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Tribology Testing

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Tribology Testing

- Introduction
- Quality Control Tests
- Tribological Testing
- Modelling Real Systems
- Specimen Pairs
- Sliding and Sliding/Rolling - Test Machines
- Sliding and Sliding/Rolling – Test Conditions
- Abrasion and Erosion Tests
- Conclusions

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Tribology Testing

What does industry think?

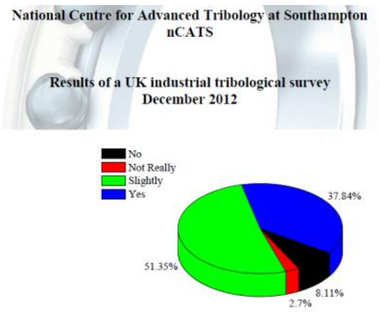

Figure 18 How good is tribological testing at informing users what they need to know.

A recent survey by nCATS gave some rather disturbing results, with just under 38% of respondents indicating that tribological testing was giving the results they needed. This is an issue that needs to be addressed.

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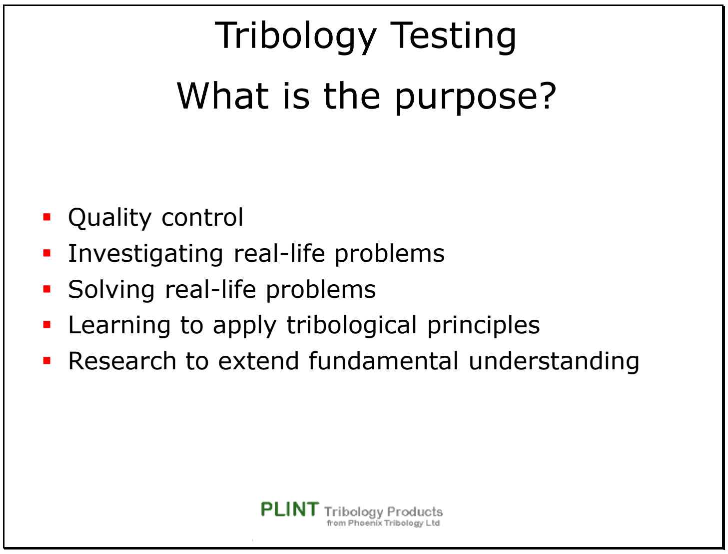

What are the purposes of tribological testing?

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

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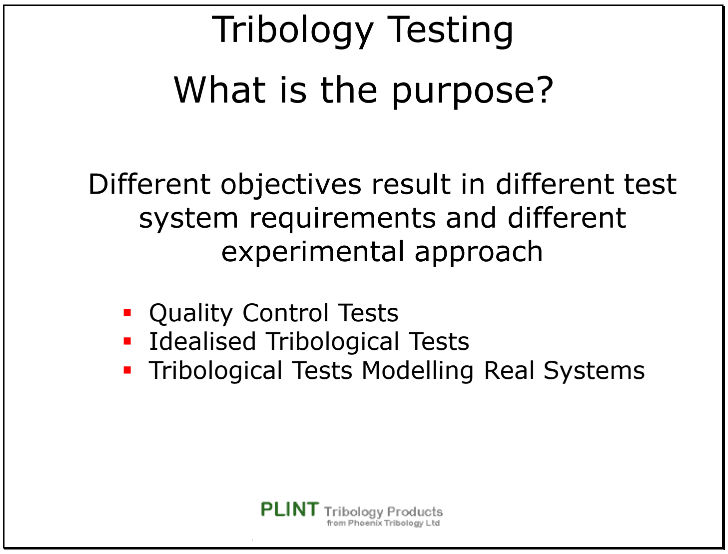

Different objectives result in different test system requirements and a different experimental approach. To help to explore the issues, three different categories will be used:

- Quality Control Tests
- Idealised Tribological Tests
- Tribological Tests Modelling Real Systems

The main focus will be on tests modelling real systems.

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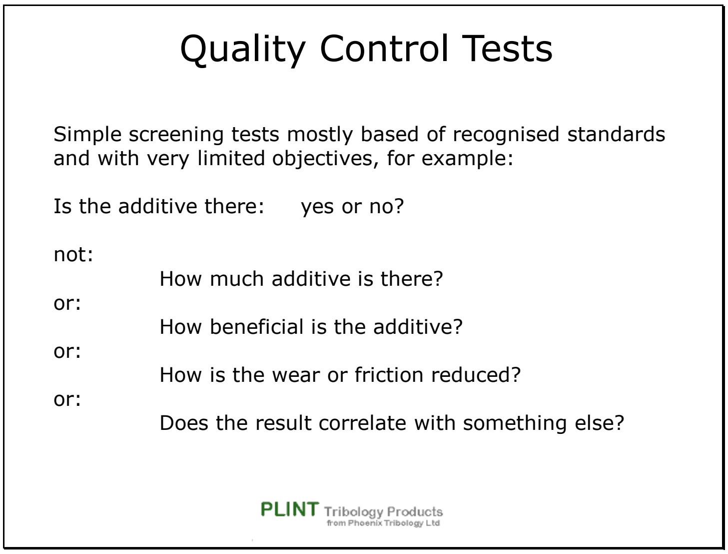

These are screening tests, mostly based on recognised standards, and with very limited objectives, for example, answering a question such as:

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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Quality Control Tests

- Results frequently far removed from modelling of and correlation with real systems
- In many cases, “wrong” standards 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 performance in real application
- Risk product designed to pass test, not work in application

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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, the ”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. There are many standard tests, including from ASTM, DIN, ISO and JASO. If you want to use these standards, follow them carefully.

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Problem with Tribology

- Friction and wear are not intrinsic material properties but are properties of the system in which the materials operate
- Properties measured in an experiment using a test machine are also a system properties
- Data generated from any properly calibrated test machine has to be valid, but only for the performance of the materials in that test machine

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Now let’s consider tribological tests.

What could be simpler than to design an experiment in which we rub a couple of bits of material together and make a few measurements? The problem is that we are not concerned with single "properties" of materials, but how those materials behave when placed together in complex systems. Friction and wear are not intrinsic material properties but are properties of the system in which the materials operate. It follows, that the properties measured in an experiment, using a test machine, are also system properties.

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Problem with Tribological Tests

- Tribological contacts are inherently stochastic, so events that occur at a certain time in one test may occur at a different time in a second test and this can have a dramatic effect on the outcome. These events are unpredictable and random and could be things such as delamination or failure of a transfer film, a fatigue failure of parent materials or any number of other things
- Each time we select a test to perform, we sometimes knowingly, but more frequently, unknowingly, accept its limitations and drawbacks

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It is important to note that tribological contacts are inherently stochastic.

Each time we select a test or test method to perform, we knowingly, but more frequently, unknowingly, accept its limitations and drawbacks

It is easy to get lost, somewhere between the real world and the laboratory. Similarly, it is difficult to project results generated in the laboratory, out into the real world.

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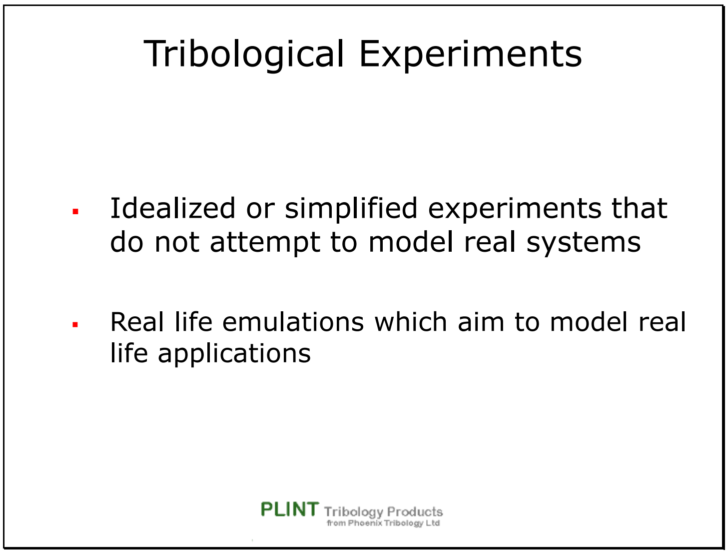

Tribology tests can sensibly be divided into two groups:

- Idealized or simplified experiments that do not attempt to model real systems
- Real life emulations, which aim to model real life applications

A good example of an idealized 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 data to predict performance in other test systems or real applications.

