Effects of test machine dynamics on tribological response
or
Why do the results from my machine not correlate with the results from someone else's machine?

George Plint
MA, CEng, FIMechE
Tribology Trust Silver Medal 2017
Phoenix Tribology Ltd
info@phoenix-tribology.com

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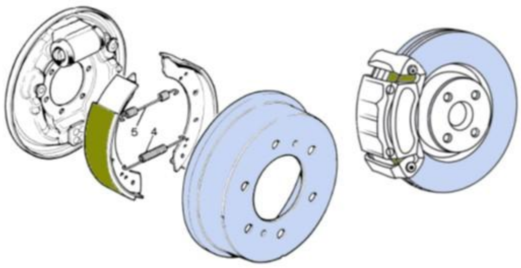
Various recently published articles have prompted me into writing some notes on tribometer dynamics, not least a discussion piece in TLT, about improving the repeatability and reproducibility of the four-ball test. I am not sure why someone might wish to do this, bearing in mind that it is a 90-year-old test of somewhat questionable validity, but said discussions did at least prompt me to consider the more general question: Why would the results from my machine not correlate with the results from someone else's machine?

And perhaps a supplementary question: how does this affect any attempts at big data analysis and the application of AI to tribology?

Czischos



Friction, wear and other tribological phenomena are system-dependent responses and not intrinsic material properties of the components of the tribological contact

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Test machine behaviour affects what the contact experiences and thus affects the evolution of friction and wear.

Professor Czischos made this point many years ago: friction, wear and other tribological phenomena are system-dependent responses and are not intrinsic material properties of the components of a given tribological contact.

The properties measured in an experiment using any tribological test machine are, by definition, system responses.

It follows that data generated from any properly calibrated test machine must be valid, but only for the response of the material pair in that particular tribo-system.

Obviously, different systems give obviously different responses.

This rather begs the question, how should we compare these different system responses, when the dynamics of said test systems have not been defined or analyzed.

Test Standards

Typically define:
- Test geometry
- Specimen preparation
- Load
- Contact pressure
- Temperature
- Lubrication or atmosphere
- Velocity
- Test duration

No mention of characteristics affecting system response:
- Stiffness and inertia of loading system
- Rate of acceleration of motor
- A.C. or D.C. drive “stiffness”
- Signal band-width and frequency response of sensors and measuring systems

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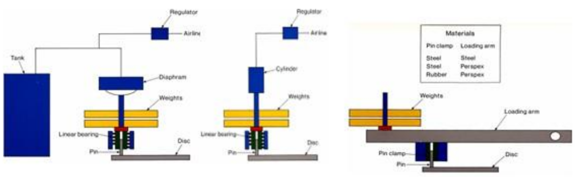
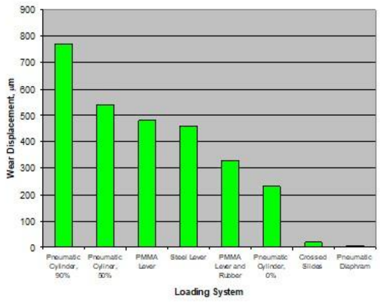
Many test standards define operational parameters, for example, specimen preparation, load, velocity, temperature, and test duration, but fail to mention key system characteristics, affecting the system response, such as the stiffness and inertia of the loading system, the rate of acceleration of a motor, the “stiffness” of the A.C. or D.C. drive, or indeed the signal band-width and frequency response of sensors and measuring systems.

In this talk I will explore some of the effects of these undefined parameters on various long established test systems. The data presented will hopefully provide an explanation for why nominally identical test procedures run on different machines, with similar test geometries, give substantially different results.

Loading System Stiffness and Inertia – Pin on Disc Machine





Work carried out by the UK National Physical Laboratory, using an in-house designed pin on disc machine, to investigate the effects of different loading systems on the wear rate of Alumina specimens, under nominally identical test conditions. A range of different mechanisms were used to apply the load

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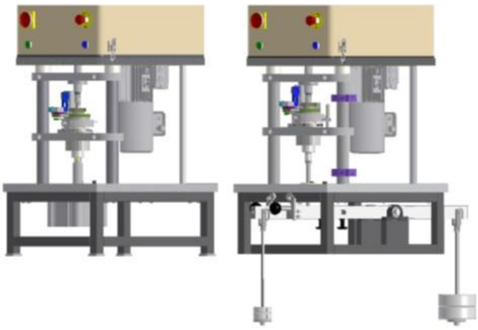
We know that, regardless of the tests we wish to perform and the configuration of machine we have chosen, mass, stiffness and method of load application all have significant influence on the results produced by a given design of test machine. How significant are these effects?

The work illustrated here was carried out by the UK National Physical Laboratory following the results of the Versailles Project on Advanced Materials and Standards. The VAMAS inter-laboratory tests had produced a factor of three in the distribution of reported friction coefficients and an order of magnitude variation in wear coefficients.

For the NPL tests, a range of different mechanisms were used to apply nominally the same load on a pin on disc machine. Substantial variations in wear rate were generated, simply by altering the dynamics of the loading system.

Loading System Stiffness & Inertia
Four-Ball Machine



Extreme pressure tests performed by the same operator, with a single machine, with interchangeable loading systems

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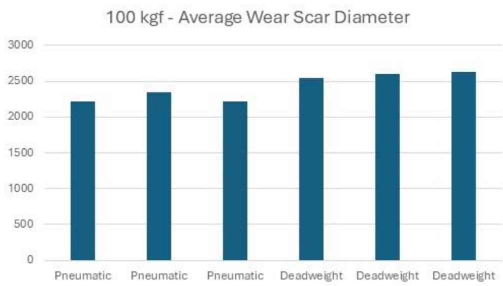
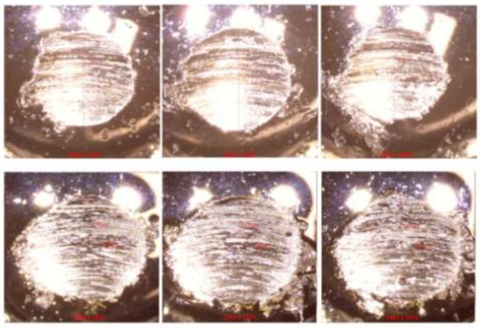
More recently, we used one of our standard sliding four-ball machine to provide a ready platform for comparing pneumatic loading with dead-weight loading. Extreme pressure tests were performed by the same operator, with a single machine, with interchangeable loading systems. Load was calibrated against a load cell inserted between the lower specimen carrier and the machine spindle, ensuring that the static load with both systems was identical. It therefore seemed reasonable to assume that any change in tribological response could sensibly be attribute to the different loading mechanisms.

Loading System Stiffness & Inertia
Four-Ball Machine





Wear scar diameter with dead-weight loading approximately 15% larger than with pneumatic loading

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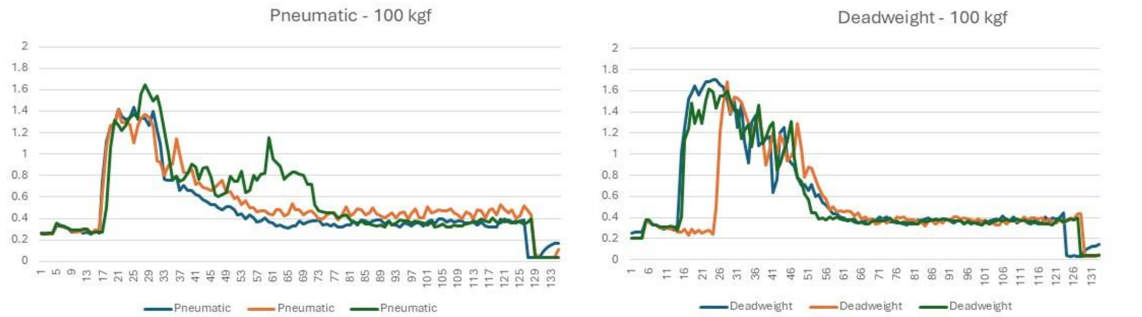
Multiple tests were run at different loads, establishing that, on average, at nominally identical loads, the wear scar diameter with dead-weight loading was approximately 15% larger than with pneumatic loading. This is an example of how the higher stiffness and inertia of the loading system gives rise to higher wear rates.

