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Engine Tribology
(including Hybrids and EVs)
Tribological Bench Tests

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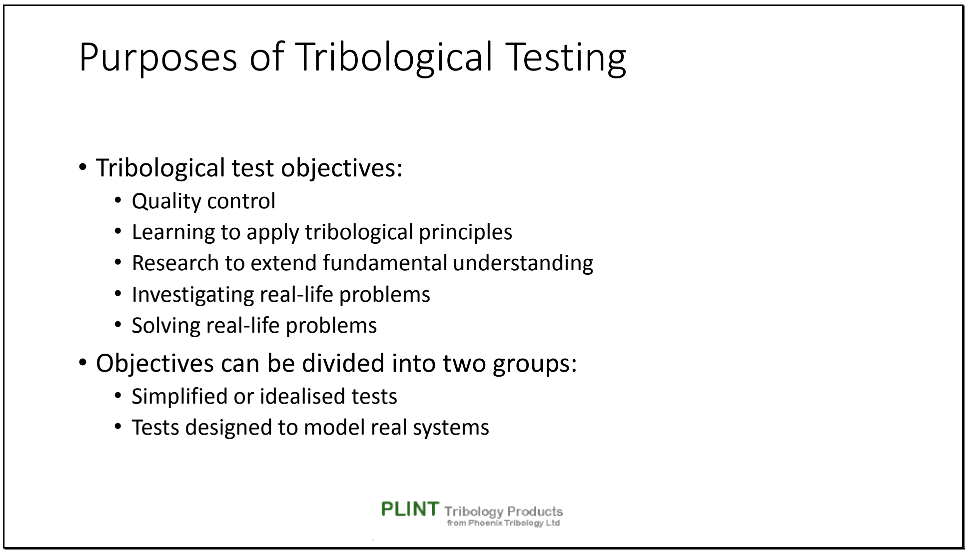

Tribological testing has a wide range of different purposes and it is important to recognise that different objectives result in different test system requirements and a different experimental approach. Here are some typical test objectives:

- Quality control
- Learning to apply tribological principles
- Research to extend fundamental understanding
- Investigating real-life problems
- Solving real-life problems

These objectives can be roughly divided into two groups:

- Simplified or idealised tribological tests

- Tribological tests design to model real systems

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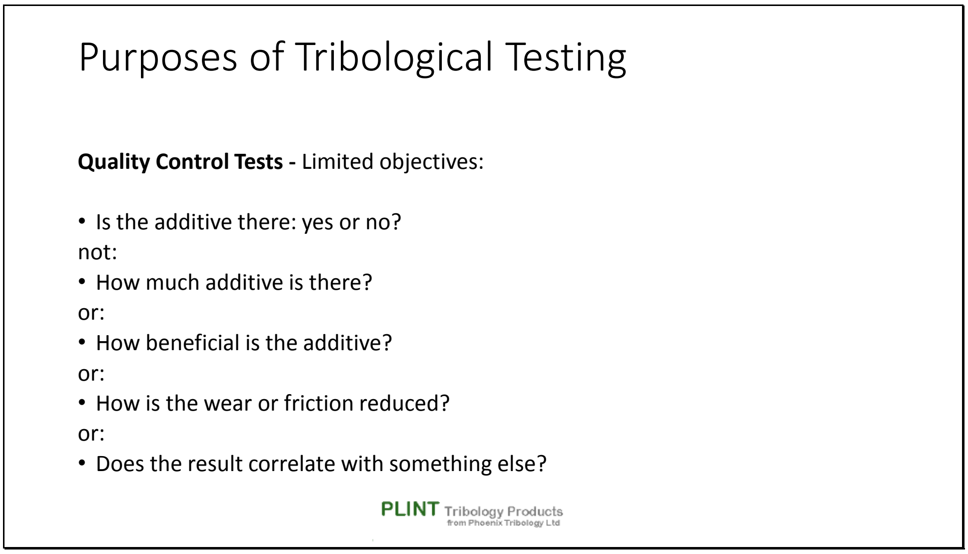

Quality control tests have a very limited purpose and are mostly based on recognised standards, and with very limited objectives, for example, answering a simple question such as:

Is the additive there: yes or no?

not:

How much additive is there?

or:

How beneficial is the additive?

or:

How is the wear or friction reduced?

or:

Does the result correlate with something else?

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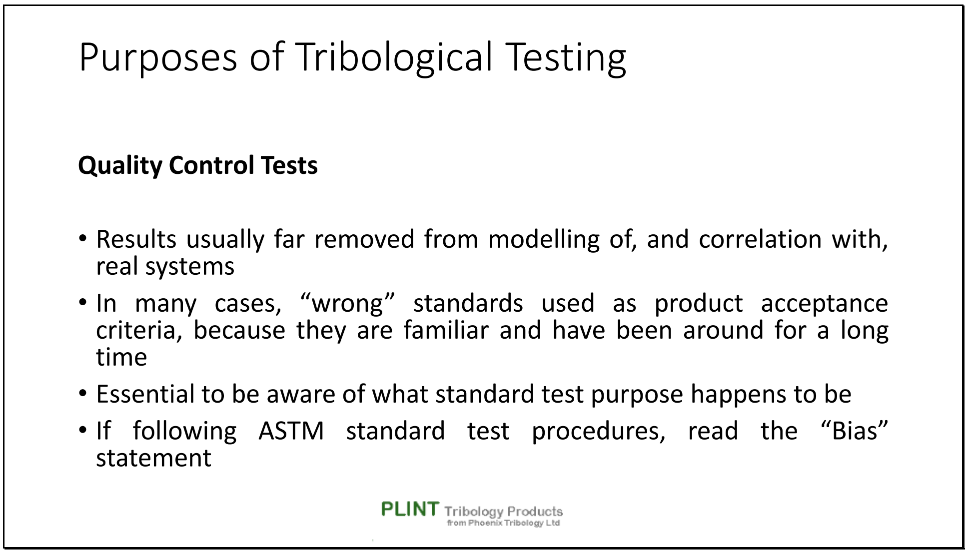

The results of many of these tests are usually far removed from modelling of, and correlation with, real systems.

In many cases, “wrong” standards are used as product acceptance criteria, because they are familiar and have been around for a long time.

By “wrong” we mean that they poorly simulate the end application so are no guide to the performance in the real application. The risk is that the product is designed to pass the test, not work in the application. Standard tests are not always used correctly!

Quality control tests have their purpose, but it is essential we are aware of what that purpose happens to be.

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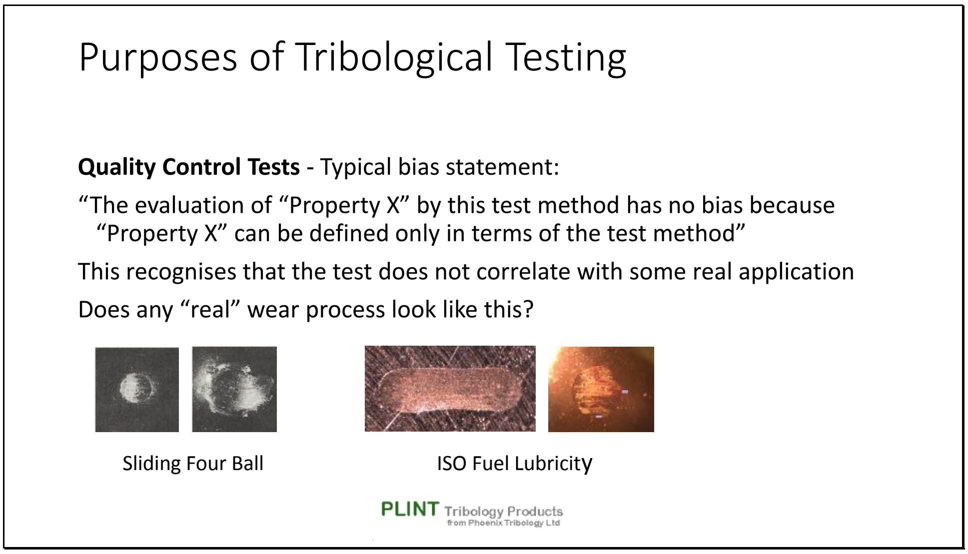

If you were to choose to follow ASTM standard test procedures you should note that the "Bias" statements will frequently state the following:

"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."

This essentially recognises that the test does not correlate with some unspecified real application.

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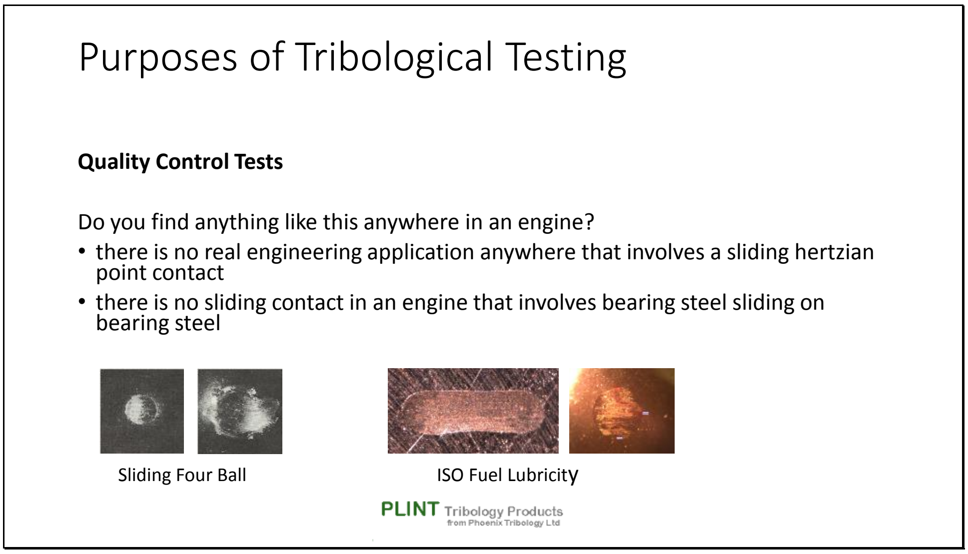

Does any "real" wear process look like this?

Do you find anything like this anywhere in an engine? The answer both questions is of course no

- there is no real engineering application anywhere that involves a sliding hertzian point contact
- there is no sliding contact in an engine that involves bearing steel sliding on bearing steel

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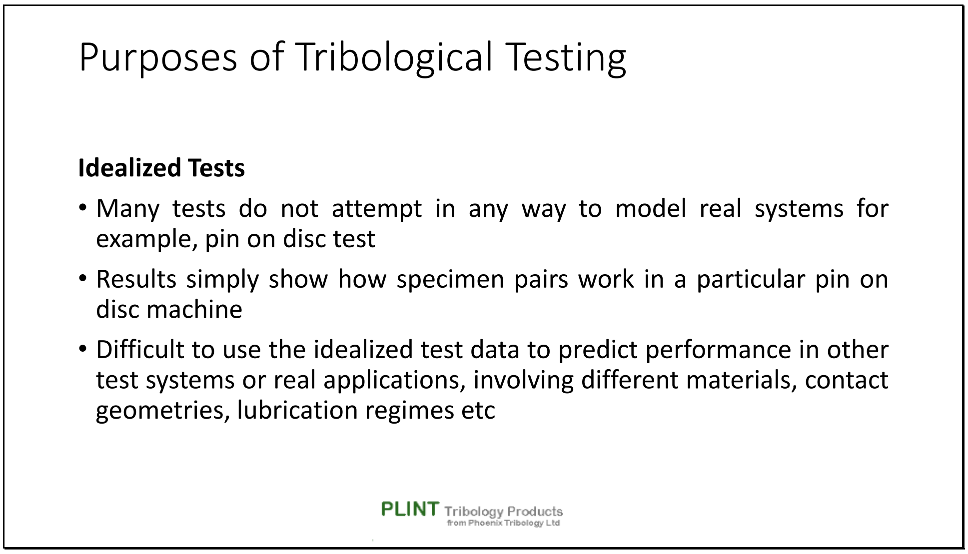

There are many idealized tests that do not attempt in any way to model real systems. A good example of an idealized tribological experiment is the pin on disc test, where the results simply show how the specimen pairs work in a particular pin on disc machine. The difficulty comes when we try to use the idealized test data to predict performance in other test systems or real applications, involving different materials, contact geometries, lubrication regimes etc.

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Purposes of Tribological Testing

Modelling Real Systems

- Tests must model the friction, wear and/or failure mechanism apparent in the real application, with test materials and variables selected and adjusted in order to obtain correlation with field data
- If wear generated in the bench test looks nothing like the wear in the real system, model likely to be wrong
- Starting point for modelling real system is to characterize the full-scale system, then see if it can sensibly be modelled in a bench test, at reduced scale and, potentially, at accelerated rate

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The challenge with real life emulations is that to be of any use the tests must model the friction, wear and/or failure mechanism apparent in the real application, with test materials and variables selected and adjusted in order to obtain correlation with field data. If the wear generated in the bench test looks nothing like the wear in the real system, the model is likely to be wrong. Hence, before we start, we need to be able to analyse what is going on in the real application and then to see if there is a sensible way to model this in a bench test. The risk here is twofold, that our original analysis and then the resulting model may both be wrong!

So the starting point for modelling a real system is to characterize the full-scale system, then see if it can sensibly be modelled in a bench test at reduced scale and, potentially, at an accelerated rate.

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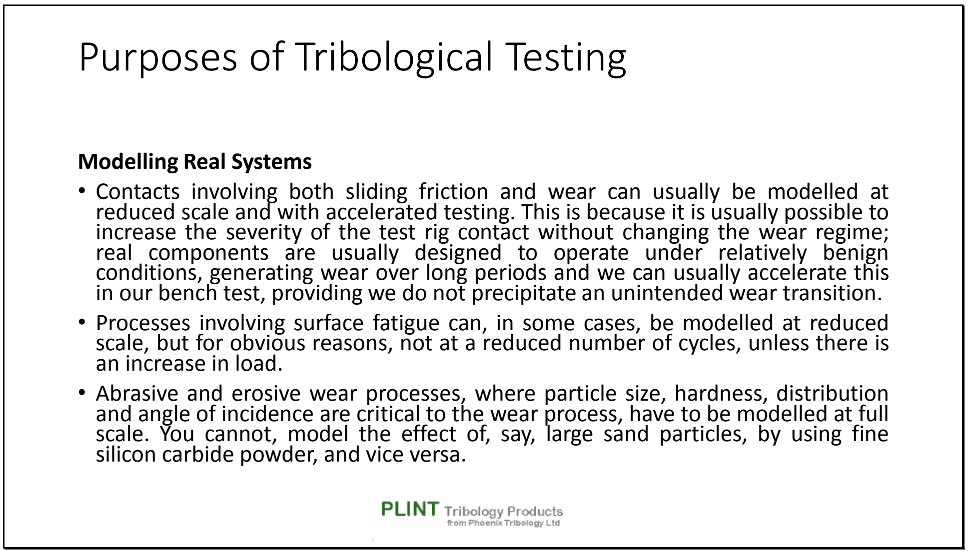

As a general rule, contacts involving both sliding friction and wear can be modelled at reduced scale and with accelerated testing. This is because it is usually possible to increase the severity of the test rig contact without changing the wear regime; real components are usually designed to operate under relatively benign conditions, generating wear over long periods and we can usually accelerate this in our bench test, providing we do not precipitate an unintended wear transition.

Processes involving surface fatigue can, in some cases, be modelled at reduced scale, but for obvious reasons, not at a reduced number of cycles, unless there is an increase in load. Examples of these processes include rolling contact fatigue and fretting.

Abrasive and erosive wear processes, where particle size, hardness, distribution, angle of incidence and, in the case of erosive wear, particle velocity, are critical to the wear process, have to be modelled at full scale. You cannot, model the effect of, say, large sand particles, by using fine silicon carbide powder, and vice versa.

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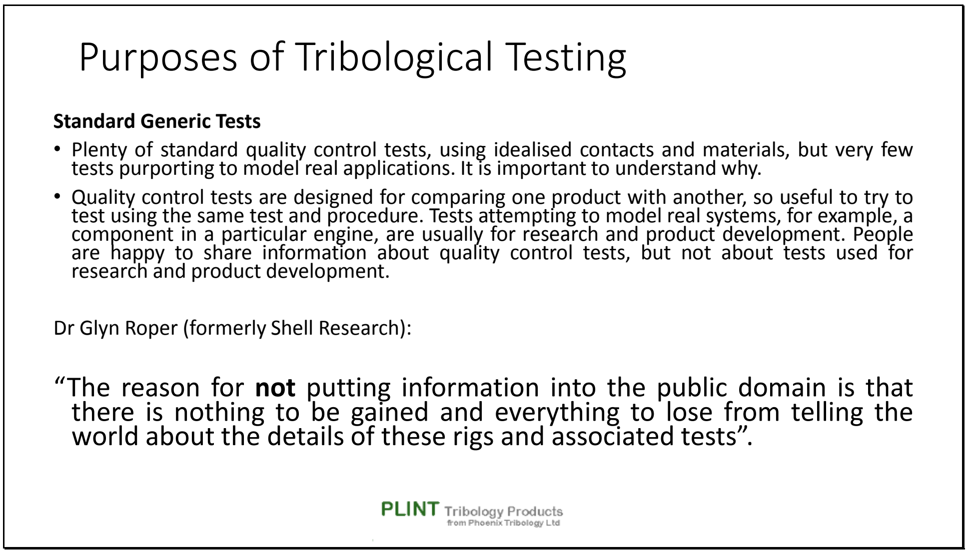

You will soon discover that there are plenty of standard tests available for quality control purposes, using idealised contacts and materials, but very few tests purporting to model tribological contacts in real applications. It is important to understand why this is.

Quality control tests are designed for comparing one product with another, so it is useful to try to test different products using the same test and procedure. Tests attempting to model real systems, for example, a specific component in a particular engine, are usually undertaken with research and product development in mind. People are happy to share information about quality control tests, but not about tests used for research and product development.

To quote Dr Glyn Roper (formerly Shell Research):

"The reason for **not** putting information into the public domain is that there is nothing to be gained and everything to lose from telling the world about the details of these rigs and associated test procedures".

