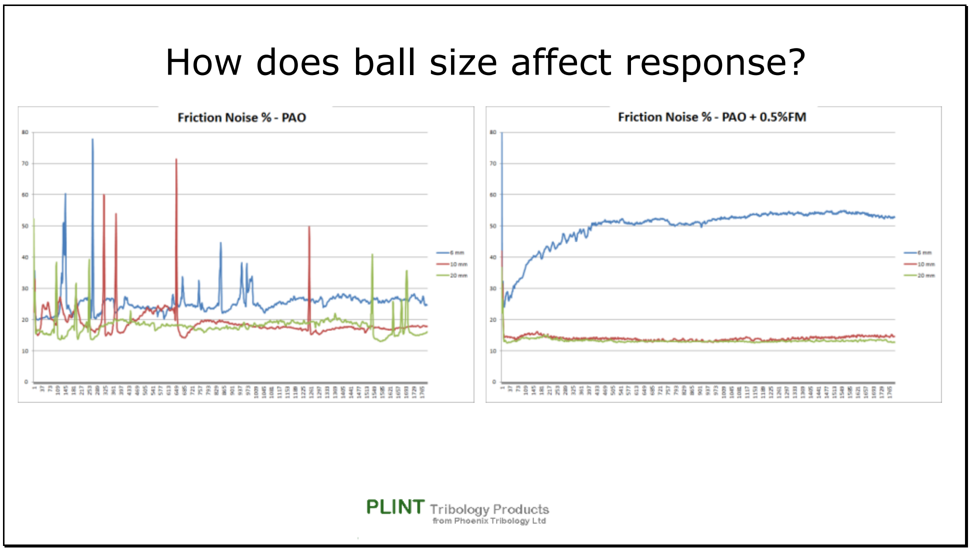


The friction coefficient traces for the tests with the base fluid produced significant friction spikes, indicating scuffing type events, for all ball sizes. This behaviour is reflected in the corresponding friction noise traces, but with the friction noise level decreasing with increasing ball size.

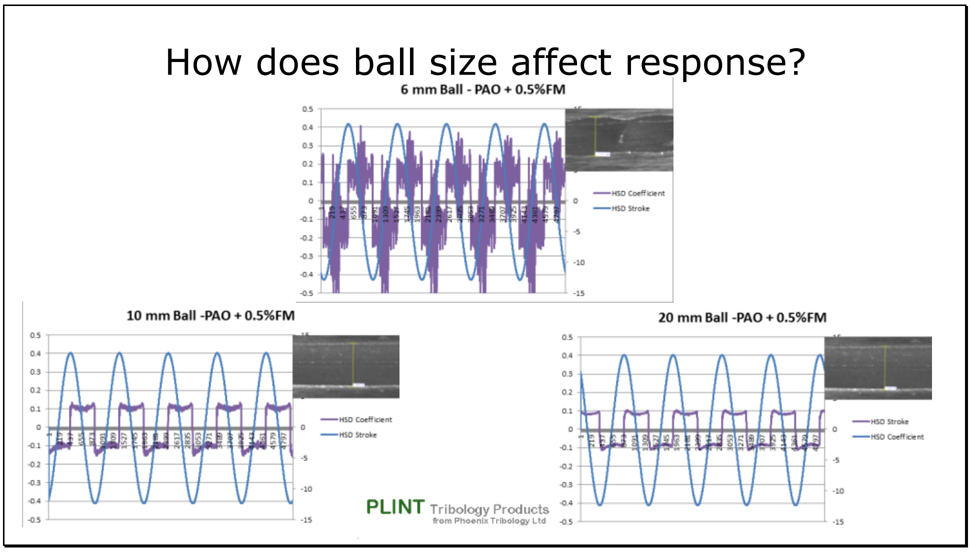
The introduction of OFM results in much steadier friction coefficients, with the 6 mm ball producing higher friction than the 10 mm and 20 mm balls, plus significantly higher friction noise.

Slide 35



Regardless of ball size, the tests run with base fluid only produced wear scars indicating junction growth and seizure, with near round wear scars on the balls.

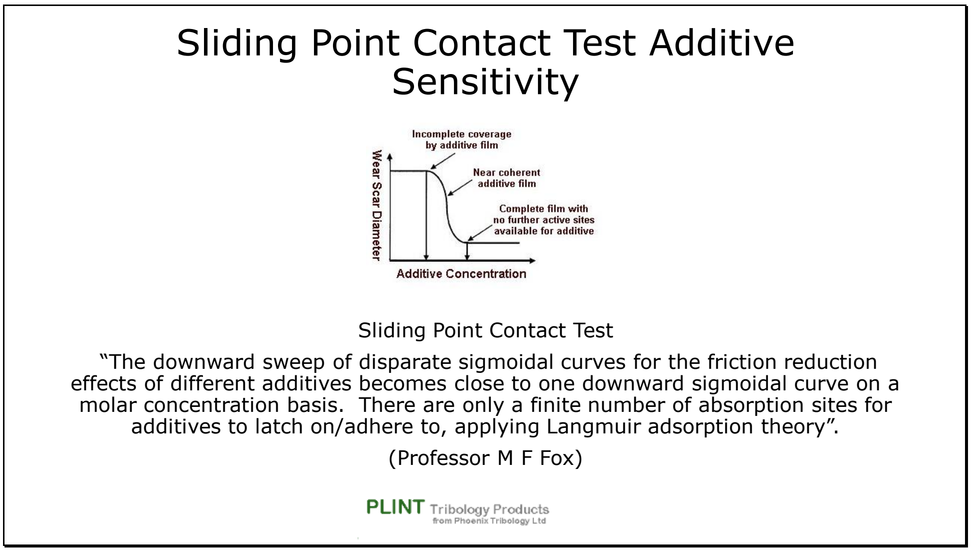
Slide 36



The tests with base fluid plus OFM produce a difference in behaviour between the 6 mm ball experiment and the 10 mm and 20 mm ball tests. In the former, plastic flow occurs on the plate specimen, resulting in a wavy edged wear scar. This results in unsteady instantaneous friction force around mid-stroke and a corresponding high value of friction noise. In all three cases, an elliptical ball scar is produced, consistent with severe adhesive wear.

It would appear that in an experiment with a hard ball running on a soft surface, the smaller the ball diameter, the greater the penetration depth and the greater the potential for plastic flow and work hardening.

Slide 37



Another feature of sliding hertzian point contact tests is that they are relatively insensitive to additive concentration. Once a complete, coherent, additive film has been formed, increased additive concentration has no further effect.

