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Lubricated Friction Testing
Sliding and Sliding/Rolling Contacts

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Tribology Trust Silver Medal 2017

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Introduction

- Many lubricated bench tests focus on boundary regime, because it is usually the wear, or failure, that arises under this regime, which limits life of components
- Well-designed systems should not normally operate under continuous boundary conditions, unless something has gone wrong
- Pure sliding conformal contacts can operate under conditions of boundary, mixed or hydrodynamic lubrication, depending on entrainment velocity
- Pure sliding non-conformal contacts usually operate under conditions of boundary or mixed lubrication, but not hydrodynamic lubrication, with frictional response substantially affected by morphology of contact
- Sliding/rolling contacts, which are by definition, non-conformal, operate under conditions of boundary, mixed or elasto-hydrodynamic lubrication

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It is perhaps not surprising that many lubricated bench tests tend to focus on the boundary regime, because it is usually the wear, or failure, that arises under this regime, which limits the life of components.

However, a well-designed system should not normally operate under continuous boundary conditions, unless something has gone wrong, for example, with the choice of lubricant.

Pure sliding conformal contacts can operate under conditions of boundary, mixed or hydrodynamic lubrication, depending on entrainment velocity.

Pure sliding non-conformal contacts usually operate under conditions of boundary or mixed lubrication, but not hydrodynamic lubrication, with frictional response substantially affected by the morphology of the contact.

Sliding/rolling contacts, which are by definition, non-conformal, operate under conditions of boundary, mixed or elasto-hydrodynamic lubrication.

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Introduction

Bench tests:

Frequently run continuously under boundary lubrication

Real systems:

Transition through boundary regime, during start-up or for a small part of operating cycle

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Whereas a bench test may run continuously under boundary lubrication, real systems will only transition through the boundary regime, during start-up or for a small part of the operating cycle. In the ring-liner contact, for example, although the contact may be running under boundary lubrication at top and bottom dead-center, a more significant part of the cycle may be operating under mixed lubrication conditions.

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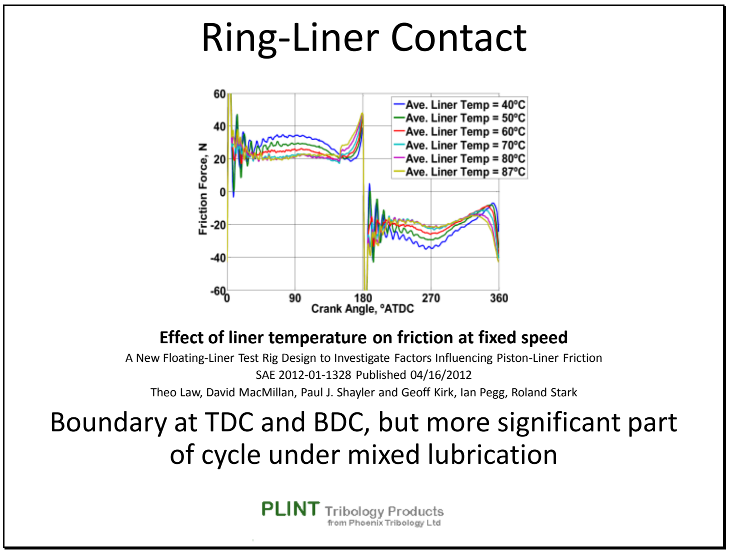

In the ring-liner contact, for example, although the contact may be running under boundary lubrication at top and bottom dead-centre, a more significant part of the cycle may be operating under mixed lubrication conditions.

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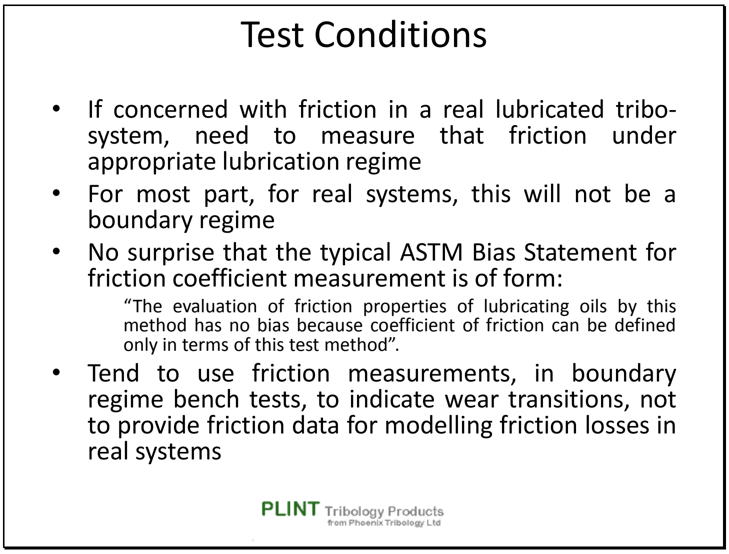

It follows, that if we are concerned with the friction in a real lubricated tribo-system, we need to measure that friction under an appropriate lubrication regime. For the most part, for real systems, this will not be in the boundary regime.

It is perhaps no surprise that the typical ASTM Bias Statement for friction coefficient measurement is of the form:

> "The evaluation of friction properties of lubricating oils +by this method has no bias because coefficient of friction can be defined only in terms of this test method".

In practice, we tend to use friction measurements, in boundary regime bench tests, to indicate wear transitions, not to provide us with friction data that can be used to model the friction losses in real systems.

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To quote one of our automotive clients:

> “Our Tier 1 suppliers provide us with lots of friction data, but it is all completely useless, as we can’t get it to correlate with anything we observe in the engine”.

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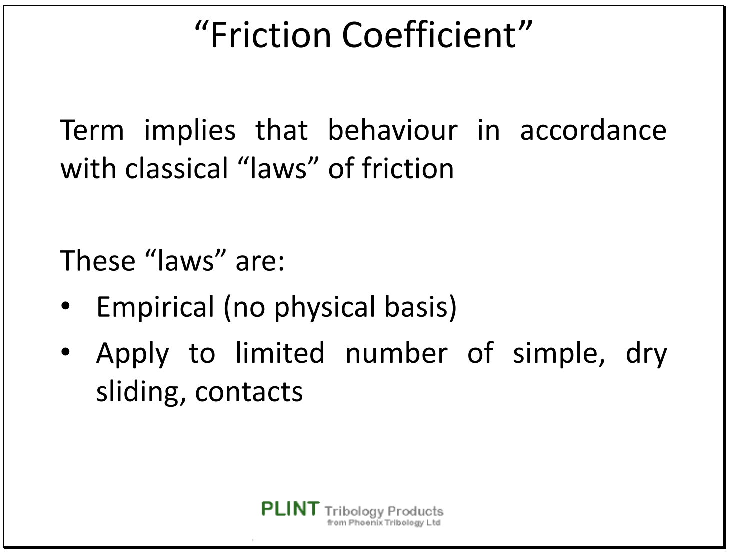

The term “friction coefficient” implies that the contact in question behaves in accordance with the Amontons-Coulomb laws of friction. It is essential to note that these empirical “laws” only apply to a limited number of simple, dry sliding, contacts; reference to any Stribeck curve should be sufficient to demonstrate that they clearly do not apply to lubricated contacts.

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Stribeck Curve

Provides useful platform for identifying operating ranges of many common systems

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Lubricated tribology is sensibly only really relevant to real systems and it is apparent that such systems operate under a variety of different lubrication regimes. The Stribeck curve provides a useful platform for identifying the operating ranges of many common systems.

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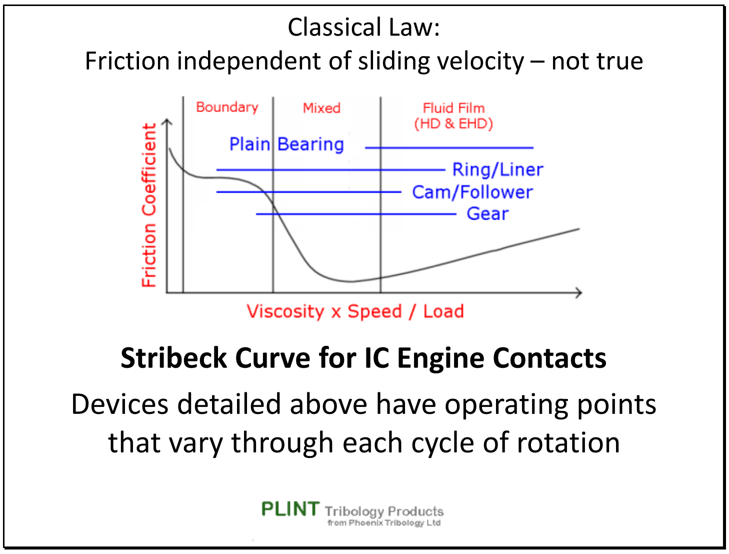

In a lubricated contact, friction is not independent of sliding velocity.

It is of course important to note that the devices detailed above, except in the case of non-IC engine plain bearings, have operating points that vary through each cycle of rotation.