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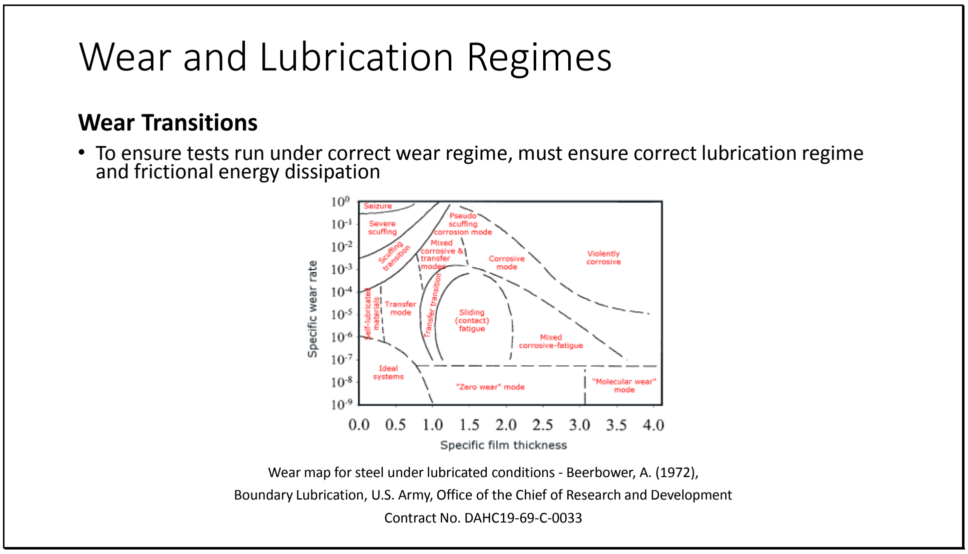

If we are to model a specific real-life wear mechanism, we need to make sure that our experiment is run under the correct wear regime.

Although there are numerous “wear mechanism maps”, (see Lim, S.C. and Ashby, M.F. (1987), Wear-mechanism maps, Acta Metall., 35(1):1-15.)), for dry sliding contacts, there are very few for lubricated contacts. One example worth noting is that proposed by Beerbower in 1972, for a steel contact under boundary lubrication.

To ensure that our tests are run under the correct wear regime, we must, in addition to numerous other parameters, ensure that we run under the correct lubrication regime, with appropriate frictional energy dissipation.

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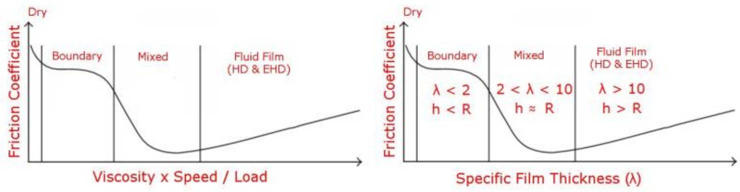
Wear and Lubrication Regimes

Lubrication Regimes



Stribeck curve:

- Specific film thickness is ratio of oil film thickness to the composite surface roughness (Lambda value)
- For values of less than 4, contact in mixed or boundary lubrication

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The specific film thickness is ratio of the oil film thickness to the composite surface roughness, otherwise known as the Lambda value. For values of less than 4, the contact will be in mixed or boundary lubrication regimes, in other words, at the lower end of the Stribeck curve.

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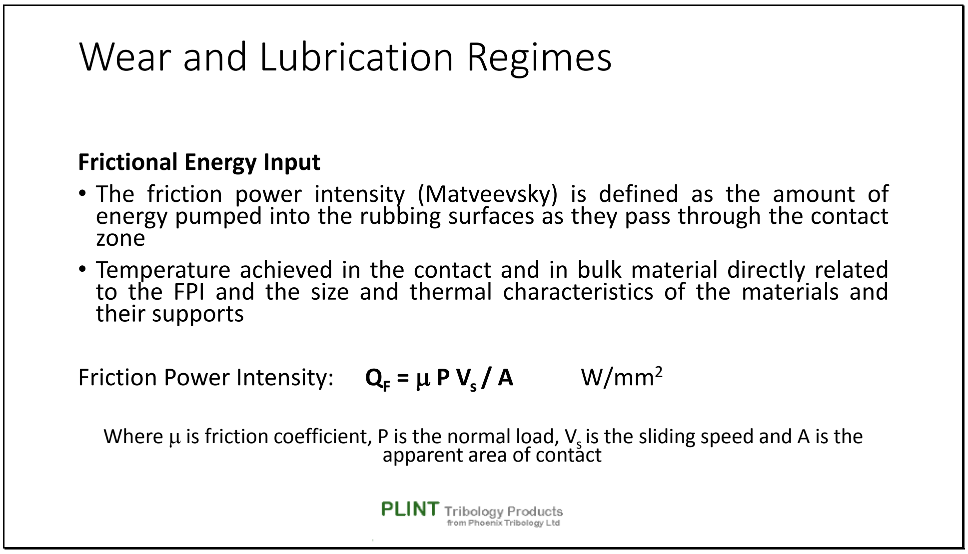

The friction power intensity (Matveevsky) is defined as the amount of energy pumped into the rubbing surfaces as they pass through the contact zone. The temperature achieved in the contact and in the bulk material is directly related to the FPI and the size and thermal characteristics of the materials and their supports.

The FPI defines only the rate of energy generation and does not take into account the timescale over which this energy can be lost to the contacting materials. This timescale clearly has implications for the amount of damage caused in the contact.

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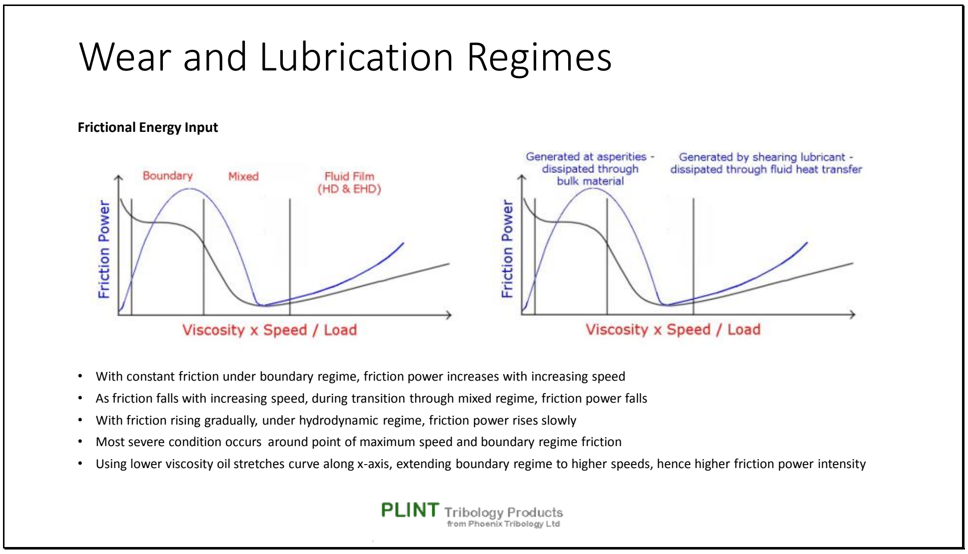

If we superimpose a friction power curve on our Stribeck curve, we get the following.

With fairly constant friction under the boundary regime, friction power increases with increasing speed. As the friction falls with increasing speed, during transition through the mixed regime, the friction power falls. With friction rising gradually with increasing speed, under the hydrodynamic regime, friction power rises steadily, from a low value.

It will be apparent that the most severe condition in terms of frictional energy input occurs somewhere around the point of maximum speed and maximum boundary regime friction. Using lower viscosity oil stretches the curve further along the x-axis, extending the boundary regime to higher speeds, thus higher friction power intensity.

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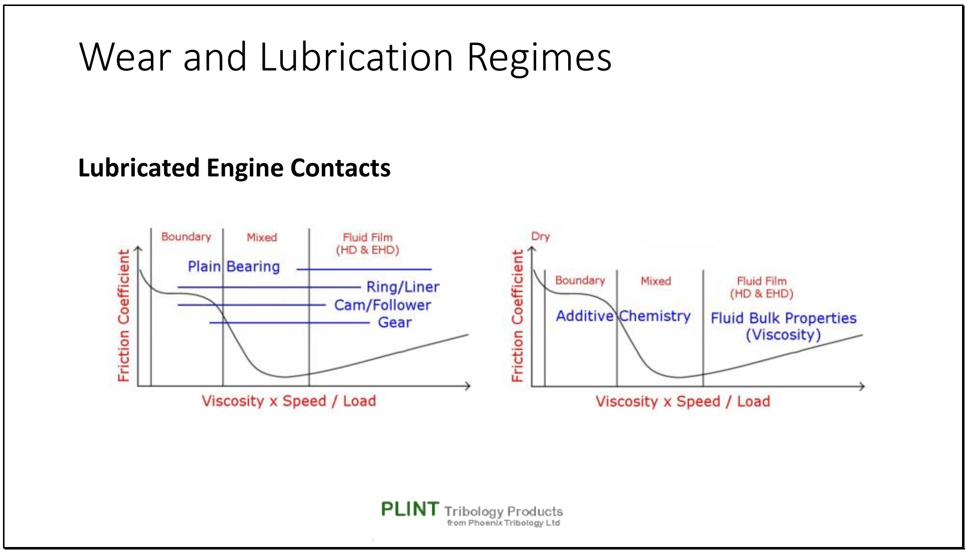

If we now replot the Stribeck curve showing where typical engine components operate, we have the following.

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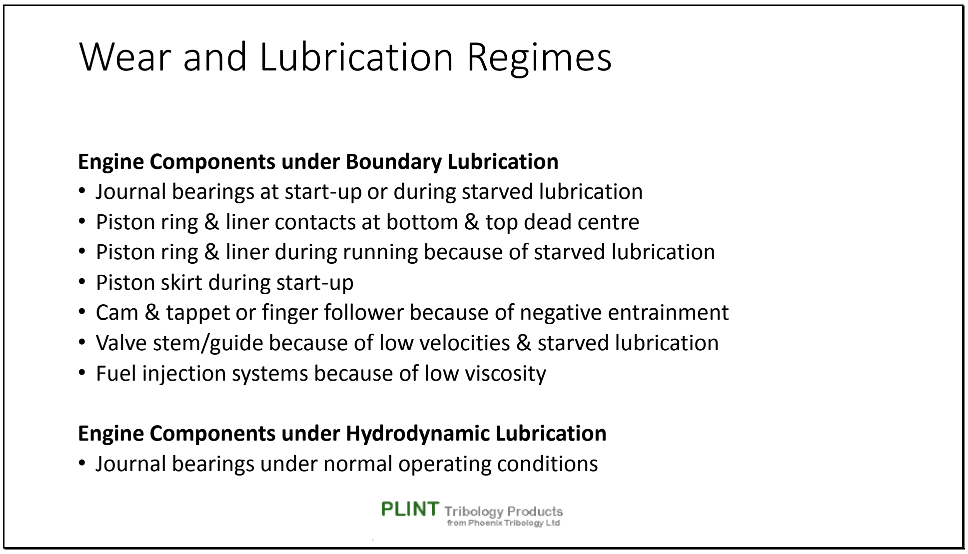

Here are some engine components running under boundary lubrication:

- Journal bearings at start-up or during starved lubrication
- Piston ring and liner contacts at bottom and top dead centre
- Piston ring and liner during running because of starved lubrication
- Piston skirt during start-up
- Cam and tappet or finger follower because of negative entrainment
- Valve stem/guide because of low velocities and starved lubrication
- Fuel injection systems because of low viscosity

Journal bearings of course run under hydrodynamic lubrication under normal operating conditions.

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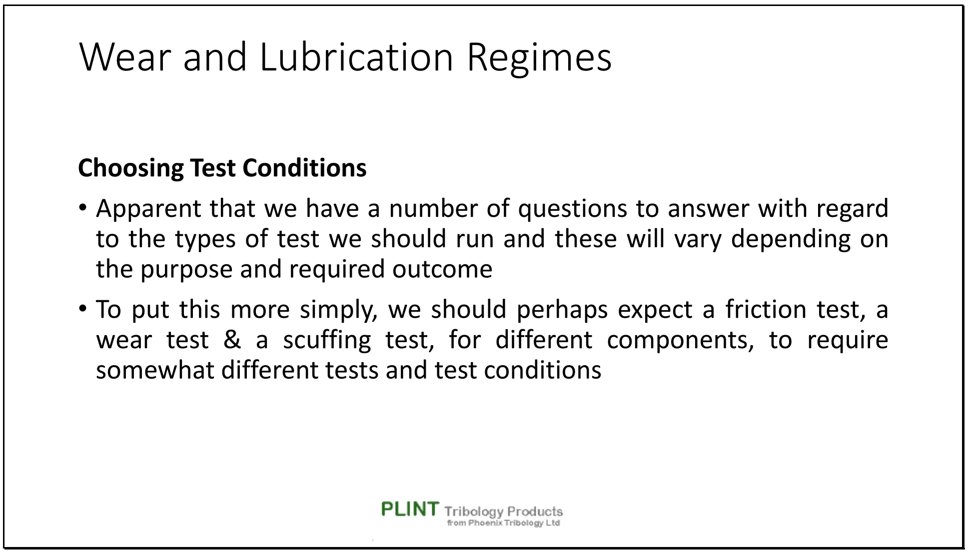

It should be apparent that we have a number of questions to answer with regard to the types of tests we should run and these will vary depending on the purpose and required outcome. To put this more simply, we should perhaps expect a friction test, a wear test and a scuffing test, for different components, to require somewhat different tests and test conditions.

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Rings and Liners

Floating Liner Rigs

- Numerous non-standard designs devised
- Rigs essentially deconstructed engines, with a cylinder liner mounted on force transducers and (usually) counter piston in place of cylinder head
- Allows measurement of friction between ring pack & liner, with limited pressurisation, but without combustion pressure
- Although friction measured directly, actual radial load on each ring is indeterminate

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Numerous designs of floating liner rig have been devised, a recent example being a unit at the University of Nottingham. The rig is essentially a deconstructed engine, with a cylinder liner mounted on force transducers and a counter piston in place of a cylinder head.

The arrangement allows measurement of friction between ring pack and liner, with a limited degree of pressurisation, but without combustion and hence combustion pressure. Although the friction is measured directly, the actual radial load on each ring is indeterminate.

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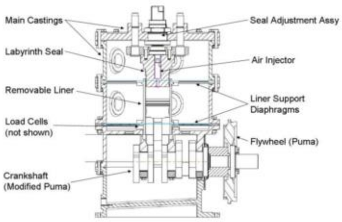
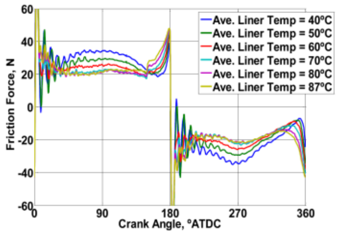
Rings and Liners

Floating Liner Rigs





A New Floating-Liner Test Rig Design to Investigate Factors Influencing Piston-Liner Friction

SAE 2012-01-1328 Published 04/16/2012

T Law, D MacMillan, PJ Shayler & G Kirk - University of Nottingham

Ian Pegg and Roland Stark - Ford Motor Co

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Rings and Liners

Reciprocating Tribometers

General purpose reciprocating tribometers used for many years in attempts to model the conditions in the ring & liner contact

- Eyre-BICERI (British Internal Combustion Engine Research Institute) Universal Wear Machine – from 1968
- Optimol SRV Machine – from 1972
- Cameron-Plint TE 77 Machine – from 1982

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General purpose reciprocating tribometers have been used for many years in attempts to model the conditions in the ring and liner contact, starting with the Eyre-BICERI (British Internal Combustion Engine Research Institute) Universal Wear Machine, in 1968, the SRV machine in 1972 and the longer stroke Cameron-Plint machine in 1982.

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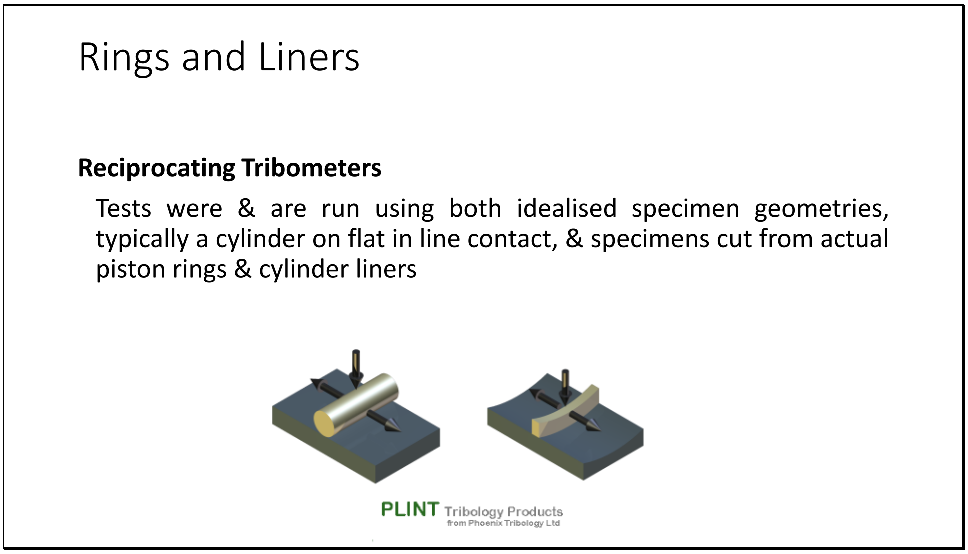

With all these machines, tests were and are run using both idealised specimen geometries, typically a cylinder on flat in line contact, and specimens cut from actual piston rings and cylinder liners.

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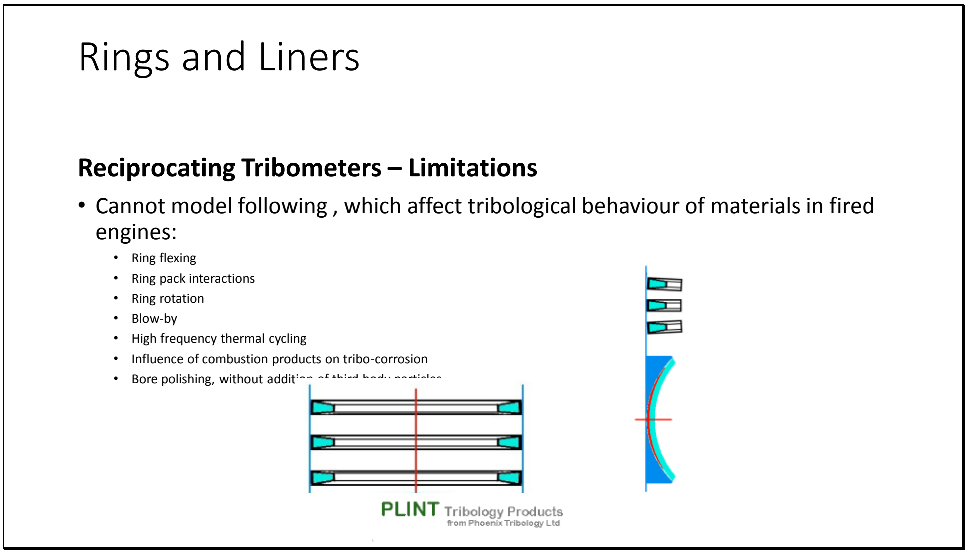

Reciprocating tribometers cannot model the following parameters, which affect tribological behaviour of materials in fired engines:

- Ring flexing
- Ring pack interactions
- Ring rotation
- Blow-by
- High frequency thermal cycling
- Influence of combustion products on tribo-corrosion
- Bore polishing, without addition of third body particles

In the engine the rings flex depending on direction of motion, combustion pressure and ring profile. It is not possible to emulate this motion in a reciprocating tribometer.