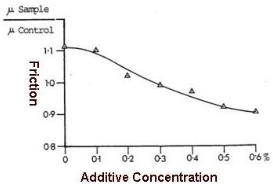
In the case of the ISO fuel lubricity test, although there is a marked difference in wear scar size for nominally high and low lubricity test samples, the test lacks the necessary sensitivity to distinguish between good candidate samples with differing amounts or types of lubricity enhancing additive.

Professor Malcolm Fox explains the issue in more scientific terms:

"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).

Slide 38

Sliding Line Contact Test Additive Sensitivity



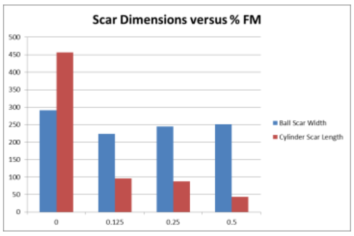
Greater sensitivity with line contact

PLINT Tribology Products
from Phoenix Tribology Ltd

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

Slide 39

Sliding Line Contact Test Additive Sensitivity



Greater sensitivity with line contact

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This is because much more of the surface is being sampled.

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Coatings

- Tribological coatings are mainly used to reduce adhesive wear and risk of seizure or cold-welding
- Test geometry of steel ball sliding on hard coated surface not sensible model of wear in practical applications because of the intensity of loading too high

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Tribological coatings are mainly used to reduce adhesive wear and the risk of seizure or cold-welding. An experiment in which steel balls are loaded in sliding contact against steel discs and against discs with various coatings demonstrates some obvious benefits.

The mutual solubility of the materials in the steel on steel contact results in poor tribological compatibility, hence seizure and galling, with correspondingly high friction. With a steel ball on a coated surface, the low friction levels indicate more or less complete suppression of adhesive friction and hence adhesive wear.

However, the test geometry of relatively soft steel ball sliding on hard coated surface is less successful as a model of wear in practical applications. This is because of the intensity of loading associated with a sliding hertzian point contact, which results in severe two-body abrasive wear of the steel ball. As abrasive wear is a much more efficient mechanism for removing material than adhesive wear, the associated work of friction is much lower, for the volume of material removed.

The following results are from lubricated tests.

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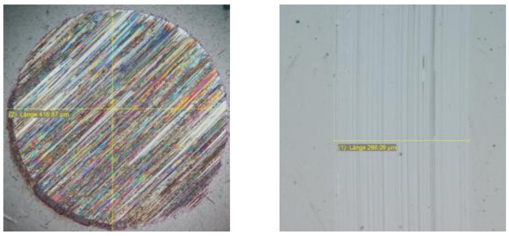
Coatings



Steel ball on DLC coating – lubricated
Mean Friction Coefficient: 0.07
Kind permission of Oerlikon Balzers

PLINT Tribology Products from Phoenix Tribology Ltd

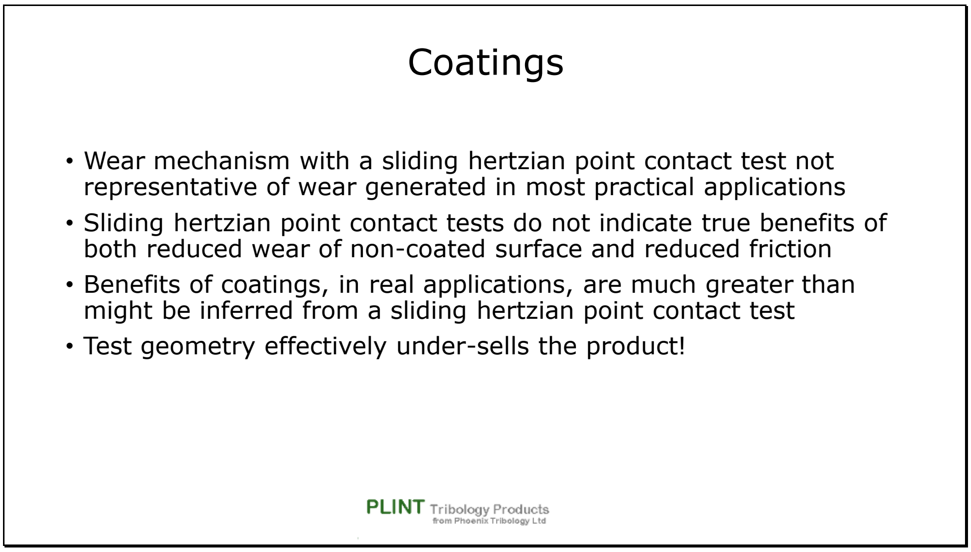
With two body abrasive wear, the volumetric loss of material from the softer surface can be considered to be proportional to the sliding distance and the intensity of loading. High intensity loading results in severe gouging of parallel grooves in the direction of sliding, with the wear substantially confined to the softer ball, with minimal wear of the coating.

Slide 42

By way of contrast, low intensity loading may result in nothing more than light scratching of the counter-face material and, in this example, satisfactory build-up of a carbon transfer film.

In addition to the significant difference in both wear mechanism and wear rate, the steady state friction coefficient with the area contact, in this example, was 0.0214, compared with a steady state friction coefficient of approximately 0.07, for the hertzian point contact test.

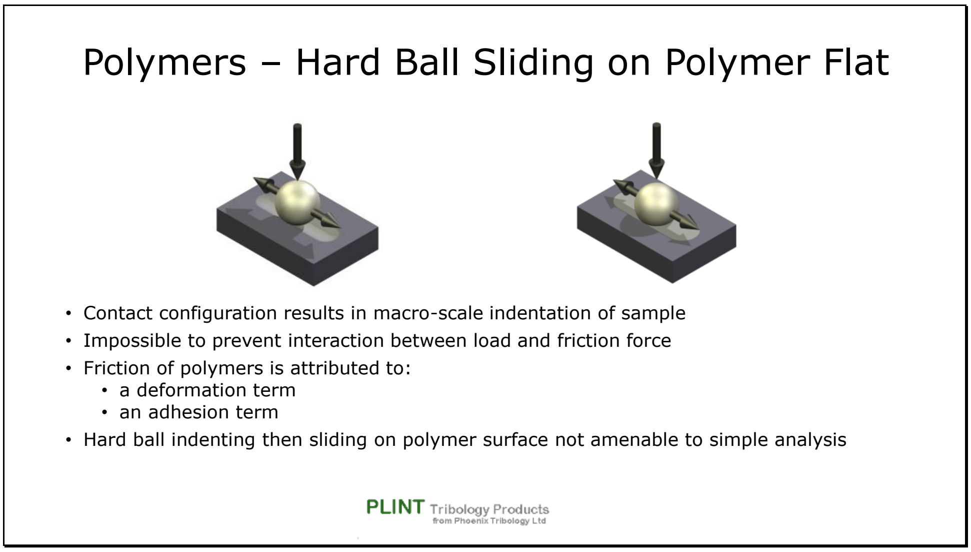
Slide 43



The wear mechanism with a sliding hertzian point contact test is not representative of the wear generated in most practical applications and although the results of steel ball on coated flat tests may indicate some of the frictional benefits of coatings, they do not indicate the true benefits of both reduced wear of the non-coated surface and reduced friction.