Loading System Stiffness & Inertia
Four-Ball Machine



Differences in frictional response with different loading systems, but hard to draw any conclusions from data - to be expected with tests that only run for 10 seconds

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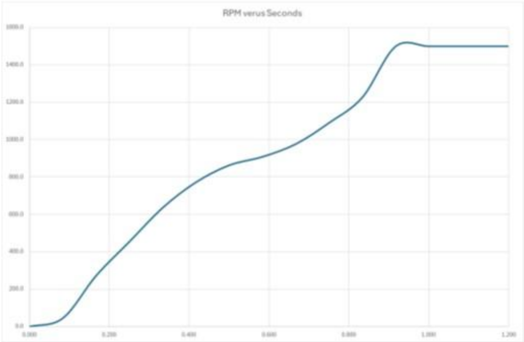
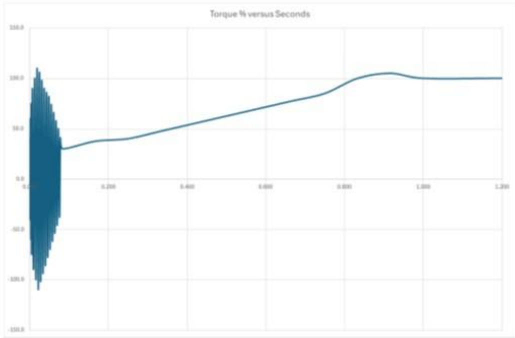
Although there were differences in frictional response with the different loading systems, it is hard to draw any firm conclusions from the data. This is perhaps to be expected with a relatively severe test procedure, that only runs for 10 seconds.

Drive System - Four Ball Machines





Original four ball machines used direct on-line starting 4-pole A.C. motors, which accelerate quite quickly, but not in a not controlled fashion

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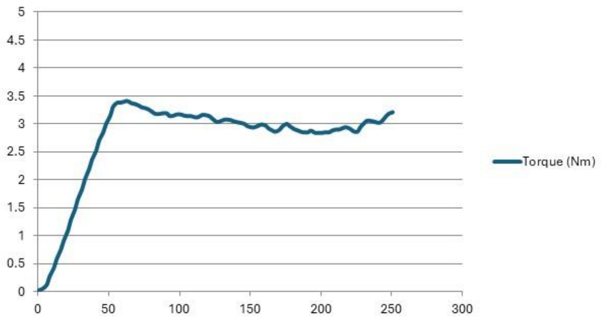
The original four ball machines used direct on-line starting 4-pole A.C. motors, which accelerate quite quickly, but not in controlled fashion. It was never considered necessary to determine how different A.C. motors might accelerate at the start of a test or to compare their performances.

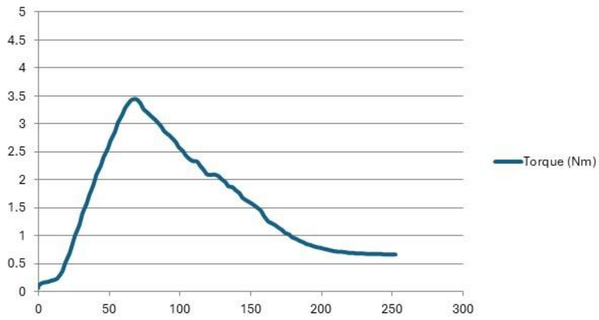
On start-up, these motors jerk themselves into motion, generating self-excited torsional vibration, with, for a short period of time, significant torque reversals. Could variations in this start-up behaviour have influenced test results, repeatability and reproducibility?

Drive System - Four Ball Machines

Torque (Nm) versus Revolutions - 0.1 s ramp
127 kg - Seizure Ball Scar



Torque (Nm) versus Revolutions - 0.5 s ramp
128 kg - Non-seizure Ball Scar



A.C Vector Motor & Drive: Tests were performed at varying loads & spindle accelerations to establish point of transition between wear & seizure

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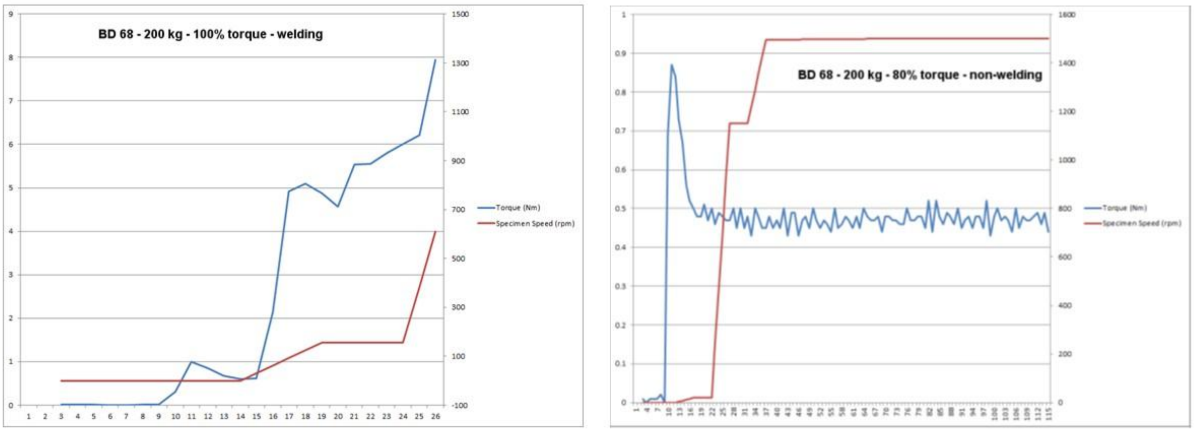
A.C. vector motors and vector drives allow precise control of speed ramps. This capability has allowed us to investigate the effect of test spindle acceleration on the severity of the four-ball EP test. The lubricant chosen in this case was Shell Tellus 22 and the ramp time for the acceleration to the test speed of 1500 rpm varied between 0.1 and 1 second.

Tests were performed at varying loads and spindle accelerations to establish the point of transition between wear and seizure. Data for two different acceleration tests, 0.1 seconds and 0.5 seconds, are given here. The faster the acceleration, the more severe the test.

Drive System - Four Ball Machines



Reducing available torque from motor, by limiting drive-current, results in slower acceleration of test spindle & hence a less severe test

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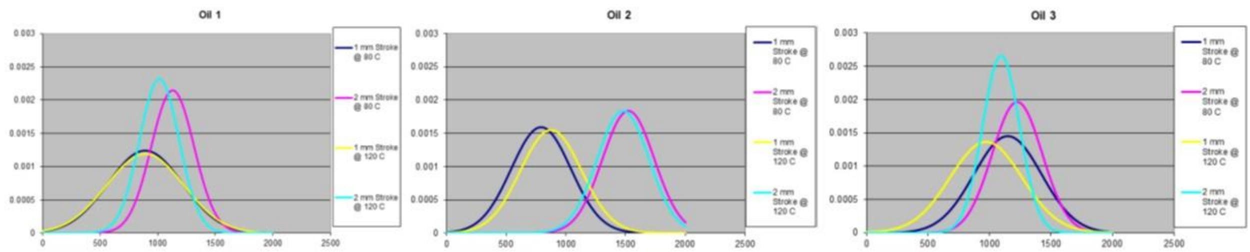
Our tests confirmed that spindle acceleration, with an electrically and mechanically stiff driving system, influences the initial seizure load. So, what would happen with a less stiff drive, in other words, with a motor where the acceleration is affected by the frictional resistance?

To investigate this, tests were run with the drive-current limit reduced, thus making spindle acceleration more dependent on friction torque resistance. Reducing the available torque from the motor, by limiting the drive-current, results in much slower acceleration of the test spindle and hence a much less severe test.

Reciprocating Drive System - Electromagnetic



ASTM D7421 - 11 Standard Test Method for Determining Extreme Pressure Properties of Lubricating Oils Using High Frequency, Linear Oscillation Test Machine

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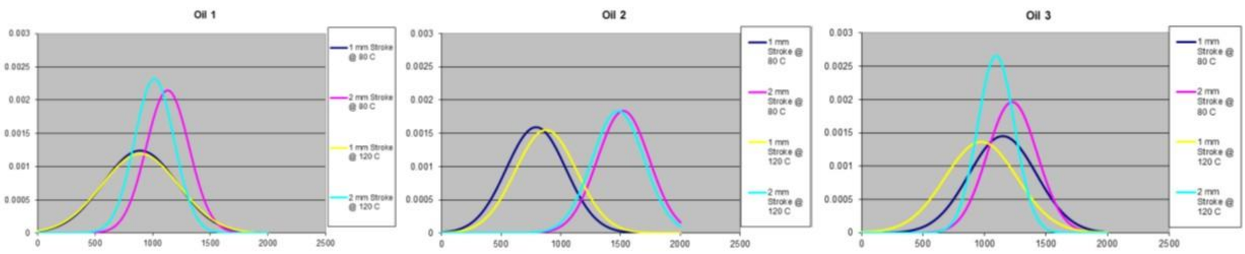
Now, what about reciprocating drive systems. Let’s start with voice coil actuators.