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Rings and Liners

Sharing of Wear Ring-Liner Model – Overlap Parameter

- A fired engine tends to produce about the same amount of wear on the ring as on the liner. In short stroke reciprocating bench tests, more wear usually occurs on the liner specimen than on the ring. This is to be expected. There is a difference in “overlap parameter” between the (long stroke) engine & the (short stroke) reciprocating rig.
- The "overlap parameter" (Czichos) is defined as 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 or reciprocating tests it is variable, but is typically less than 0.05.

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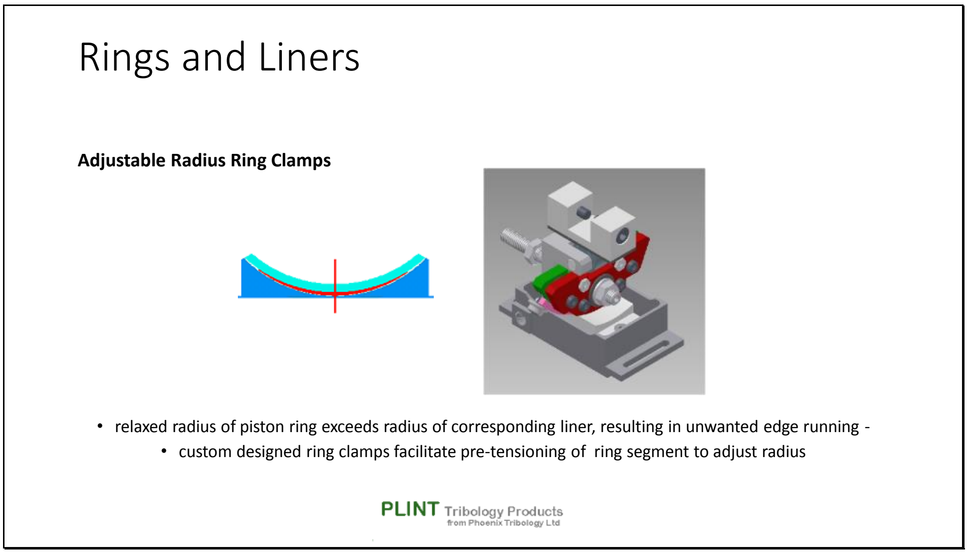

Another issue with ring and liner samples in reciprocating tribometers is that the relaxed radius of the piston ring exceeds the radius of the corresponding liner, resulting in unwanted edge running.

This issue can usually be addressed by using custom designed ring clamps, which facilitate pre-tensioning of the ring segment to adjust the radius to match the liner section.

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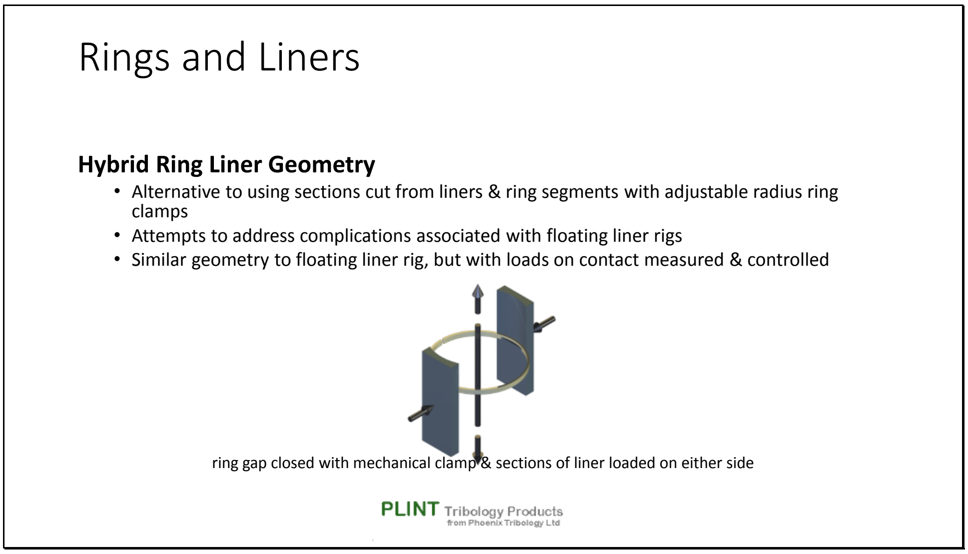

An alternative to using sections cut from liners and ring segments with adjustable radius ring clamps and that attempts to address some of the complications associated with floating liner rigs, is to use a complete ring, with the ring gap closed with a mechanical clamp. A section of liner is then loaded on either side of the ring, which is reciprocated. The result is a rig with similar geometry to a floating liner rig, but with the loads on the contact measured and controlled.

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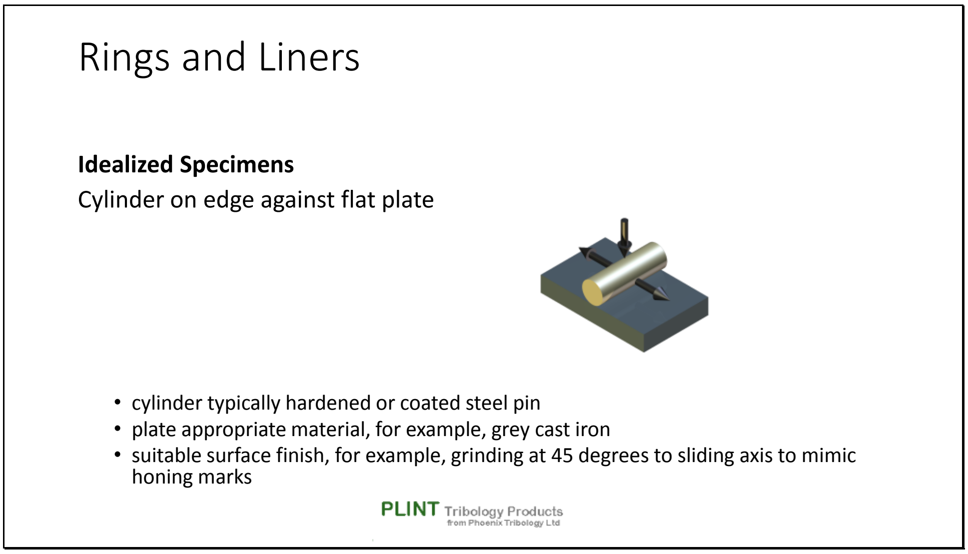

The most commonly used “idealized” ring and liner model used on reciprocating tribometers is the cylinder on edge against a flat plate. The cylinder will typically be of some form of harden or coated steel and the plate of appropriate material, for example grey cast iron, with suitable surface finish. In an attempt to mimic honing marks, grinding the plate surface at 45 degrees to the sliding axis is sometimes used.

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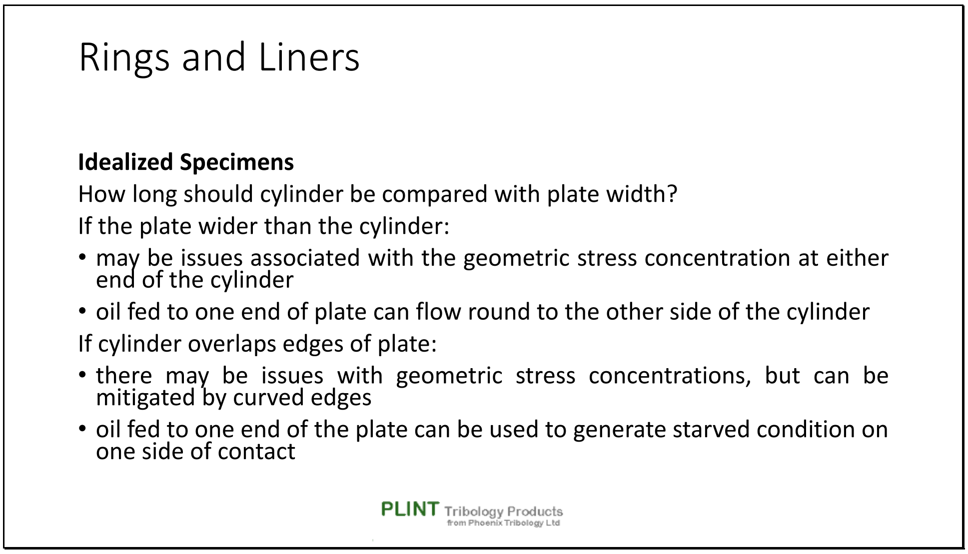

One of the questions to be addressed is how long the cylinder should be compared with the width of the plate.

If the plate is wider than the cylinder there may be issues associated with the geometric stress concentration at either end of the cylinder.

Oil fed to one end of the plate can flow round to the other side of the cylinder.

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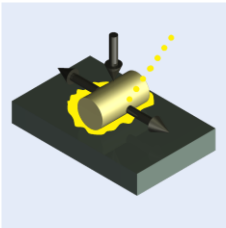
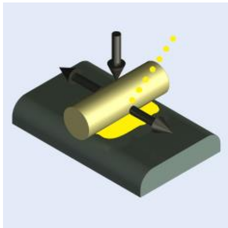
Rings and Liners

Idealized Specimens





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If the cylinder overlaps the edges of the plate, there may be issues with geometric stress concentrations, but these can be mitigated by having curved edges.

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Rings and Liners

Idealized Specimens

Curved edges means that edges of the plate do not have to be parallel, making wedge shaped specimen possible, giving stroke-wise pressure variation, hence different wear patterns

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Having curved edges means that the edges of the plate do not have to be parallel, making a wedge shaped specimen possible, such that there is stroke-wise pressure variation.

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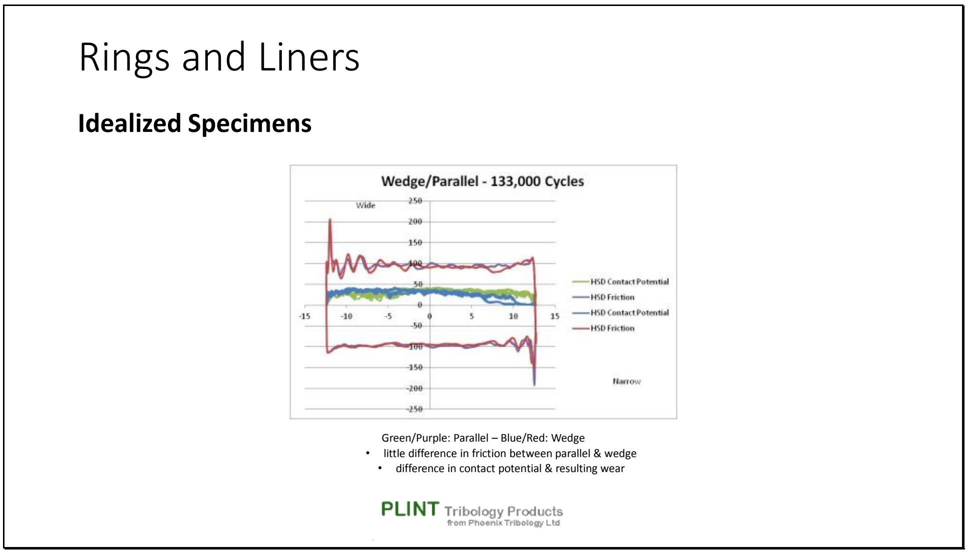

There is very little difference in friction between parallel and wedge shaped specimens, running under identical conditions. There is however a difference in contact potential and the resulting wear.

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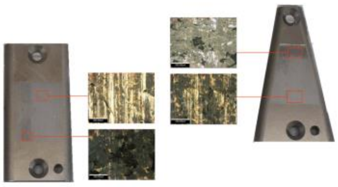
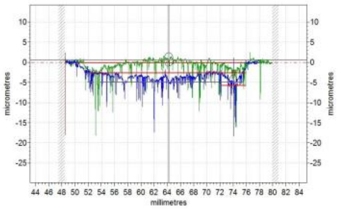
Rings and Liners

Idealized Specimens – Different Wear Patterns





Green – Parallel / Blue - Wedge

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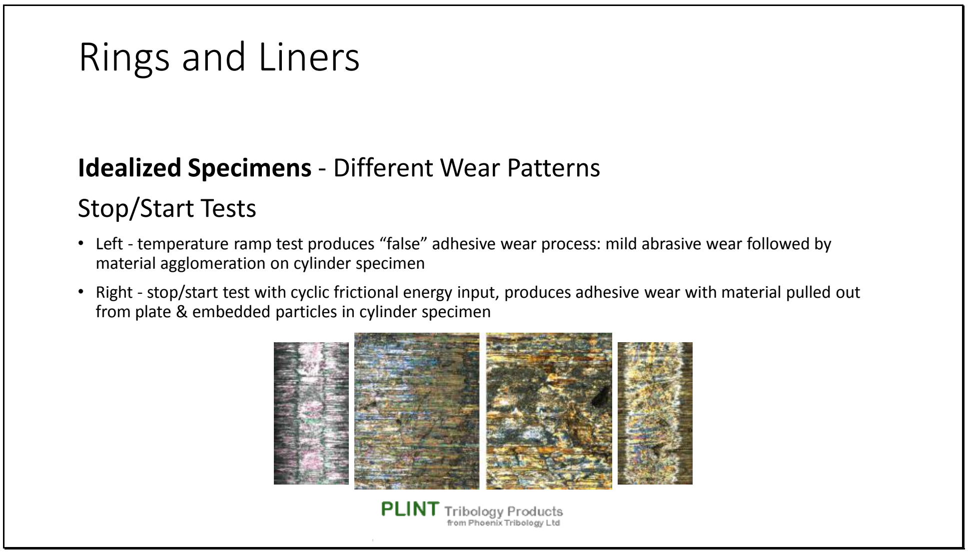

Different test procedures ca generate very different wear mechanisms. In this example, a temperature ramp test produces what one might term a "false" adhesive wear process; this is in fact mild abrasive wear followed by material agglomeration on the cylinder specimen.

A stop/start test using identical specimens and lubricant, with the temperature gradient the right way round and cyclic frictional energy input, produces adhesive wear with material pulled out from the plate surface and embedded in the cylinder specimen.

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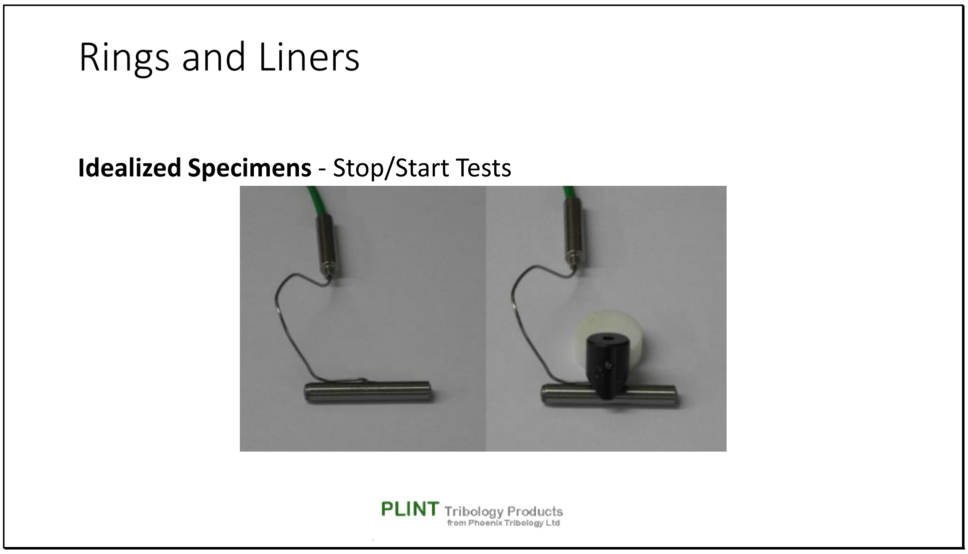

Inserting a thermocouple in the cylinder specimen allows the difference in temperature between the cylinder and the plate to be measured during the stop-start cycle.

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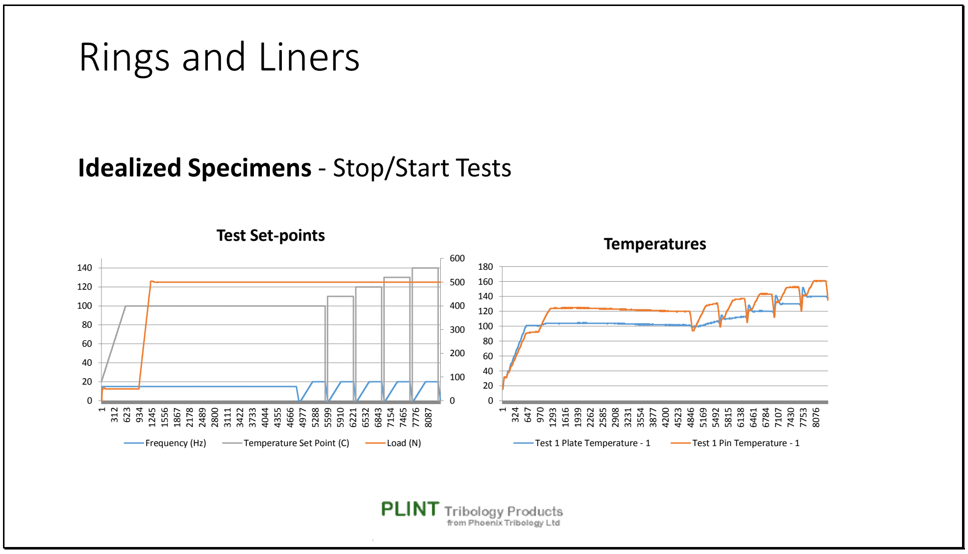

In this test, the plate specimen temperature is controlled and the speed cycled. At rest, the temperature of the cylinder falls below the temperature of the plate. When motion starts, frictional heating causes the plate temperature to rise a few degrees above the temperature set-point, however, in due course, the cylinder temperature rises substantially higher. When motion stops, the cylinder temperature falls back below the plate temperature and the cycle can be repeated.

