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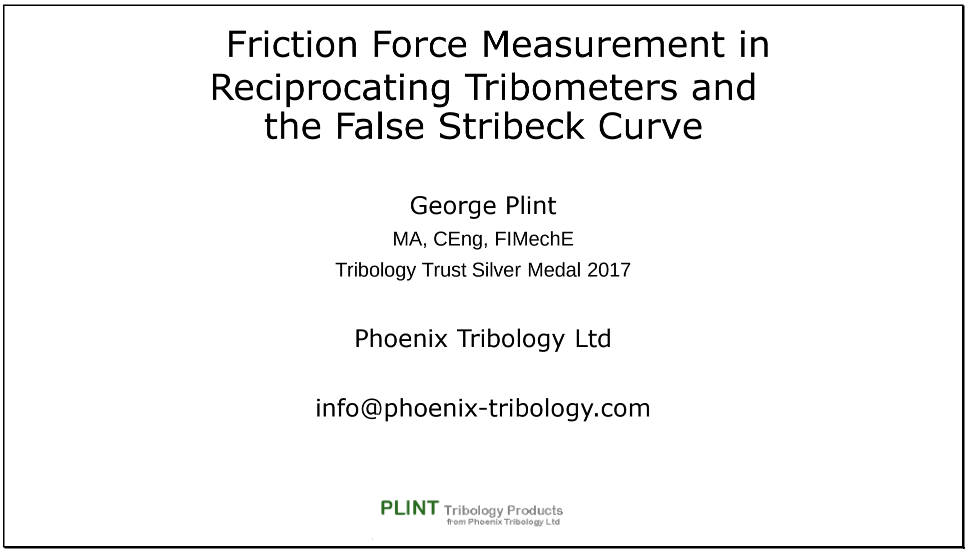

At STLE in 2011, I gave a talk on measurement of friction in reciprocating tribometers during one of the tribo-testing sessions. It obviously struck a chord with some people, but sadly not that many. Since then, I have come across numerous examples of people claiming to show a Stribeck like friction-velocity response, in their lubricated friction experiments, with contact geometries and velocities that simply could not, under any circumstances, operate in anything other than a boundary lubrication regime, in other words, under conditions where friction is more or less independent of sliding velocity.

A recent example of a claimed Stribeck like response was made by one of the presenters at the 2019 ASTM Lubricated Friction Workshop, however, when I asked the presenter whether he had attempted to calculate the lubricant film thickness for his experiments, to confirm whether indeed there was some form of hydrodynamic or elasto-hydrodynamic separation of the surfaces, the answer was no; actually, the answer was that he did not know how to. So, he was claiming a Stribeck like response from his experiments, but without either attempting to calculate or measure the lubricant film thickness.

One final thing convinced me that it would be a good idea to revisit this subject. I searched the numerous test standards I have, covering dry and lubricated friction measurement, and could find no reference to terms such as “frequency response”, “resonance”, “signal bandwidth”, “filtering”, etc, in other words, nothing to do with the tricky issue of measurement of dynamic forces. Is that because it’s all too difficult?

In this talk, I am going to cover all these issues, plus look at their differing effects on the generation of both Stribeck curves and force-displacement loops.

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Friction Force Measurement

- Signal bandwidth
- Resonance
- Filtering
- Stiffness of measuring system
- Effect on tribological measurements

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Measurement of a dynamic force presents a number of challenges, frequently ignored, when the dynamic force involved is associated with a tribological experiment.

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 friction in tribological experiments are rarely analysed. This is of particular concern in reciprocating experiments, where claimed effects are not actual friction effects, but measurement artefacts or system resonance.

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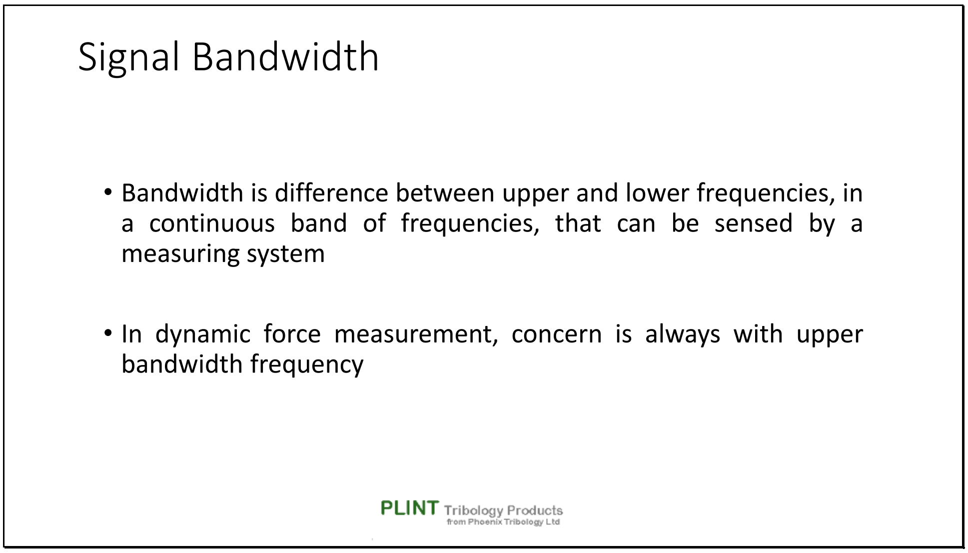