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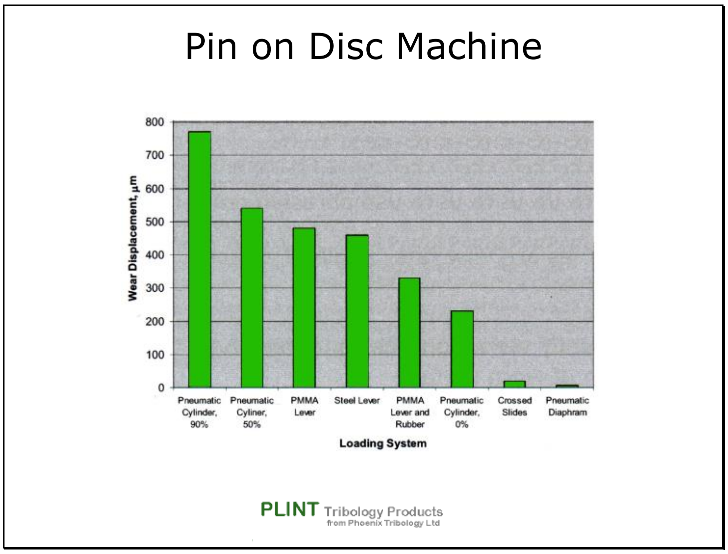

The chart here, from the UK National Physical Laboratory, shows the influence of varying the stiffness and inertia of the loading system, on a pin on disc machine, running under nominally identical test conditions.

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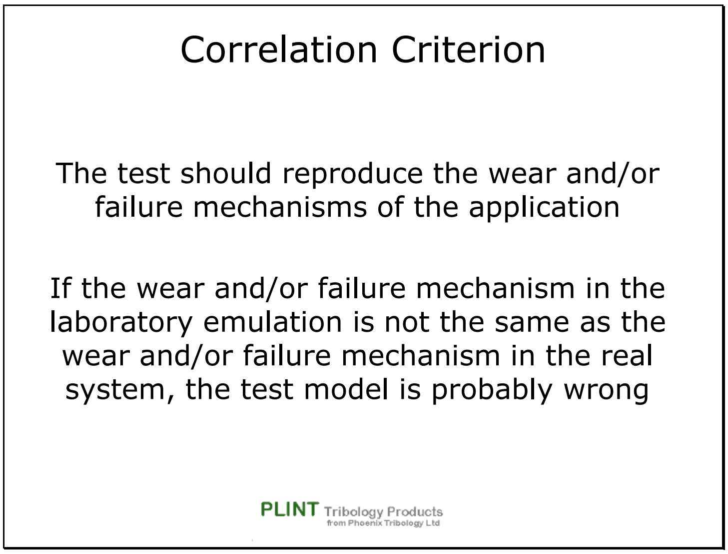

The challenge with real life emulations is that to be of any use the tests must model the friction, wear and/or failure mechanism apparent in the real application, with test variables selected and adjusted in order to obtain correlation with field data. If the wear generated in the bench test looks nothing 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!

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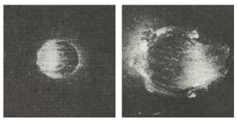
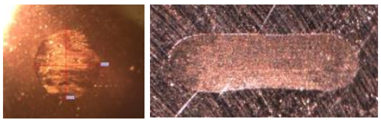
Correlation Criterion



Sliding Four Ball



ISO Fuel Lubricity

Does anything real look like this?

Examples of typical Quality Control Test!

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Does any “real” wear process look like this?

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Modelling Real Systems

- Contacts involving both sliding friction and wear can be modelled at reduced scale and with accelerated testing
- Processes involving surface fatigue can in some cases be modelled at reduced scale, but not at a reduced number of cycles, unless there is an increase in load
- Abrasive and erosive wear processes, where particle size, distribution, angle of incidence and, for erosive wear, particle velocity, are critical to the wear process, have to be modelled at full scale

If we can correctly characterize the full-scale system to be modelled, the easier it becomes to ensure that the bench tests we run will provide useful information. The type of wear process will, to a large extent, govern whether it can be modelled at reduced scale and whether accelerated testing is valid. 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.

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.

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Comparison of Idealized & Real Contacts

Continuous Sliding Point Contact	Continuous Sliding Line Contact	Continuous Sliding Area Contact
Pencil	Vane Pump	Journal Bearing
Felt Tipped Pen	Positive Displacement Flow Meter	Thrust Bearing
		Face Seal
		Lip Seal
		Tool Face
		Drill String

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A review of any collection of papers published on tribological testing will invariably show a great enthusiasm for tests involving sliding point contacts. This is because they are easy to set up and allow simple measurements to be made of the nominal wear scar. The disadvantage is that such tests invariably involve high hertzian contact pressures, which may be unrepresentative of contacts in real applications; there are no real engineering applications involving sliding point contacts.

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Comparison of Idealized & Real Contacts

Intermittent Sliding Point Contact	Intermittent Sliding Line Contact	Intermittent Sliding Area Contact
	Draw Bead	Draw Bead
	Die Press	Die Press
		Clutch Plate
		Brake Disc
		Brake Drum
		Lead Screw
		Cables/Wires
		Screw Thread
		Bullets/Shells

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Oscillating Sliding Point Contact	Oscillating Sliding Line Contact	Oscillating Sliding Area Contact
		Ring/Liner Contact
		Linear Seal
		Rod End Bearing
		Hip Joint
		Push Rod
		Piston Skirt/Liner
		Gudgeon Pin
		Piston Rod Packing
		Gland Packing
		Heat Exchanger Tube
		Shaft Coupling
		Bolted Flange
		Bolted Joint
		Machine Slide-way
		Rudder Post

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Comparison of Idealized & Real Contacts

Sliding/Rolling Point Contact	Sliding/Rolling Line Contact	Sliding/Rolling Area Contact
Toroidal Transmission	Gear Contact	Car Tyre
	Rail/Wheel Interface	Belt Drive
	Cam/Tappet	Rolling Mill
	Cam/Finger Follower	Knee Joint
	Gear Pump	Tractor Tyre
		Tank Track
Rolling Point Contact	**Rolling Line Contact**	**Rolling Area Contact**
Ball Bearing	Roller Bearing	Car Tyre
	Rail/Wheel Interface	
	Cam/Roller Follower	

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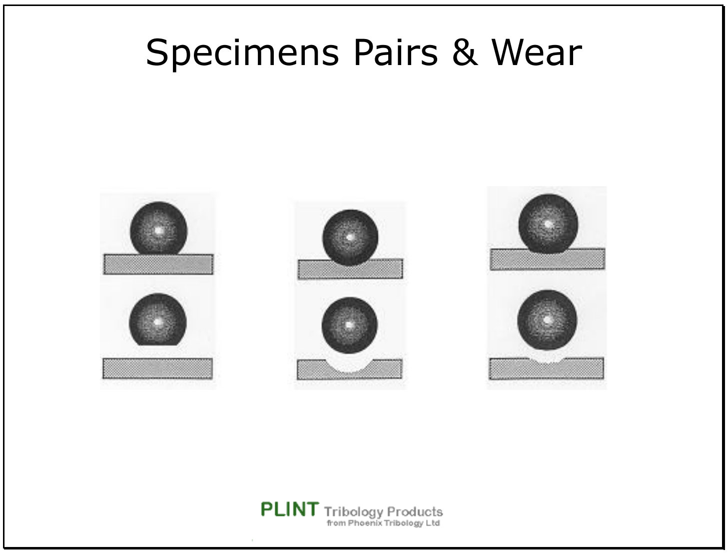

The first issue to address in designing a test is which way round, in terms of relative hardness, to have the specimen pair. Traditionally, many wear tests have involved running a soft pin or ball on a hard disc or plate. Under these conditions, the wear occurs on the softer material, sometimes accompanied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

Measurement of material lost from the softer pin or ball is relatively easy. It should however be remembered that if material has been transferred to the disc or plate, its mass may increase.

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

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

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

In real machines, we frequently find contacting materials of similar hardness, with the result that wear is shared between the two contacting surfaces. The only solution here is to measure the wear on both surfaces, not forgetting that, if the materials are different, the wear rate will still be dependent on which material is used for the pin or ball and which is used for the disc or plate. This is because the energy inputs are different for the two specimens.

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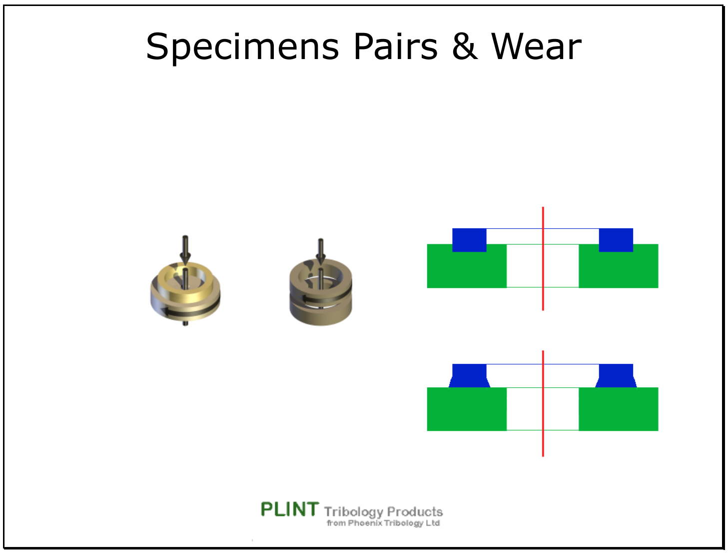

With area contact specimens, there are similar considerations. For example, the thrust washer test configuration provides a continuous contact on both test surfaces and thus avoids any of the leading edge problems associated with pin on disc or pin on plate configurations.