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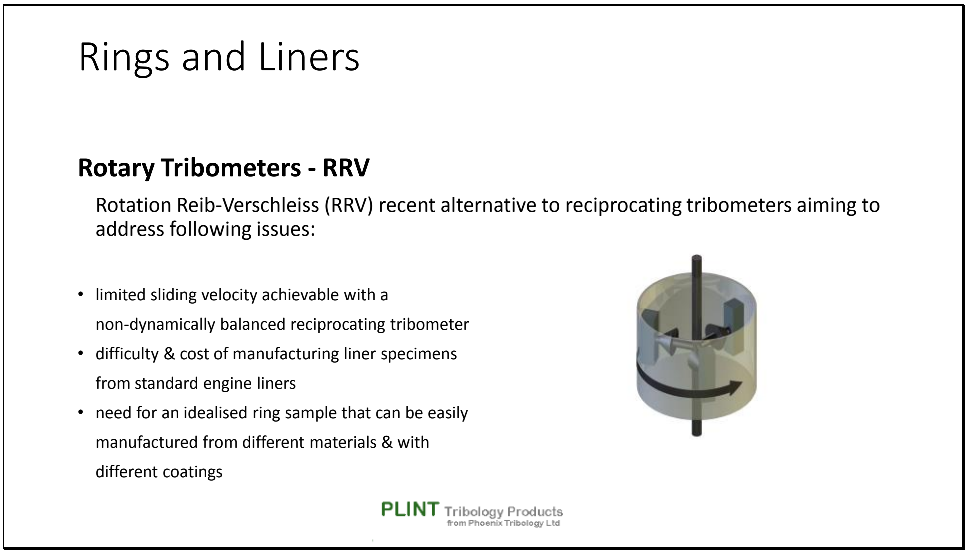

The Rotation Reib-Verschleiss (RRV) test rig is a more recent alternative to the assorted reciprocating tribometers and aims to address a number of issues including:

- The limited sliding velocity achievable with a non-dynamically balanced reciprocating tribometer

- The difficulty and cost of manufacturing liner specimens from standard engine liners

- The need for an idealised ring sample that can be easily manufactured from different materials and with different coatings

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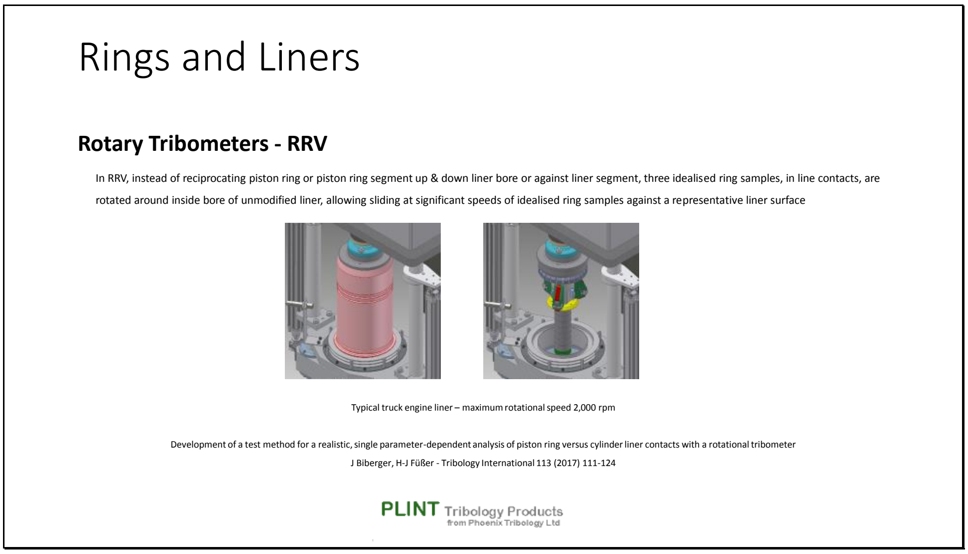

In the RRV, instead of reciprocating a piston ring or piston ring segment up and down a cylinder liner bore or against a liner segment, three idealised ring samples, in the form of line contacts, are rotated around the inside of the bore of an unmodified liner. The rig thus allows sliding at constant speed of idealised ring samples against a representative liner surface.

In practice, for tests using complete liners, it is preferable to have the ring sample carrier static and rotate the liner.

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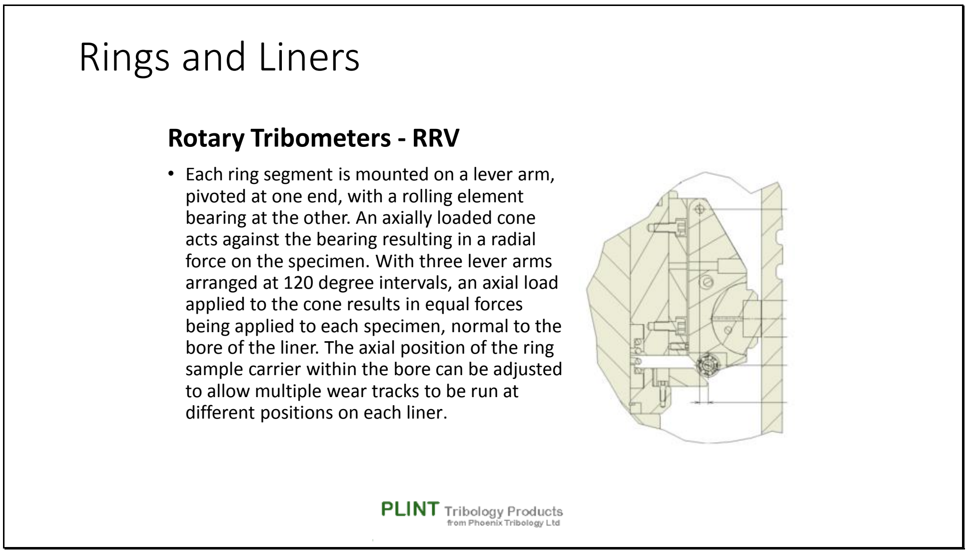

Each ring segments is mounted on a lever arm, pivoted at one end, with a rolling element bearing at the other. An axially loaded cone acts against the bearing resulting in a radial force on the specimen. With three lever arms arranged at 120 degree intervals, an axial load applied to the cone results in equal forces being applied to each specimen, normal to the bore of the liner.

The axial position of the ring sample carrier within the bore can be adjusted to allow multiple wear tracks to be run at different positions on each liner.

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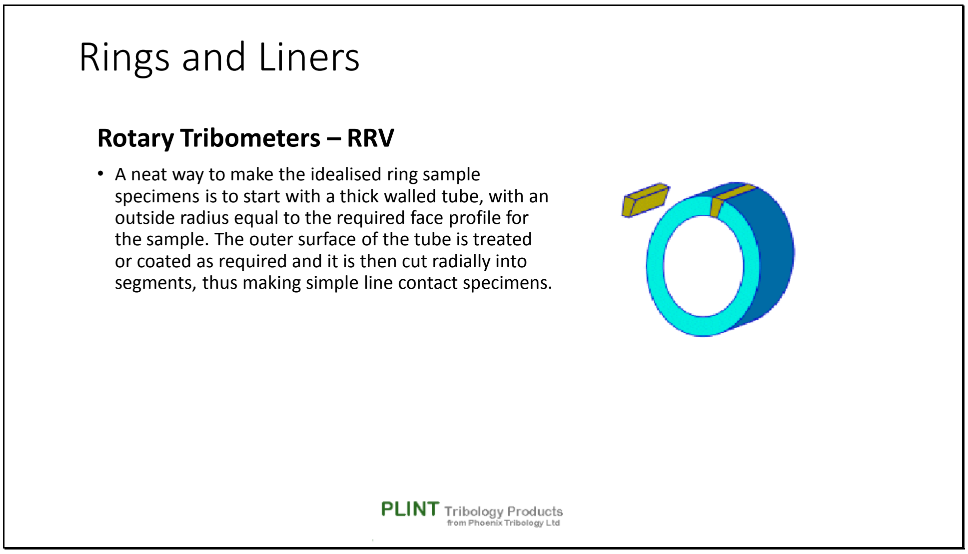
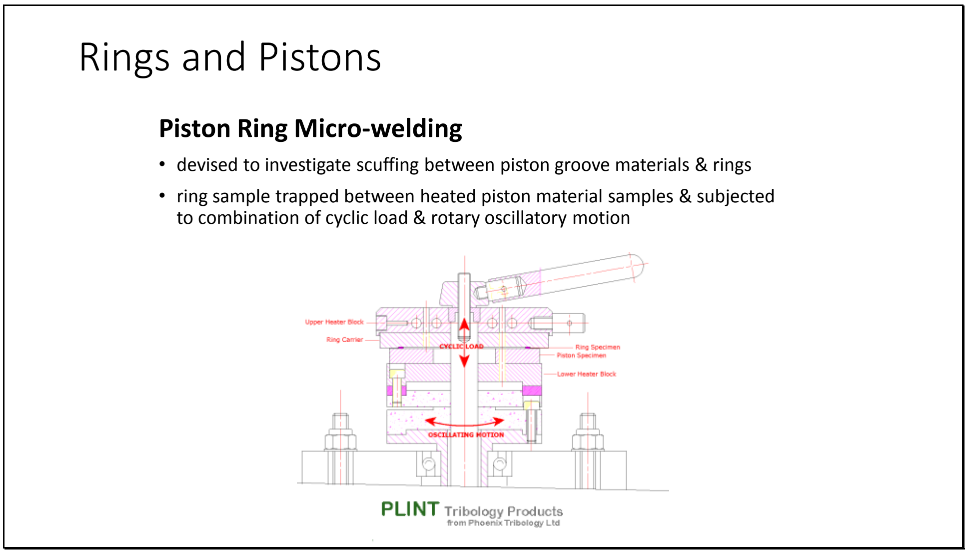


A neat way to make the idealised ring sample specimens is to start with a thick walled tube, with an outside radius equal to the required face profile for the sample. The outer surface of the tube is treated or coated as required and it is then cut radially into segments, thus making simple line contact specimens.

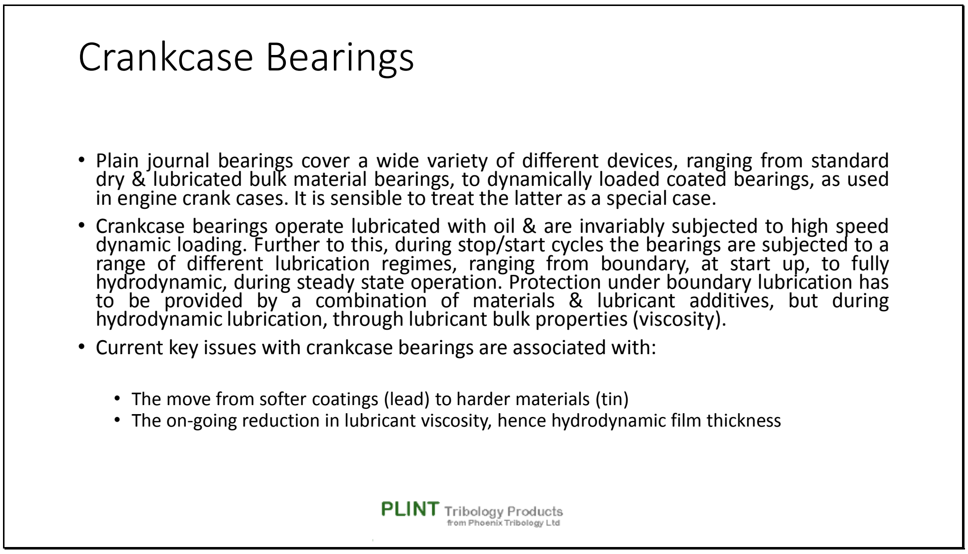
A derivative of the RRV, compatible with a standard rotary tribometer, uses a stationary section of cylinder liner and a rotating ring sample carrier. This arrangement works satisfactorily with smaller bore diameter liners. It does, of course, require the liner to be cut into short lengths, which can prove problematic.

Slide 40



The piston ring micro-welding test was devised to investigate scuffing between piston groove materials and rings. A ring sample is trapped between heated piston material samples and subjected to a combination of cyclic load and rotary oscillatory motion.

Slide 41



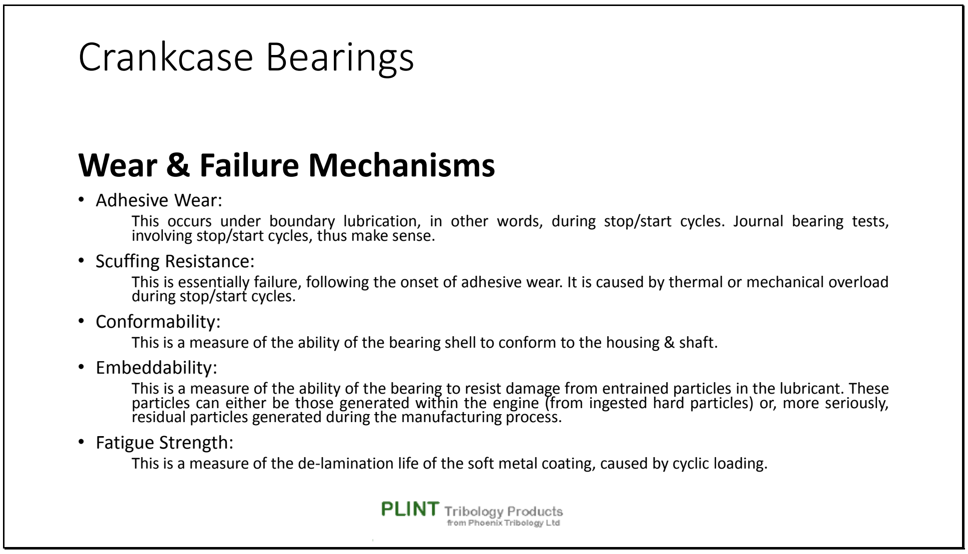
Plain journal bearings cover a wide variety of different devices, ranging from standard dry and lubricated bulk material bearings, to dynamically loaded coated bearings, as used in engine crank cases. It is sensible to treat the latter as a special case.

Crankcase bearings operate lubricated with oil and are invariably subjected to high speed dynamic loading. Further to this, during stop/start cycles the bearings are subjected to a range of different lubrication regimes, ranging from boundary, at start up, to (hopefully) fully hydrodynamic during steady state operation. Protection under boundary lubrication has to be provided by a combination of materials and lubricant additives, but during hydrodynamic lubrication, through lubricant bulk properties (viscosity).

Current key issues with crankcase bearings are associated with:

- The move from softer coatings (lead) to harder materials (tin).
- The on-going reduction in lubricant viscosity, hence hydrodynamic film thickness.

Slide 42



The kinds of wear and failure mechanisms endured by crankcase bearings include:

Adhesive Wear:

This occurs under boundary lubrication, in other words, during stop/start cycles. Journal bearing tests, involving stop/start cycles, thus make sense.

Scuffing Resistance:

This is essentially failure, following the onset of adhesive wear. It is caused by thermal or mechanical overload during stop/start cycles.

Conformability:

This is a measure of the ability of the bearing shell to conform to the housing and shaft.

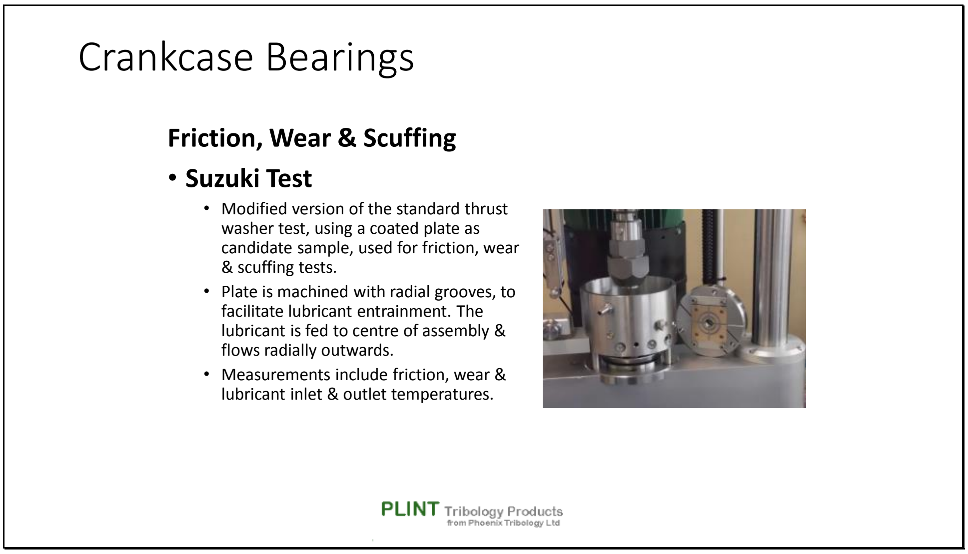
Embeddability:

This is a measure of the ability of the bearing to resist damage from entrained particles in the lubricant. These particles can either be those generated within the engine (from ingested hard particles) or, more seriously, residual particles generated during the manufacturing process.

Fatigue Strength:

This is a measure of the de-lamination life of the soft metal coating, caused by cyclic loading.

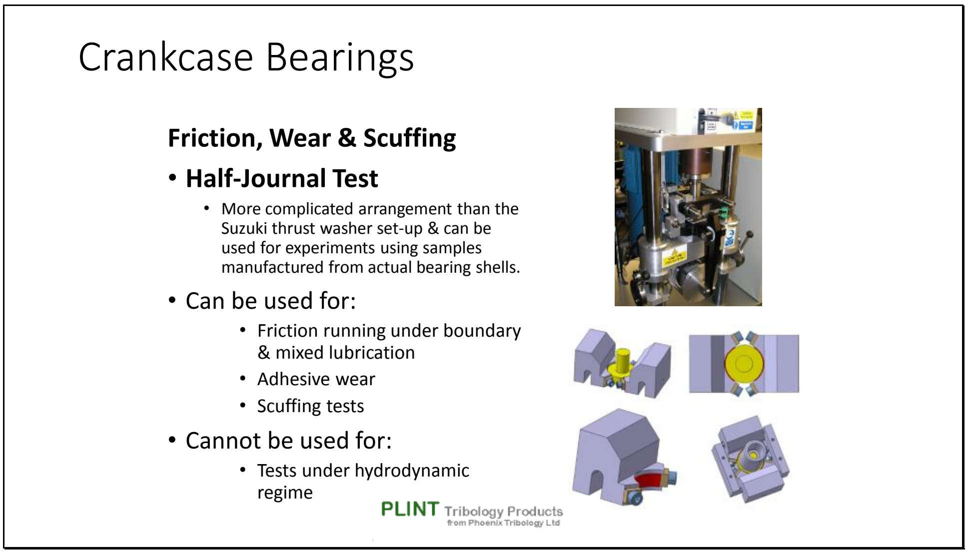
Slide 43



The Suzuki test is a modified version of the standard thrust washer test geometry, using a coated plate as the candidate sample. It is used for friction, wear and scuffing tests.

The plate is machined with radial grooves, to facilitate lubricant entrainment. The lubricant is fed to the centre of the assembly and flows radially outwards. In addition to measuring friction and sometimes wear, the difference in lubricant inlet and outlet temperature is a useful measure.