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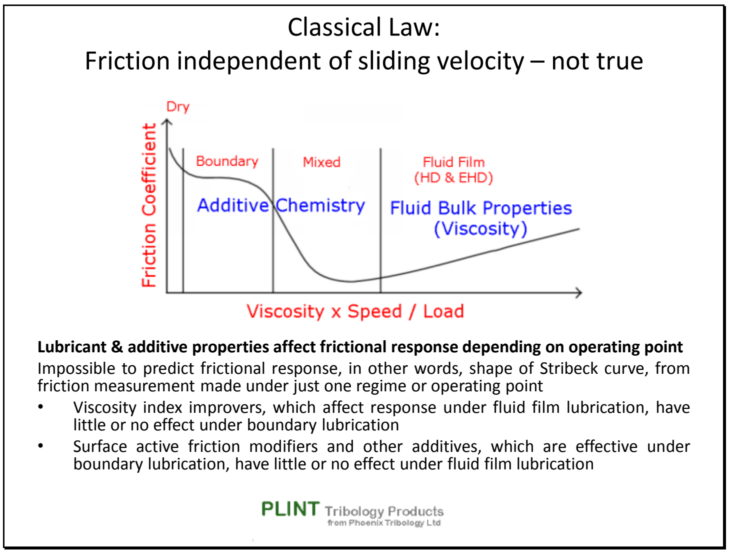

Lubricant and additive properties affect frictional response, depending where we are on the Stribeck curve.

It is important to note that it is more or less impossible to predict the frictional response, in other words, the shape of the Stribeck curve, from a friction measurement made under just one regime or operating point. This is because different properties are at play, for example: viscosity index improvers, which affect response under fluid film lubrication, have little or no effect under boundary lubrication and, similarly, surface active friction modifiers and other additives, which are effective under boundary lubrication, have little or no effect under fluid film lubrication.

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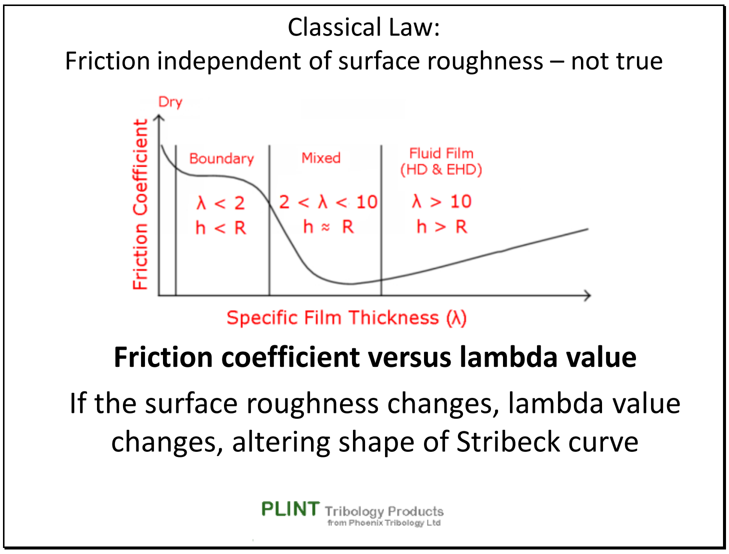

Friction is not independent of surface roughness.

Plotting friction coefficient against specific film thickness, in other words the ratio of film thickness to R_a or, more typically, R_q, provides a useful alternative form of the Stribeck curve. If the surface roughness changes, the lambda value changes; this alters the shape of the Stribeck curve.

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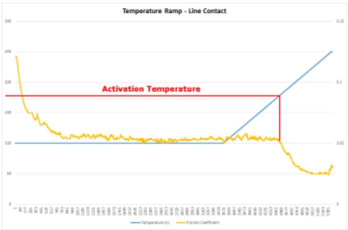
Classical Law:

Friction is independent of temperature – not true



Effect of temperature on additive activation

- Boundary and mixed: frictional response affected by lubricant additives, additive activation, adsorption and desorption processes, which are driven by contact temperature
- Hydrodynamic regimes: temperature of bulk fluid entering contact determines viscosity, hence frictional response

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In a lubricated contact, friction is not independent of contact temperature.

In the boundary and mixed regimes, where frictional response is affected by lubricant additives, additive activation, adsorption and desorption processes, are driven by contact temperature.

In the hydrodynamic regimes, temperature of the bulk fluid entering the contact determines the viscosity.

Changing the temperature, whether contact temperature or bulk fluid temperature, alters the shape of the Stribeck curve. In effect, one could plot a series of isothermal Stribeck curves, if one were so inclined.

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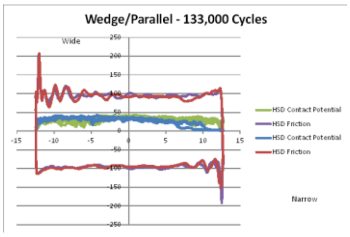
Classical Law:
Friction independent of apparent area of contact – true (?)



Experiments with fixed and variable area specimens

Green/Purple: Parallel – Blue/Red: Wedge

- More or less true for contacts operating under boundary lubrication
- Not true for contacts operating under mixed or hydrodynamic lubrication

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The law that states that friction is independent of apparent area of contact is more or less true for contacts operating under boundary lubrication.

It is not true for contacts operating under mixed or hydrodynamic lubrication.

In this example, a line contact specimen is run against a parallel sided specimen and against a tapered specimen under steady load. With the latter, the contact pressure is double at the narrow end of the stroke, compared with the wide end. The doubling of contact pressure does not appear to affect the frictional response.

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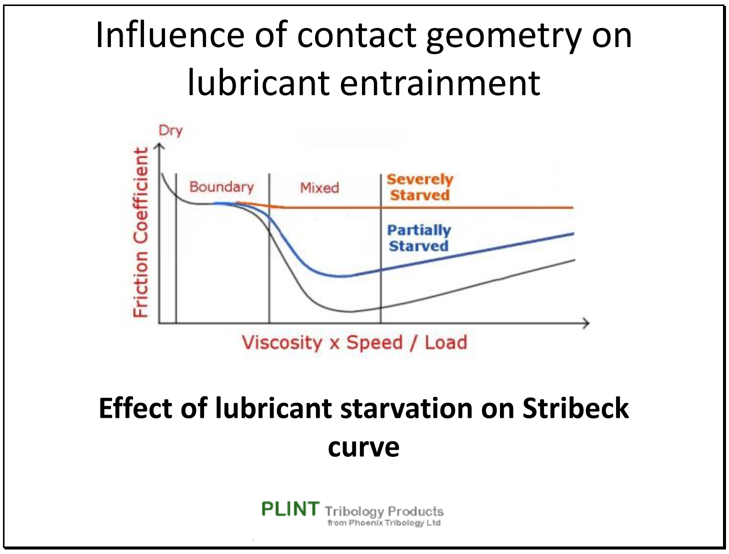

Elasto-hydrodynamic and hydrodynamic lubrication, in addition to requiring the necessary lubricant entrainment velocity, also requires an adequate supply of lubricant and a mechanism for the lubricant to be delivered into the contact. Without these, the contact may end up running under starved lubrication. In the limit, without an adequate supply of lubricant, the contact ends up running, regardless of speed, under boundary lubrication. This is one mechanism that can obviously lead to scuffing, as the speed is increased.

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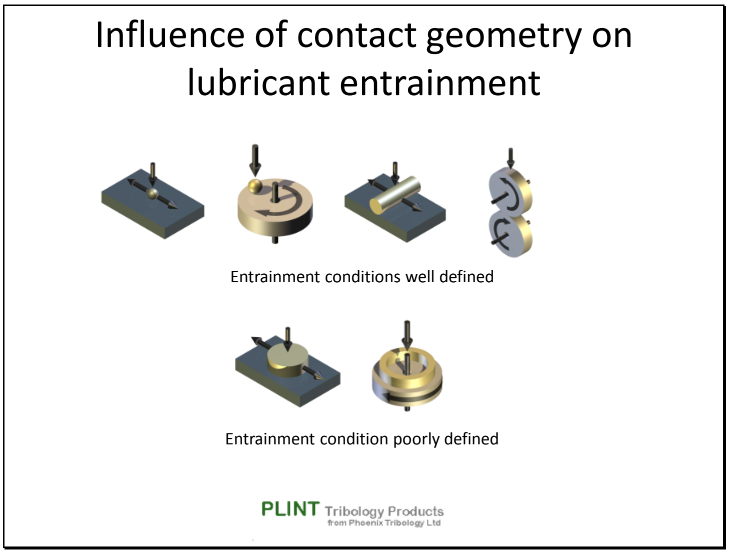

The different contact configurations for sliding or sliding/rolling tests affect lubricant entrainment in different ways.

With point contacts, entrainment conditions are well defined but the contact area and shape may change significantly during test, altering entrainment geometry.

Similarly, with line contacts, entrainment conditions are well define, however, with area contacts, entrainment condition are very poorly defined – how does lubricant get into contact?