We do not have to go far, indeed, we do not have to do any experiments, to discover examples of effects of reciprocating drive system stiffness. There is plenty of information readily available for analysis in ASTM standards, in particular, the precision and reproducibility statements and the published data from inter-laboratory tests.

This example comes from ASTM D7421 - 11 Standard Test Method for Determining Extreme Pressure Properties of Lubricating Oils Using High Frequency, Linear Oscillation Test Machine. Table 1 gives results for tests run on three different oil samples at either 1 mm or 2 mm stroke. Plotting standard deviation gives these graphs.

Reciprocating Drive System - Electromagnetic



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

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Analysis of the data demonstrates that:

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

This is a good example of a system response, in this case, the response of the method of actuation to changes in stroke or resisting force. This is indeed a “system” response, but it has more to do with the method of actuation than the fundamentals of the tribo-contact. We are, in effect, using the tribo-contact as a friction brake dynamometer to characterize the performance of the electro-magnetic actuator.

Drive System –
Electromagnetic versus Mechanical

	Mechanical	Electro-magnetic
High Lubricity Reference:		CEC DF-92-02
Mean wear scar dia. (MWSD)	426	404
Corrected wear scar dia. (WS1,4)	443	420
Low Lubricity Reference:		CEC DF-70-00 SF 006
Mean wear scar dia. (MWSD)	692	619
Corrected wear scar dia. (WS1,4)	706	633
Range:		
MWSD (high – low)	266	215
WS 1,4 (high – low)	263	213

ASTM D6079 fuel lubricity tests, using mechanically driven short stroke reciprocating tribometer & PCS HFRR

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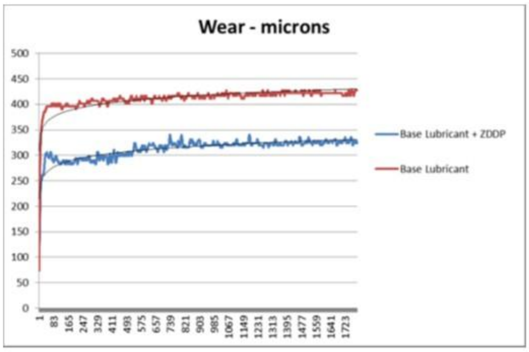
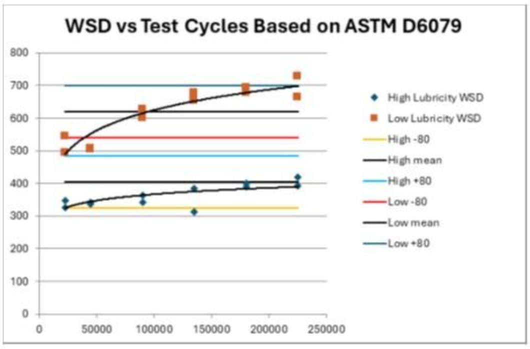
Tests using a mechanically driven short stroke reciprocating tribometer, showed that such devices consistently produce a larger wear scar than the PCS HFRR instrument. The difference in wear scar size is particularly marked for the low lubricity reference fluid. This arises because of differences in the method of actuation: the HFFR uses an electro-magnetic oscillator, which is a force generating as opposed to a displacement generating device. The resulting stroke length may vary as the frictional resistance of the contact varies and the control system adjusts the driving force to compensate.

These differences do not represent a problem as the specified test is a comparative test and the result is a simple offset bias, with the mechanically driven reciprocating tester producing a larger discrimination between fuels with different lubricities, compared with the HFRR.

Drive System –
Sliding Hertzian Point Contact





Most of the wear (or plastic deformation) occurs at beginning of test. Once the difference in wear between samples has been established, number of cycles over which test is run is somewhat arbitrary

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It is perhaps worth noting that in a sliding hertzian point contact test, most of the wear (or plastic deformation) occurs at the very beginning of the test. Once the difference in wear between candidate samples has been established, the number of cycles over which the test is run is somewhat arbitrary.

ASTM D6079 gives a reproducibility figure of 80 microns as the (approximately) 95% confidence level. Plotting the average wear-scar measurements for tests of different duration indicate that most of the measurements fall within the reproducibility limits, indicating that once the initial wear has taken place, further cycles result in very limited additional wear.

An acceptable result can be achieved within the limits of the standard, running a mechanically oscillated test, for the required time, at any frequency from 20 to 50 Hz!

Force Measuring System Resonance

- Force measuring system measures more than just friction or load
- All force measuring systems have a natural frequency, giving rise to resonance
- Resonance causes a measurement error
- Depending on excitation frequency, resonance error may be amplified or attenuated

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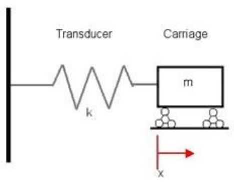
Now, let us move on to force measuring systems.

All force measuring systems have a natural frequency, giving rise to resonance and resonance causes a measurement error. Depending on excitation frequency, the resonance error may be amplified or attenuated.

Natural frequency is the frequency at which a system will vibrate after an initial disturbance. It is determined by the system's physical properties, such as mass and stiffness.

Resonant frequency is the frequency at which a system will vibrate with the largest amplitude when an external force is applied. When the frequency of the external force matches the system's natural frequency, resonance occurs, leading to a large amplitude of vibration.

Transmissibility of Spring-Mass System



$$m\ddot{x} = -kx$$

$$\ddot{x} + \frac{k}{m}x = 0$$

$$\therefore x = A\sin\left(\sqrt{\left(\frac{k}{m}\right)}t\right) + c$$

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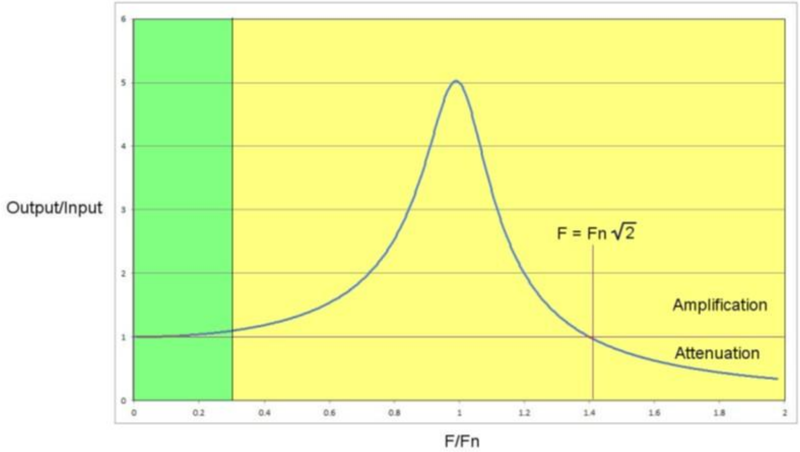
Any force measuring system, connected to a known mass, such as specimen assembly, not only measures some externally applied force, such as load or friction, but will also act as an accelerometer, so will sense any externally excited vibration.

An oscillating force, applied at the resonant frequency of a dynamic system, will cause the system to oscillate at a much higher amplitude than when the same force is applied at non-resonant frequencies.

Transmissibility of Spring-Mass System



In order to keep measuring errors low, dynamic force measuring systems should not be used at frequencies above about 0.3 of its resonant frequency

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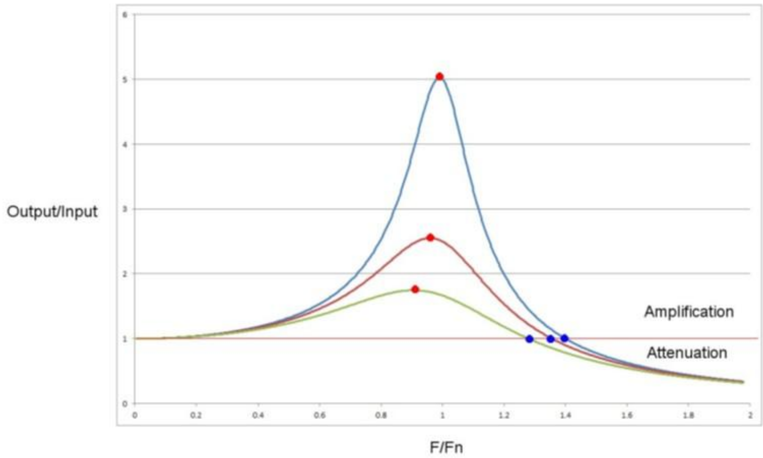
The transfer function of a measuring system is the ratio of the output of the system to the input to the system. The transfer functions of measuring systems for load or friction, in a tribological experiment, are rarely analyzed or specified.