The bandwidth of a measuring system is the difference between upper and lower frequencies, in a continuous band of frequencies, which can be sensed by the system. In dynamic force measurement, concern is always with upper bandwidth frequency.

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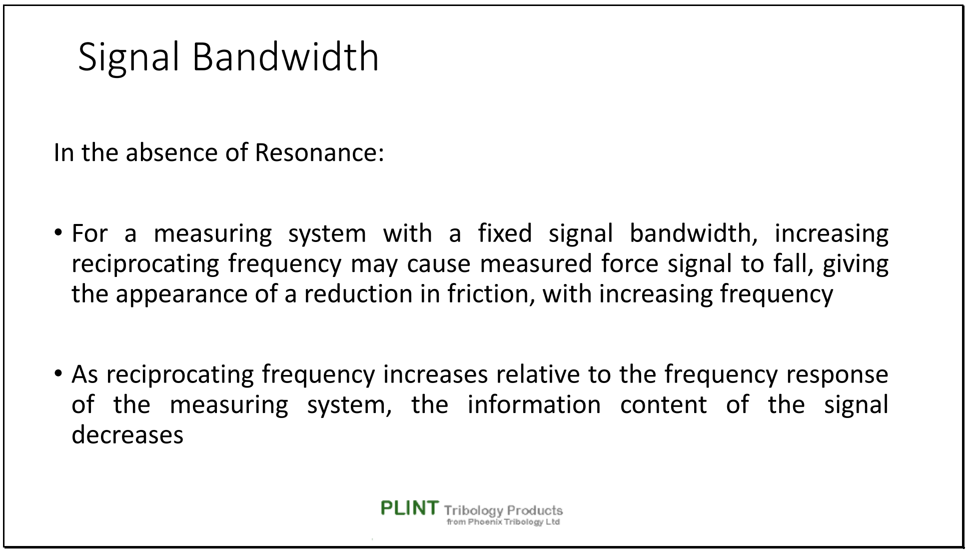

For a measuring system with a fixed signal bandwidth, in the absence of resonance, increasing the reciprocating frequency causes the measured force signal to fall, giving the appearance of a reduction in friction, with increasing frequency.

As reciprocating frequency increases relative to the frequency response of the measuring system, the information content of the signal decreases, which is what causes the effect.

If we now add in measuring system resonance, things get a bit more complicated.

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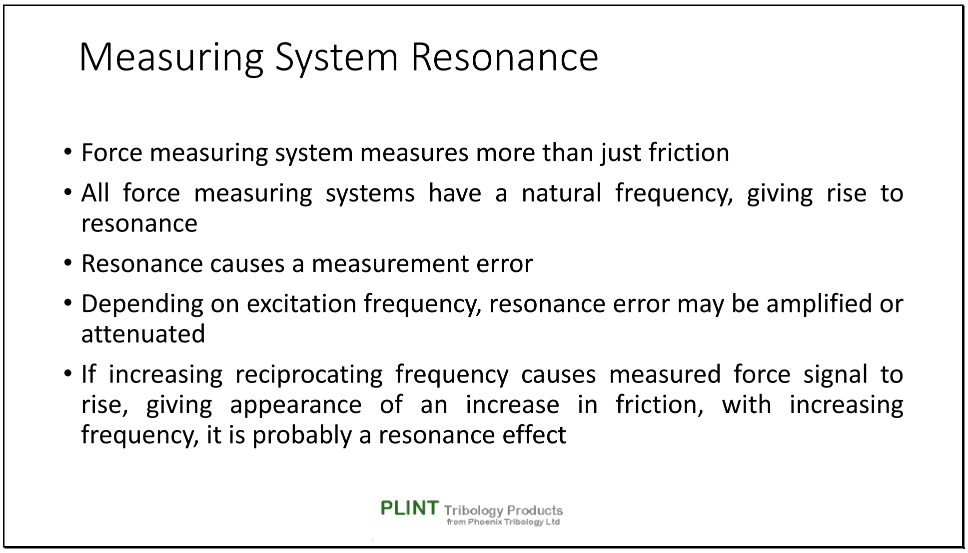

You may have seen examples in published papers, where, somewhat unexpectedly, the nominal friction force signal in a reciprocating test appears to go up, as the reciprocating frequency increases. This is much more likely to be as a result of system resonance, than some tribological effect.