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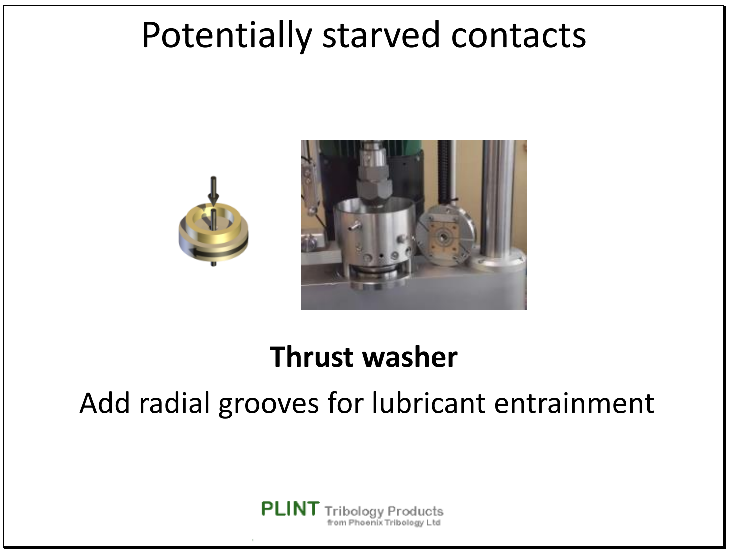

The thrust washer geometry is effectively a sliding face seal, in other words, a system designed to prevent lubricant from getting from one side of the contact to the other. For perfectly flat surfaces, lubricant entrainment is impossible.

To facilitate lubricant entrainment, radial grooves must be machined in one surface, as in various designs of plain thrust bearing and in the JASO Suzuki test geometry.

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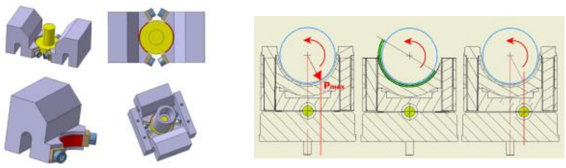
Potentially starved contacts



Partial journal bearing

Only works for tests requiring starved lubrication

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A partial journal bearing test geometry looks promising for lubricated friction tests, however, it only works satisfactorily for tests requiring starved lubrication.

In a journal bearing, the point of peak pressure is not on the centre line. With a half journal bearing contact configuration (which includes conforming block on ring), this results in the inlet being closed, preventing lubricant entering the bearing contact. Designers of partial journal bearings address this problem by designing bearings with the required “pre-load” and “off-set”.

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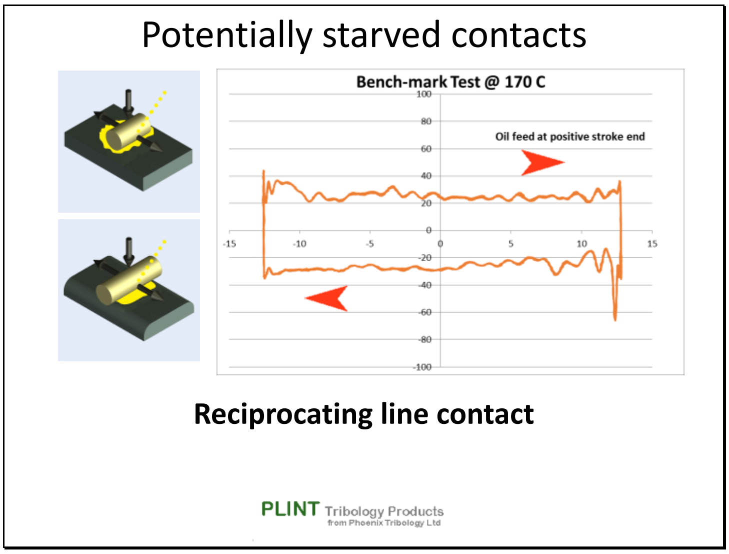

It is possible to produce an asymmetrical contact in a reciprocating test, by not allowing the lubricant to spread to both sides of a line contact. This produces starved lubrication in the sliding direction away from the lubricant feed and fully flooded lubrication in the opposite direction, hence different regimes, depending on the direction of sliding, hence a direction dependent frictional response.

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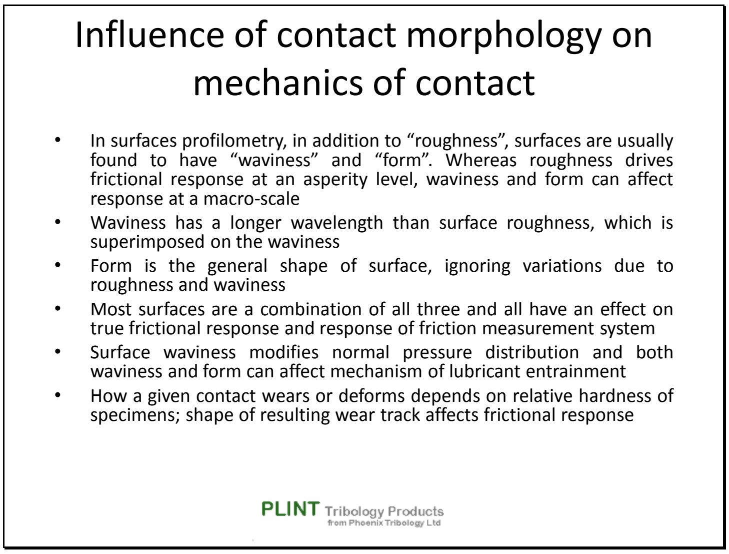

Now let’s consider the influence of contact morphology on the mechanics of the contact.

Greenwood and Williamson provides the basis for the modern understanding of the roll of surface roughness, hence asperity interaction, in frictional response.

The term nominally flat in the title refers to the fact that the reference plane in their model is flat, but with superimposed surface roughness.

In surfaces profilometry, in addition to “roughness”, surfaces are usually found to have “waviness” and “form”. Whereas roughness drives the frictional response at an asperity level, waviness and form can affect the response at a macro-scale.

Waviness has a longer wavelength than surface roughness, which is superimposed on the waviness.

Form is the general shape of the surface, ignoring variations due to roughness and waviness.

Most surfaces are a combination of all three and all have an effect on the true frictional response and the response of the friction measurement system.

Surface waviness modifies the normal pressure distribution and both waviness and form can affect the mechanism of lubricant entrainment.

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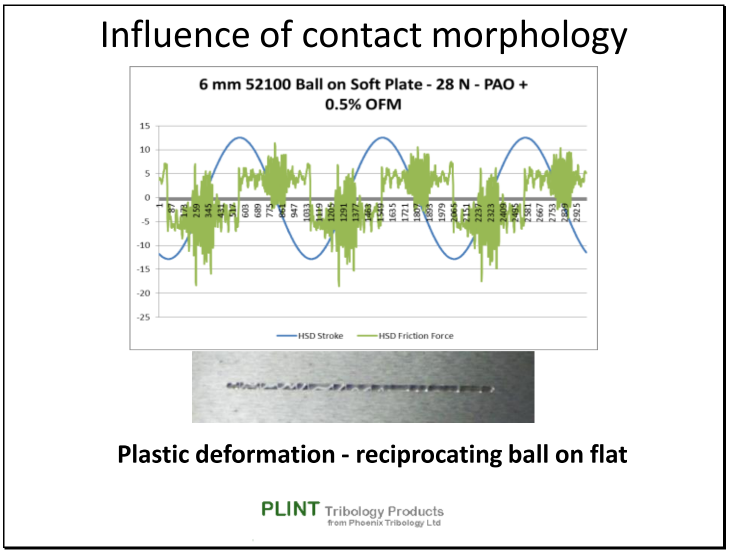

How a given contact wears or deforms depends on the relative hardness of the specimens. The shape of the resulting wear track affects the frictional response.

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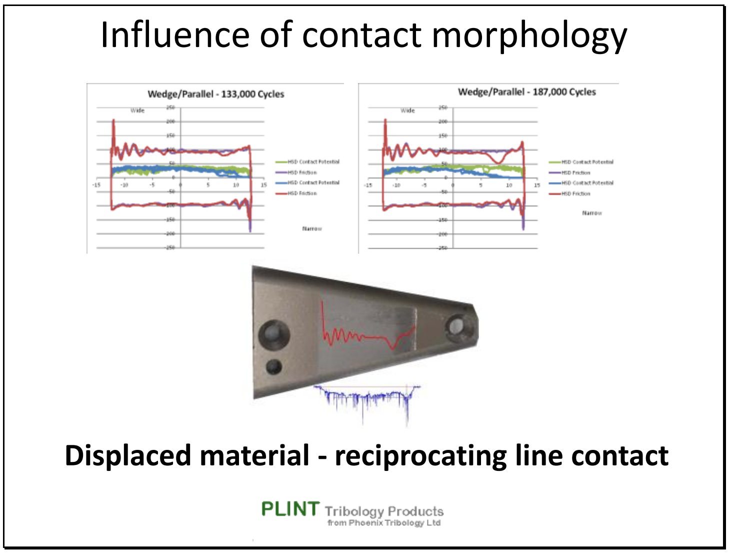

The dip in friction trace towards the stroke end, with the wedge specimen, is associated with a change in surface topography.