However, even with a continuous contact, edge effects must be considered, especially if one specimen has a smaller outer diameter and larger inner diameter than the other.

If the smaller outer diameter specimen (the upper specimen in the sketch) is harder than the larger diameter specimen, it will cut into surface of the latter and the frictional behaviour of the contact will be dominated by circumferential edge effects.

If the smaller upper specimen is softer than the larger diameter lower specimen, the edge effect is removed, but elastic deformation of the softer material may result in a change in apparent area of contact.

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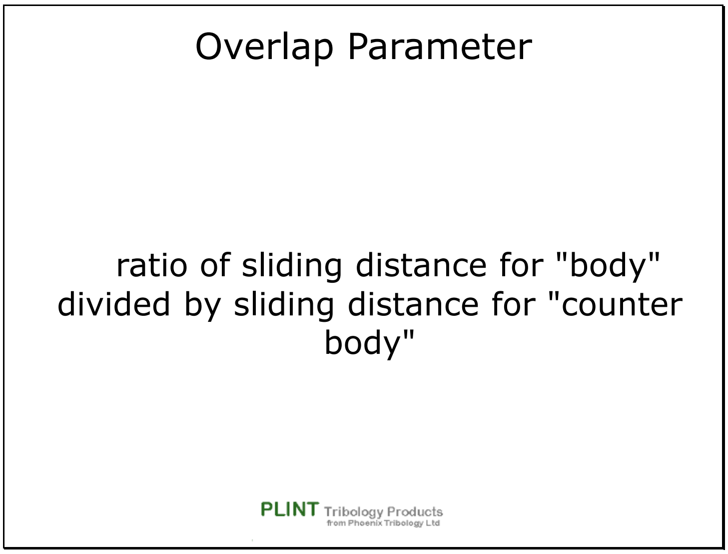

Now let’s move on to the overlap parameter.

If we have a 10 mm diameter pin running on a 100 mm circumference disc track, then in one revolution, a point on the pin experiences 100 mm of sliding. However, a similar point on the disc sees only a single pass of the pin, hence a sliding distance of just 10 mm. Double the track circumference and the point on the pin sees 200 mm sliding per revolution whereas the point on the disc still only sees 10 mm.

Hence, in this example, changing the track diameter has a direct impact on how the sliding distance, hence the wear, is shared between the two surfaces. It also means that running repeat tests at different track diameters, at the same surface speed on the same disc, will generate different wear rates.

By contrast, with the thrust washer arrangement, the sliding distance for a point on either sample has to be the same. This probably makes it a better arrangement for testing many materials, unless, of course, we wish deliberately to confine the majority of wear to one surface.

The "overlap parameter" (Czichos) is defined as the ratio of sliding distance for "body" divided by sliding distance for "counter body". For the thrust washer this is 1, for fretting tests it is close to 1, but for pin on disc tests it is variable, but is typically less than 0.05. The overlap parameter also applies for reciprocating tests, but here there is not the temptation to use the equivalent of different pin on disc track diameters, as one would sensibly keep the stroke the same and index the specimen plate sideways to run a fresh wear track.

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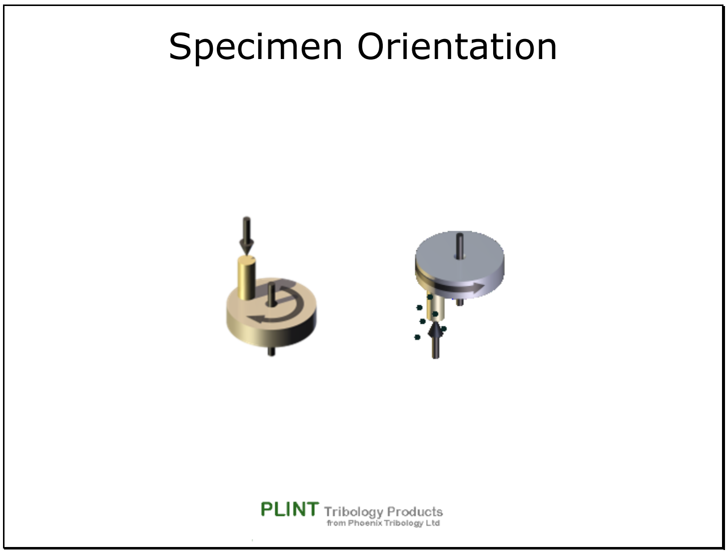

Let's think a bit about specimen orientation.

If we run a pin on disc machine with the pin loaded onto the disc from above, any wear debris generated will tend to accumulate on the surface.

This will give different behaviour from exactly the same configuration turned upside down.

In this case, the debris will fall off the disc surface, giving different friction and wear behaviour.

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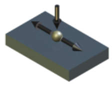
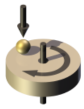
Specimen Configurations
Advantages & Disadvantages

Point Contact:

- Positive:
  - Alignment easy
  - Entrainment conditions well defined
- Negative:
  - High contact pressure at start & potential failure condition
  - Contact area changes significantly during test
  - Not good with hard ball on soft disc/plate (unless modelling forming)





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Now let's consider the advantages and disadvantages of different specimen configurations.

We have three basic contact configurations for our sliding or sliding/rolling tests.

With point contact tests, we have easy alignment of specimens, and in lubricated tests, the entrainment conditions are well defined. There are negatives however; high contact pressure at start of the test is unavoidable and this may generate potential failure conditions. Contact area changes significantly during test and of course, as indicated earlier, this is not a very good model of wear processes if we have, say, a hard ball running on a soft surface, unless our interest is in modelling forming processes.

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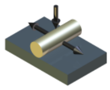
Specimen Configurations

Advantages & Disadvantages

Line Contact:

- Positive:
  - Alignment not difficult
  - Entrainment conditions well defined
- Negative:
  - In sliding tests, contact area changes significantly during test



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With the line contact, alignment is still not difficult and entrainment conditions are well defined. The only small negative point is that in sliding tests, the contact area changes during the test, but obviously not as much as happens with a sliding hertzian point contact.

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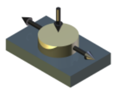
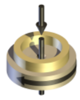
Specimen Configurations
Advantages & Disadvantages

Area Contact:

- Positive:
  - No variation in contact pressure during test
  - Higher loads achievable generating higher friction force
- Negative:
  - Alignment difficult without special tooling
  - Entrainment condition poorly defined





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With area contacts we have no variation in contact pressure during test, if run at constant load. Higher loads are achievable, generating higher friction forces, making measurement of friction easier.

The negative with area contact specimens is that they can be difficult to align, unless we use special tooling, and a major negative with lubricated tests is that the entrainment condition are poorly defined.

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Contact Pressure

Orders of Magnitude

- Steel ball 10 mm diameter
  - Point Contact
- Steel cylinder 10 mm diameter and same mass
  - Line Contact – cylinder on edge
  - Area Contact – cylinder

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Now let’s think a little bit about contact pressures and orders of magnitude.

In this example, we compare the pressure generated by a 10 mm diameter steel ball on a steel flat, by a 10 mm diameter cylinder in line contact and with the same mass as the ball, that makes the length of the cylinder 6.67 mm, and the same cylinder, on its flat end, providing an area contact.

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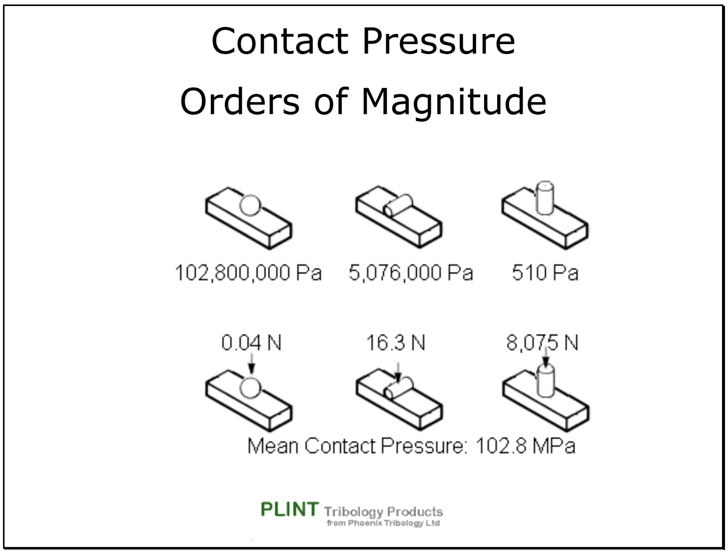

The mean contact generated by the ball is just less than 103 MPa, for the cylinder, 5 MPa, whereas the cylinder on end, in area contact, it is just 510 Pa.