Slide 44



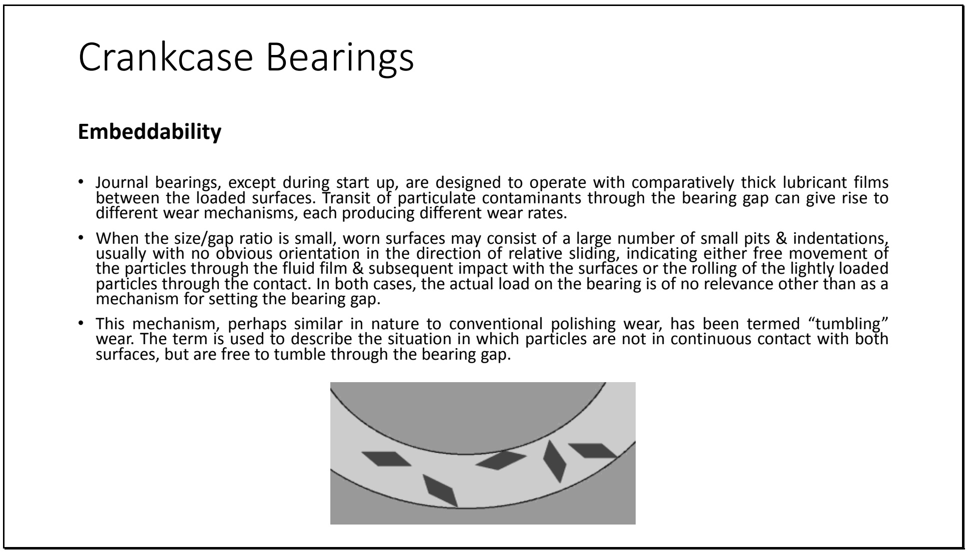
The half-journal test geometry is a more complicated arrangement than the Suzuki thrust washer set-up and can be used for experiments using samples manufactured from actual bearing shells.

This arrangement can be used for:

- Friction running under boundary and mixed lubrication
- Adhesive wear
- Scuffing tests

Despite the potential for rotating at high speeds, this geometry cannot be used for tests under hydrodynamic regimes, because it is not possible to duplicate the necessary lubricant entrainment conditions associated with a full journal bearing.

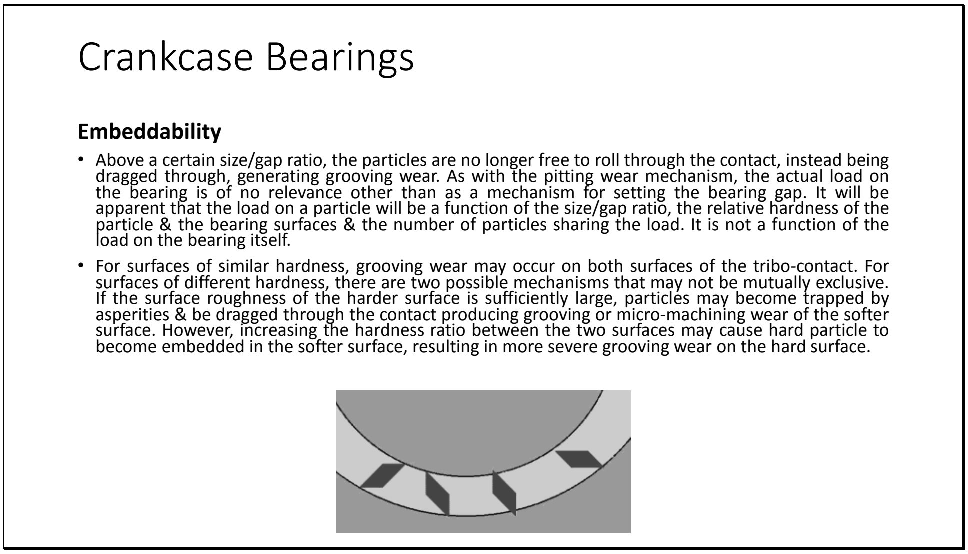
Slide 45



Journal bearings, except during start up, are essentially designed to operate with comparatively thick lubricant films between the loaded surfaces. Transit of particulate contaminants through the bearing gap can give rise to different wear mechanisms, each producing different wear rates.

When the size/gap ratio is small, worn surfaces may consist of a large number of small pits and indentations, usually with no obvious orientation in the direction of relative sliding, indicating either free movement of the particles through the fluid film and subsequent impact with the surfaces or the rolling of the lightly loaded particles through the contact. In both cases, the actual load on the bearing is of no relevance other than as a mechanism for setting the bearing gap. This mechanism, perhaps similar in nature to conventional polishing wear, has been termed “tumbling” wear. With polishing wear, we would expect the free particles to roll through the tribo-contact in continuous contact with both sides. The term tumbling is used to describe the situation in which particles are not in continuous contact with both surfaces, but are free to tumble through the bearing gap.

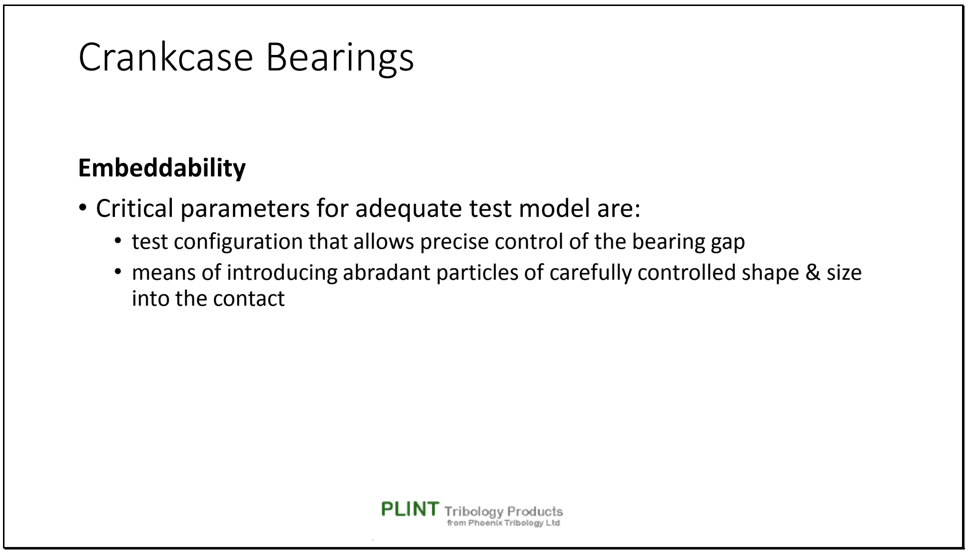
Slide 46



Above a certain size/gap ratio, the particles are no longer free to roll through the contact, instead being dragged through, generating grooving wear. As with the pitting wear mechanism, the actual load on the bearing is of no relevance other than as a mechanism for setting the bearing gap. It will be apparent that the load on a particle will be a function of the size/gap ratio, the relative hardness of the particle and the bearing surfaces and the number of particles sharing the load. It is not a function of the load on the bearing itself.

For surfaces of similar hardness, grooving wear may occur on both surfaces of the tribo-contact. For surfaces of different hardness, there are two possible mechanisms that may not be mutually exclusive. If the surface roughness of the harder surface is sufficiently large, particles may become trapped by asperities and be dragged through the contact producing grooving or micro-machining wear of the softer surface. However, increasing the hardness ratio between the two surfaces may cause hard particle to become embedded in the softer surface, resulting in more severe grooving wear on the hard surface.

Slide 47



The critical parameters for an adequate test model are therefore:

- A test configuration that allows precise control of the bearing gap
- A means of introducing abradant particles of carefully controlled shape and size into the contact

It will be noted that load (either static or dynamic) is not considered of importance except in as much as it may provide a mechanism for setting the bearing gap.

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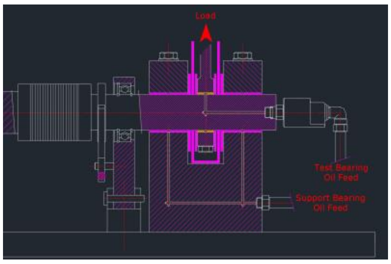
Crankcase Bearings

Embeddability - Possible Test Systems



Difficult to control total mass, size & angularity of particles passing through contact

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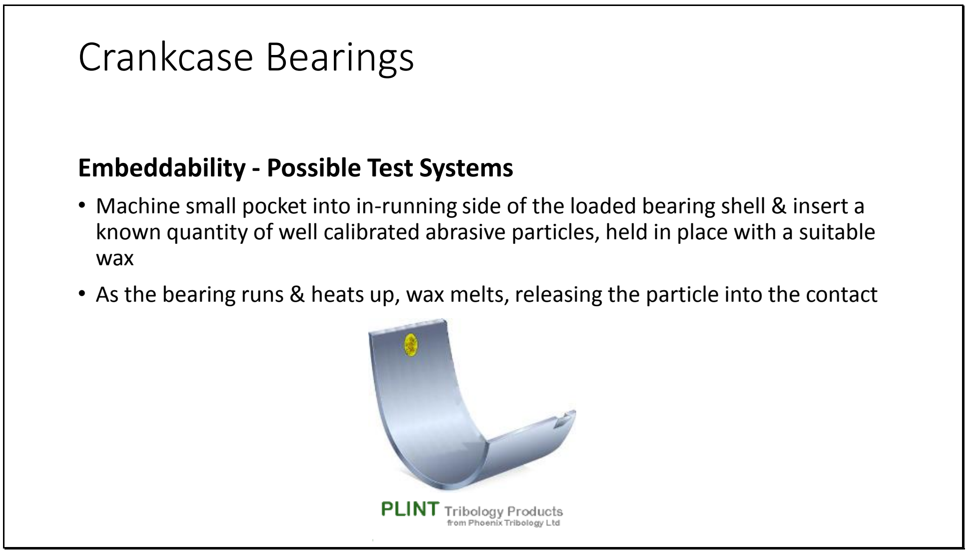
The key challenge is the injection of particles in a controlled fashion into the test bearing. As with any tests involving abrasive particles, there are uncertainties associated with how long the particles remain in the contact and avoid comminution.

Ideally, one would like to control the total mass, size and angularity of the particles passing through the contact.

Slide 49



One way of achieving this is to machine a small pocket into the in-running side of the loaded bearing shell and insert a known quantity of well calibrated abrasive particles, held in place with a suitable wax.

As the bearing runs and heats up, the wax melts, releasing the particles into the contact.

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Crankcase Bearings

Fatigue Strength - Cyclic Load Test Rigs

- Rigs can be divided into two basic categories:
  - Machines Applying Pulsating Alternating Loads - Full Wave Load Cycle
  - Machines Applying Pulsating Loads Between Zero & Maximum in one direction –
    Half Wave Load Cycle
- Dynamic loads generated mechanically, hydraulically, servo-hydraulically & by resonance pulsator

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Fatigue strength cyclic load test rigs can be divided into two basic categories:

- Machines Applying Pulsating Alternating Loads - Full Wave Load Cycle
- Machines Applying Pulsating Loads Between Zero and Maximum in one direction - Half Wave Load Cycle

Dynamic loads can be generated mechanically, hydraulically, servo-hydraulically and by resonance pulsator.

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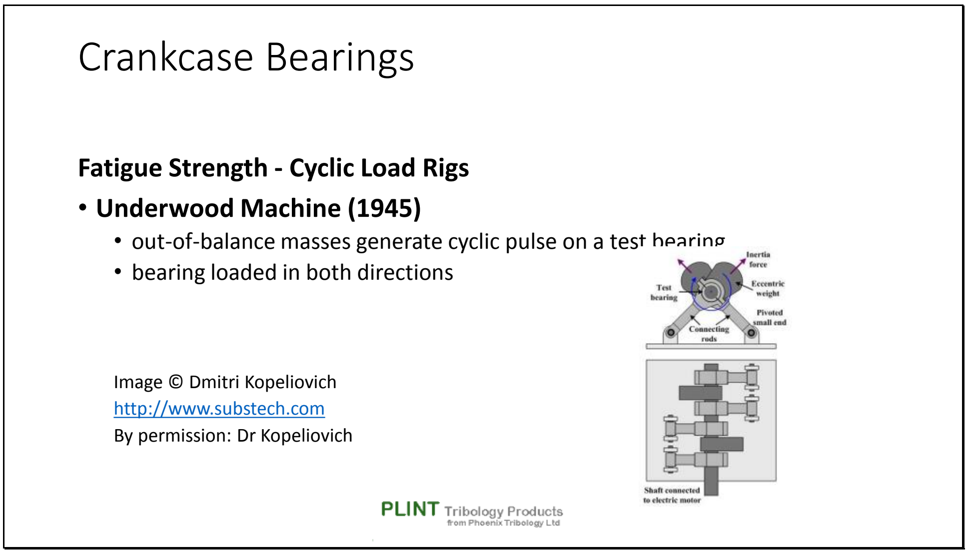
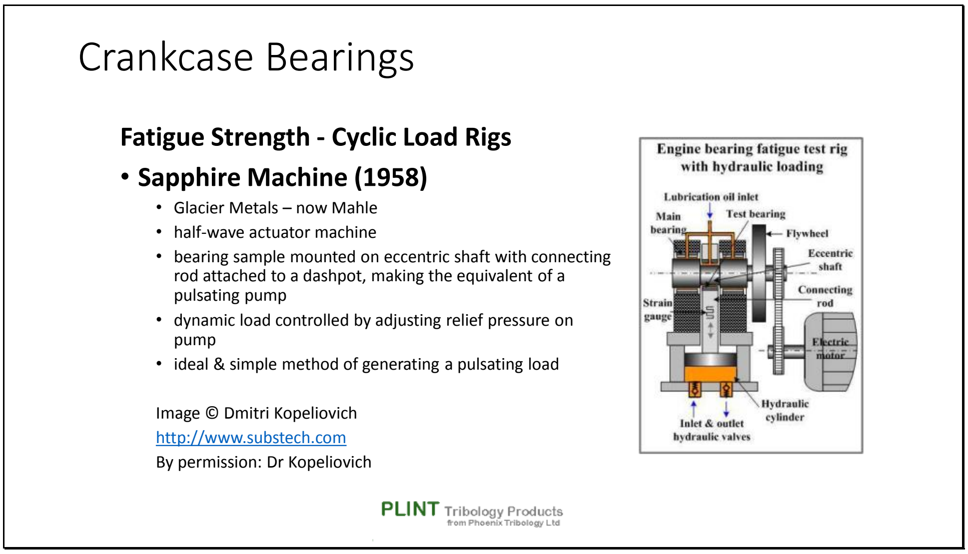


This machine, which is still used in various guises, involves out-of-balance masses generating a cyclic pulse on a test bearing. The bearing is loaded in both directions.

Slide 52



This Sapphire machine is a half-wave actuator machine. A journal bearing sample is mounted on an eccentric shaft with a connecting rod attached to the equivalent of a dashpot, making the equivalent of a badly pulsing pump. This is however an ideal and simple method of generating a pulsating load.

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Crankcase Bearings

Fatigue Strength - Cyclic Load Rigs

- **Pulsator Rigs**
  - Resonant frequency devices: electromagnetic drive is used to excite a specimen assembly at its natural frequency, which, for a given system stiffness, is adjusted by varying the mass of the assembly.
  - Although the dynamic load cannot be directly controlled in real-time, pulsators are a very reliable, energy efficient, but less flexible alternative to a servo-hydraulic actuator based test system.
  - Pulsators tend to operate at much higher frequencies than servo-hydraulic systems.
  - Current suppliers of pulsators include ZwickRoell (Vibrophore) & SincoTec (Universal Resonance Pulsator).

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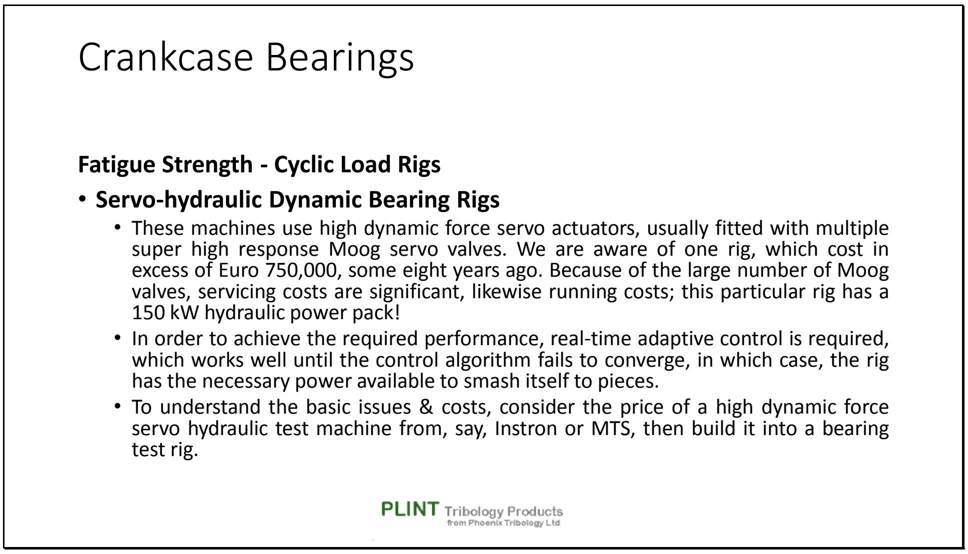
Puslator rigs are resonant frequency devices. An electromagnetic drive is used to excite a specimen assembly at its natural frequency, which, for a given system stiffness, is adjusted by varying the mass of the assembly.

Although the dynamic load cannot be directly controlled in real-time, pulsators are a very reliable, energy efficient, but less flexible alternative to a servo-hydraulic actuator based test system. Furthermore, pulsators tend to operate at much higher frequencies than servo-hydraulic systems.

Current suppliers of pulsators include ZwickRoell (Vibrophore) and SincoTec (Universal Resonance Pulsator)

Slide 54



Servo-hydraulic dynamic bearings rigs use high dynamic force servo actuators, usually fitted with multiple super high response Moog servo valves. We are aware of one single actuator rig, which cost in excess of Euro 750,000, some eight years ago. Because of the large number of Moog valves, servicing costs are significant, likewise running costs; this particular rig has a 150 kW hydraulic power pack!

In order to achieve the required performance, real-time adaptive control is required, which works well until the control algorithm fails to converge, in which case, the rig has the necessary power available to smash itself to pieces.

To understand the basic issues and costs, consider the price of a high dynamic force servo hydraulic test machine from, say, Instron or MTS, then build it into a bearing test rig.