It is worth noting that in terms of frictional response, humps have a bigger effect than dips!

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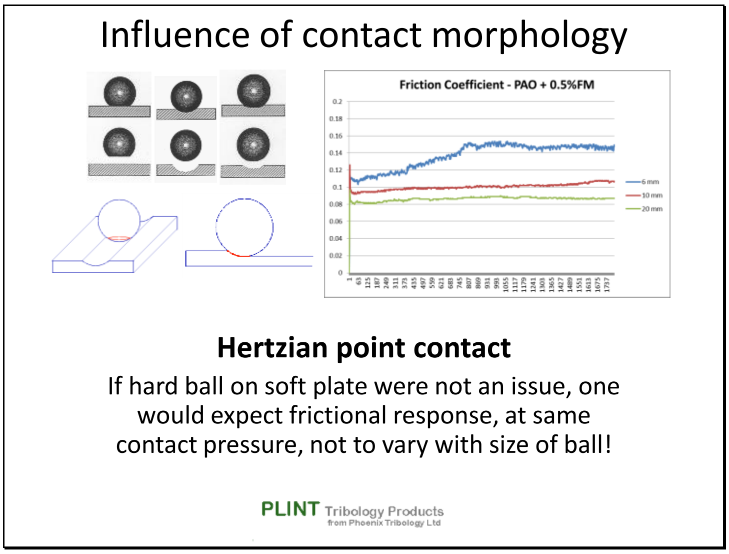

With a soft ball running on a hard surface, the contact patch remains flat and round; this is readily amenable to analysis of the forces acting on the contact. This is not the case with a hard ball sliding in a conforming wear groove on a softer surface.

In this case, we would expect an elliptical contact patch. 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 in many reciprocating ball on flat tests: ball wear scars showing an elliptical wear scar with grooving in the direction of motion, plus end of stroke witness marks, which lack directionality. What affect this change of contact geometry has on frictional response is not clear or readily analysable.

If hard ball on soft plate were not an issue, one would expect the frictional response, at the same contact pressure, not to vary with size of test ball. It does!

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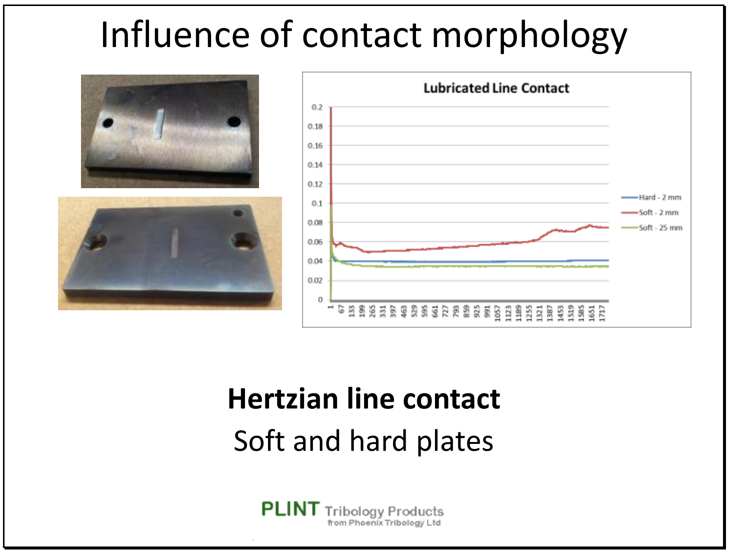

Similar effects can be seen with a cylinder on flat line contact. At short strokes the hard cylinder on soft plate can result in much higher friction than the same cylinder on a hard plate. In this example, with the former, a hole has been dug in the surface; with the latter, the contact patch remains essentially flat.

If the test is repeated with the soft plate at a longer stroke, such that the wear on the plate is distributed over a larger area, thus reducing the linear wear depth, the friction response more closely matches that achieved with the hard plate. The frictional response is clearly affected by the flatness of the resulting wear scar.

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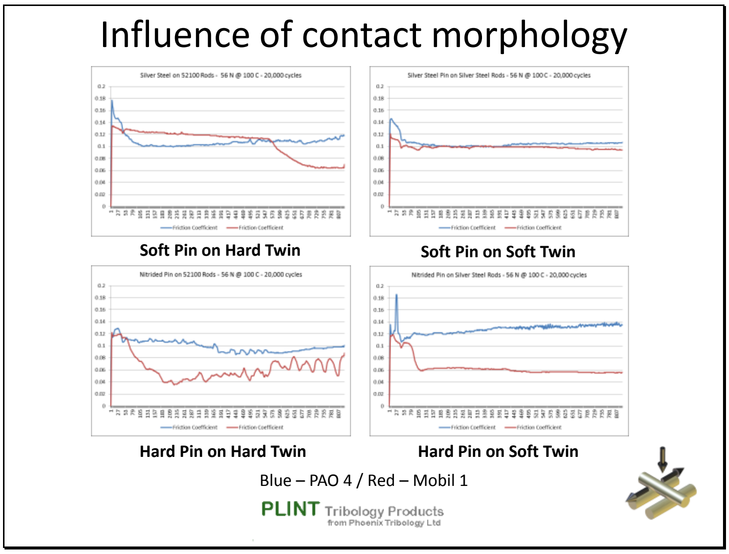

The pin on twin geometry provides very clear examples of the effect of wear track shape, resulting from a combination of contact geometry and relative specimen hardness, on frictional response.

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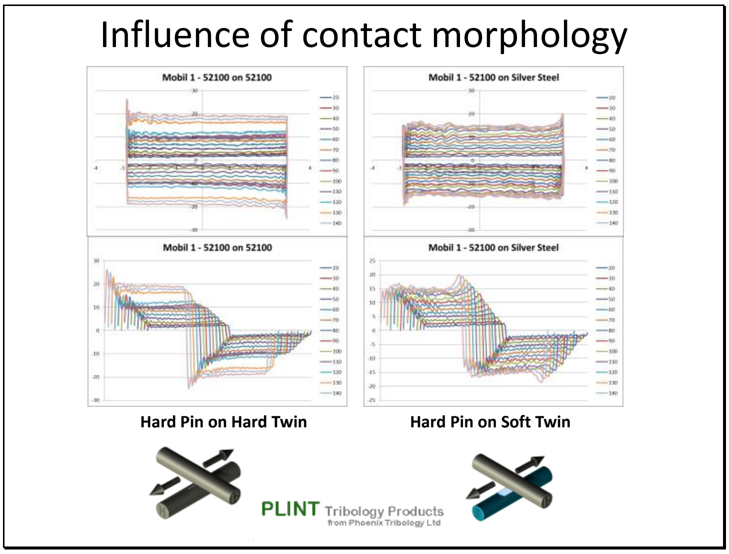

Differences in the shape of the friction loop/instantaneous friction signal for hard on hard and hard on soft illustrates the effects of contact morphology on frictional response.

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

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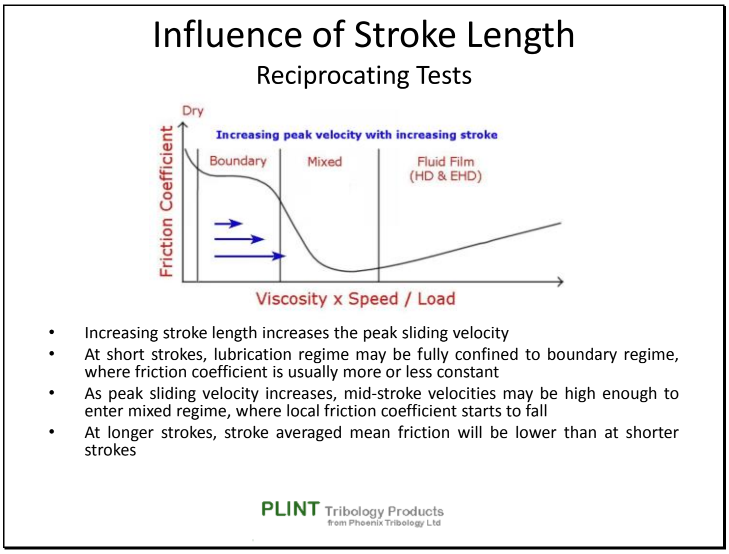

In exactly the same way that real contacts (gears, ring-liner etc) operate across lubrication regimes, the same happens with reciprocating tribometers.

For a given reciprocating frequency, increasing the stroke length increases the peak sliding velocity. At short strokes, the lubrication regime may be fully confined to the boundary regime, where the friction coefficient is usually more or less constant. As the peak sliding velocity increases, the mid-stroke velocities may be high enough to enter the mixed regime, where the local friction coefficient starts to fall. It follows that at longer strokes, hence higher sliding velocities, one would expect the stroke averaged mean friction to be lower than at shorter strokes. This phenomenon can be observed in practice.