The benefits of the coatings, in real applications, are much greater than might be inferred from a sliding hertzian point contact test. This test geometry effectively under-sells the product! It follows that to demonstrate the benefits of a coating, in the majority of its intended applications, only tests involving low intensity loading are appropriate.

Slide 44



A practical issue with experiments involving a hard ball sliding against a polymer flat is that, if the contact configuration results in macro-scale indentation of the sample, it is impossible to prevent interaction between the load and friction force, hence achieve fully deterministic load application and friction force measurement. This is because the frictional contact is not flat, so the resulting friction vector is somewhat indeterminate with respect to the load application and measurement axes.

The friction of polymers is attributed to two sources, a deformation term and an adhesion term. The deformation term is meant to refer to the force associated with the sliding of a hard counter-face asperity over a polymer surface, hence an unambiguous tribological response. It is important not to confuse the use of the word deformation with the mechanical interaction produced as a result of indentation.

A hard ball indenting then sliding across a polymer surface is not amenable to simple analysis. To illustrate the key issues with this test geometry it is perhaps best to consider a simpler two-dimensional geometry such as a hard cylinder sliding against a soft flat. This model then becomes broadly similar to that applied to “draw bead” or “strip reduction” tests.

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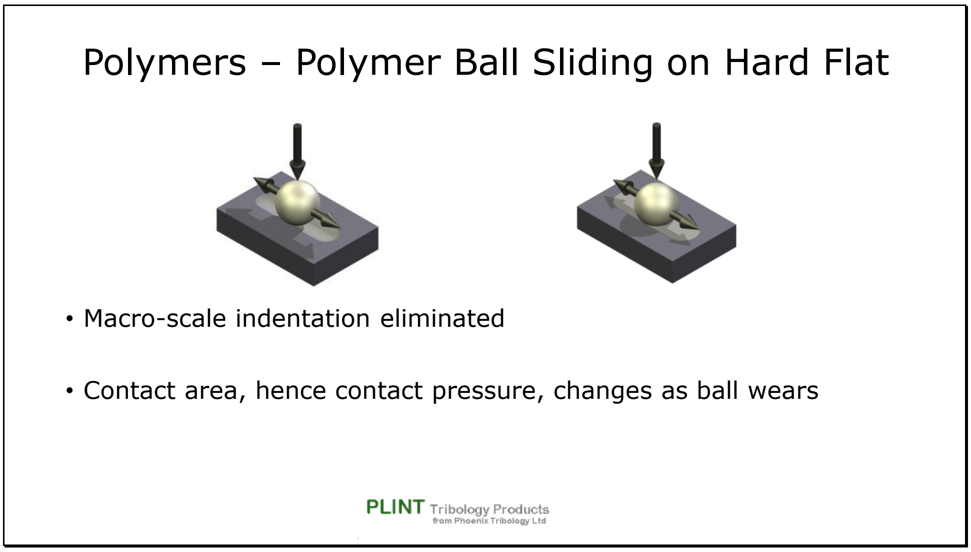
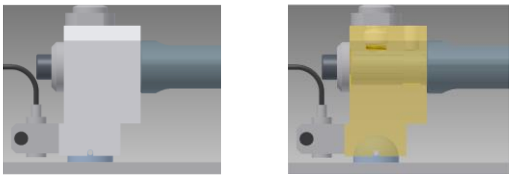


Of course, these problems disappear if we flip the specimens and run with soft ball on hard flat. For the most part, rather than trying to manufacture polymer balls, dome ended pins are the better option. Of course, we still have the issue of the area of contact, hence contact pressure, changing as the pin wears, but this is probably acceptable for a simple screening test. For more comprehensive research applications, an area contact is perhaps preferable.

Slide 46

Polymers – Area Contacts



Deformation component of friction definitively confined to that associated with the surface roughness of counter-face

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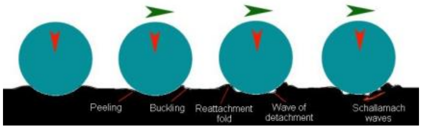
The issue with a flat ended pin is, of course, that with the frictional contact cantilevered at the end of a pin, potentially of limited stiffness, the contact may not wear flat. This problem can be overcome by making specimens in the form of a self-aligning sliding bearing.

With this type of geometry, the deformation component of friction is definitively confined to that associated with the surface roughness of the counter-face. A relatively rough surface may give rise to both abrasive wear and surface fatigue, whereas a smooth surface will promote adhesive wear and material transfer to the counter-face surface. Adhesive wear is a much less efficient mechanism for removing material from a surface, compared with abrasive wear, so the temperature with the former mechanism is likely to be a lot higher than with the latter.

Slide 47

Rubber Friction



Schallamach Waves

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Under dry conditions, with a rubber or elastomer in sliding contact with a smooth, rigid, counter-face, friction coefficients can be very high, 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.

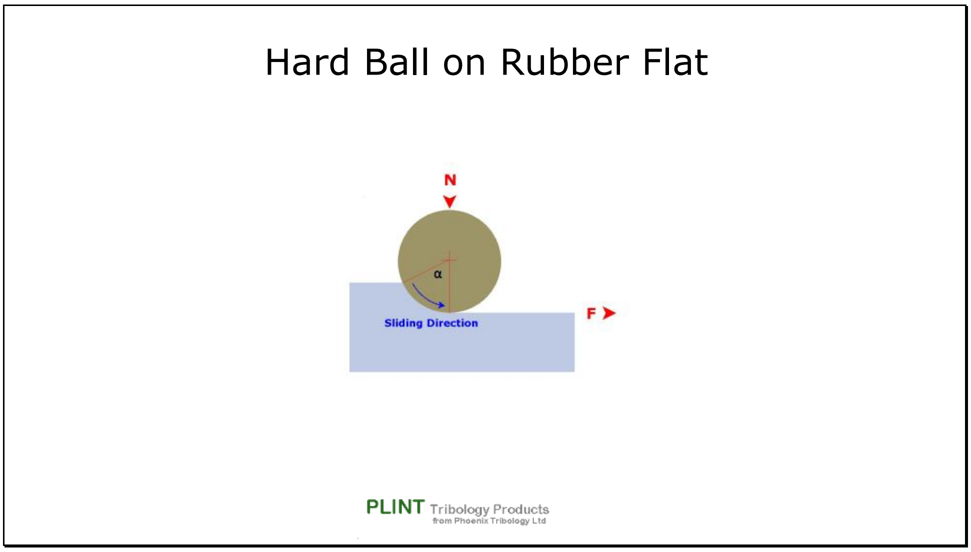
At the trailing edge, there is a requirement to peel apart the contact. Local recovery and slip can give rise to re-attachment of the rubber at the outermost edge of the contact. The process is cyclic and gives rise to variations in friction.