The closer the excitation frequency gets to the natural frequency, the greater the measurement error, with the output signal considerably amplified with respect to the input signal. As the excitation frequency exceeds the natural frequency, the amplification progressively declines until the excitation frequency equals the natural frequency x Root 2. At this point, the output signal becomes progressively more attenuated.

The disheartening thing about this is that we can only have absolute confidence that our dynamic force measurement is free of errors due to resonance at excitation frequencies of nought and natural frequency x Root 2!

It is generally accepted that, to keep measuring errors low, a force measuring system should not be used at frequencies above about 0.3 of its resonant frequency. It is standard practice to apply input filtering to limit the signal bandwidth accordingly.

Transmissibility of Spring-Mass System



Increasing damping reduces natural frequency and introduces a phase shift

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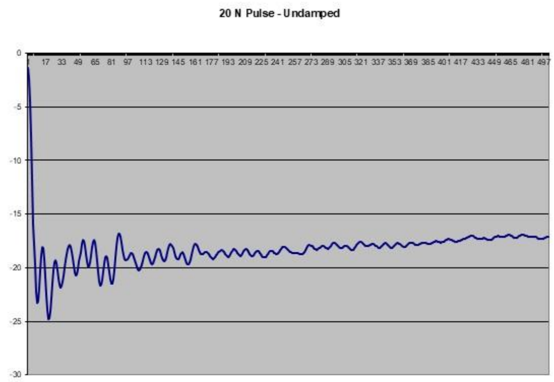
It is worth noting that damping has two effects on the system; it reduces the amplitude of the resonance, but it also reduces the natural frequency of the system.

Friction Force Dynamic Response – TE 77



- Reversal of friction force at beginning of stroke “plucks” force measuring system rather like plucking a string
- Magnitude of resulting oscillations function of magnitude of “plucking” force
- Perturbation frequency function of system natural frequency

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The graph shows an unfiltered friction signal from a standard TE 77 High Frequency Friction Machine, running at 15 Hz and 15 mm stroke.

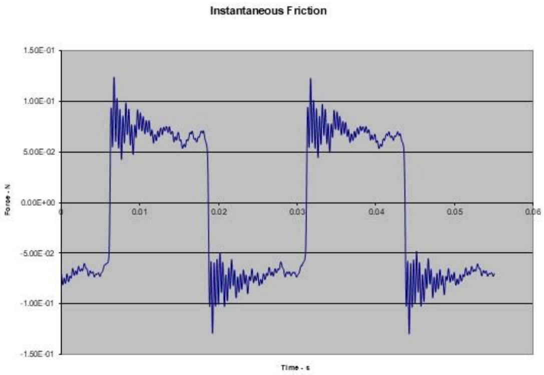
The transition from static friction to dynamic friction at the beginning of each stroke and the resulting resonance is apparent in all unfiltered reciprocating tribometer friction force measurements, assuming that the measuring system has sufficient signal bandwidth.

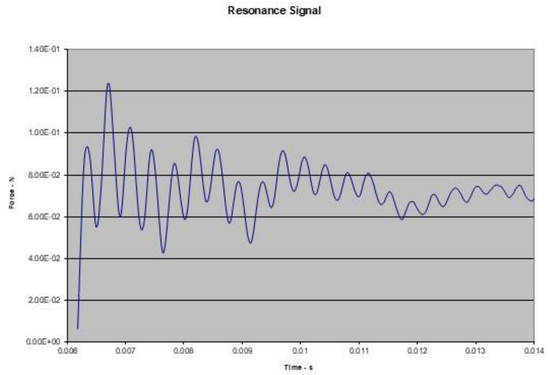
Whereas the initial peak signal at the start of the stroke may indicate the limiting static friction, the subsequent oscillating spikes are not friction effects! The signal perturbation cannot be anything other than a resonant harmonic force giving rise to oscillation.

Reversal of the friction at the beginning of the stroke effectively “plucks” the force measuring system, rather like plucking a string. The magnitude of the resulting oscillations is of course a function of the magnitude of the “plucking” force.

The rate of decay of the resulting vibration signal is a function of the inertia of the sample assembly, the stiffness of the transducer, both of which tend to be constant, and the variable damping coefficient of the system, which is a function of the friction in the tribo-contact.

Friction Force Dynamic Response - SRV





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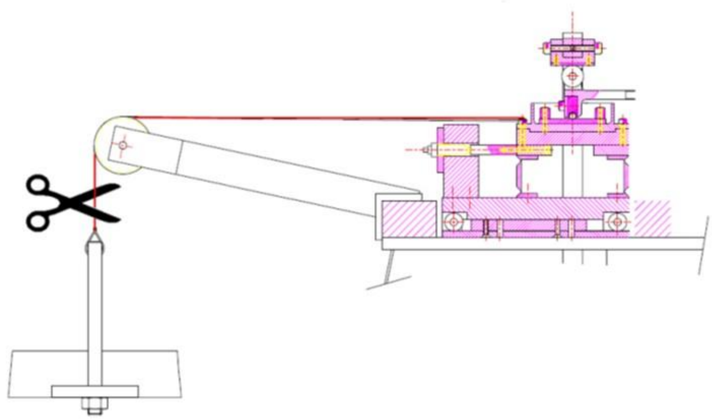
Here is an unfiltered friction trace from an SRV3, showing similar behaviour.

Friction Force Dynamic Response



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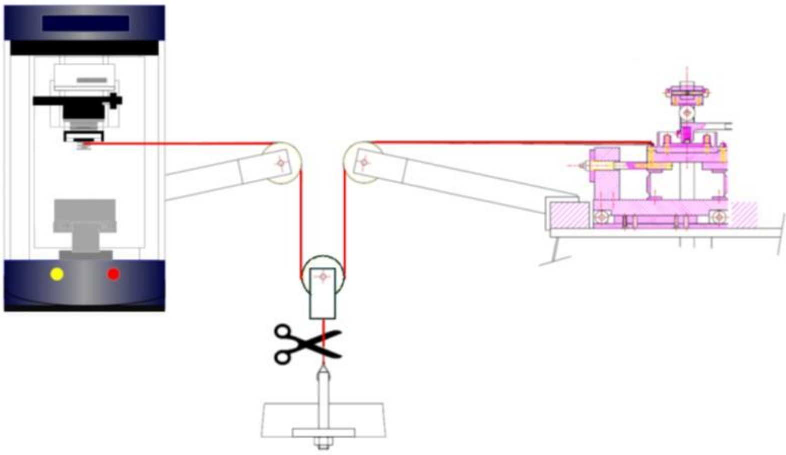
Confirmation of this resonance effect and measurement of the test assembly’s natural frequency can by demonstrated by a simple experiment: the system response to a step change input. A mass is attached to the transducer assembly by means of a string and pulley arrangement. Cutting the string imposes a step change in input.

This is normally done without specimens in contact. You can, however, repeat the experiment, with specimens in contacts, to explore the damping effect of friction on the system response.

Friction Force Dynamic Response



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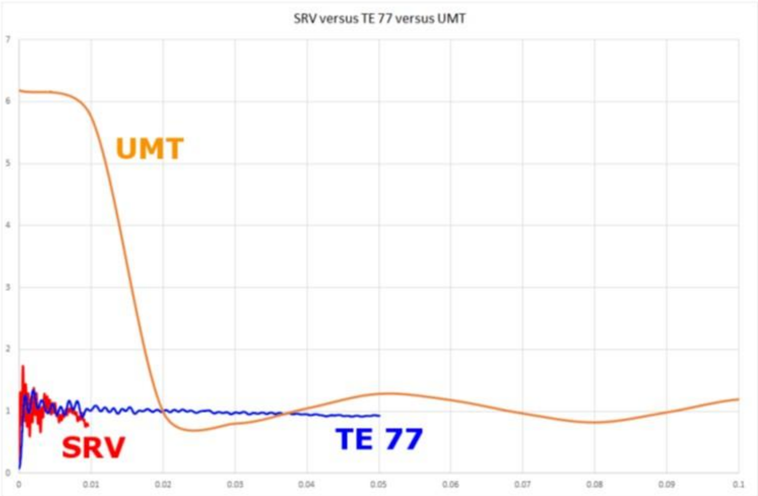
We can, if we feel so inclined, directly compare the response of two different machines to the same input, with the set-up shown here.