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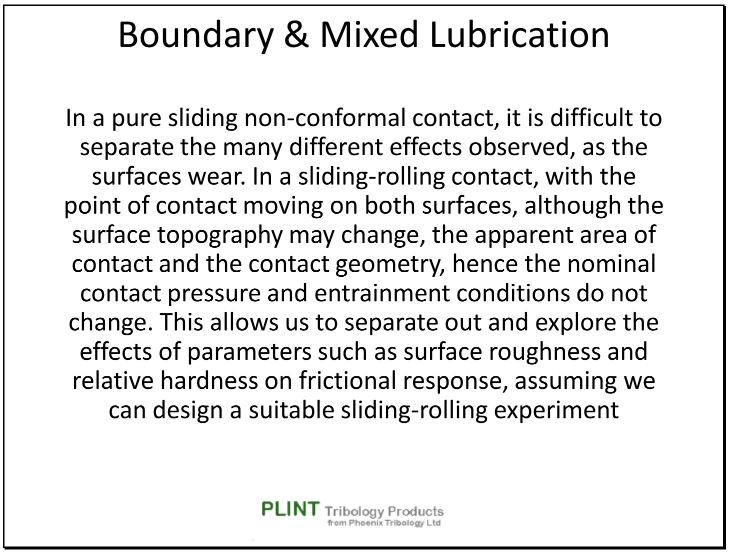

Now let's consider sliding-rolling contacts under boundary and mixed lubrication.

In a pure sliding non-conformal contact, it is difficult to separate the many different effects observed as the surfaces wear. In a sliding-rolling contact, with the point of contact moving on both surfaces, although the surface topography may change, the apparent area of contact and the contact geometry, hence entrainment conditions, do not change. If we can design a suitable sliding-rolling experiment, we can separate out and explore the effects of parameters such as surface roughness and relative hardness.

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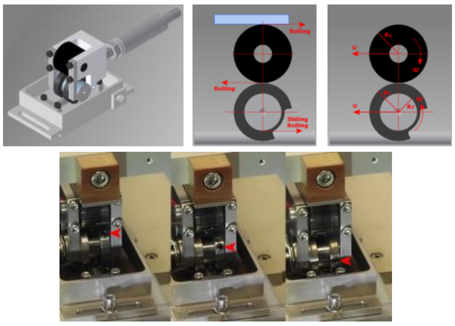
Boundary & Mixed Lubrication



Simple sliding-rolling line contact geometry

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This adapter, designed specifically for the purpose, uses a simple and low cost roller specimen, running on a flat rail. Two discs are secured to the roller by grub screws, with the inner faces of the discs locating on either side of the rail, providing axial guidance. Because it is a sliding-rolling contact, the contact geometry does not change as the test progresses, as happens in a simple sliding hertzian contact test. The lubricant entrainment conditions thus remain constant, as does the nominal contact pressure.

The roller locates between two guide forks, projecting down from a reciprocating head. The upper surface of the discs is in rolling contact with a needle roller cam follower, which is loaded from above via a loaded running plate. The resulting slide-roll ratio is determined by the diameter of the roller sample and of the discs.

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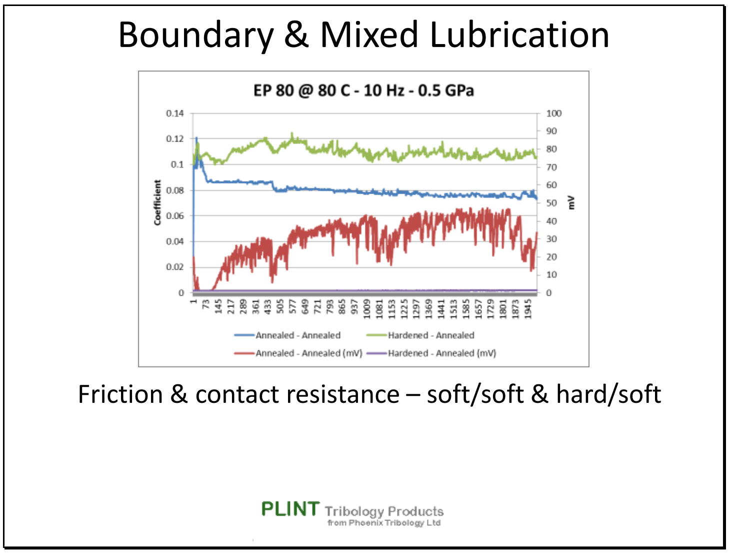

With a harden roller running on a soft plate, the hard roller abrades the softer surface producing a significant amount of oxide debris in the oil. The hard ground surface remains rough throughout, with friction levels typically associated with boundary lubrication. The ploughing (or two body abrasive) component of friction is dominant.

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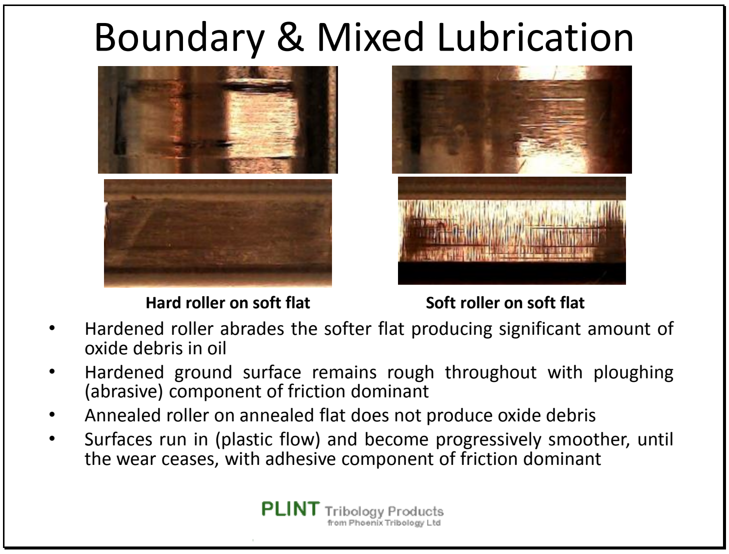

The soft roller running on soft flat does not produce oxide debris. The surfaces run in (with a degree of plastic flow) and become progressively smoother, with mean friction coefficient, instantaneous friction force and contact resistance consistent with mixed lubrication. In this case, the adhesive component of friction is dominant.

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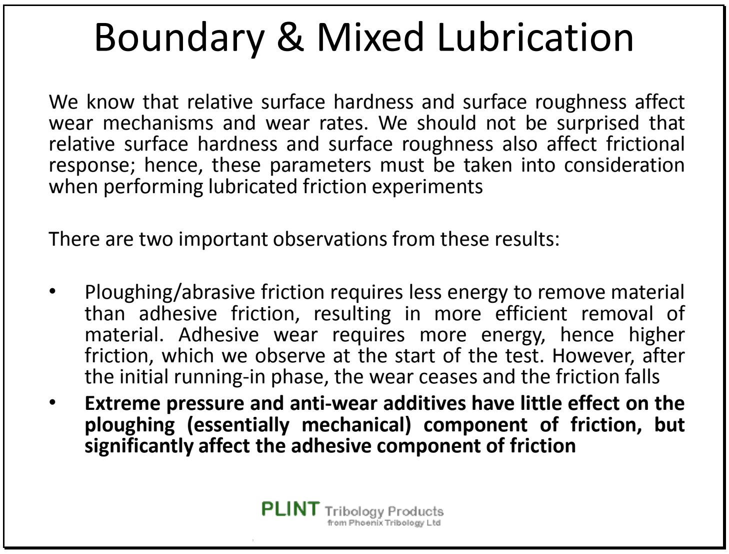

We know that relative surface hardness and surface roughness affect wear mechanisms and wear rates. We should not be surprised that relative surface hardness and surface roughness also affects frictional response; hence, these parameters must be taken into consideration when performing lubricated friction experiments.

There are two further observations that can be made from these experiments:

- Ploughing/abrasive friction requires less energy to remove material than adhesive friction, resulting in more efficient removal of material. Adhesive wear requires more energy, hence higher friction, which we observe at the start of the test. However, after the initial running-in phase, the wear ceases and the friction falls.

- The extreme pressure and anti-wear additives have little effect on the ploughing (essentially mechanical) component of friction, but significant effect on the adhesive component of friction.

The results of the sliding-rolling tests provide insight into what might be happening with the pure sliding tests, for example, with a soft pin on a hard twin.

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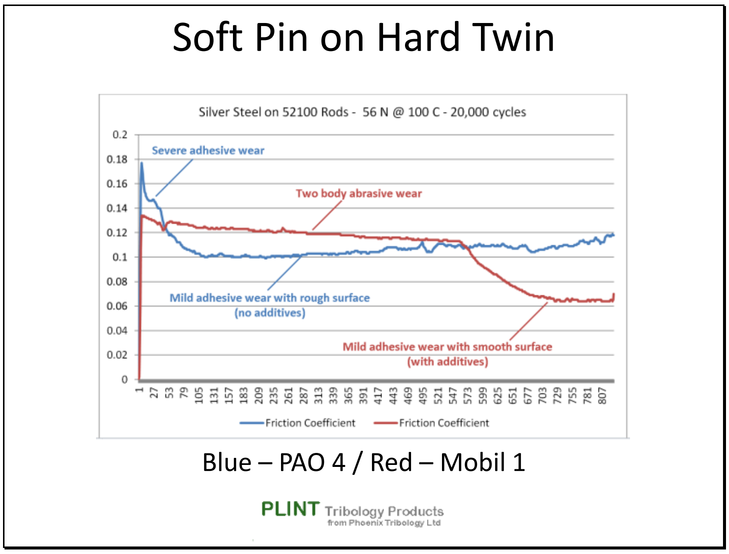

With the fully formulated oil, we observe mild abrasive wear transitioning to mild adhesive wear.