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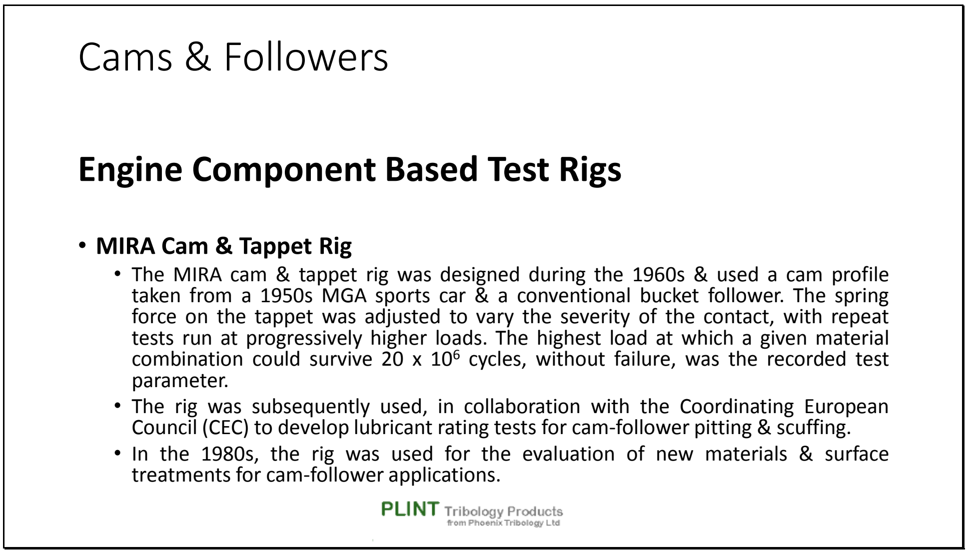

The MIRA cam and tappet rig was designed during the 1960s and used a cam profile taken from a 1950s MGA sports car and a conventional bucket follower. The spring force on the tappet was adjusted to vary the severity of the contact, with repeat tests run at progressively higher loads. The highest load at which a given material combination could survive 20 x 10^6 cycles, without failure, was the recorded test parameter.

The rig was subsequently used, in collaboration with the Coordinating European Council (CEC) to develop lubricant rating tests for cam-follower pitting and scuffing.

In the 1980s, the rig was used for the evaluation of new materials and surface treatments for cam-follower applications.

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Cams & Followers

Engine Component Based Test Rigs

- **VW-PV-5106 Cam & Tappet Rig**
  - Modern equivalent of the MIRA rig.

Image © KST Motorenversuch GmbH & Co KG

A modern equivalent of the MIRA rig is the VW-PV-5106 Cam and Tappet Rig.

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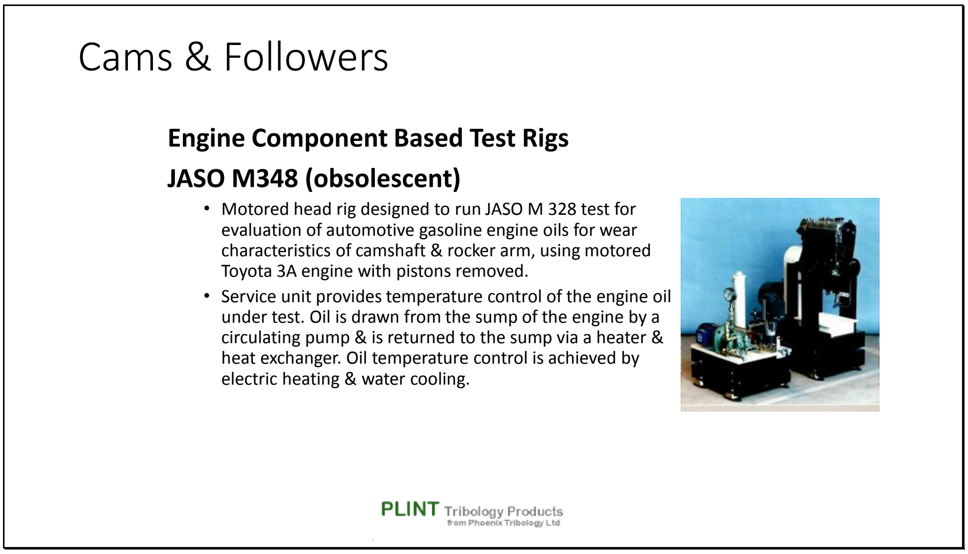

This test rig is designed to run the standard JASO M 328 test for the evaluation of automotive gasoline engine oils for wear characteristics of camshaft and rocker arm. In this configuration the rig is fitted with a Toyota 3A engine. The rig comprises three modules, the engine unit, the service unit and the control unit.

The service unit provides temperature control of the engine oil under test. Oil is drawn from the sump of the engine by a circulating pump and is returned to the sump via a heater and heat exchanger. Oil temperature control is achieved by electric heating and water cooling.

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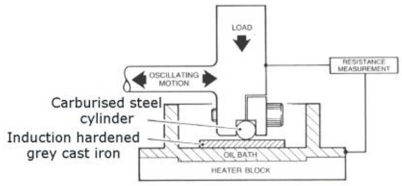
Cams & Followers

Idealised Test Geometries

- **Reciprocating**
  - Simple reciprocating tests using appropriate materials have been used for screening lubricant additives, pre-engine test or more sophisticated rig tests; the latter include those involving sliding-rolling contacts.



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Simple reciprocating tests using appropriate materials have been used for screening lubricant additives, pre-engine test or more sophisticated rig tests; the latter including those involving sliding-rolling contacts.

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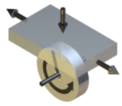
Cams & Followers

Idealised Test Geometries

- **Sliding-rolling**
  - **Asymmetric slide-roll arrangement – Reciprocating Amsler**
    - Involves plate reciprocating on rotating roller giving asymmetric variation in entrainment velocity, similar to that in a cam & tappet contact
    - Claimed correlation with various engine tests, however, minimal information in public domain to support this



Review & evaluation of lubricated wear in simulated valve train contact conditions - GW Roper & JC Bell

SAE952473, Fuels & Lubes Meeting & Exposition, Toronto, Ontario, 16-19 Oct 1995 - SAE Publication: “Recent snapshots & insights into lubricant tribology”, SP-1116, pp67-83

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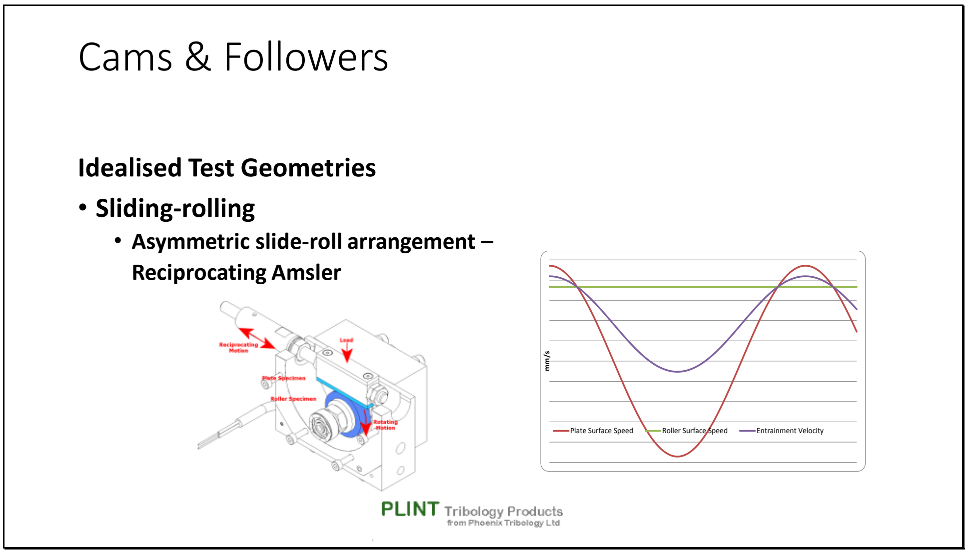
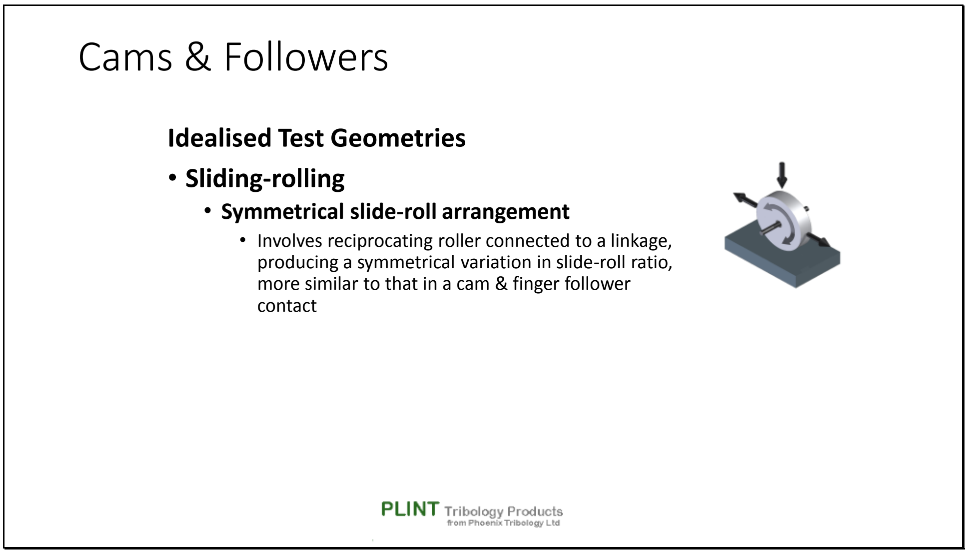
Two such tests have been extensively used, with claimed correlation with various engine tests, however, there is minimal information in the public domain to support this claim.

Slide 60



The first such rig was the “reciprocating Amsler”, which involves a plate reciprocating on a rotating roller. This gives an asymmetric variation in entrainment velocity, similar to that in a cam and tappet contact.

Slide 61



An alternative arrangement, involving a reciprocating roller connected to a linkage, produces a symmetrical variation in slide-roll ratio, more similar to that in a cam and finger follower contact.

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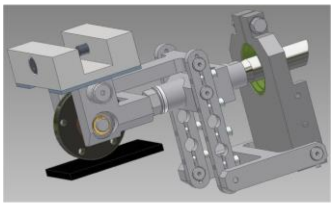
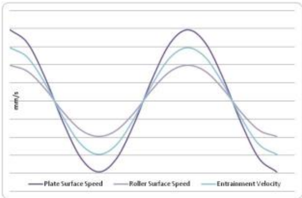
Cams & Followers

Idealised Test Geometries

- **Sliding-rolling**
  - **Symmetrical slide-roll arrangement**





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With all these idealized test geometries, the correct materials must be used.

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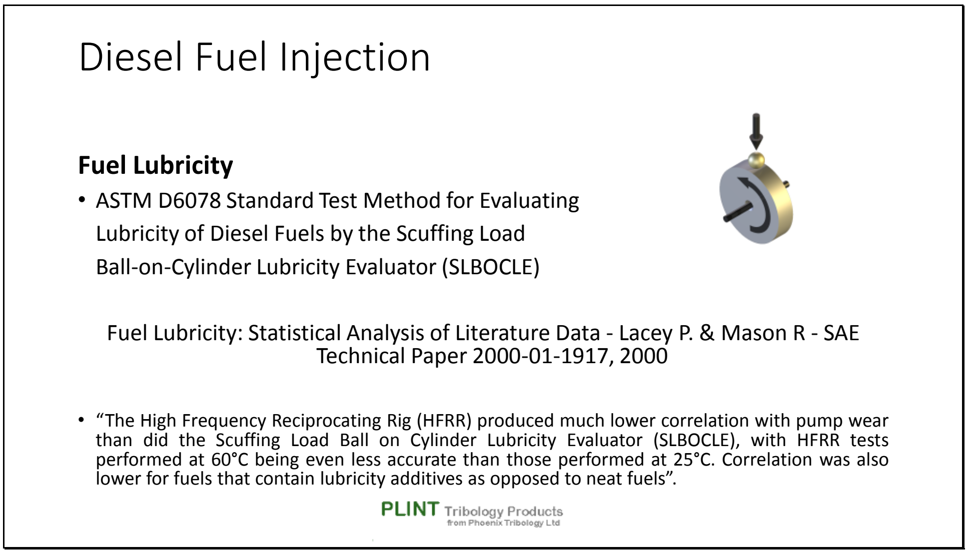

Various diesel full lubricity tests exist. These, however, do not use representative materials for test samples.

Methods include ASTM D6078 Standard Test Method for Evaluating Lubricity of Diesel Fuels by the Scuffing Load Ball-on-Cylinder Lubricity Evaluator (SLBOCLE)

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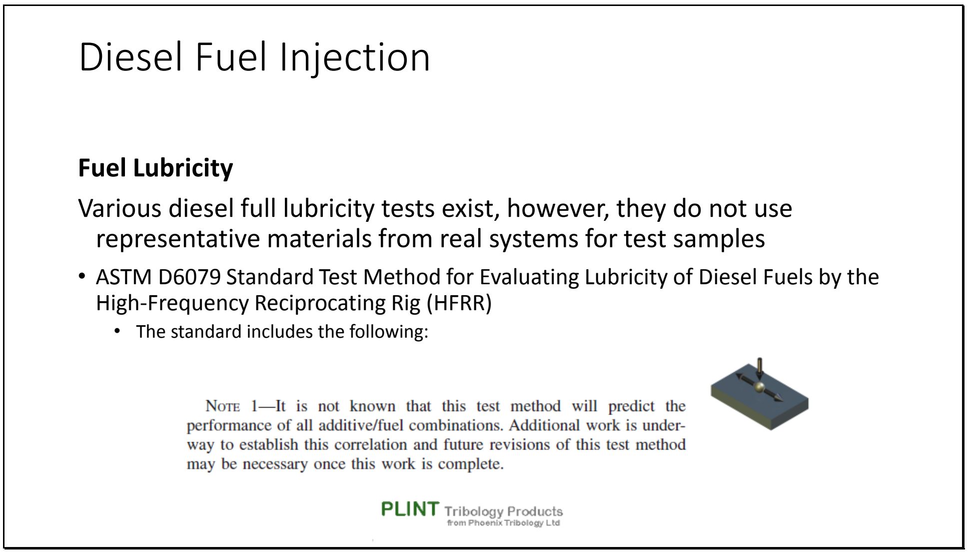

And ASTM D6079 Standard Test Method for Evaluating Lubricity of Diesel Fuels by the High-Frequency Reciprocating Rig (HFRR)

The standard includes the following:

Note: It is not known that this test method will predict the performance of all additive/fuel combinations. Additional work is underway to establish this correlation and future revisions of this test method may be necessary once this work is complete.

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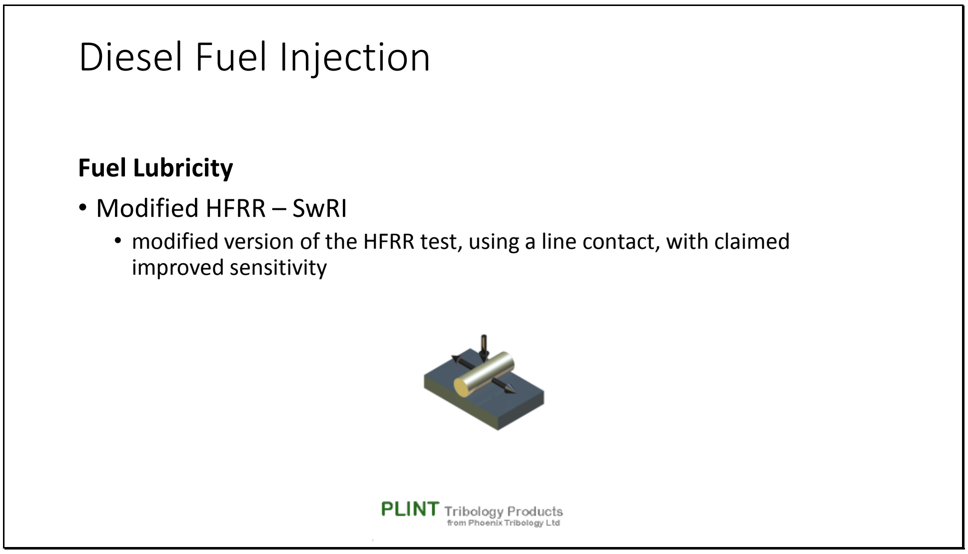
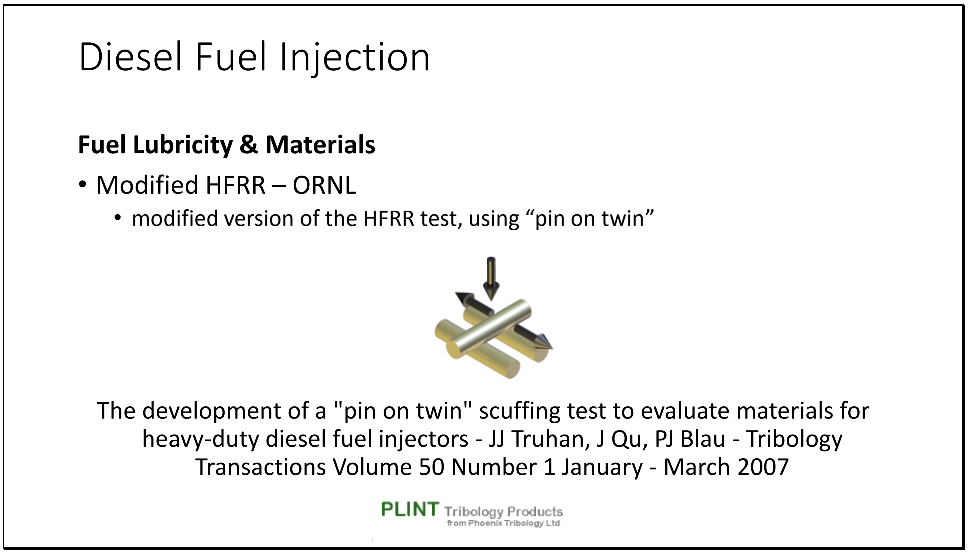


SwRI have been working on a modified version of the HFRR test, using a line contact, with claimed improved sensitivity.

Slide 66



ORNL devised an alternative reciprocating test geometry, the “pin on twin”.

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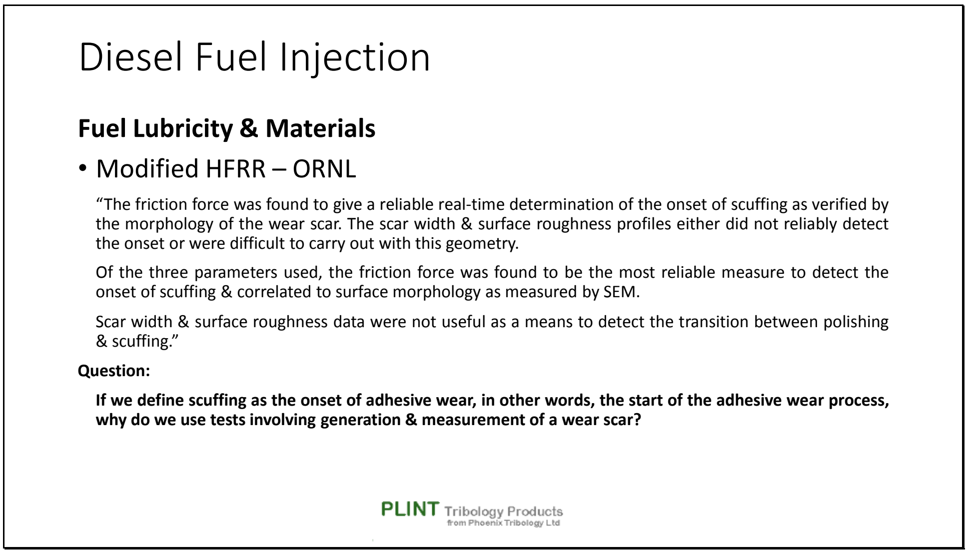

This raises an interesting point: if we define scuffing as the onset of adhesive wear, in other words, the start of the adhesive wear process, why do we use tests involving generation and measurement of a wear scar?