Friction Force Dynamic Response



- TE 77: 830 Hz
- SRV: 2730 Hz
- UMT: 20 Hz

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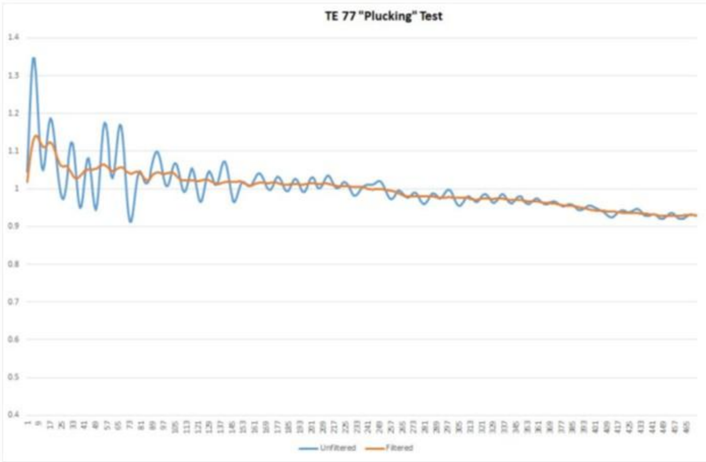
Here are comparative results for a TE 77, an SRV3 and a UMT, showing the response and resonant frequencies of the three different measuring systems.

The UMT has a much lower resonant frequency than either the TE 77 or the SRV. This is because the device has a relatively high mass specimen assembly, cantilevered off a relatively low stiffness force transducer.

The SRV3 has the highest natural frequency because the specimen is attached directly to a very high stiffness piezo transducer.

Dynamic Response - Effect of Filtering



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We can “tidy up” a signal by applying filtering; this effectively limits the signal bandwidth of the measuring system, removing the higher frequency components we wish to hide.

Dynamic Response – Sampling Rate

- Same characteristic filter on all channels of high frequency systems
- Sample at high enough a rate to preserve information in original signal
- Nyquist: minimum acceptable sampling rate is twice maximum frequency of interest
- Gain bandwidth product is estimate of the information carrying ability of the system
- Sampling rate of system should be well matched to its signal bandwidth
- Aliasing occurs when sampling rate is too low for frequency response of system

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Having discussed the response of our force transducers and how we might clean up the signal it is probably worth mentioning something about sampling rate and the possibility of aliasing.

If we are going to use filters, the same characteristic filter should be used on all channels, especially in high frequency systems, to ensure that the information from different channels can be directly correlated and are not subject to differing time delays.

When sampling the signals processed by a filter it is important to sample at high enough a rate to preserve the information in the original signal. The Nyquist Sampling Theory indicates that the minimum acceptable sampling rate is twice the maximum frequency of interest. The measured amplitudes of signals at half the sampling rate are attenuated to 64% of their true value.

Estimating the gain bandwidth product (an estimate of the information carrying ability of the system) shows that once an adequate sampling rate has been chosen, sampling faster than that may produce more data but not more information!
The sampling rate of the system should be well matched to its signal bandwidth to preserve information content.

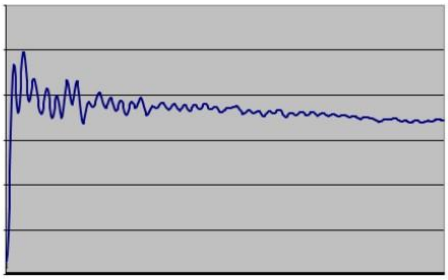
Aliasing occurs when the sampling rate is too low for the frequency response of the system. As a simple example of aliasing, consider sampling the levels of illumination by looking out of the window, at the same time, just once per day. If the observations were always made at midnight, the collected data would imply that it was always dark outside. If the observations were always made at midday, the collected data would imply that it was always light outside.

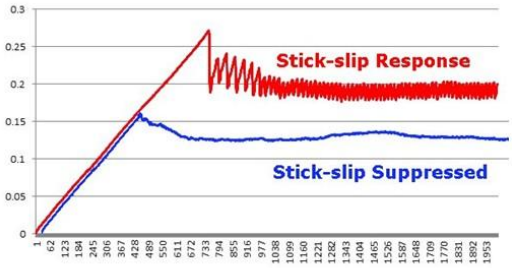
Now, assuming that there are twelve hours of darkness and twelve hours of light, consider the effect of taking three samples per day, at eight hours interval. The observations would indicate twice as much light as dark, or vice versa.

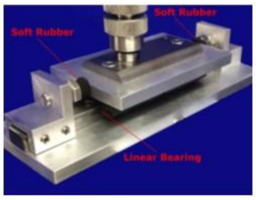
As the signal is varying twice per day (from day to night) the absolute minimum sampling rate to avoid aliasing would be four equi-spaced observations per day. In practice, a bit more than four samples per day would be a safer option, so a bit more than the Nyquist theory suggests.

How often is published friction data supported by any form of meaningful analysis of measuring system bandwidth, filtering and sampling rate?

Resonance versus Stick-Slip







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It is important not to confuse resonance with stick-slip. The former is essentially a symmetrical wave form, whereas stick-slip is not.

We can promote stick-slip behaviour by including a degree of compliance in our system. This emphasizes the deflection during the stick phase and the rapid decrease in friction during the slip phase. We end up with a saw tooth, not a symmetrical wave form.

This is a nice example of altering the system response by altering the stiffness in the friction plane.

Force Measuring Systems

FOR A GIVEN FORCE, THE VALUE MEASURED WILL DEPEND ON THE PROPERTIES OF THE MEASUREMENT SYSTEM

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It will be apparent that for a given true force input, the measured output value will depend on the properties of the complete measurement system.

Model Test Systems

Friction - Contact	Fc		Pressure	P
Friction - Measured	Fm		Load Spring Stiffness	S
Load - Contact	Lc		Inertia - Motor Armature	Jm
Electro Magnetic Torque	EMT		Inertia - Disc & Carrier	Jd
Electro Magnetic Force	EMF		Inertia - Motor & Flywheel	Jf
Inertia	J		Torque - Shaft	Ts
Mass	M		Mass - Specimen Carrier	Mc
Stiffness	K		Mass - Voice Coil Armature	Ma
Damping Coefficient	C			

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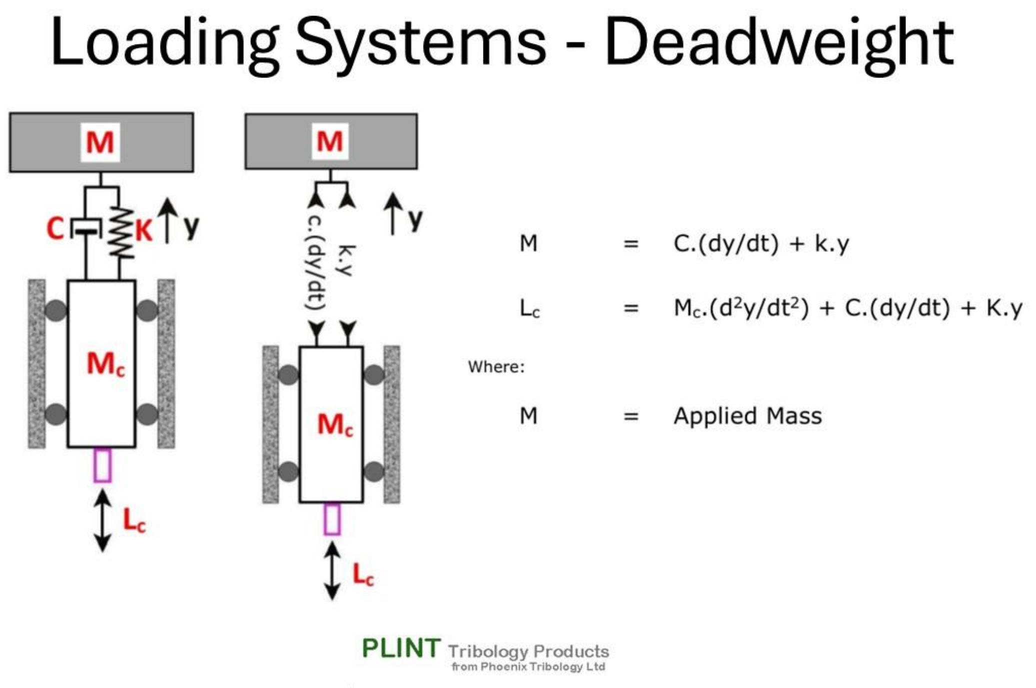
Having given numerous examples of test system response, based on simple demonstration experiments and analysis of published data, we can now give some thought as to how we might model the dynamic response of our test systems.