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.

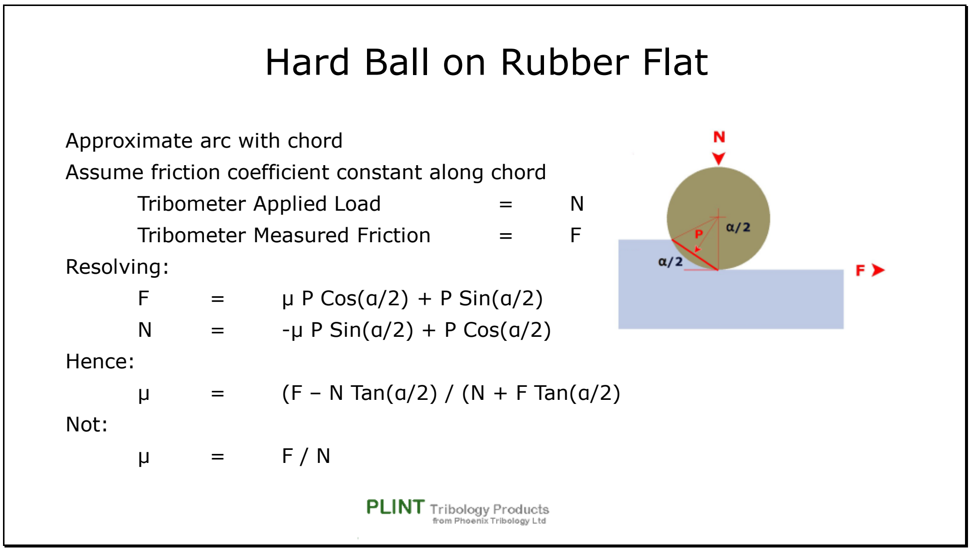
These factors in turn depend on other parameters, for example, the time dependent behaviour for the rubber or, in the case of liquid lubricated contacts, lubricant entrainment conditions and squeeze film effects.

Slide 48



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

Slide 49



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:

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

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

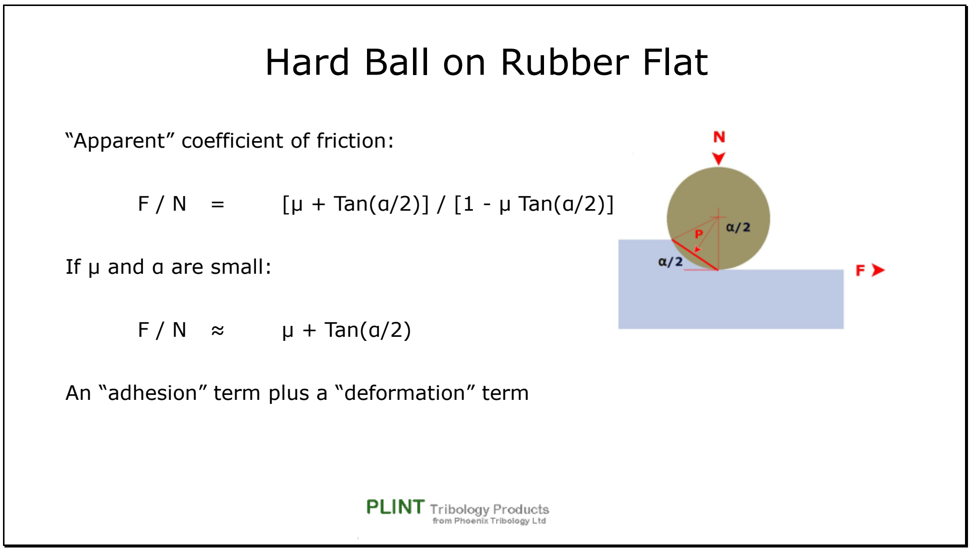
Hence:

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

Not:

$$\mu = F / N$$

Slide 50



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.

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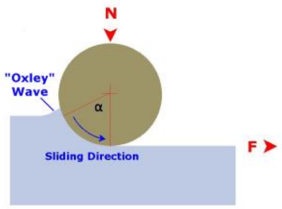
Hard Ball on Rubber Flat



Cannot be steady state solution:

Ball must move up, to bring the contact point to original surface level, 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)

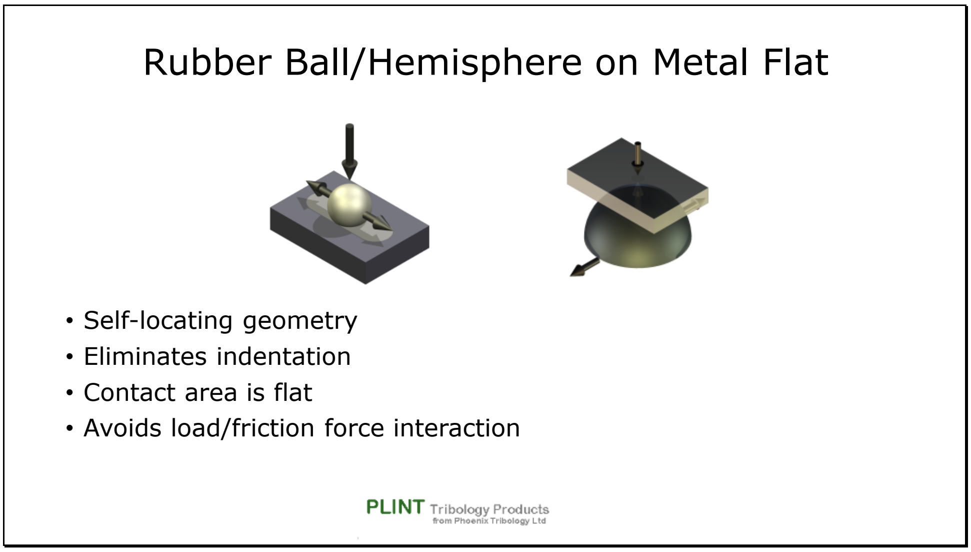
PLINT Tribology Products from Phoenix Tribology Ltd

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)