So here we have orders of magnitude difference in contact pressures.

It is worth noting that if we wanted to run a hertzian point contact test with our ball at less than 103 MPa, we would have to make the ball levitate.

If we now wanted to generate the same mean contact pressure as we get with the ball, under its own mass, we would have to apply about 16 N to our line contact specimen, but about 8 kN to our pin area contact specimen.

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Contact Pressure
Orders of Magnitude

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

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We should remind ourselves that in the case of the point contact we may not be justified in using the Hertzian contact equations. For these to be true, it is assumed that the contact is elastic, that the contact zone is flat, that there are no shear stresses in the contact zone and that the contact radius is much smaller than the radius of the ball. This will not be the case if we have a hard ball loaded against a softer flat, where plastic deformation may take place. In this case, we cannot sensibly use the Hertz equations and instead must use an elasto-plastic finite element model to evaluate the contact conditions. Such models lead to the perhaps slightly unexpected conclusion that increasing the load on our hard ball simply increases the size of the plastic zone. We effectively have the equivalent of a sliding Brinell hardness test.

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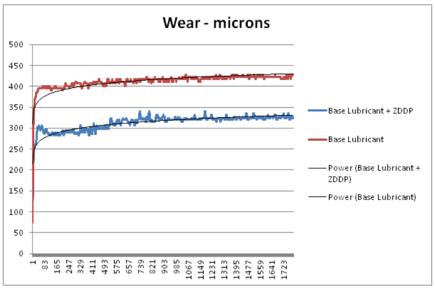
Contact Pressure and Wear



Comparative wear displacement,
back-calibrated from post-test scar measurement
Sliding Point Contact Test

We should not forget that a rapid increase in contact area, caused by wear or plastic deformation, will result in significant changes in contact pressure. This is particularly the case with experiments involving sliding hertzian point contacts.

This example shows the evolution of wear and the resulting change in contact pressure at the start of a sliding hertzian point contact test.

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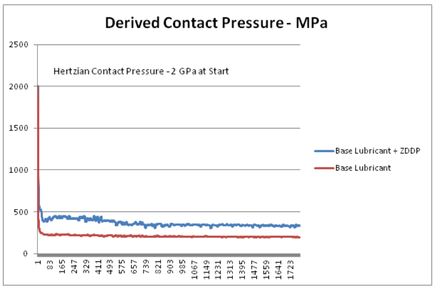
Contact Pressure and Wear



Do not get these differences in contact pressure in a real machine:
the contact pressure in a gear, cam or ring liner contact does not alter just because we decide to test two different additives

We do not get these differences in contact pressure in a real machine: the contact pressure in a gear, cam or ring liner contact does not alter just because we decide to test two different additives

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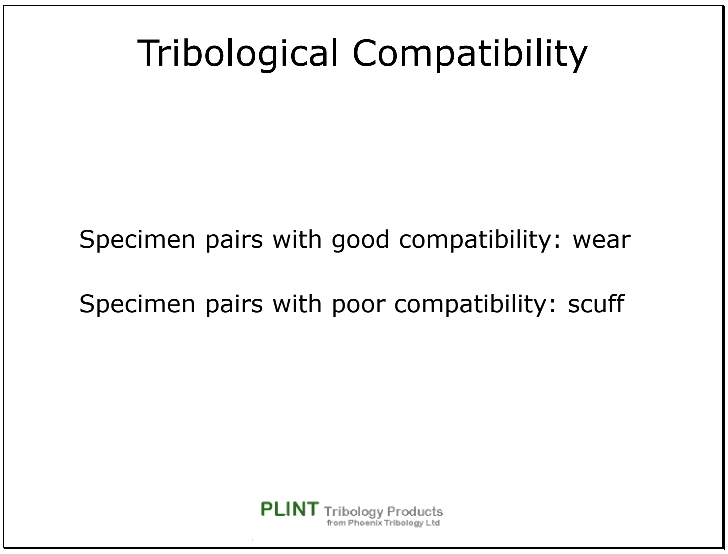

The term “tribological compatibility” denotes the reluctance of contacting surfaces to form strong interfacial bonds. The tribological compatibilities of metals do not correlate perfectly with other properties, although the extent of mutual solubility is often suggested as a guide. Those combinations which show negligible solid solubility in each other usually form tribologically compatible pairs.

Identical metal pairs are completely mutually soluble and thus show poor tribological compatibility. Poor tribological compatibility usually indicates a tendency to galling.

In sliding bench tests, specimen pairs with good compatibility will tend to wear, whereas pairs with poor compatibility will tend to scuff or gall. This raises the question as to what material combinations we should be choosing for a given test. It is of course worth noting that in real applications involving sliding contacts, it is normal to use materials with good tribological compatibility, hence different materials, whereas with pure rolling contacts, such as bearings, it is of course acceptable to use identical materials.

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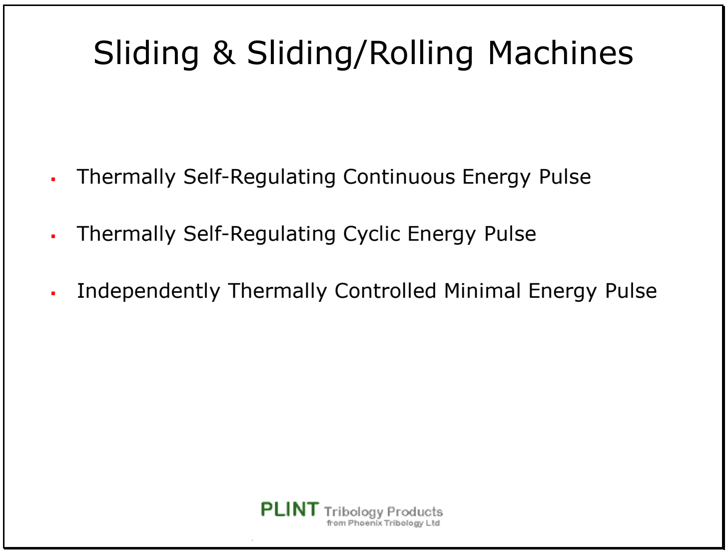

Teasing out the various points discussed so far, it should be apparent that we have a number of questions to answer with regard to the types of test we should run and these will vary depending on the purpose and required outcome. To put this more simply, we should perhaps expect a friction test, a wear test and a scuffing test to require somewhat different test conditions.

There are many friction and wear test machines. Selection will depend on what kind of test you wish to perform. Test rigs can usefully be divided into three categories, each describing fundamentally different tribological systems.

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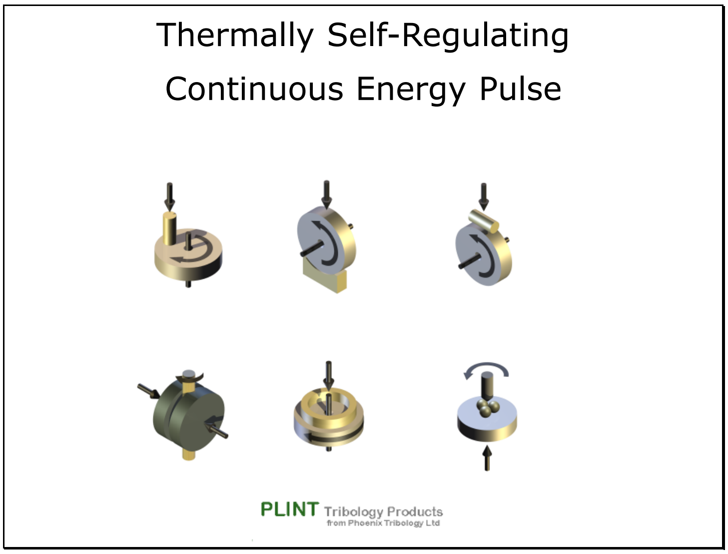

Thermally self-regulating continuous energy pulse machines are those in which the point of contact is stationary with respect to one of the specimens and subject to constant speed uni-directional sliding. The test configuration defines the thermal conditions in the contact and the contact temperature is self-regulating and cannot be controlled as an independent variable.