Here, the additives are preventing adhesive wear, so material is removed efficiently by two body abrasive wear and the surfaces become smoother, in a controlled fashion, finally producing a transition to mild adhesive wear, at which point, the surface active additives are effective. Adhesive component of friction dominates, because the surfaces are smooth.

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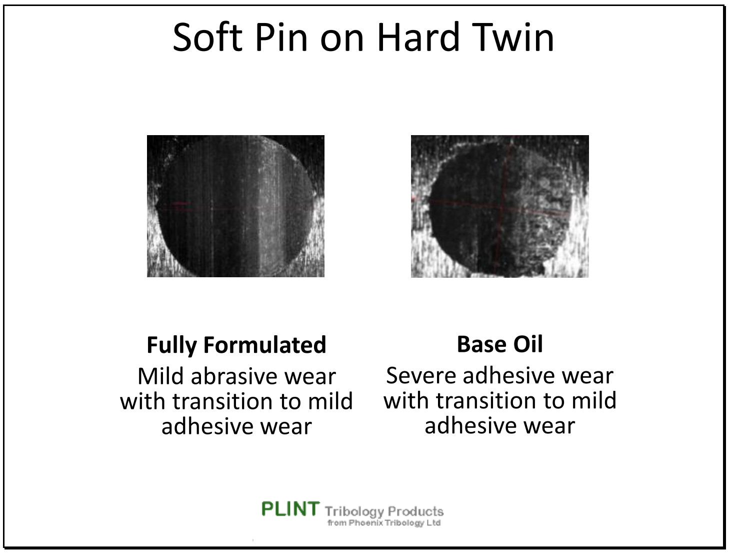

With the base oil, we observe severe adhesive wear, transitioning to mild adhesive wear.

Severe adhesive wear cannot be prevented, because there are no additives. This results in a rapid wear process at the beginning of the test with material transfer and work hardening of the surfaces. Because the resulting surfaces are rough, the frictional response is dependent on both the adhesive component of friction and the ploughing (abrasive) component.

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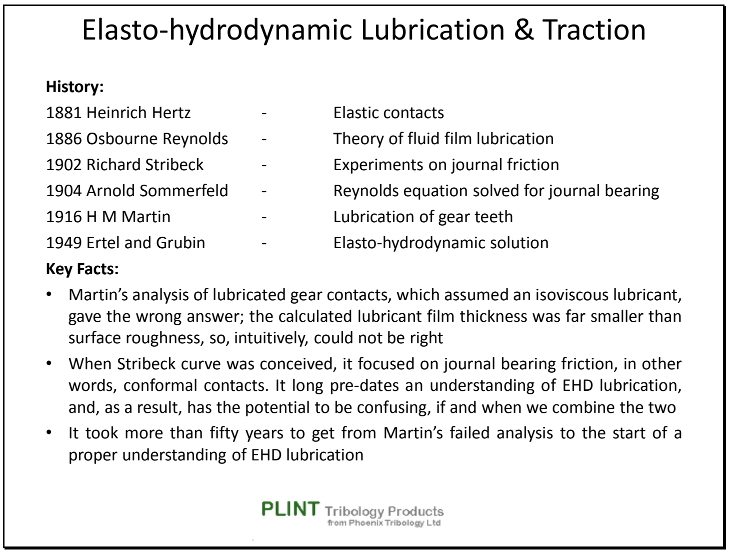

Moving further along the Stribeck curve, our sliding-rolling contact finally arrive at the elasto-hydrodynamic regime. The simplest introduction to EHD lubrication is to start with a bit of history, which will put many things in context:

1881 Heinrich Hertz	-	Elastic contacts
1886 Osbourne Reynolds	-	Theory of fluid film lubrication
1902 Richard Stribeck	-	Experiments on journal friction
1904 Arnold Sommerfeld	-	Reynolds equation solved for journal bearing
1916 H M Martin	-	Lubrication of gear teeth
1949 Ertel and Grubin	-	Elastohydrodynamic solution

There is one key fact to note, which is that H M Martin's analysis of lubricated gear contacts, which assumed an isoviscous lubricant, gave the wrong answer; the calculated lubricant film thickness was far smaller than the surface roughness, so, intuitively, could not be right.

The other point to note is that when the Stribeck curve was conceived, it focused on journal bearing friction, in other words, conformal contacts. It long pre-dates our understanding of EHD lubrication, and, as a result, it has the potential to be confusing, if and when we combine the two ideas.

The final point to note is that it took more than fifty years to get from Martin's failed analysis to the start of a proper understanding of EHD lubrication.

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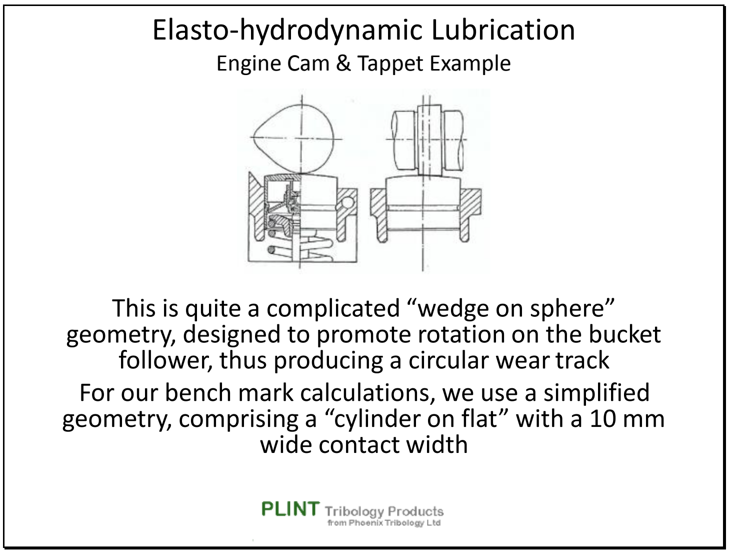

The following example, from the (now defunct) Cambridge Tribology Course, helps to illustrate the point.

This is quite a complicated “wedge on sphere” geometry, designed to promote rotation on the bucket follower, thus producing a circular wear track. For our bench mark calculations, we use a simplified geometry, comprising a “cylinder on flat” with a 10 mm wide contact width.

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Elasto-hydrodynamic Lubrication

Engine Cam & Tappet Example

Base Circle Radius:	18	mm
Base Circle Load:	10	N
Nose Radius:	5	mm
Nose Load:	540	N
Contact Width:	10	mm
Lift:	9	mm
E*:	115×10^9	Pa
Speed:	1500	rpm

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Elasto-hydrodynamic Lubrication

Engine Cam & Tappet Example

Calculate Peak Hertz Pressure:

P_0	=	$(WE^* / LR)^{0.5}$	
P_{0-Base}	=	45	MPa
P_{0-Nose}	=	0.63	GPa

Lubricant Film Thickness [isoviscous plus rigid solids – Martin]:

h_c	=	$4.90\ (\bar{U}\eta RL / W)$		
$\bar{U}$	=	*Entrainment Velocity (assume half sliding velocity)*		
$\bar{U}_{Base}$	=	1.4	ms^{-1}	
$\bar{U}_{Nose}$	=	2.1	ms^{-1}	
h_{Base}	=	1.24	µm	
h_{Nose}	=	0.01	µm	**(cannot be right)**

Low Hertz pressure at base circle suggests Martin’s equation valid
High Hertz pressure at nose suggests Martin’s equation not valid

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Low Hertz pressure at base circle suggests Martin’s equation valid.

High Hertz pressure at nose suggests that Martin’s equation is not valid and elasticity of steel and lubricant pressure sensitivity quantified by the pressure viscosity index α must be taken into account, hence the film thickness must be re-calculated using a suitable EHD equation.