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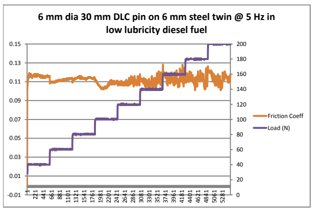
Diesel Fuel Injection

Fuel Lubricity & Materials

- Pin on twin geometry has been used for testing DLC coated pins, with diesel fuels
- Coatings tend to fail, not wear



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The pin on twin geometry has been used for testing DLC coated pins, with diesel fuels. The coatings tend to fail, not wear.

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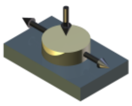
Diesel Fuel Injection

Fuel Lubricity & Materials

- DLC coated specimens have also been tested using reciprocating area contacts, with similar dimensions, loads & motions to those in actual pumps
- The arrangement achieves similar results to more complicated, pump based rigs



DLC coated specimens have also been tested using reciprocating area contacts, with similar dimensions, loads and motions to those in actual pumps.

This arrangement achieves similar results to more complicated, pump based rigs.

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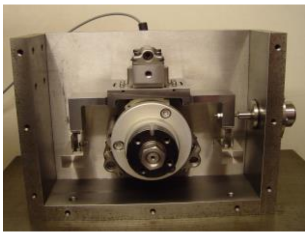
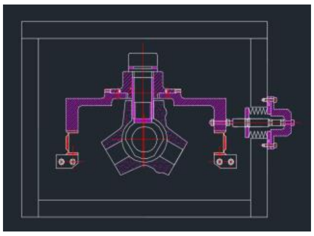
Diesel Fuel Injection

Fuel Lubricity & Materials

- Floating cylinder pump rig





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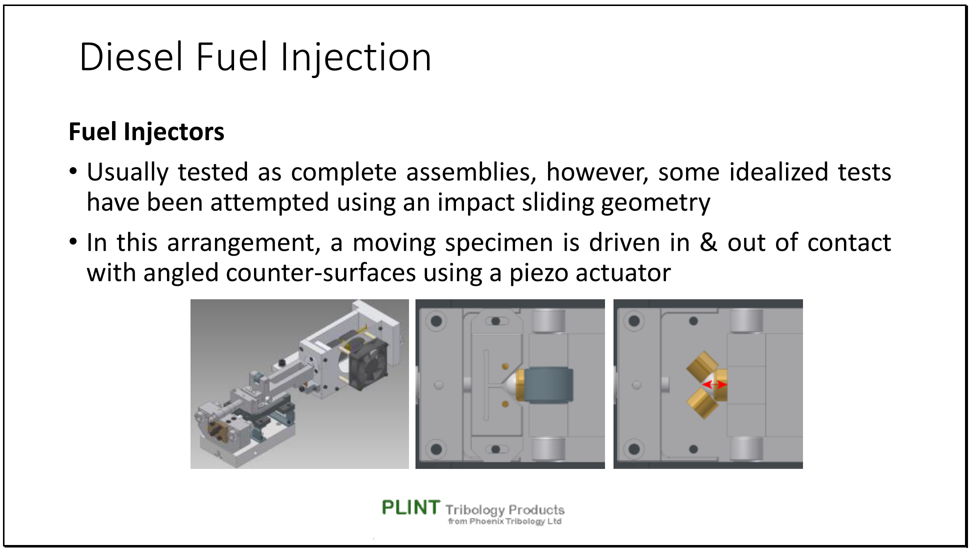
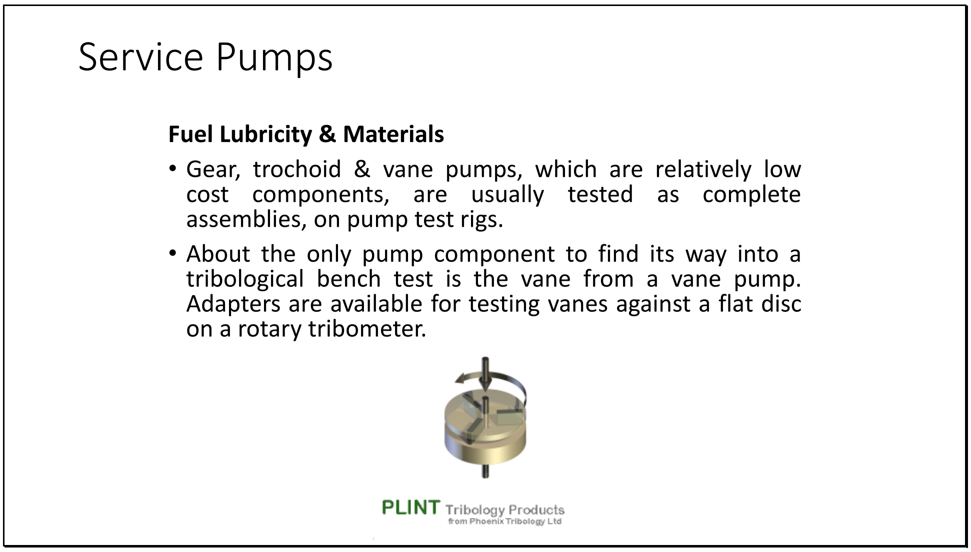


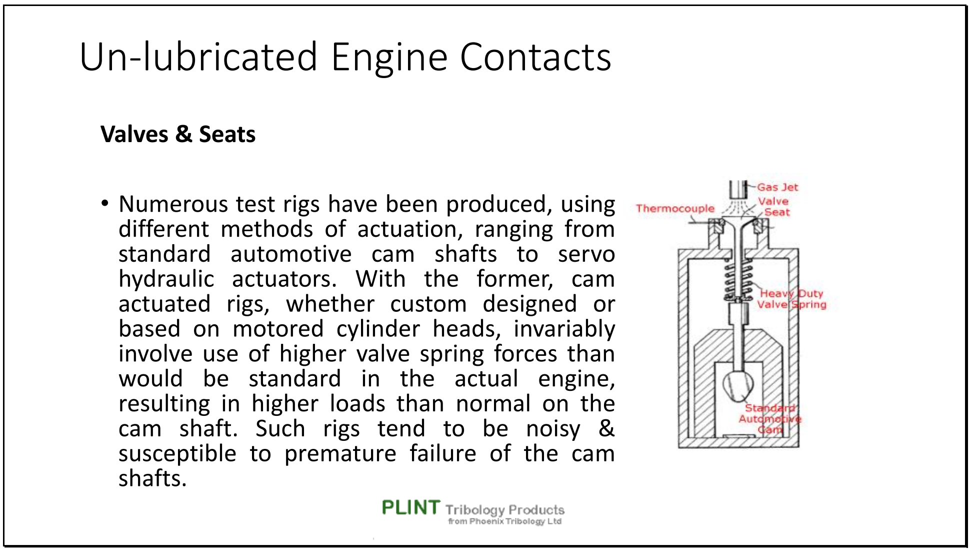
Fuel injectors are usually tested as complete assemblies, however, some idealized tests have been attempted using an impact sliding geometry. In this arrangement, a moving specimen is driven in and out of contact with angle counter-surfaces, using a piezo actuator.

Slide 72



Gear, trochoid and vane pumps, which are relatively low cost components, are usually tested as complete assemblies, on pump test rigs. About the only pump component to find its way into a tribological bench test is the vane from a vane pump. Adapters are available for testing vanes against a flat disc on a rotary tribometer.

Slide 73



Not all components in engines are lubricated. Un-lubricated engine contacts include:

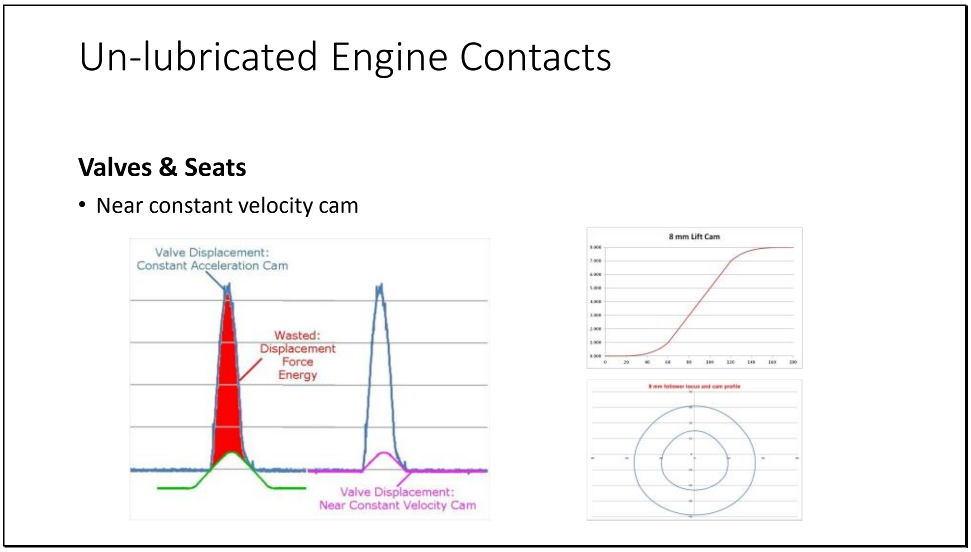
- Valve and seats– impact sliding
- Valve and seats – valve head flexing
- Turbocharger guide vanes – severe oxidative wear

For a comprehensive review of “valve recession”, the wear of the valve-seat interface, plus associated test rigs, read the following:

Automotive Engine Valve Recession. Roger Lewis & Rob Dwyer-Joyce - Wiley-Blackwell (23 November 2001) ISBN-10: 186058358X - ISBN-13: 978-1860583582

Numerous different test rigs have been produced, using a range of different methods of actuation, ranging from standard automotive cam shafts to servo hydraulic actuators. With regard to the former, cam actuated rigs, whether custom designed or based on motored cylinder heads, almost invariably involve the use of higher valve spring forces than would be standard in an actual engine, resulting in higher load than normal on the cam shaft. Such rigs tend to be very noisy and susceptible to premature failure of the cam shafts.

Slide 74



Lewis and Dwyer-Joyce came to the conclusion that the key parameter affecting valve recession was the velocity of the valve at the point of impact with the seat.

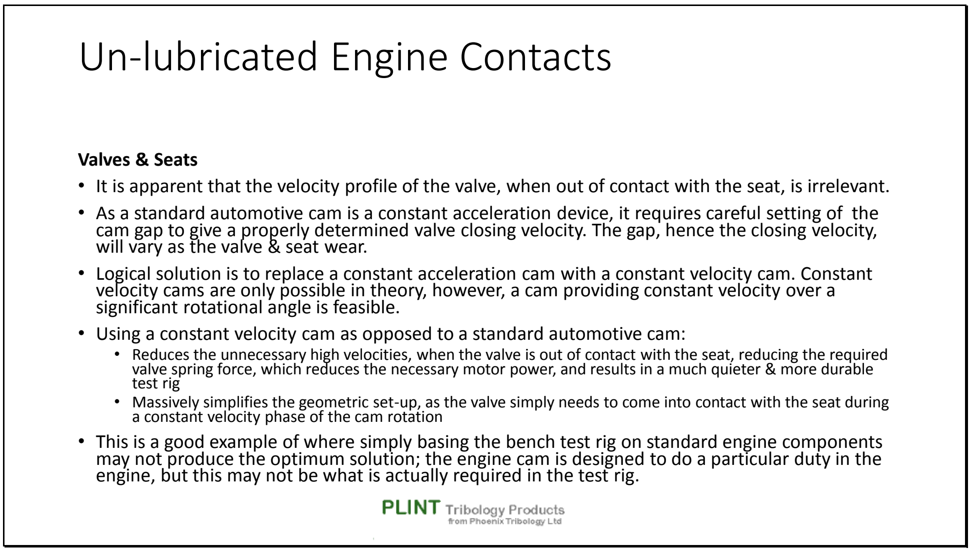
Following this thought to the logical conclusion, it is apparent that the velocity profile of the valve, when substantially out of contact with the seat, is irrelevant. Furthermore, as a standard automotive cam is more or less a constant acceleration device, it requires careful setting of the cam gap to give a properly determined valve closing velocity. This gap, hence the closing velocity, will vary as the cam and seat wear.

The logical solution is to replace a constant acceleration cam with a constant velocity cam. Constant velocity cams are of course only possible in theory, however, a cam providing constant velocity over a significant rotational angle is perfectly feasible.

Using a constant velocity cam as opposed to a standard automotive cam:

> Reduces the unnecessary high velocities, when the valve is out of contact with the seat, which:
>
> > reduces the required valve spring force, which reduces the necessary motor power, and:
> >
> > > results in a much quieter and more durable test rig

Slide 75



It also massively simplifies the geometric set-up, as the valve simply needs to come into contact with the seat during a constant velocity phase of the cam.

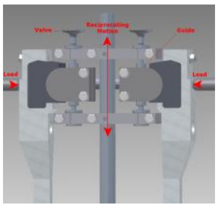
This is a good example of where simply basing the bench test rig on standard engine components may not produce the optimum solution; the engine cam is designed to do a particular duty in the engine, but this may not be what is actually required in the test rig.

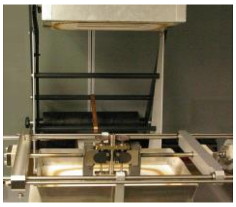
Slide 76

Un-lubricated Engine Contacts

Valve Guides

- Valve guide rigs usually involve reciprocating a valve stem, loaded on one side of a valve guide, within a furnace





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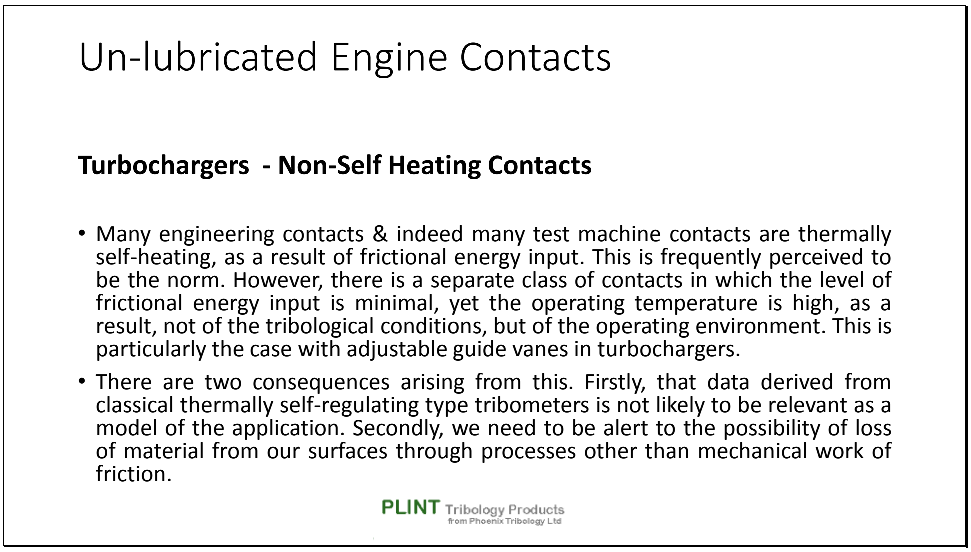
Valve guide rigs usually involve reciprocating a valve stem loaded on one side of a standard valve guide, within a furnace.

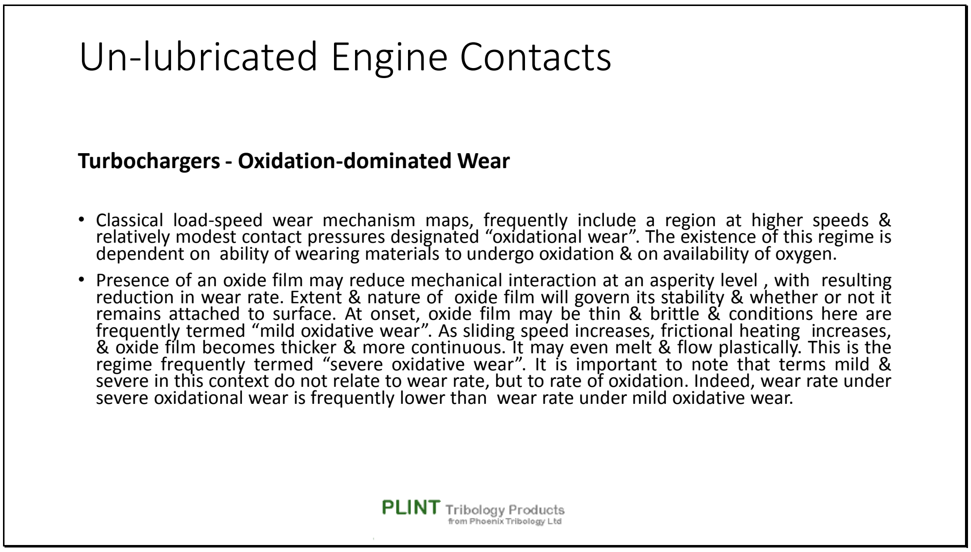
Slide 77



Many engineering contacts and indeed many test machine contacts are thermally self-heating, as a result of frictional energy input. This is frequently perceived to be the norm. However, there is a separate class of contacts in which the level of frictional energy input is minimal, yet the operating temperature is high, as a result, not of the tribological conditions, but of the operating environment. This is particularly the case with adjustable guide vanes in turbochargers.

There are two consequences arising from this. Firstly, that data derived from classical thermally self-regulating type tribometers is not likely to be relevant as a model of the application. Secondly, we need to be alert to the possibility of loss of material from our surfaces through processes other than mechanical work of friction.

Slide 78



A classical load-speed wear mechanism map, will frequently include a region at higher speeds and relatively modest contact pressures designated "oxidational wear". The existence of this regime is dependent on the ability of the wearing materials to undergo oxidation and on the availability of oxygen.