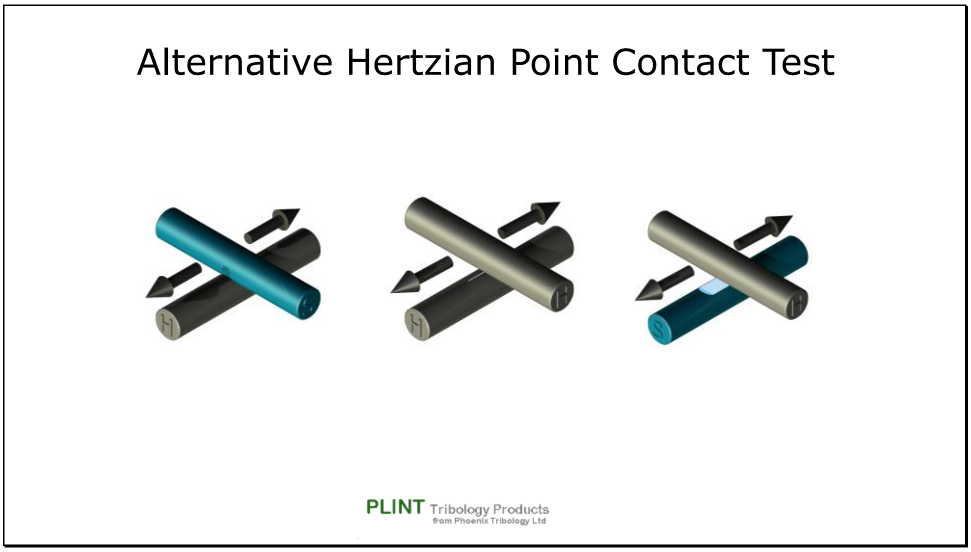
Switching the material pairs around to give rubber ball or hemisphere on hard flat, in other words a self-locating geometry, eliminates indentation thus avoiding load/friction force interaction; the contact area is flat.

Slide 52



Depending on the rubber under test, uni-directional sliding can result in high wear rates and pattern abrasion, with macro-scale ridges forming in the surface. Testing the same material in reciprocating sliding may suppress the formation of pattern abrasion and generate lower wear rates, with a much finer scale surface roughness produced. This is intrinsic abrasion, as defined by Schallamach.

Slide 53

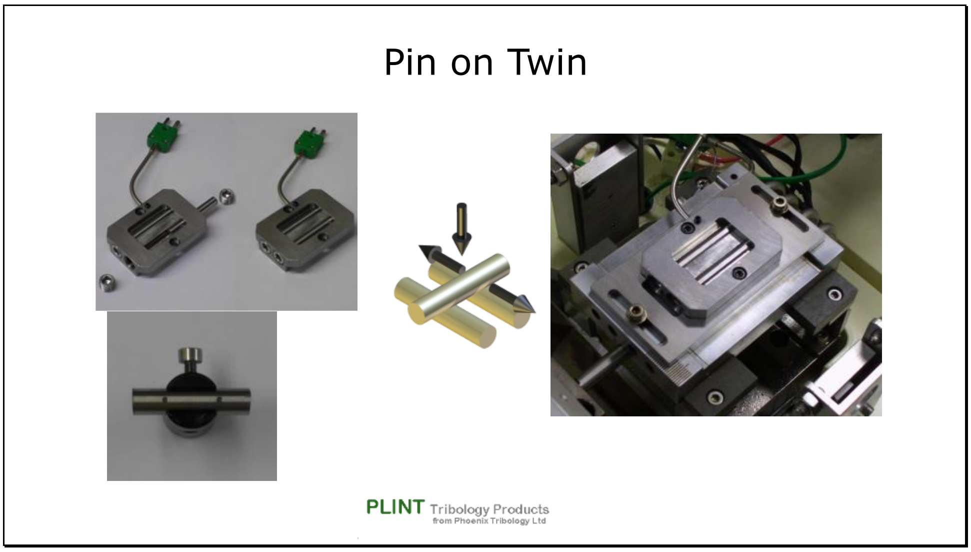


I think we can safely say that we have arrived at the point where we can summarise the problems of tribological experiments involving a ball sliding on a flat surface as follows:

- They provide poor models for real contacts
- It is very difficult to analyse and understand the wear processes in the contact
- For lubricated tests, they are relatively insensitive to variations in additive concentration
- Measurement of ball wear scars is fraught with difficulty and uncertainty

A ball sliding on a flat surface is, however, not the only way to produce a sliding hertzian point contact, if that is what you want. A crossed cylinder arrangement with equal diameter cylinders produces the same contact as a ball of the same diameter on a flat. For various reasons, the geometry seems to produce much more accessible and understandable results.

Slide 54

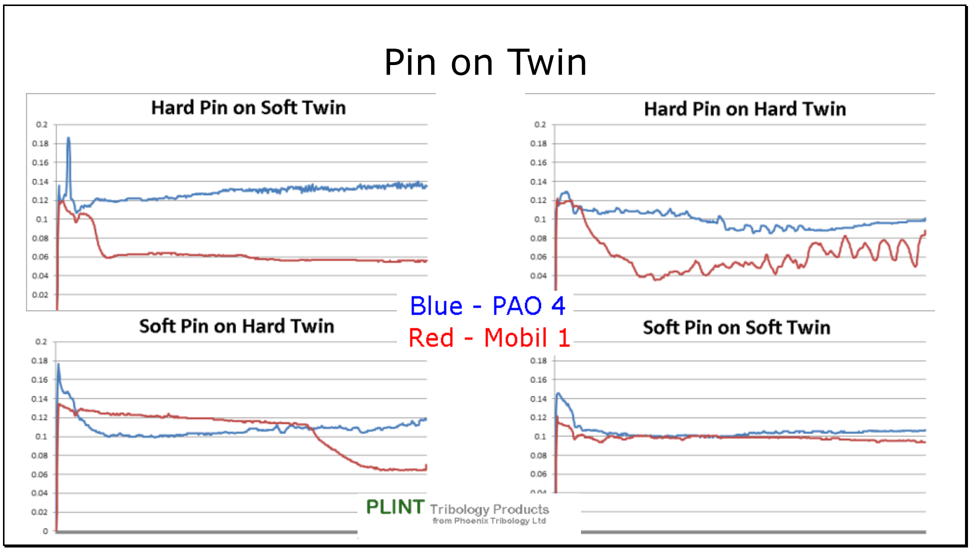


For practical purposes, reciprocating crossed cylinder experiments are best performed using the pin on twin geometry, first used by Truhan, Qu and Blau, at Oakridge National Laboratories.