The table here outlines some of the parameters we might wish to include in our models.



Let us start with loading systems.

This is a general model of a deadweight loading system, with vertical excitation generated by the contact. The associated free-body diagram on the right, from which we derive the following equations:

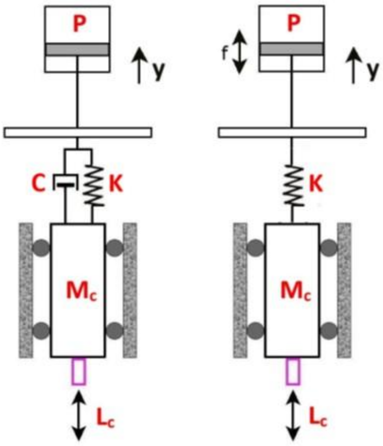
$$M = C.(dy/dt) + k.y$$

$$L_c = M_c.(d^2y/dt^2) + C.(dy/dt) + K.y$$

Where:

M = Applied Mass

Loading System - Pneumatic



Non-velocity dependent damping may come from friction in the cylinder piston rings, with the compressed air acting as a spring

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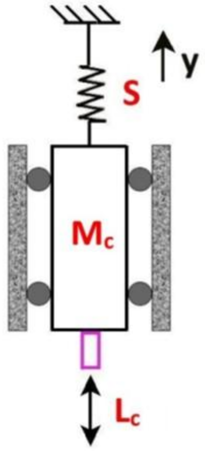
This is a general model of a pneumatic loading system, with vertical excitation generated, once again, by the tribological contact. In practice, non-velocity dependent damping may come from friction in the cylinder piston rings, with the compressed air acting as a spring.

Loading System - Compressed Spring



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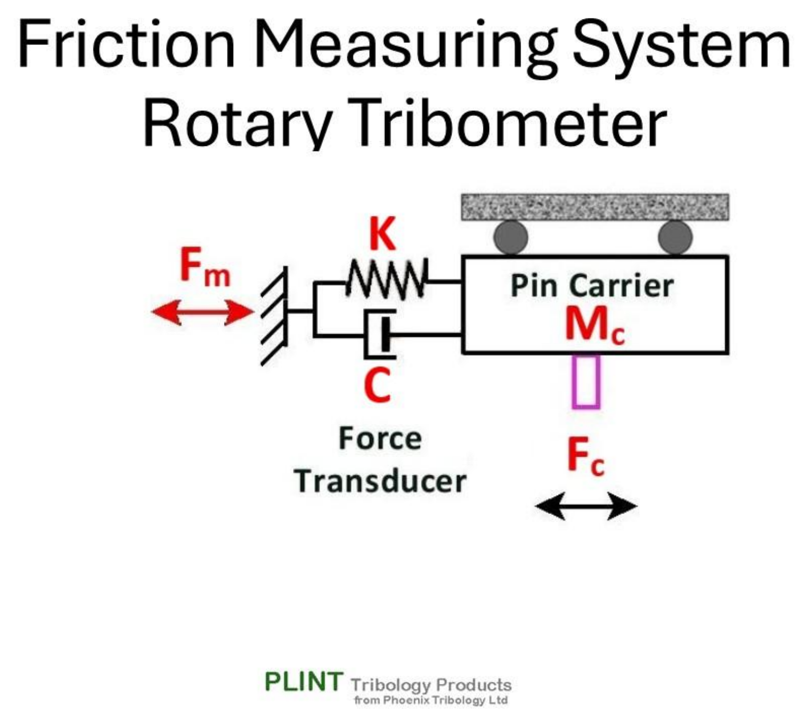
This is what a spring loading system might look like.

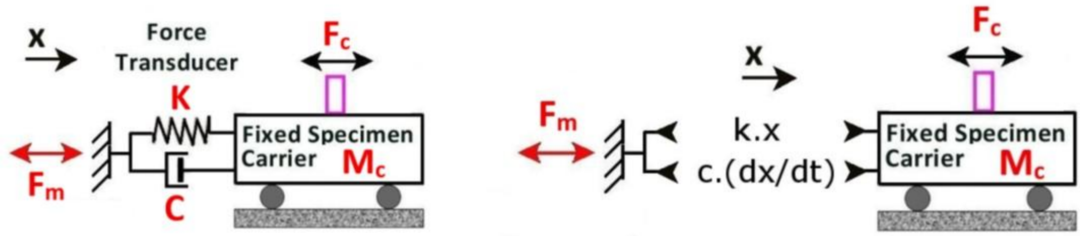


Let us now consider what the friction measuring systems might look like.

This model assumes that friction in the rotary tribometer is measured at the pin and not with an in-line torque transducer. System excitation F_c is the time dependent friction generated in the contact.

Friction Measuring System Reciprocating Tribometer

Contact friction affects damping coefficient



$$F_m = C.(dx/dt) + K.x$$

$$F_c = M_c.(d^2x/dt^2) + C.(dx/dt) + K.x$$

$$F_c = \mu.L_c$$

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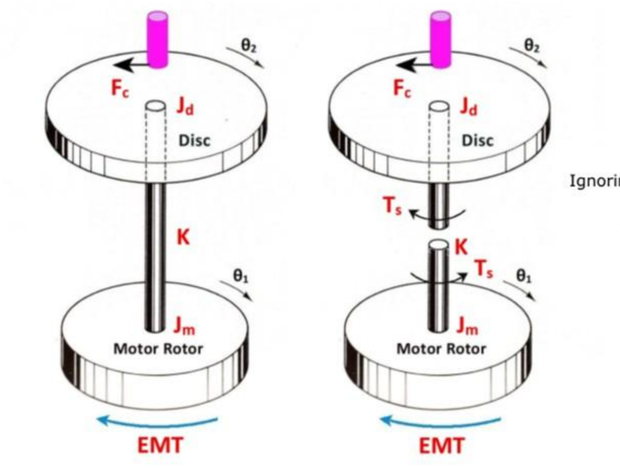
This is a typical arrangement for sensing friction in reciprocating tribometers, along with its free body diagram.

It is important to note that in the friction plane the contact friction affects the damping coefficient, hence the damping coefficient is a function of the friction in the contact: C = f(F_c). We saw this illustrated in the earlier practical experiments.

Drive System - Rotary



Ignoring a torsion damping term:

$$T_s = J_d.(d^2\theta_2/dt^2) + K.(\theta_1 - \theta_2) + F_c.R$$

$$EMT = J_m.(d^2\theta_1/dt^2) + K.(\theta_1 - \theta_2) + T_s$$

Not "tethered" to a fixed point

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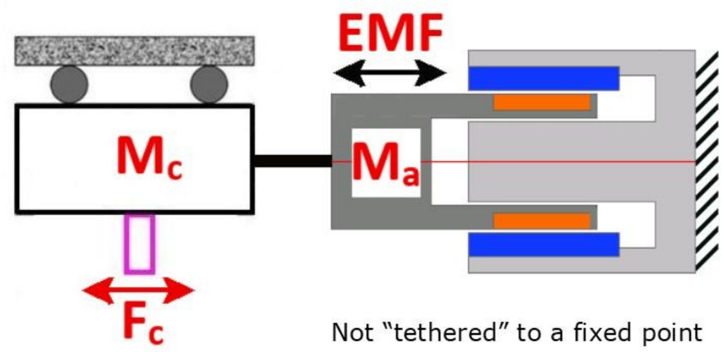
Unlike the friction and loading systems, the elements of the motor drive system are not tethered to a fixed point. There are two mechanisms for exciting torsional vibration: periodic fluctuations in motor torque and fluctuations in frictional response of the tribo-contact. The latter could be caused either by actual tribological effects or by fluctuations in load caused by the loading system dynamics.