The presence of an oxide film may reduce the mechanical interaction at an asperity level in the contact, with a resulting reduction in wear rate. The extent and nature of the oxide film will govern its stability and whether or not it remains attached to the surface. At onset, the oxide film may be thin and brittle and the conditions here are frequently termed "mild oxidative wear". As the sliding speed is increased, the frictional heating in the contact increases, and the oxide film becomes thicker and more continuous. It may even melt and flow plastically. This is the regime frequently termed "severe oxidative wear". It is important to note that the terms mild and severe in this context do not relate to the wear rate, but to the rate of oxidation. Indeed, the wear rate under severe oxidational wear is frequently lower than the wear rate under mild oxidative wear.

Slide 79

Un-lubricated Engine Contacts

Turbochargers - Oxidation-dominated Wear

- Many of contacts of interest in turbochargers involve formation & failure of beneficial oxide films, but these, instead of being generated by the classical mechanisms for oxidative wear described in the previous slide, are generated by external temperature in the local environment. It follows that wear rates may well be higher the lower the operating temperature & that thermal cycling may result in repeated formation & failure of beneficial oxide films; as the components cool, for example, during engine braking, the oxide layers crack & spall off leaving surfaces poorly protected. It follows that tests to model this type of wear & failure mechanism should use low frictional energy input contacts & externally generated temperature cycling.

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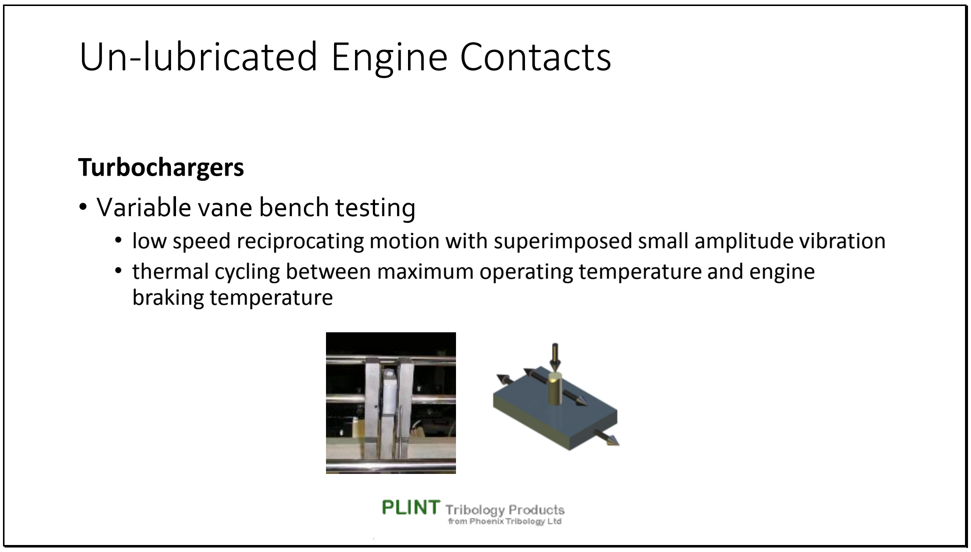
Many of the contacts of interest in turbochargers will involve the formation and failure of beneficial oxide films, but these, instead of being generated by the classical mechanisms for oxidative wear described above, are generated by the external temperature in the local environment.

Slide 80



It follows that wear rates may well be higher the lower the operating temperature and that thermal cycling may result in repeated formation and failure of beneficial oxide films; as the components cool, for example, during engine braking, the oxide layers crack and spall off leaving surfaces poorly protected. It follows that tests to model this type of wear and failure mechanism should use low frictional energy input contacts and externally generated temperature cycling.

Slide 81

Hybrids and EVs

Pure EVs:

Goodbye to:

- Piston rings and cylinder liners
- Piston skirts and cylinder liners
- Cams and followers
- Cams and tappets
- Valve stems and guides
- Crankshaft bearings
- Little end bearings

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The automotive internal combustion engine has kept tribologists busy for many years now. It is important to consider what will happen with the accelerating shift towards electric vehicles and hybrids. Many significant tribological contacts will cease to be that important. Bear in mind how quickly the reciprocating aero-engine disappeared at the end of the second world war with the advent of the gas turbine.

Slide 82

Hybrids and EVs

Hybrids:

Hello to:

- Low cost, lightly stressed, gasoline engines

Goodbye to:

- Cars with diesel engines

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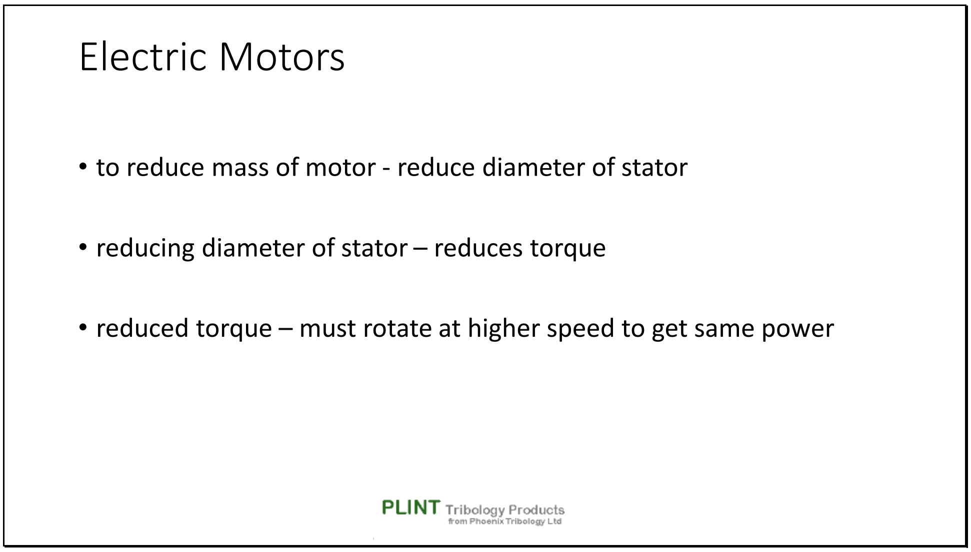
With the advent of hybrids, we would anticipate a shift use low cost, lightly stressed, gasoline engines. Diesel engines in cars will probably become a thing of the past, as the combined cost of diesel engine and electric drive will be uncompetitive.

Slide 83



Electric motors as the source of motive power will generate new tribological challenges.

Firstly, to reduce the mass of the motor, we have to reduce the diameter of the stator; reducing the diameter of the stator reduces the torque generated. In order to produce the same power output, at this reduced torque, the speed of rotation of the motor must be increased.

Slide 84

EVs – Gear-box Input Speeds

- Many current pure EVs have motor input speeds typically of zero to 12,000 rpm, with the target to reduce motor size still further, with motor speeds increasing to 20,000 or more rpm.

- Motors still have to be attached to wheels that rotate as speeds of zero to approximately 1,000 rpm. However this is done, it is impossible to avoid having some form of gear-box, with an input speed of up to 12,000 or, in the future, 20,000 rpm.

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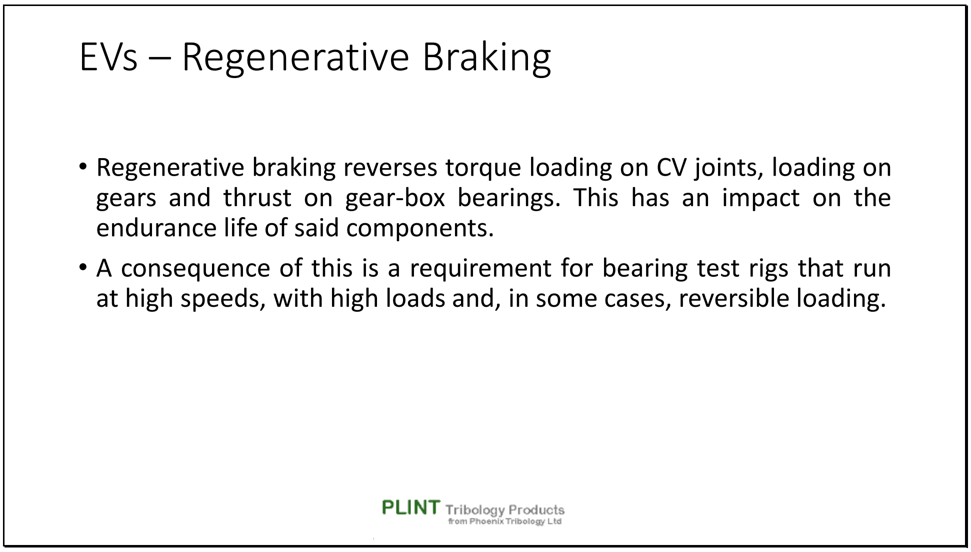
Many current pure EVs have motors with a maximum output speed of 12,000 rpm. The motors still have to be attached to wheels that typically rotate as speeds varying from 0 to approximately 1,000 rpm. However this is done, it is impossible to avoid having some form of gear-box, with an input speed of up to 12,000 rpm. With reduced mass and stator diameter motors, that input speed is going to go up to perhaps 20,000 rpm or higher.

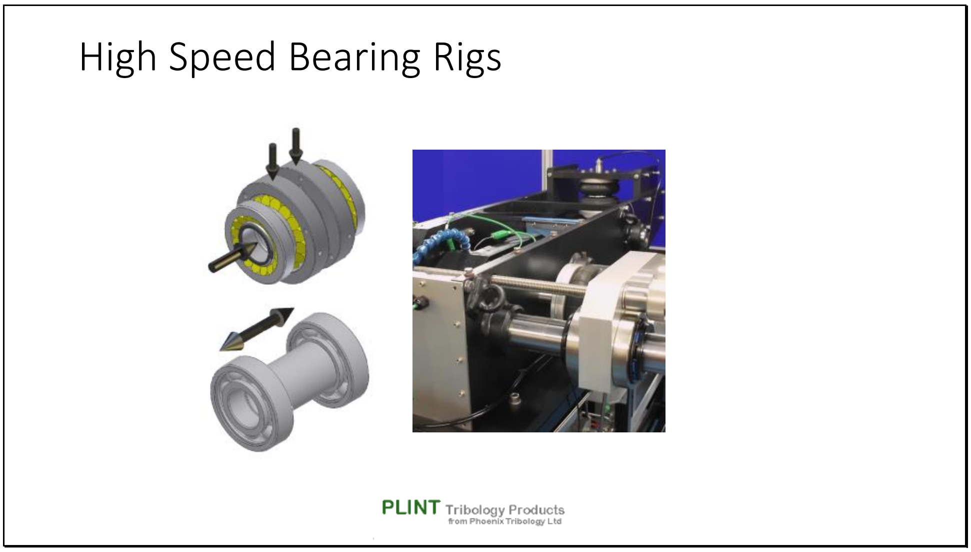
Slide 85



Now consider the effects of regenerative braking. This reverses the torque on CV joints and gears and the thrust on gear-box bearings.

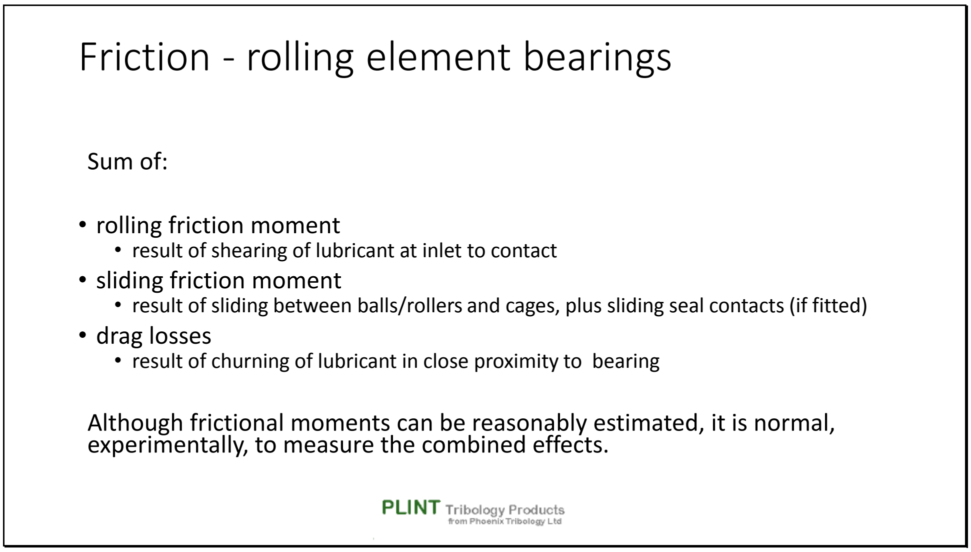
Gear-boxes in EVs will invariably be much simpler than the multi-speed gear-boxes associated with IC engine vehicles, however, both gears and bearings will be required to run at much higher speeds and be capable of coping with the reverse loading issue.

Slide 86



This has led to a requirement for high-speed bearing test rigs in which a combination of radial and axial loading can be applied, with, where necessary, bi-directional loading.

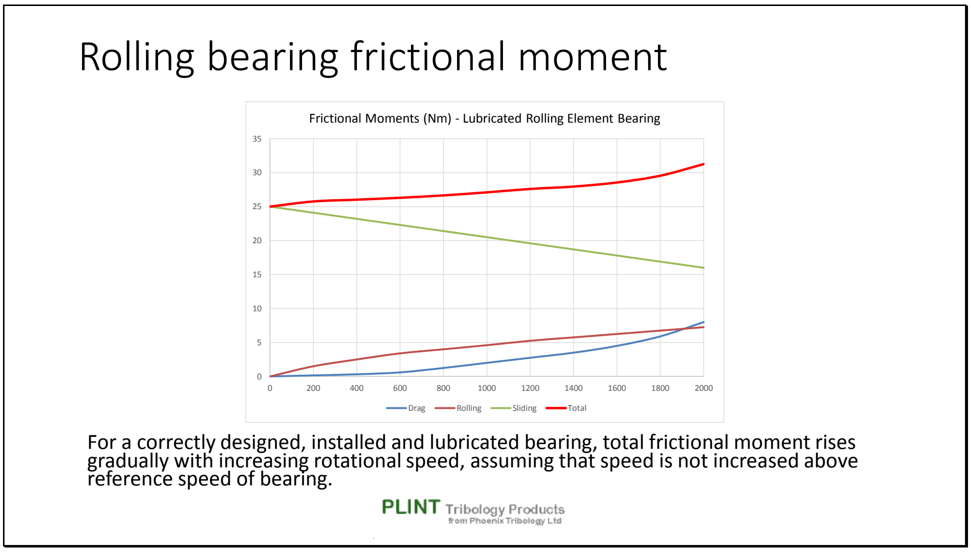
Slide 87



The frictional moment in rolling element bearings is the sum of the rolling friction moment, the sliding friction moment and the drag losses. The rolling friction moment is the result of shearing of the lubricant at the inlet to the contact. The sliding friction moment is the result of sliding between balls/rollers and cages, plus sliding seal contacts (if fitted). The drag losses are the result of churning of the lubricant in close proximity to the bearing.

Although these frictional moments can be reasonably estimated, it is normal, experimentally, to measure the combined effects.

Slide 88

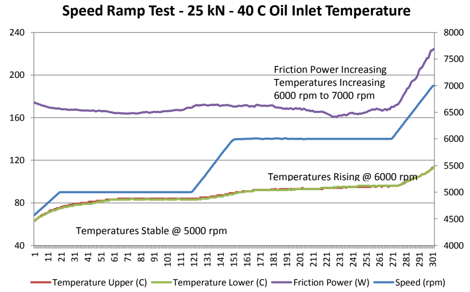


For a correctly designed, installed and lubricated bearing, the total frictional moment rises gradually with increasing rotational speed, assuming that said speed is not increased above the reference speed of the bearing.

If we exceed the limiting speed for a given load, differential expansion causes the bearing resistance to increase, with potential run-away to failure.

Slide 89

Bearing exceeding limiting speed

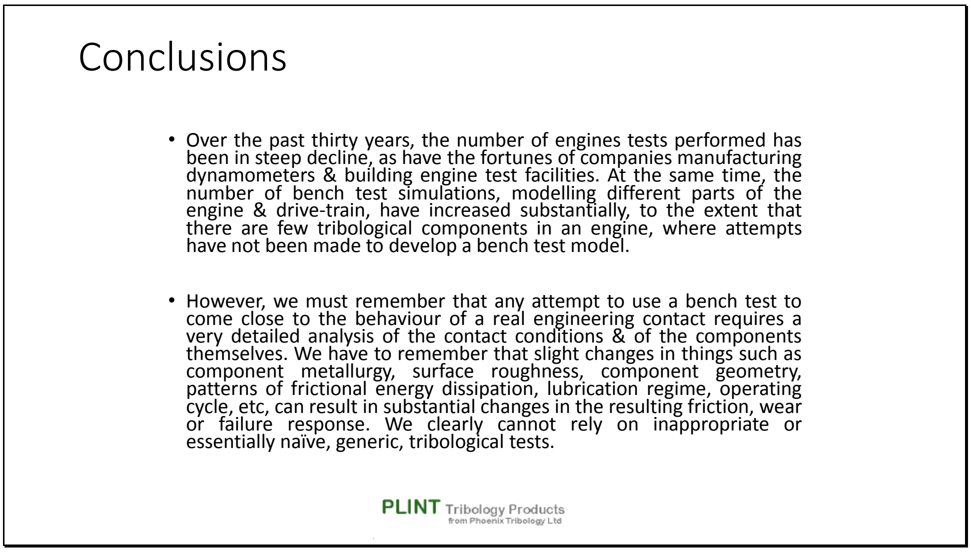


Limiting speed is speed at which bearing can achieve a steady-state operating temperature, in other words, where frictional energy dissipation does not exceed capacity of bearing system to dissipate heat. If exceeded, differential expansion causes bearing resistance to increase, with potential run-away to failure.

In these circumstances, increasing the oil flow or reducing the oil inlet temperature, in an attempt to remove more heat, does not work; it simply increases the rolling and drag friction moments.

This is the essential reason why measurement of rolling element bearing frictional moments is only really of interest at high rotational speeds. It is perhaps understandable why it has become of significant interest to electric vehicle manufacturers.

Slide 90



Almost 30 years ago, a paper was published with the title:

Routine Engine Tests - Can We Reduce Their Number? Dr M A Plint, Dr A F Alliston-Greiner - Petroleum Review, July 1990, p. 368-370

Over the past thirty years, the number of engines tests performed has been in steep decline, as have the fortunes of companies manufacturing dynamometers and building engine test facilities. At the same time, the number of bench test simulations, modelling different parts of the engine and drive-train, have increased substantially, to the extent that there are few tribological components in an engine, where attempts have not been made to develop a bench test model.

However, we must remember that any attempt to use a bench test to come close to the behaviour of a real engineering contact requires a very detailed analysis of the contact conditions and of the components themselves. We have to remember that slight changes in things such as component metallurgy, surface roughness, component geometry, patterns of frictional energy dissipation, lubrication regime, operating cycle, etc, can result in substantial changes in the resulting friction, wear or failure response. We clearly cannot rely on inappropriate or naïve, generic, tribological tests.

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