> The development of a "pin on twin" scuffing test to evaluate materials for heavy-duty diesel fuel injectors
>
> J J Truhan, J Qu, P J Blau
>
> Tribology Transactions Volume 50 Number 1 January - March 2007

The pin on twin geometry, although starting with high hertzian contact pressure, is self-locating and, depending on relative specimen hardness, can produce crisp wear scars, which are much easier to measure than the typical ball on flat type scars. The use of simple cylindrical samples has a significant impact on specimen costs. The geometry has the added advantage of generating two wear scars per test, thus providing two wear data points.

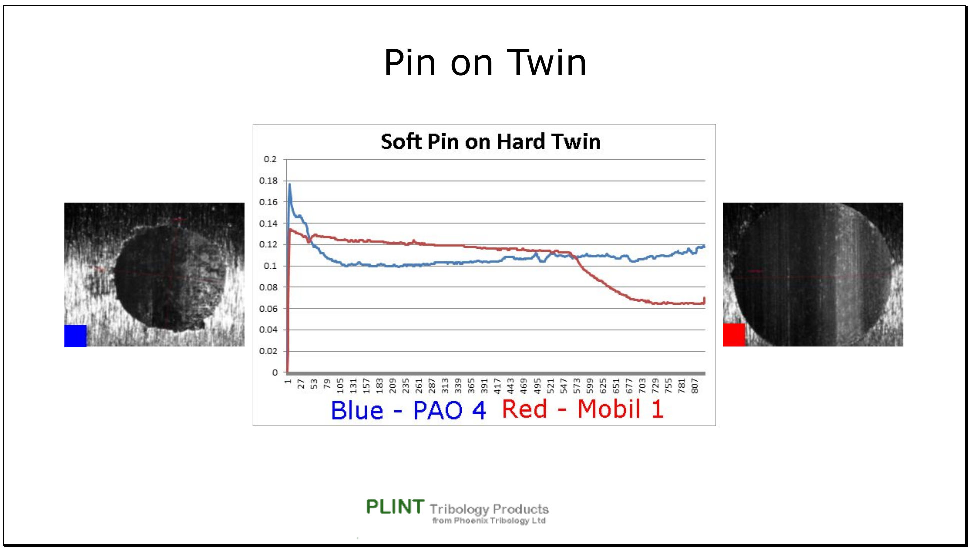
Slide 55



In this series of tests, 6 mm diameter rod samples were used, with a load of 56 N, hence 26 N per contact, so producing the same contact pressure as the previous 6 mm ball on flat experiments. PAO 4 and Mobil 1 were used as fluid samples.

It is clear that different hardness combinations produce very different frictional responses.

Slide 56

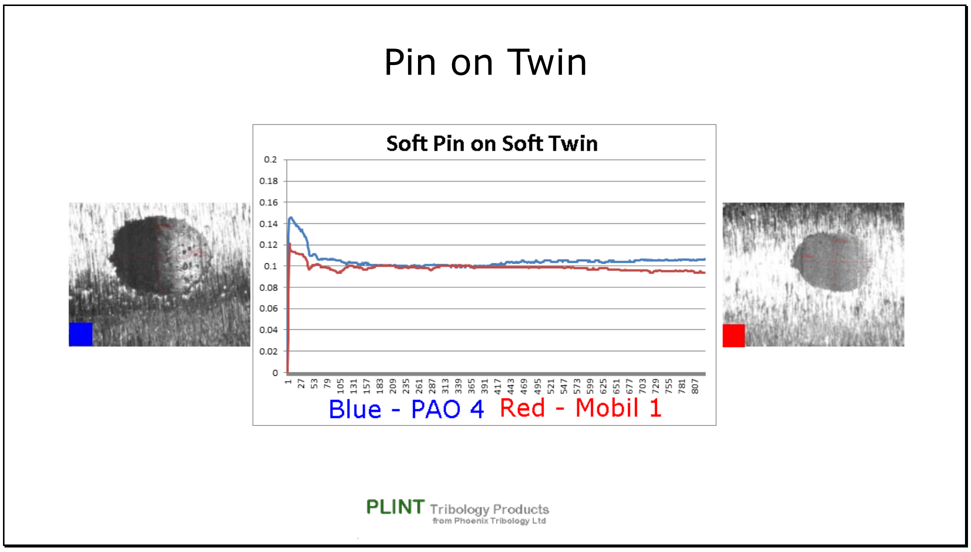


With a soft pin on a hard twin, despite providing very clear wear scars on the soft pin, the wear with the fully formulated oil is much higher than for the base fluid, however, the friction traces are very different, perhaps indicating very different running-in and wear processes.

The base oil wear scar shows signs of adhesive wear and the friction trace indicates initial severe adhesive wear with a transition to mild adhesive wear.

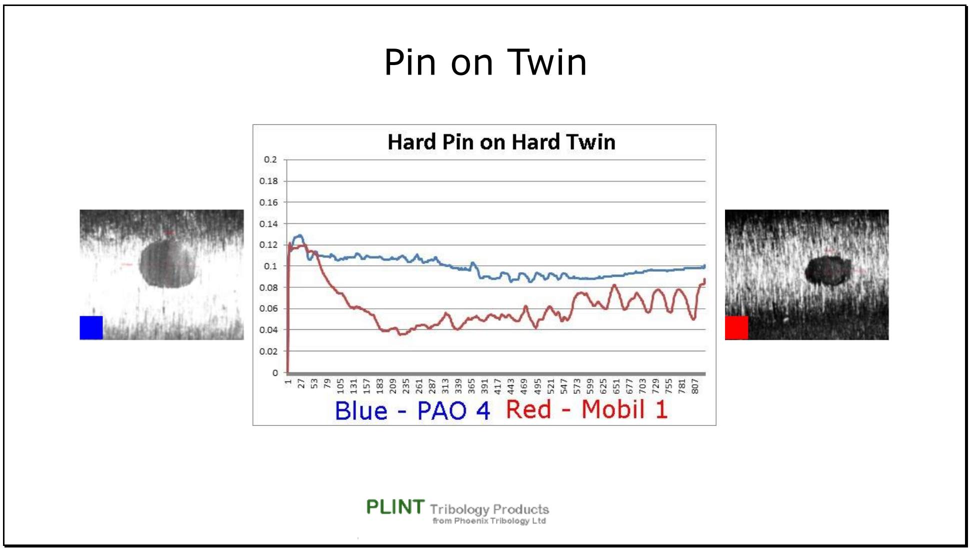
By contrast, with the fully formulated oil, adhesive wear is suppressed, leaving abrasive wear, which efficiently removes material from the surface, producing a smooth, conforming wear scar. The abrasive wear mechanism is not affected by the lubricant additives. At some point, the contact becomes large enough and smooth enough to be supported by the lubricant film and the abrasive wear process ceases and the friction falls.