There are few real life applications in which this type of motion occurs, especially with sliding hertzian contacts, and these machines thus do not normally provide adequate models of most real systems. Instead of brief rubbing episodes frequently repeated, the machines subject one specimen to continuous rubbing and the associated temperature field dominates. These machines are widely used because they are simple. Examples include pin on disc, block on ring, crossed cylinder and sliding 4-ball test machines. Not surprisingly, data generated by such machines rarely correlates with field data. However, the majority of existing standards in sliding wear use these uni-directional sliding machines.

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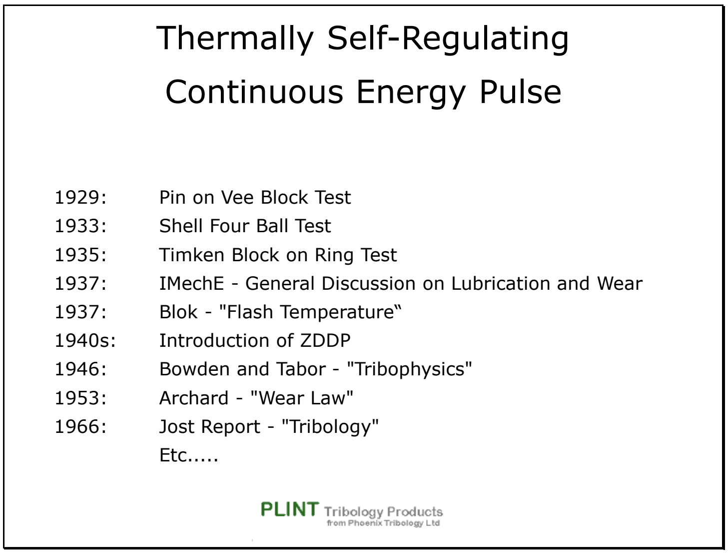

To understand their historical context, consider the following:

1929:	Pin on Vee Block Test
1933:	Shell Four Ball Test
1935:	Timken Block on Ring Test
1937:	IMechE - General Discussion on Lubrication and Wear
1937:	Blok - "Flash Temperature"
1946:	Bowden and Tabor - "Tribophysics"
1953:	Archard - "Wear Law"
1966:	Jost Report - "Tribology"

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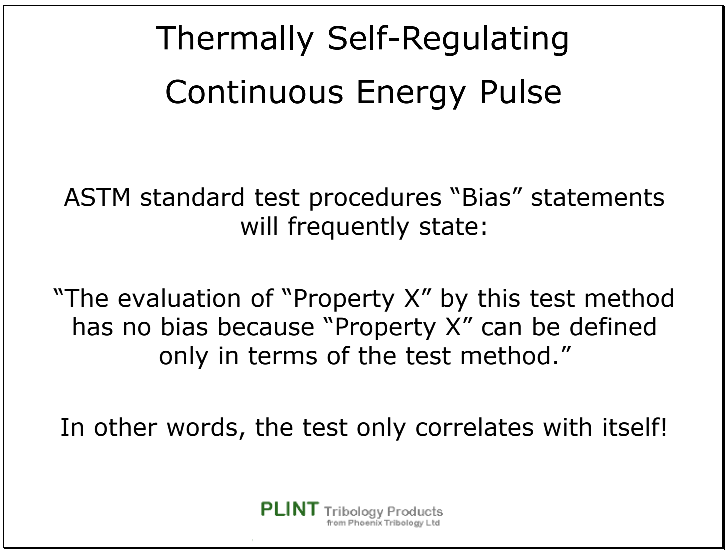

If you were to choose to follow ASTM standard test procedures you should note that the “Bias” statements will frequently state the following:

> “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.”

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Thermally Self-Regulating
Continuous Energy Pulse

- Point of contact stationary with respect to one of the specimens and subject to constant speed uni-directional sliding
- Uses:
  - Fundamental wear studies of materials
  - Quality control
- Advantages:
  - Simple and cheap specimens
  - Wear mapping and parametric studies

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Historically, these machines have been used for fundamental wear studies of materials and coatings in dry conditions. Because they can usually be run over a wide range of loads and speeds, wear mapping and parametric studies may readily be performed.

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Thermally Self-Regulating Continuous Energy Pulse

- Limitations:
  - For lubricant additive tests, entrainment conditions result in requirement for heavily loaded contacts
  - Increasing load to achieve film failure may bring to significance specimen material properties
  - Energy Pulse for the fixed point of contact specimen is continuous and associated temperature field dominates
  - Test configuration defines the thermal conditions in the contact
    - In plastics, limit is melting point
    - With ceramics, heat/quench cycle can generate thermal fatigue
    - For liquid lubricants, temperature drives chemical reactions

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They are less successful for liquid lubricated tests where the entrainment conditions associated with constant speed sliding usually results in a requirement for heavy loads to overcome hydrodynamic lubrication and to promote mixed or boundary lubrication or film failure. Increasing the load to achieve film failure usually results in thermally induced scuffing failure; that is fine if scuffing is your interest, but not good if you are interested in wear.

Further to this, for fully formulated lubricants, temperature drives the chemical reactions of additives with specimen surfaces, changing the friction and wear conditions in the contact. In these machines we have no control of, or satisfactory means of measuring, the contact temperature.

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Thermally Self-Regulating
Real System Models

Thermally self-regulating, with continuous or intermittent sliding:
thrust washers, face seals, clutches, brakes, journal bearings

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There are, of course, real systems that are thermally self-regulating, with continuous or intermittent sliding, including thrust washers, face seals, clutches, brakes and journal bearings.

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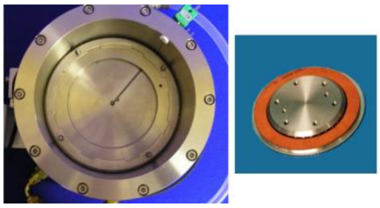
Thermally Self-Regulating
Real System Models



Low pressure, area contacts and most will either have:

- intermittent sliding (clutches and brakes)
- continuous sliding with hydrodynamic lubrication (lubricated plain bearings)
- continuous sliding with very light loads (dry bearings and seals)

None of these “real” systems involve sliding hertzian point contacts

These are all low pressure, area contacts and most will have either have:

- intermittent sliding (clutches and brakes)
- continuous sliding with hydrodynamic lubrication (lubricated plain bearings)
- continuous sliding with very light loads (dry bearings and seals)

None of these “real” systems involve sliding hertzian point contacts.

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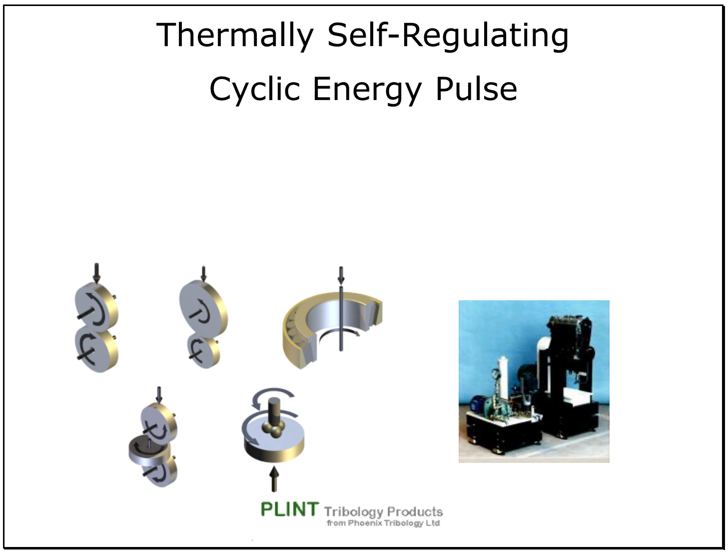

Thermally self-regulating cyclic energy pulse machines are those where the point of contact moves with respect to both contacting surfaces and there is a close approximation to the motion in actual machine components (for example, gears, cams, joints and mechanisms).

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Thermally Self-Regulating
Cyclic Energy Pulse

- Includes component test machines using idealized or standardized components such as gears, cam/follower and rolling element bearings
- Designed to emulate real contact conditions and typically operate under conditions broadly similar to those found in practical applications
- Contact temperature cannot be independently controlled

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These include a number of component test machines, using idealized or standardized components such as gears, cam/follower and rolling element bearings.

These machines are essentially designed to emulate real contact conditions and typically operate under conditions broadly similar to those found in practical applications. For all intents and purposes these machines are "full scale" and are hence emulators of the real contact.

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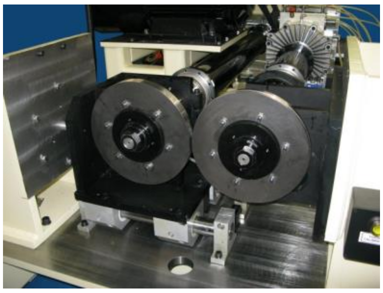
Thermally Self-Regulating

Cyclic Energy Pulse

Two-roller Machine



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The most generalized version of this type of device is the two-roller machine. Here two specimen rollers are loaded together. If they are rotated at the same surface speed the motion is pure rolling and such machines are used to study pitting failure (rolling contact fatigue) caused by the cyclic stressing of the surfaces. Rolling element bearing test rigs and rolling four ball machines perform a similar function.