Reciprocating Electromagnetic Drive



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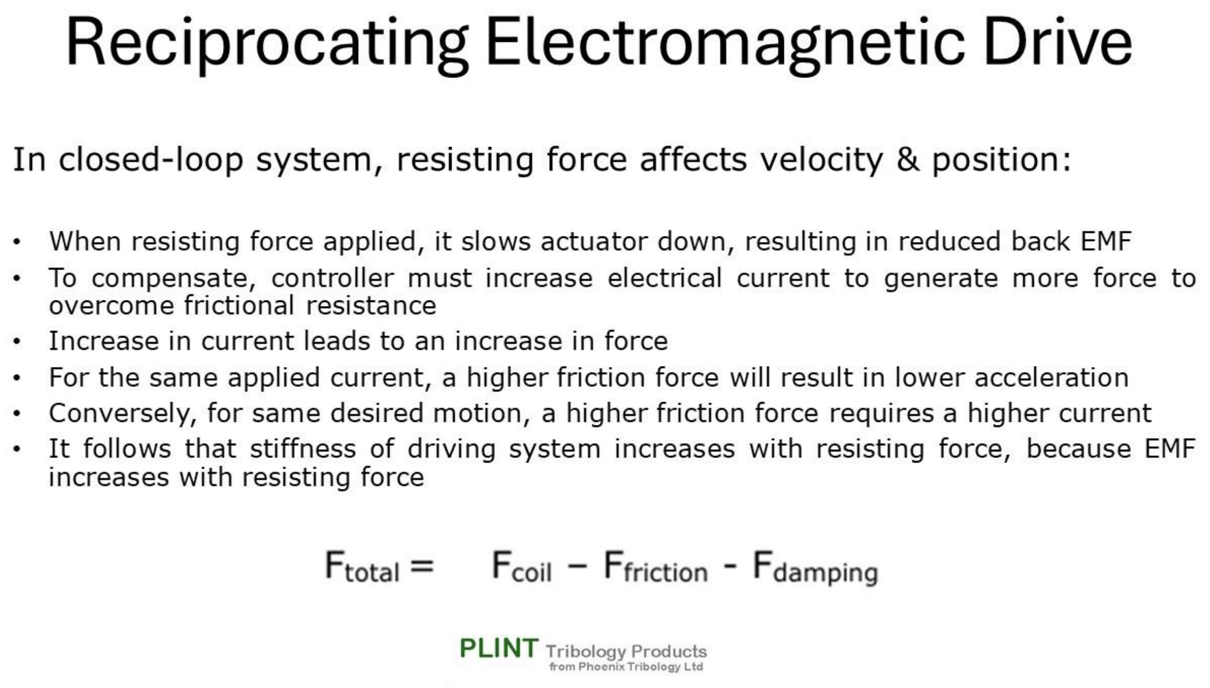
Let us now consider a reciprocating electro-magnetic drive.

In the case of a voice coil actuator (VCA) system, the moving component, hence moving specimen, is not rigidly tethered to a fixed point, but is supported on flexural elements and electro-magnetically coupled to the fixed component.

The output of a VCA varies significantly with resisting force and performance is governed by a balance of forces and electrical properties, with the constant Lorentz force it produces being counteracted by the resisting friction force.



In a closed-loop system, a resisting force will affect the actuator's velocity and position.

- When a resisting force is applied, it will slow the actuator down, resulting in a reduction in back EMF.
- To compensate, a controller must increase the electrical current to generate more force to overcome frictional resistance.
- An increase in current leads to an increase in force.

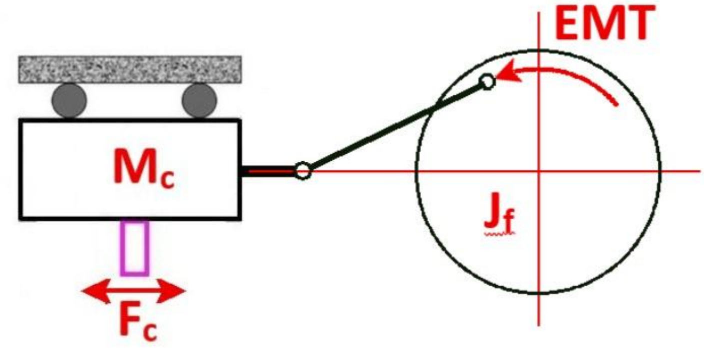
The motion equation for a VCA can be represented as:

$$F_{total} = F_{coil} - F_{friction} - F_{damping}$$

F_{total} is the net force available for acceleration or deceleration. This means that for the same applied current, a higher friction force will result in lower acceleration. Conversely, for the same desired motion, a higher friction force requires a higher current. It follows that the stiffness of the driving system increases with resisting force because EMF increases with resisting force.

The interaction between the VCA and the resisting force determines the overall system dynamics. This helps to explain the behaviour described in our earlier experimental analysis.

Reciprocating Mechanical Drive



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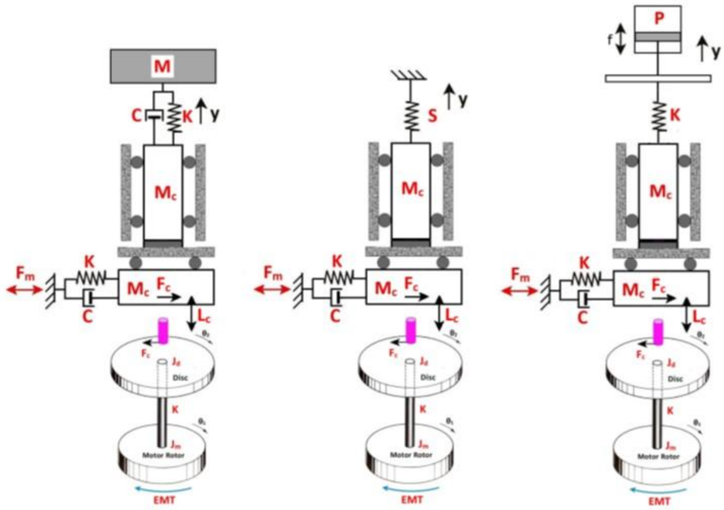
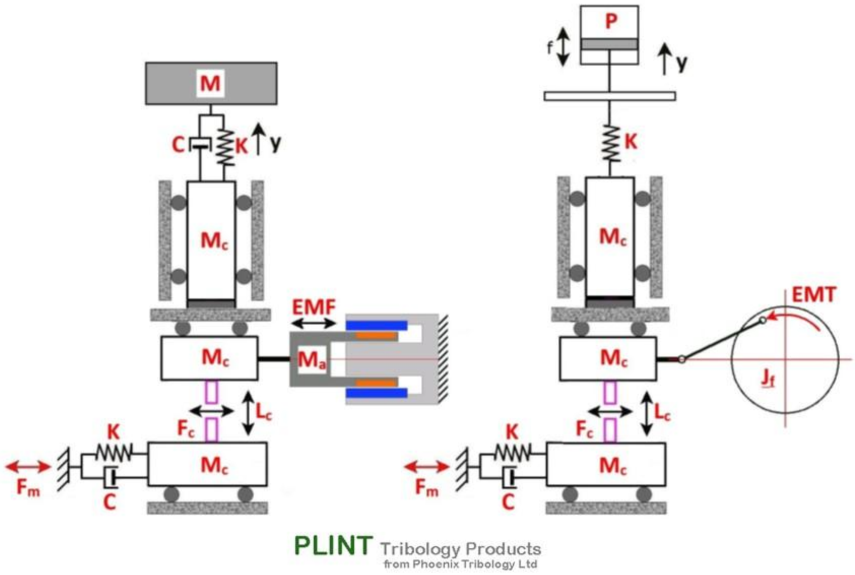
Mechanical drive systems are essentially tethered displacement generators and are usually designed with sufficient inertia, in the form of a flywheel, not to be susceptible to variations in resisting force.

Complete Pin on Disc Systems



Combining the three elements, the loading system, the friction measuring system and the driving system, can result in systems as illustrated above, the dynamic response of which will depend on the mechanical characteristics of each element and the three sources of perturbation, noting that the latter are not independent variables.

Complete Reciprocating Tribometers



This is how reciprocating tribometers might be modelled.

Load and force measuring systems are usually de-coupled with these machines and low inertia, heavily damped, loading systems are preferred; dead-weight loading systems can only sensibly be used at low reciprocating frequencies.