Slide 57



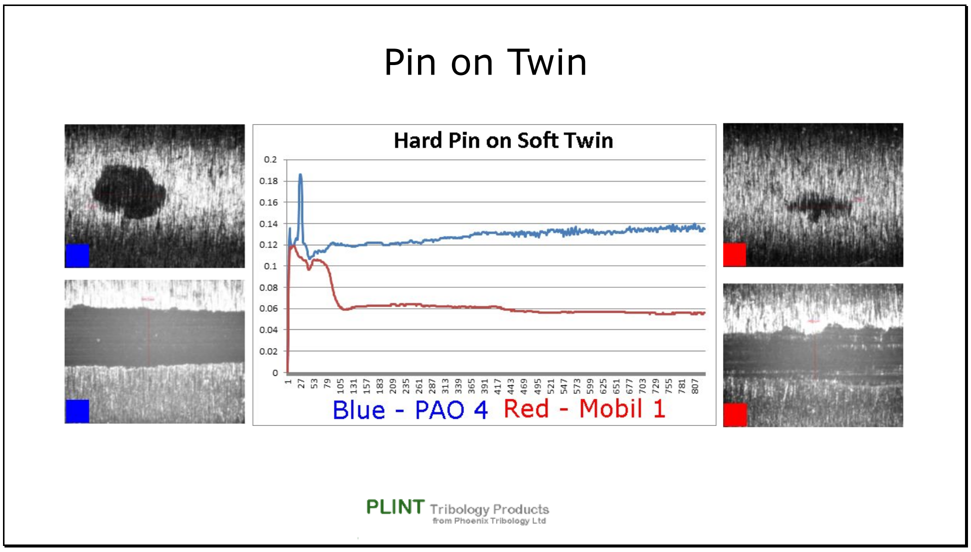
With a soft pin on soft twin, although the base fluid produces more wear than the fully formulated oil, there is little difference in friction, except during the initial running-in phase. In both cases, we have an adhesive wear process, with the additives in the fully formulated oil having an obvious beneficial effect on wear.

Slide 58



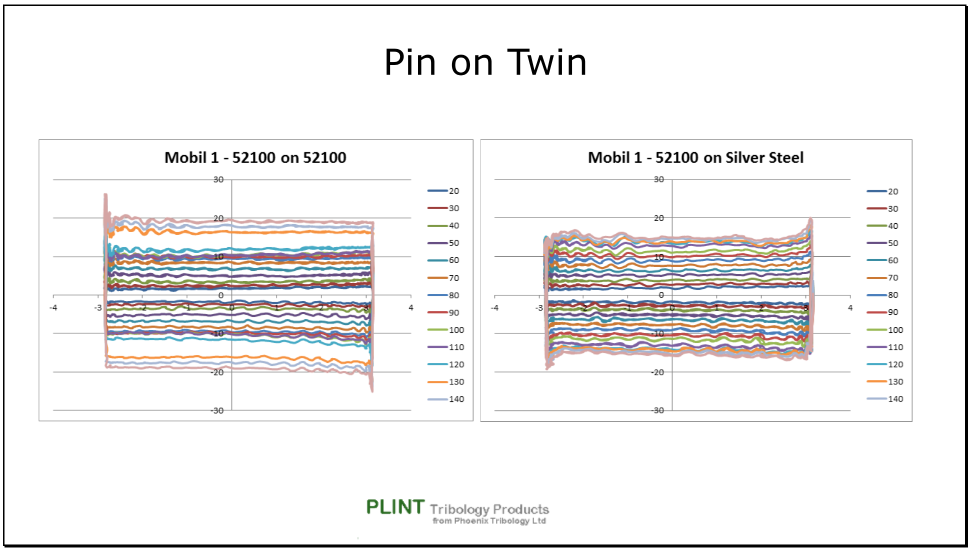
With hard pin on hard twin, the base fluid produces more wear than the fully formulated oil and there is a significant difference in friction. It is not immediately clear what combination of abrasive and adhesive wear is occurring, but it is likely that the tests start with a predominantly abrasive wear process and then transition to an adhesive wear process, with the fully formulated oil additives limiting the effects of adhesive friction.

Slide 59



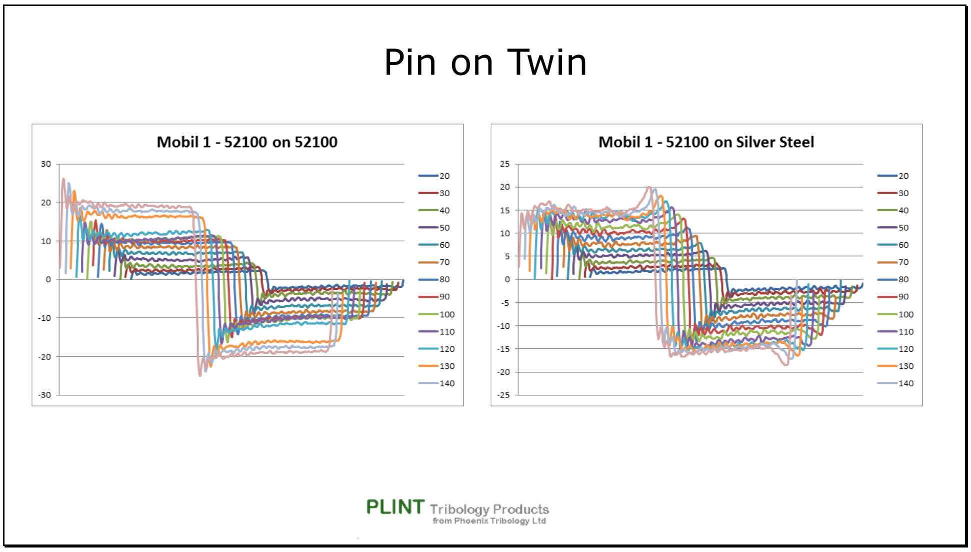
Tests with hard pin on soft twin result in different frictional response with the fully formulated oil producing much lower friction than the base fluid. The friction trace for the fully formulated oil, after initial running in, is both lower and smoother than with the base fluid. With both samples, the wear scars on the softer twin samples are readily identifiable with the fully formulated oil producing a much smaller wear scar, compared with the base fluid. The wear scars on the pin sample are not readily measurable, with the base fluid scars showing witness marks corresponding to stroke end contact and the fully formulated lubricant producing witness marks indicative of a line contact.