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Elasto-hydrodynamic Lubrication

Engine Cam & Tappet Example

Elasticity of steel and lubricant pressure sensitivity quantified by pressure viscosity index must be taken into account, hence film thickness must be re-calculated using a suitable EHD equation

Lubricant Film Thickness (Ertel-Grubin Equation):

$$\frac{\bar{h}}{R} = 1.37\left(\frac{\eta_0 \alpha 2\bar{U}}{R}\right)^{3/4}\left(\frac{E^* RL}{W}\right)^{1/8}$$

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Elasto-hydrodynamic Lubrication

Engine Cam & Tappet Example

Assuming:

$\eta_0 =$ 0.01 Pa s

$\alpha =$ 2×10^{-8} Pa^{-1}

h_{Nose} = 0.186 µm **(more realistic)**

Calculate Lambda Value:

Assuming:

R_a	=	0.3	µm
λ_{Base}	=	4.13	Mixed Regime
λ_{Nose}	=	0.62	Boundary Regime

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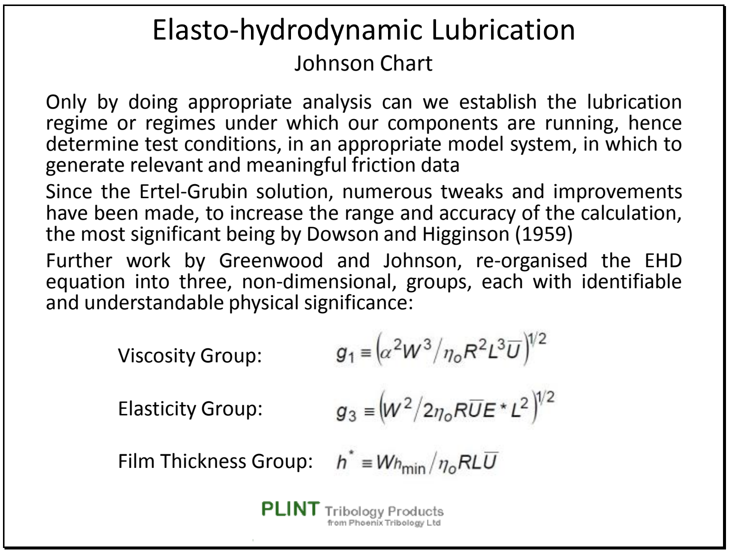
Only by doing the appropriate analysis can we establish the lubrication regime or regimes under which our components are running, hence determine test conditions, in an appropriate model system, in which to generate relevant and meaningful friction data.

Slide 40



Since the Ertel-Grubin solution, numerous tweaks and improvements have been made, to increase the range and accuracy of the calculation, the most significant being by Dowson and Higginson (1959).

Further work by Greenwood and Johnson, re-organised the EHD equations into three, non-dimensional, groups, each with identifiable and understandable physical significance. These comprise:

Viscosity Group: $$g_1 \equiv \left(\alpha^2 W^3 / \eta_o R^2 L^3 \overline{U}\right)^{1/2}$$

Elasticity Group: $$g_3 \equiv \left(W^2 / 2\eta_o R\overline{U}E^* L^2\right)^{1/2}$$

Film Thickness Group: $$h^* \equiv W h_{min} / \eta_o R L \overline{U}$$

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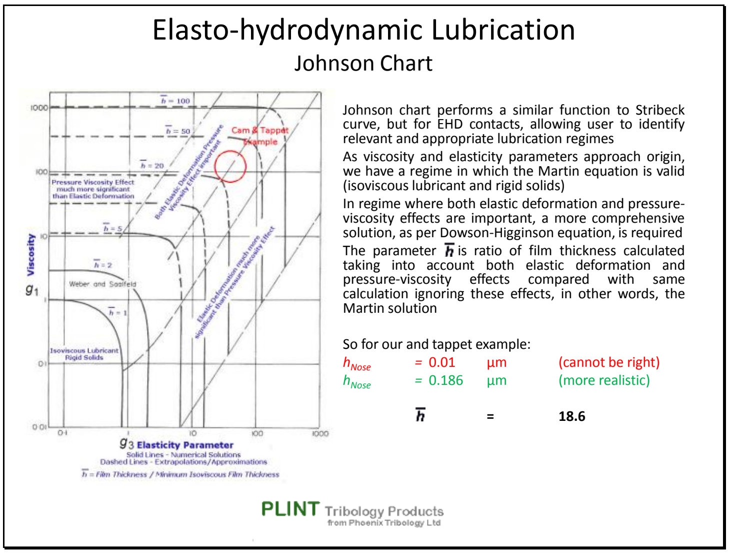

In 1970, Johnson introduced the concept of the “Johnson chart”, which performs a similar function to the Stribeck curve, but for EHD contacts. This allows the user to identify relevant and appropriate lubrication regimes.

As the viscosity and elasticity parameters approach the origin, we have a regime in which the Martin equation is valid (isoviscous lubricant and rigid solids).

In the regime where both elastic deformation and pressure-viscosity effects are important, a more comprehensive solution, as per the Dowson-Higginson equation, is required.

The parameter $\bar{h}$ is the ratio of the film thickness calculated taking into account both elastic deformation and pressure-viscosity effects compared with the same calculation ignoring these effects, in other words, the Martin solution.

So for our cam and tappet example:

h_{Nose}	=	0.01	µm	(cannot be right)
h_{Nose}	=	0.186	µm	(more realistic)
$\bar{h}$	**=**	**18.6**		

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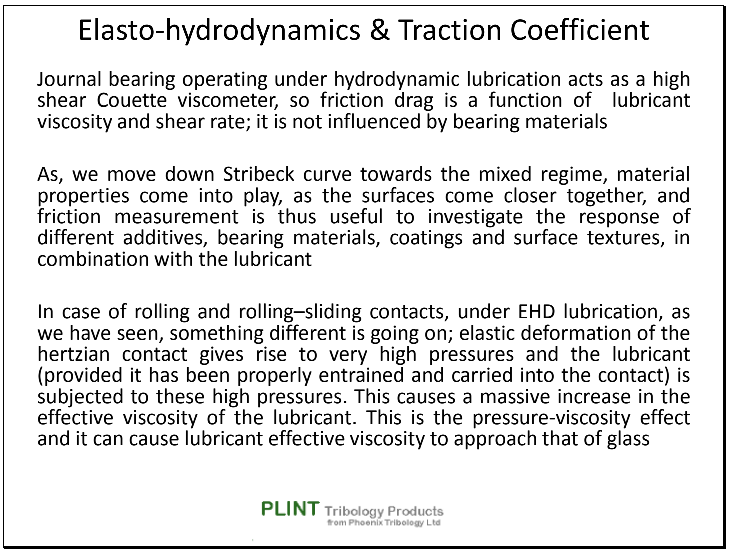

But why is this so important if we are only interested in friction/traction in a contact running under EHD lubrication?

A journal bearing operating under hydrodynamic lubrication acts as a high shear Couette viscometer, so the friction drag is a function of the lubricant viscosity and the shear rate; it is not influenced by the bearing materials. As, we move down the Stribeck curve towards the mixed regime, material properties come into play, as the surfaces come closer together, and friction measurement is thus useful to investigate the response of different additives, bearing materials, coatings and surface textures, in combination with the lubricant.

In the case of the rolling and rolling–sliding contacts, under EHD lubrication, something different is going on. Elastic deformation of the hertzian contact gives rise to very high pressures and the lubricant (provided it has been properly entrained and carried into the contact) is subjected to these high pressures; this causes a massive increase in the effective viscosity of the lubricant. This is the pressure-viscosity effect and it can cause the lubricant effective viscosity to approach that of glass.

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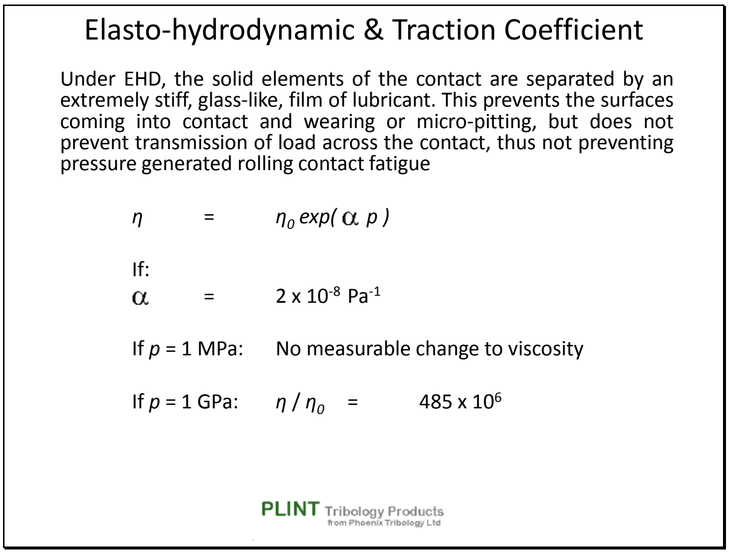

So, under EHD, the solid elements of the contact are separated by an extremely stiff, glass-like, film of lubricant. This prevents the surfaces coming into contact and wearing or micro-pitting, but obviously does not prevent the transmission of load across the contact, thus not preventing pressure generated rolling contact fatigue. It is useful to note that the thickness of the EHD lubricant film is pretty insensitive to applied load, implying that the harder it is squeezed, the stiffer it gets.

When it comes to traction between the two sides of the contact, we are clearly dealing with transmission of a shear force through a film of given thickness and (pressure enhanced) effective viscosity.