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Two-roller Machine
Slide/Roll Ratio

Slide/Roll Ratio	**=**	**Sliding Velocity/Rolling Velocity**
Where:		
Sliding Velocity	=	$\lvert U_1 - U_2 \rvert$
Rolling Velocity	=	$\frac{1}{2} (U_1 + U_2)$
Slide/Roll Ratio%	**=**	$\mathbf{200 \times \lvert U_1 - U_2 \rvert / (U_1 + U_2)}$
Note:		
Slip%	**=**	$\mathbf{100 \times (U_1 - U_2) / U_1}$

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If the rollers are rotated with an enforced surface speed difference between them, the device can be tuned to emulate the conditions found in various real machine elements. We define this speed difference as the slide/roll ratio.

The test configuration in these machines, as with the previous category, defines the thermal conditions in the contact. The contact temperature is thus self-regulating and cannot be controlled as an independent variable.

By altering the Slide/Roll ratio and the Sliding Velocity in the contact, by varying the speed of the two rollers, we can vary the wear and failure mechanism from rolling contact fatigue at pure rolling or low slide/roll ratios, to wear at moderate slide/roll ratios and low sliding speeds, through to catastrophic scuffing failure at high sliding velocities.

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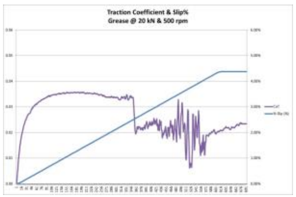
Two-roller Machine

Example Data



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In essence, changing the sliding speed changes the contact temperature and hence what happens to the surfaces and changing the rolling velocity changes the amount of lubricant entering the contact. By simply altering the test parameters, we can produce a range of different wear and failure mechanisms from the same test configuration.

There are a few standards for rolling or slide/roll testing but they refer more or less exclusively to the performance of the lubricant rather than the materials.

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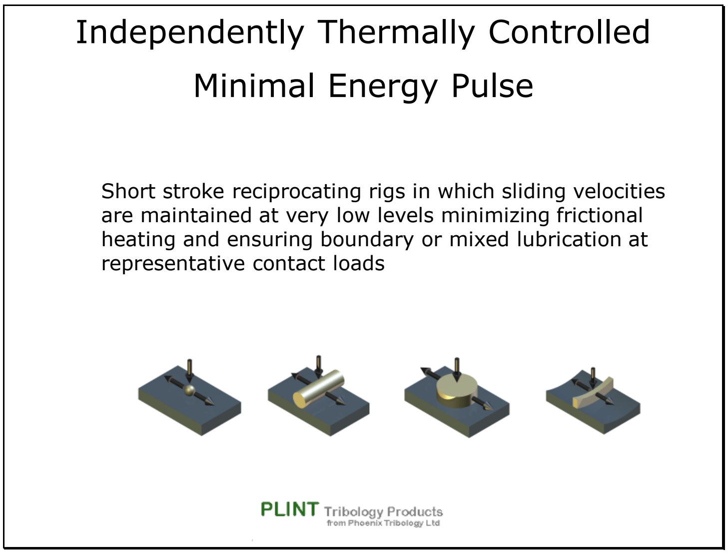

Independently thermally controlled minimal energy pulse machines are the short stroke reciprocating rigs, in which, although sliding velocities may be low, the rate of events (specimen passes or asperity contacts) may be high. These are machines in which sliding velocities are maintained at low levels in order to minimize frictional heating and to ensure, in the case of lubricated tests, boundary lubrication at representative contact loads.

By minimizing frictional heating we have the opportunity to control the contact temperature by controlling the bulk temperature of the test specimens, thus allowing contact temperature to be controlled as an independent variable. High repetition rates can be achieved without significant increases in sliding speed and corresponding loss of control of contact temperature.

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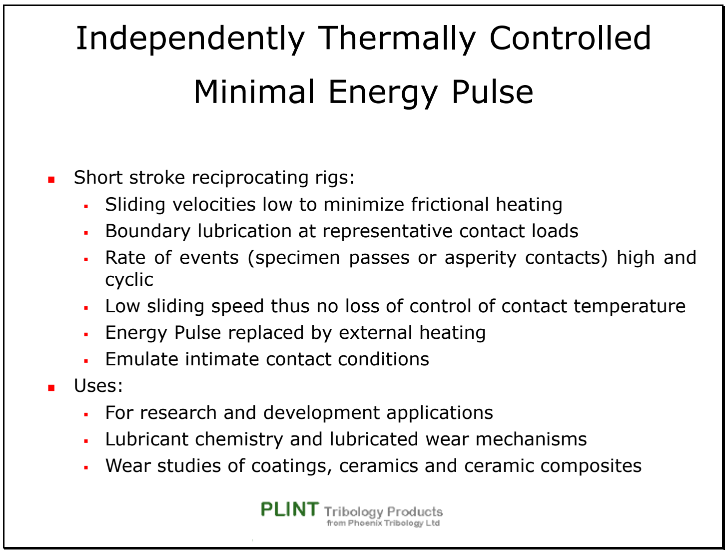

These devices (except in the case of the piston ring on liner contact near end stroke) do not model exactly the real contact to be investigated, but aim to emulate the intimate contact conditions, in a controllable and accessible way.

One of the principal advantages of the reciprocating test is that the direction of motion and the direction of surface finish can be the same. With the uni-directional specimens, for example, pin on disc, the orientation of grain structure or surface finish as presented to the pin varies as the disc rotates.

Historically these rigs have been used for fundamental and applications studies in the lubricants field with a particular emphasis on the lubricant chemistry and lubricated wear mechanisms. These rigs have an obvious similarity to the motion experienced by many practical components that have cyclic energy inputs, such as gears and cams and followers. They are also used for wear studies of coatings, ceramics and ceramic composites.

There are an increasing number of standards based on short stroke reciprocating tests covering dry and lubricated wear of ceramics, metals and ceramic composites, measurement of friction, wear and extreme pressure properties of lubricating greases and measurement of the diesel fuel lubricity.

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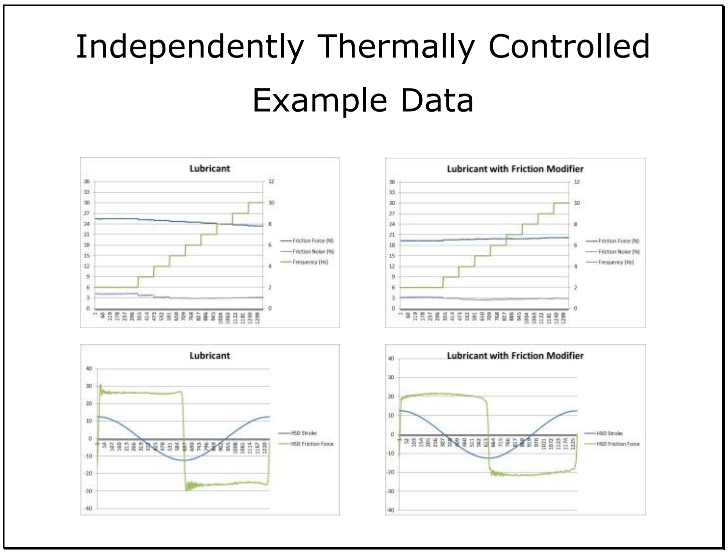

As with the sliding/rolling machines, we can generate a whole range of different wear and failure mechanisms in the boundary or mixed lubrication regime, not by altering the sliding speed as with the thermally self-regulating devices, but by varying the contact temperature by external heating.

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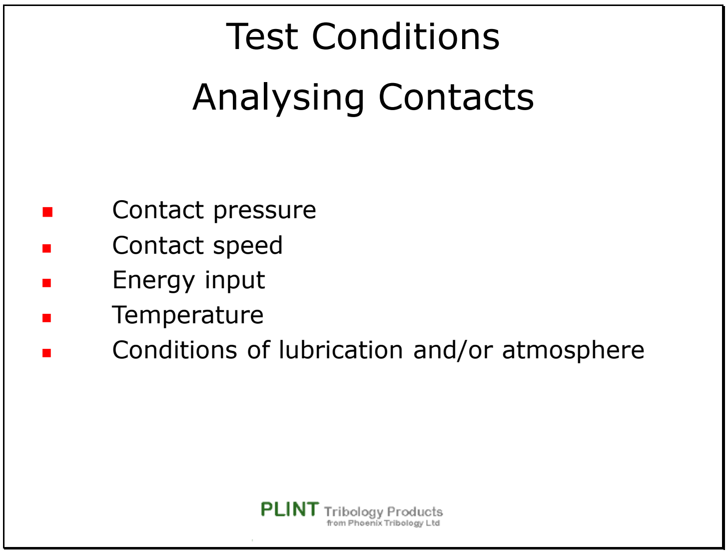

Before attempting to design an experiment, we must ensure that we have properly analysed and understood the tribological conditions to be modelled. The contacting environment is usually defined in terms of:

- Contact pressure
- Contact speed
- Energy input
- Temperature
- Conditions of lubrication and/or atmosphere

None of these are completely straightforward to define, either for the practical contact or in the test machine model. When building a test model, considerations of scale are paramount. It is hazardous simply to attempt to define the "real life" conditions (load, speed, temperature etc) and apply them to a small test piece on a test machine. The first concern is with the bulk temperatures reached by the test specimens.