Inadequate Friction Models

Having modelled our test system, we could produce a finite element model, but only if we know the constitutive relationships between load and friction and the constitutive relationship between friction and sliding velocity

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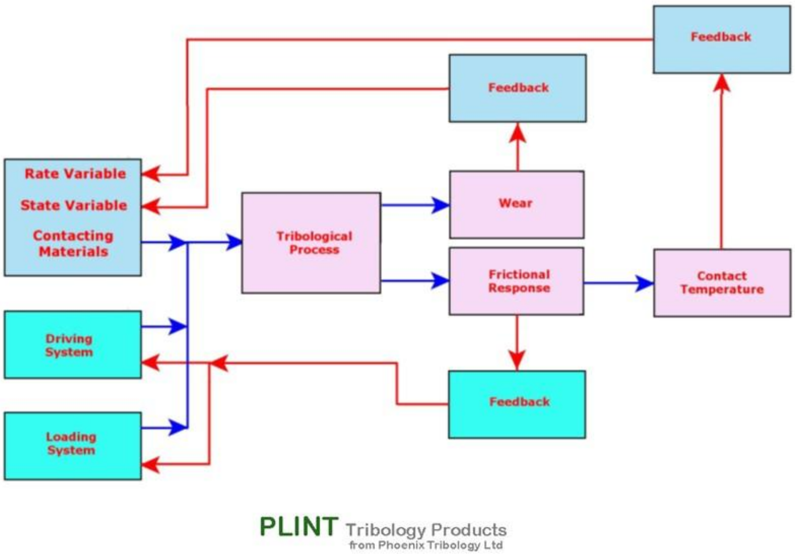
Having defined our model test systems, with the necessary mechanical information regarding mass, stiffness, inertia etc, we could in theory produce a finite element model of our test system. We have, however, managed to avoid one key requirement: the thorny subject of the constitutive relationship between load and friction and the constitutive relationship between friction and sliding velocity.

It will be apparent that tribo-contacts in general, even with significant stiffness in the friction plane, are likely to be subject to small amplitude vibrations. These may not only affect the frictional response but may also affect the dynamic load.

We know the limitations of the Amontons-Coulomb laws of friction; in view of the dynamics of our diverse test systems, how far can we progress without replacing Coulomb's law with some form of rate-and-state friction model?

What might our model look like?



It is clear that with both a tribo-contact and a tribo-test systems, we are dealing with systems with internal feedback.

This schematic shows a possible feedback mechanism for a friction and wear coupled test system.

A rate-and-state friction model describes friction as dependent on the instantaneous slip rate and a state variable, representing the changing morphology of the contact. The model comprises two equations: one for the friction coefficient, which includes direct (velocity-dependent) and evolution (state-dependent) effects, and a second for the evolution of the state variable itself, which clearly must vary with wear and temperature.

Of course, it may just be that the concept of state variable is just a more complicated way of saying that the surfaces in our experiment wear and, as a result, so the frictional response changes.

In practice, we don’t really know how and why friction varies – topography, damage, surface chemistry, development of interface layers, whatever. We can however assume that changes in surface topography will translate into excitation normal to the surface, hence in the loading axis.

Non-existent Wear Models

Having modelled our test system, we could produce a finite element model, but only if we know the constitutive relationship between load, friction and wear.

We have empirical understanding, but no constitutive equations at all!

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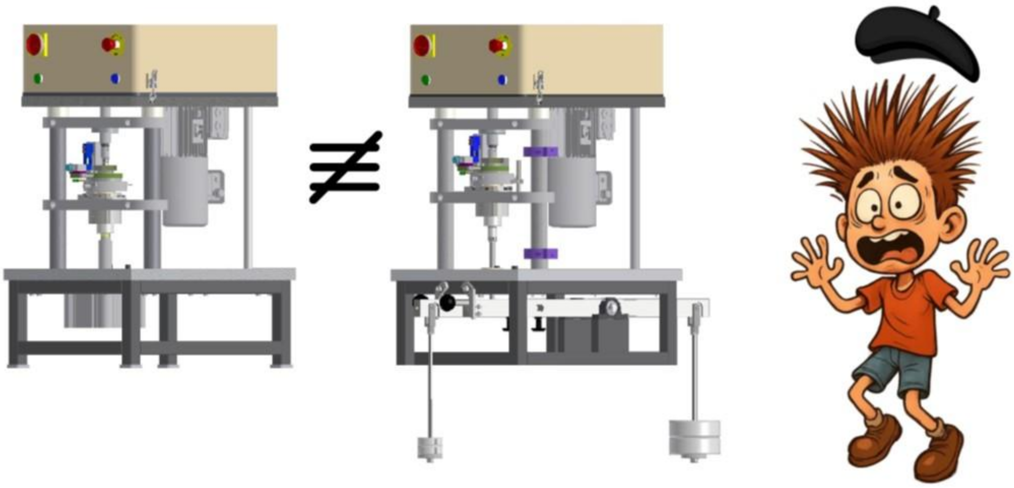
Of course, when it comes to wear, we have empirical understanding, but no constitutive equations at all.

Without viable constitutive equations for friction and wear, our modelling exercise comes to an abrupt halt!

Conclusion



Conclusion - Dynamics

- Stiffness & inertia affect both system friction & wear response
- Higher loading system stiffness & inertia give rise to higher dynamic contact pressures hence higher wear rates
- Lower stiffness & inertia loading systems, rather like a well-designed suspension system, reduce potential for generating high dynamic load forces
- Mass of specimen carrier & stiffness of measuring system both affect frictional response with higher mass & lower stiffness potentially resulting in friction induced vibration, stick-slip & other forms of instability
- System stiffness plays a critical role in the occurrence of friction-induced vibrations & instability

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Thinking about the dynamics, stiffness and inertia affect both system friction and wear response. In the case of load application, higher stiffness and inertia typically gives rise to higher wear rates, including fatigue induced wear mechanisms such as rolling contact fatigue. This is because higher stiffness and inertia give rise to higher dynamic contact pressures. Lower stiffness and inertia loading systems, rather like a well-designed suspension system, reduce the potential for generating high dynamic loads.

When it comes to friction, the mass of the specimen carrier and the stiffness of the measuring system both affect the frictional response of the tribo-contact, with higher mass and lower stiffness potentially resulting in friction induced vibration, stick-slip and other forms of instability.

We cannot of course afford to ignore force measurement system bandwidth and signal processing.

Conclusion - Feedback

- System dynamics in loading axis & friction plane are not mutually independent, because it is a friction-coupled system
- There is an internal feedback mechanism comprising both the tribo-contact itself & the overall machine dynamics; we are dealing with a closed loop, not an open loop system
- If the feedback loops are negative in nature, response will be to damp outputs & prevent change, whereas if the feedback loops are positive, tendency will be to amplify outputs, giving rise to instability

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Thinking about system feedback, it seems potentially attractive to treat the system dynamics in the loading axis and the friction plane as mutually independent, however, this is clearly unrealistic, because frictional response is dependent on load.

We have a friction and wear coupled system; there is an internal feedback mechanism comprising both the tribo-contact itself and the overall machine dynamics; we are dealing with a closed loop, not an open loop system. It follows that if the feedback loops are negative in nature, the response will be to damp outputs and prevent change, whereas if the feedback loops are positive, the tendency will be to amplify outputs, giving rise to instability.

Conclusion

Challenge can be summed up very simply:

- we specify test conditions but make no reference to system response of our tribometer, however, it is the system response that we are recording
- if we are to compare results from different tribometers, we need to start by comparing response of said tribometers
- data sets must be curated on basis of test system response and system bias before ever attempting any big data analysis of results

Significant implications for the validity of big data models & the application of AI to tribology

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How we should compare data generated by one test machine with that generated by another nominally similar device is moot.

Our problems are further exacerbated by designing experiments based on inadequate friction “laws” and practically non-existent wear laws.

The challenge can be summed up very simply; we specify the test conditions for our experiment but make no reference to the system response of our tribometer, yet it is the system response that we are recording.

Logically, if we are to compare results from different tribometers, we need to start by comparing the response of said tribometers. If we do not do this, our diverse tribometers run the risk of becoming random result generators.

We need to understand the spread of results in terms of friction and wear as a function of system dynamics, and then perhaps work out when to be worried about comparing data from different devices and when we can just be relaxed about it.

It follows that data sets must be curated on the basis of system response and system bias, as well as test procedure, before ever attempting any big data analysis of published results.

These issues have significant implications for the validity of big data analysis and the application of AI to tribology.

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