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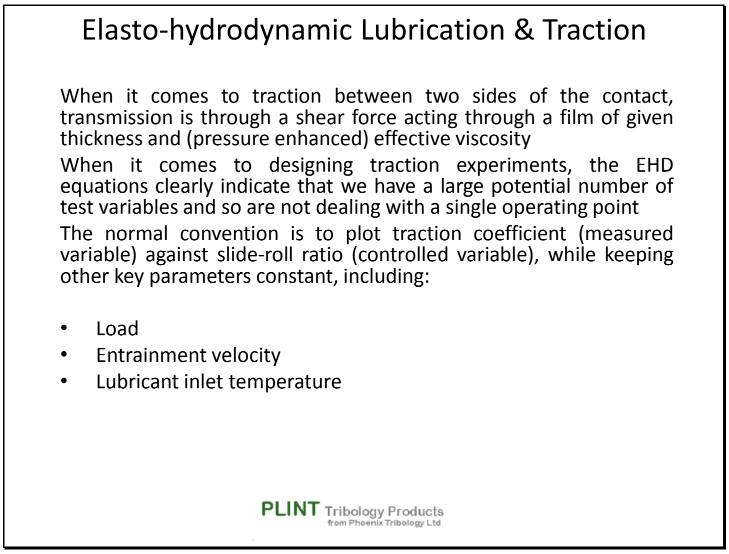

When it comes to designing traction experiments, the EHD equations clearly indicate that we have a large potential number of test variables; we are not dealing with a single operating point!

The normal convention is to plot traction coefficient (measured variable) against slide-roll ratio (controlled variable), while keeping other key parameters constant.

These include:

- Load
- Entrainment Velocity
- Lubricant Inlet Temperature

Step-wise variation of these parameters then allows three families of traction curves to be plotted, to investigate the effects of different loads, entrainment velocities and lubricant inlet temperatures on the resulting traction coefficient.

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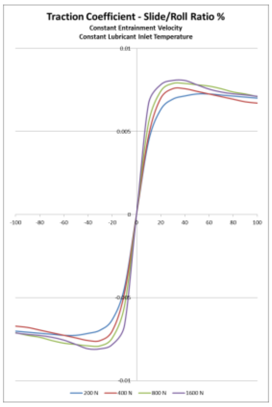
Elasto-hydrodynamic Lubrication & Traction



Key point to note: if we wish to run experiments at constant entrainment velocity, hence nominally constant lubrication regime, we have to vary the speed of both test rollers, not just one

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A key point to note is that if we wish to run experiments at constant entrainment velocity, hence nominally constant lubrication regime, we have to vary the speed of both test rollers, not just one.

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Stribeck Curve on Two Roller Machine

Slide-Roll Ratio	5 %		Constant			
Roller 1 - Diameter	25 mm		Constant			
Roller 2 - Diameter	50 mm		Constant			
Entrainment Velocity	U1+U2	U1-U2	U1	N1	U2	N2
m/s	m/s	m/s	m/s	rpm	m/s	rpm
0	0	0	0.000	0.00	0.000	0.00
0.1	0.2	0.005	0.100	76.38	0.095	36.33
0.2	0.4	0.01	0.200	152.77	0.190	72.66
0.3	0.6	0.015	0.300	229.15	0.285	108.99
0.4	0.8	0.02	0.400	305.54	0.380	145.32
0.5	1	0.025	0.500	381.92	0.476	181.65
0.6	1.2	0.03	0.600	458.31	0.571	217.98
0.7	1.4	0.035	0.700	534.69	0.666	254.30
0.8	1.6	0.04	0.800	611.08	0.761	290.63
0.9	1.8	0.045	0.900	687.46	0.856	326.96
1	2	0.05	1.000	763.84	0.951	363.29

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We require the same approach if we want to run experiments at the same slide-roll ratio and vary the entrainment velocity. By doing this, we can generate an EHD Stribeck curve.

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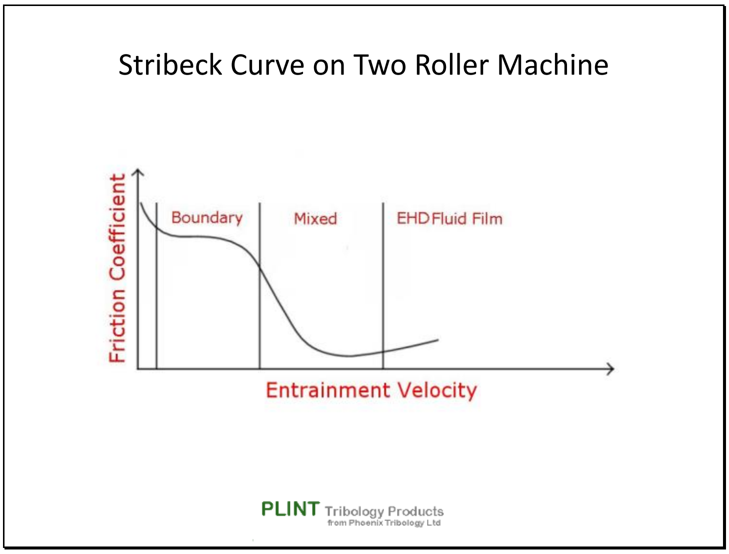

The friction or traction coefficient plotted against entrainment velocity.

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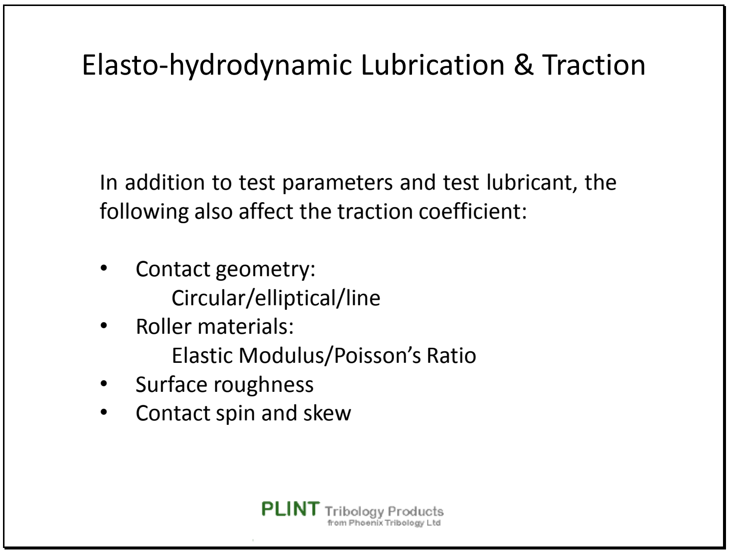

In addition to the test parameters and the test lubricant, the following also affects the traction coefficient:

- Contact geometry (circular, elliptical or line)
- Roller materials (Elastic Modulus/Poisson’s Ratio)
- Surface roughness
- Contact spin and skew

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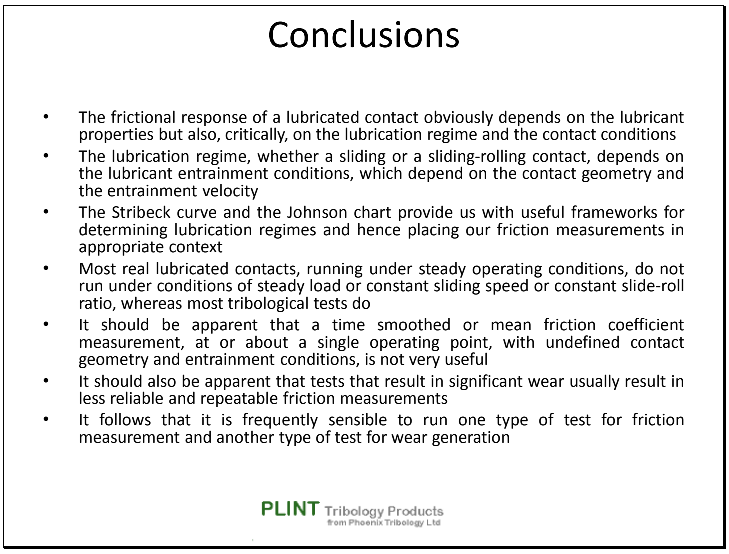

The frictional response of a lubricated contact obviously depends on the lubricant properties but also, critically, on the lubrication regime and the contact conditions. The lubrication regime, whether a sliding or a sliding-rolling contact, depends on the lubricant entrainment conditions. The lubricant entrainment conditions depend on the contact geometry and the entrainment velocity. The Stribeck curve and the Johnson chart provide us with useful frameworks for determining lubrication regimes and hence placing our friction measurements in the appropriate context.

Although friction measurement under boundary lubrication provides a useful indication of wear transitions, it is usually not much help if we are concerned with frictional losses in most lubricated sliding or sliding-rolling contacts.

Most real lubricated contacts, running under steady operating conditions, do not run under conditions of steady load or constant sliding speed or constant slide-roll ratio. Most tribological tests do.

It should be apparent that a time smoothed or mean friction coefficient measurement, at or about a single operating point, with undefined contact geometry and entrainment conditions, is not very useful.

It should also be apparent that tests that result in significant wear usually result in less reliable and repeatable friction measurements. It follows that it is frequently sensible to run one type of test for friction measurement and another type of test for wear generation.

The many parameters that affect the frictional response of a lubricated contact make the very notion of a lubricated friction coefficient more or less meaningless.

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