Slide 60



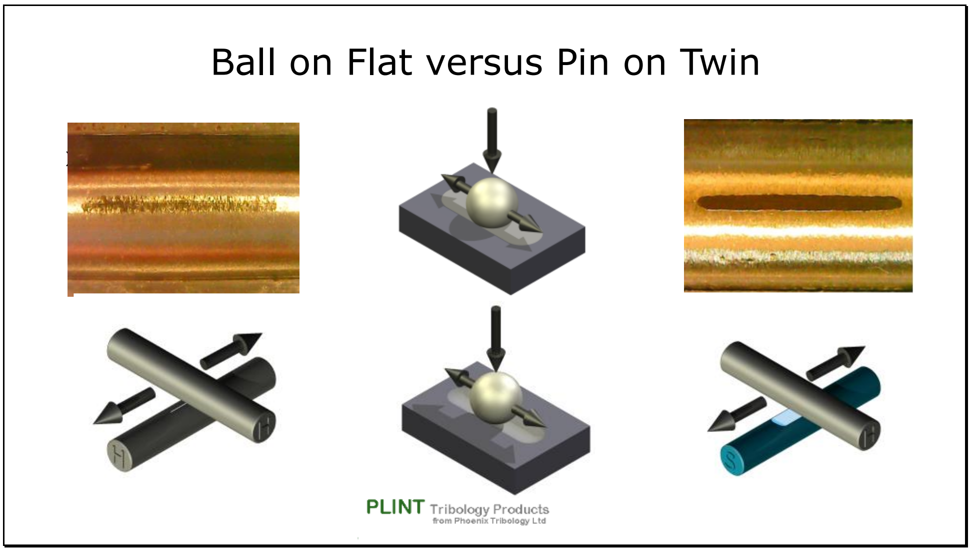
As well as the wear and the time smoothed friction, there may be things to learn from the instantaneous friction traces. This slide shows the distinctive differences in shape of the friction loops for hard on hard and hard on soft.

Slide 61



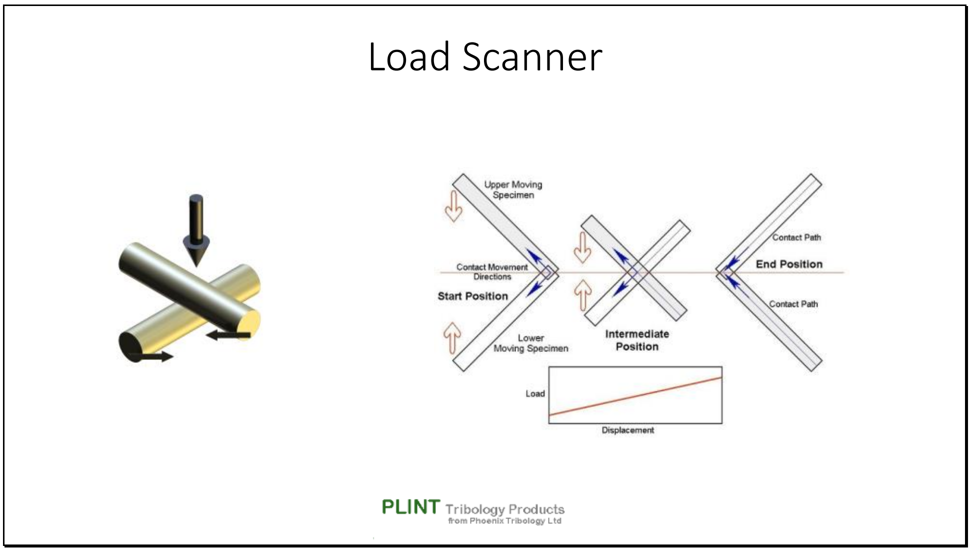
The hard on hard specimens produce a well-defined spike and essentially square subsequent trace, indicating sliding over a flat, unworn, surface. The hard on soft specimens do not start with a spike, but end with a rise in friction. This is as a result of the specimen wear at the ends of the stroke; the moving specimen starts the stroke by sliding down a slope and ends by sliding up a slope.

Slide 62



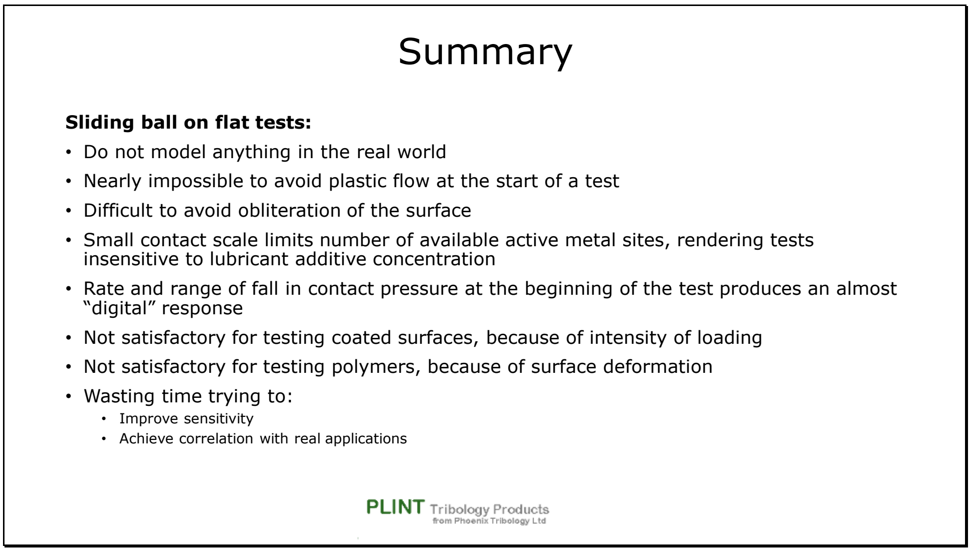
Unlike the stroke end for the hard ball on soft flat, which is a three-dimensional “pocket”, with the crossed cylinder hard pin on soft pin, we have a two-dimensional slope.

Slide 63



The “Load Scanner” geometry, invented by the Uppsala University, achieves an equivalent sliding hertzian point contact geometry to the pin on twin, but avoids many of the issues with highly localised wear or deformation, by having the point of contact move on both surfaces.

Slide 64



Conclusions:

In this paper I have attempted to explain what it is that is produced in various different sliding point contact tests. Who could have imagined how complicated it all is?

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 and frequently with not much idea of the wear mechanism or mechanisms involved. When it comes to friction, it should be clear that there is more to measuring friction and analysing frictional response, than simply recording some unanalysed, time smoothed, force signal.

Whether such measurements are meaningful 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:

With sliding point contact tests:

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 sliding ball on flat tests “Sliding Brinell Hardness Tests”!

The crossed cylinder pin on twin geometry may provide a more accessible and understandable geometry for those determined to run sliding hertzian point contact tests.

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