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Contact Temperature

- Strongly influenced by width of contact
- Responsible for temperature reached at surface
- Many wear & friction effects
- Much higher than bulk temperature

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This average contact temperature will always be higher than the bulk temperature of the material as measured by some sensor embedded in the specimens, some distance from the contacting surfaces.

The contact temperature is of particular importance when considering lubricated contacts.

Slide 50

Lubricant Additive Chemistry

Types of Boundary Additive Film

Physisorped	-	70 to 150 C
Chemically Reacted	-	170 to 240 C

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There are two basic forms of additive protection available to surfaces that are either temporarily or permanently in intimate contact, physisorped and chemically reacted. The former, associated with the migration of polar molecules to the active metal sites on the surface, typically occurs at contact temperatures in the range of 70°C to 150°C.

The chemically reacted layers are associated with actual chemical reaction between additives and the active metal sites on the surface. These are frequently extreme pressure (or anti-scuffing) additives and are usually activated at temperature above 170°C. With conventional lubricant additive packages, these layers frequently fail at contact temperatures above 240°C.

It is worth noting that there is often a region of tribological distress, sometimes known as the Temperature Distress Gap, between the physisorped and the chemically reacted regimes.

Changes in running conditions or test parameters, for example changing load or frequency of events, will usually cause a temporary change in the consistency of these protective layers or films, indicating some kind of limited dynamic stability.

The rate of formation of the chemically reacted films in particular is considered to be a direct function of the contact temperature, following the Arrhenius equation, and the repetition rate of the motion. In other words, one needs temperature for activation and for controlling the rate of reaction plus a finite time for the chemistry to take effect.

Common sense and chemical intuition suggest that the higher the temperature, the faster a given chemical reaction will proceed. For tribologists, it is important to remember that this is an exponential relationship with reaction rates accelerating with increasing temperature.

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Energy Dissipation

- Wear occurs in conjunction with the dissipation of frictional energy in the contact and is always accompanied by a rise in temperature
- All wear processes influenced by temperature, including:
  - formation of oxides
  - transformation of microstructure
  - formation or break-down of additive or tribochemical films
  - melting of surfaces (PV limit of the material)
  - thermal stress induced failure
- Frictional energy is generated by the combination of load and sliding speed and its distribution and dissipation is influenced by other contact conditions such as size and relative velocity
- Different patterns of energy dissipation give different wear patterns

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All wear processes are influenced by temperature, be that the formation of oxides on surfaces, the transformation of microstructure, the formation or break-down of lubricant additive or other tribo-chemical films, the melting of the surface (the PV limit of the material) or thermal stress induced failure. To be more specific, wear occurs in conjunction with the dissipation of frictional energy in the contact and this is always accompanied by a rise in temperature.

The frictional energy is generated by the combination of load and sliding speed and its distribution and dissipation is influenced by other contact conditions such as size and relative velocity. Different patterns of energy dissipation will give different wear patterns. Two more global parameters have been shown to be valuable in defining these conditions in sliding wear.

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Friction Power Intensity

- FPI represents amount of energy pumped into rubbing surfaces as they pass through contact zone
- Temperature achieved in contact and in bulk material directly related to FPI and size and thermal characteristics of materials and their supports
- Defines only rate of energy generation and does not take into account timescale over which this energy can be lost to contacting materials
- Timescale clearly has implications for amount of damage caused in the contact

Friction Power Intensity: $\mathbf{Q_F = \mu\ P\ V_s / A}$ W/mm^2

μ : friction coefficient, P: load, $V_{s:}$ sliding speed, A: apparent area of contact

Practical contacts have FPIs in range 5,000 to 20,000 W/mm^2

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The friction power intensity (Matveevsky) is simply 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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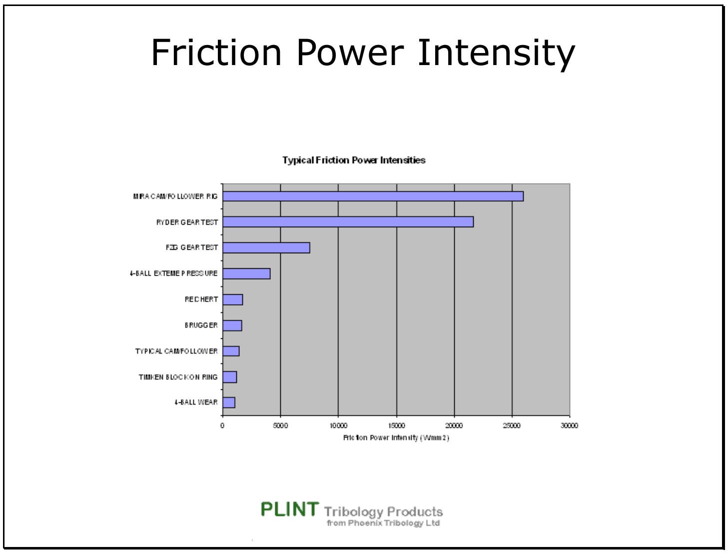

Practical contacts have FPIs in the range 5,000 to 20,000 W/mm^2.

The bar charts shows the FPIs for a number of different sliding and sliding-rolling test machines. It will be noted that the pure sliding machines, the continuous energy pulse machines, have much lower FPIs than the machines in which the point of contact moves on both surfaces.

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Friction Power Intensity

It should be noted that Friction Power Intensity (FPI) has the same dimensions as PV.

PV: Pa ms^{-1} $\quad \frac{M\,L}{T^2} \cdot \frac{1}{L^2} \cdot \frac{L}{T} = \frac{M}{T^3}$

FPI: W/mm^2 $\quad \frac{M\,L}{T^2} \cdot \frac{L}{T} \cdot \frac{1}{L^2} = \frac{M}{T^3}$

The FPI is equivalent to the PV limit multiplied by the friction coefficient.

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The Pressure-Velocity (PV) Limit is a commonly used measure for defining the performance limits of polymers. This limit is in effect the melting point of the material. Thermal collapse limits the contact at high velocity, whereas mechanical strength limits the contact at high pressure.

Note that Friction Power Intensity (FPI) has the same dimensions as PV.

The FPI is equivalent to the PV limit multiplied by the friction coefficient.

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Energy Pulse

Product of FPI and contact transit time, EP takes into account length of time during which material is subjected to energy input during its transit of contact zone, where t_t is transit time in seconds

Energy Pulse: $E_P = \mu P V_s t_t / A$ J/mm²

EP is analogous to Archard Wear Law, however, it uses the friction force rather than applied load, which is perhaps more logical as it takes into account the rubbing conditions (but assumes that the friction coefficient can be measured)

Archard Wear Law: $\Delta V = k P V_s t_t / A$ mm³

EP can be regarded as an incremental contribution to wear or surface damage in contact. Sum of EPs can be used as a measure of total wear

Correct analysis of EP in real contact and subsequent modelling in experimental design significantly enhances probability of achieving a satisfactory emulation

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The Energy Pulse is the product of the FPI and the contact transit time. The EP therefore takes into account the length of time during which the material is subjected to energy input during its transit of the contact zone.

The Energy Pulse is analogous to the Archard Wear Law, however, the Energy Pulse equation uses the friction force rather than the applied load. This is perhaps more logical as it takes into account the work done in the contact.

Each Energy Pulse can be regarded as an incremental contribution to wear or surface damage in the contact. The sum of the Energy Pulses can be used as a measure of the total wear.

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Energy Pulse

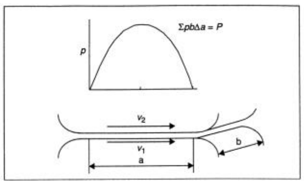
EP **=** μ **P Vs t_t / A** Jmm^{-2}

where

μ	=	friction coefficient	
P	=	applied load	N
Vs	=	relative sliding velocity	ms^{-1}
A	=	area of contact	mm^2
t_t	=	transit time	s

The transit times for the contact are:

Upper body:	t_t	=	a / v_2
Lower body:	t_t	=	a / v_1



Correct analysis of the EP in the real contact and subsequent modelling in the experimental design significantly enhances the chances of achieving a satisfactory emulation of sliding and combined sliding and rolling contacts.

It is important to note that in many machine components there can be very high FPIs but, because the contact durations are short, the EP is low and hence the incremental damage is low.

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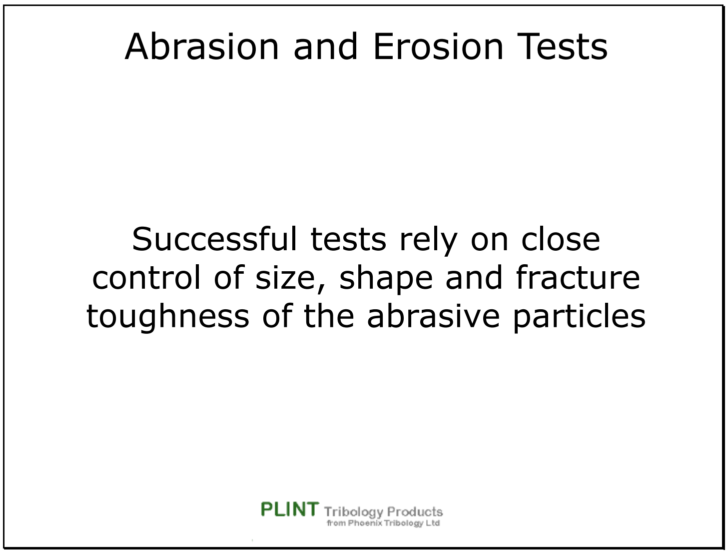

Abrasive and erosive testing of materials requires different types of machines from those discussed so far, as a free body is usually involved. Although the size of test samples may be small compared with real life applications, satisfactory data can only be generated by matching conditions within the real life process, hence these are effectively “real life” and not scale model test devices.

Successful tests rely on close control of particle size, shape and fracture toughness of the abrasive particles, and can only be achieved by careful sourcing and grading.

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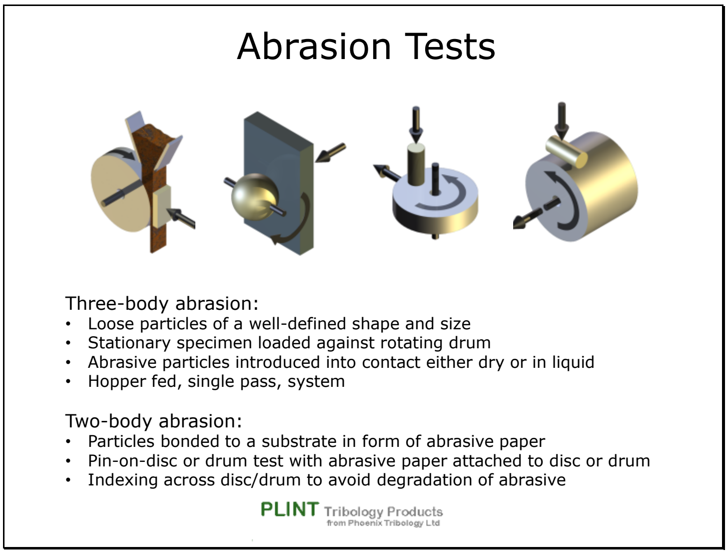

Abrasion tests either use loose particles of a well-defined shape and size or particles bonded to a substrate in the form of abrasive paper. The former results in three-body and the latter in two-body abrasion.

Three body abrasion test rigs typically comprise a stationary specimen loaded against a rotating drum with abrasive particles introduced into the contact either dry or with a liquid transport medium.

The use of hopper fed systems for abrasive particles improves the uniformity of supply, which in turns enhances control of the particle loading on the contact. To avoid contaminating the abrasive particles, whether dry or wet, with wear debris from the test specimens, it is normal to use a single pass system and not re-circulate the abradant.

In two-body abrasion rigs the principal problem is controlling the condition of the abrasive paper. If a pin-on-disc test is carried out with abrasive paper attached to the disc, the paper rapidly degrades and becomes clogged with wear particles. Indexing the pin across the disc in a spiral pattern, thus ensuring that fresh abrasive paper comes into contact with the pin at all times, overcomes this problem.

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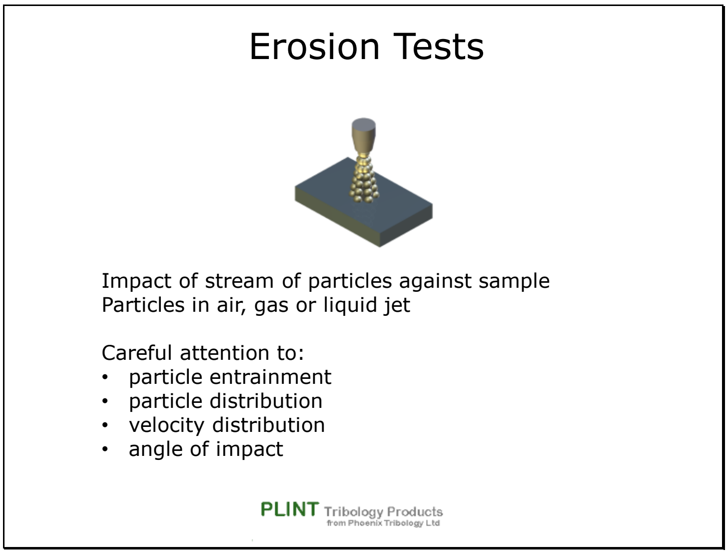

Erosion testers involve the impact of a stream of particles against a test sample. The particles may be introduced in an air or liquid jet. The use of hopper fed systems for abrasive particles improves the uniformity of supply, which in turns enhances control of the particle loading on the contact. Careful attention to particle entrainment, distribution and velocity distribution is essential to the generation of repeatable results.

There are a number of factors to do with the choice of particle and particle stream. The particle velocity and angle of impact both affect the dynamics of the erosive particle and the wear mechanism produced. The sharpness or angularity of the particle also affects the wear process.

A number of standards exist for abrasion and erosion, usually specific to a material or product group. All the standards define the grit to be used and wear rates are profoundly affected by the relative hardness of the grit to the material being abraded. This is an area where current research is leading to better laboratory test methods.

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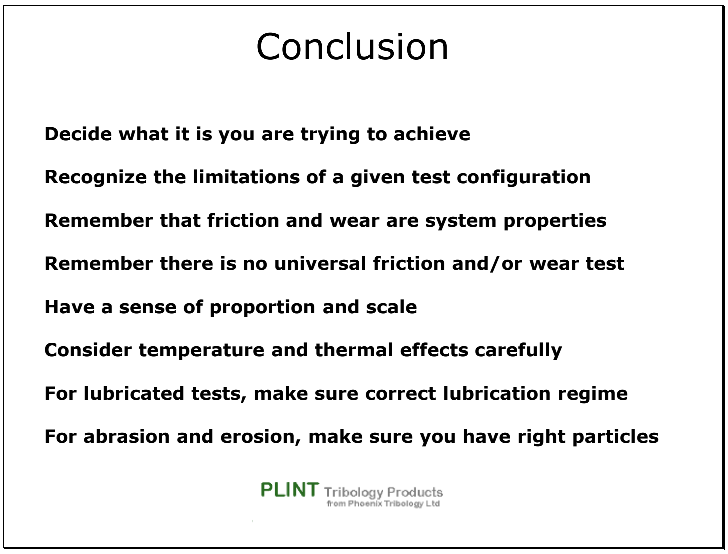

There are a few general rules worth remembering both before designing an experiment or a test machine and indeed once we have generated results:

There is no universal friction and/or wear test; there are far too many different wear mechanisms and frictional responses to consider.

Decide what it is you are trying to achieve, when you start thinking about your experiment.

Remember that friction and wear are system properties and that your test machine or rig is a system in itself.

Recognize the inherent limitations of a given test regime. You will have made assumptions in developing your model and you must understand what effect they may have on your results.

Have a sense of proportion and scale; don't push too much frictional energy into small specimens.

Consider specimen configurations and materials carefully; don't just go for the most convenient and easiest to source materials, because they may be inappropriate for the system you wish to model. Bear in mind that there is no bearing steel in an engine, for example.

Consider temperature and thermal effects carefully.

If you are performing lubricated tests, make sure you are operating under the correct lubrication regime, in other words you are at the correct point, and the point you want to be, on the Stribeck curve.

If you are performing abrasion or erosion tests, make sure you use the right abrasive materials.

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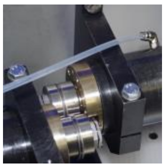
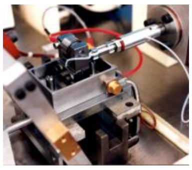
Summary





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