The point is that a dynamic force measuring system is sensitive to more than just friction. 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.

It is clear that many “claimed” friction effects, including the apparent generation of Stribeck curves, are not actually real friction effects, but are either measurement artefacts or system resonance.

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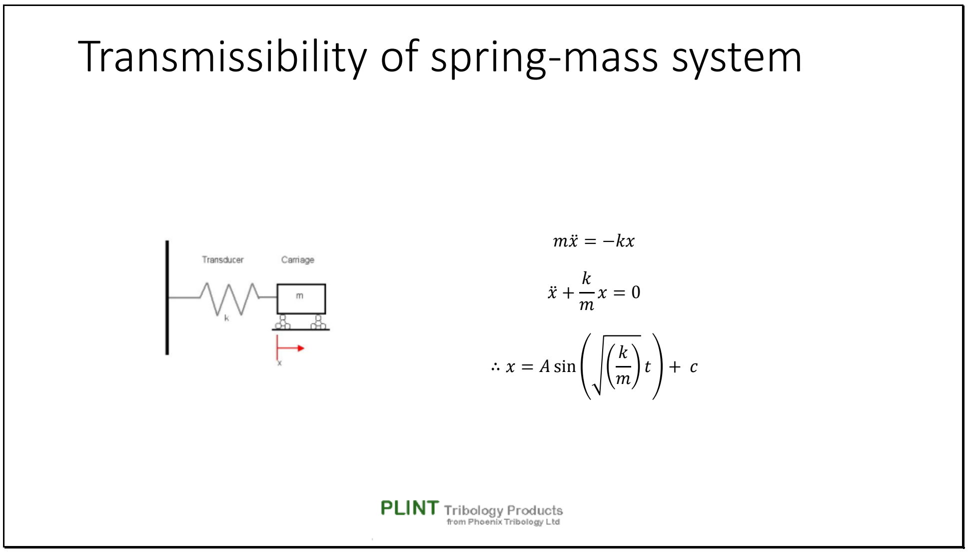

Any force measuring system, connected to a known mass, such as specimen assembly, not only measures some externally applied force, such as load or friction, but will also acts as an accelerometer, so will sense any externally generated vibration.

In addition to this, the assembly will have a natural frequency, which is the frequency at which resonance occurs. 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.

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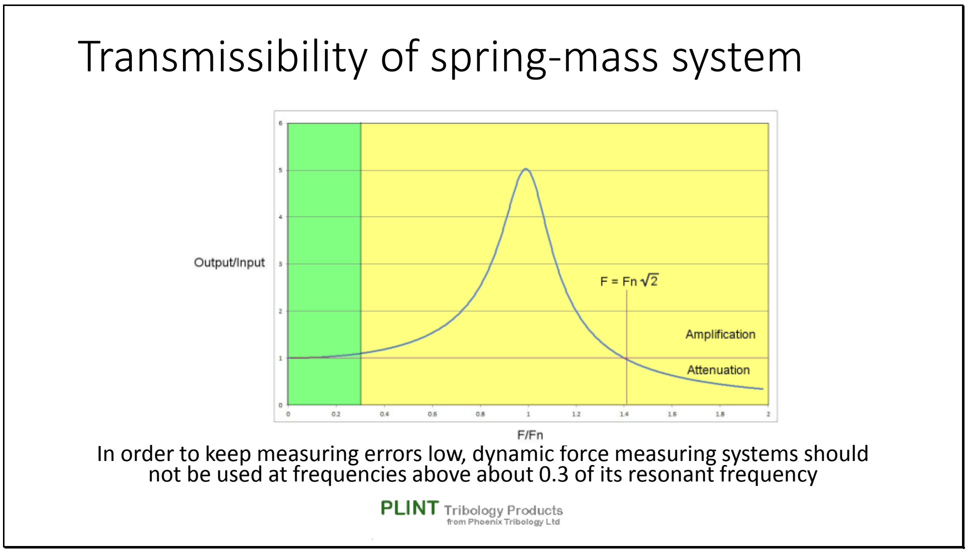

With any dynamic force measuring system, we are in fact dealing with what vibration engineers term the transmissibility of a spring-mass system. At low excitation frequencies, the force transducer will give a value that is slightly higher than predicted; for the most part, we have learnt to live with these small errors.

However, the closer the reciprocating or 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 zero and natural frequency x Root 2!

It is generally accepted that, in order 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.

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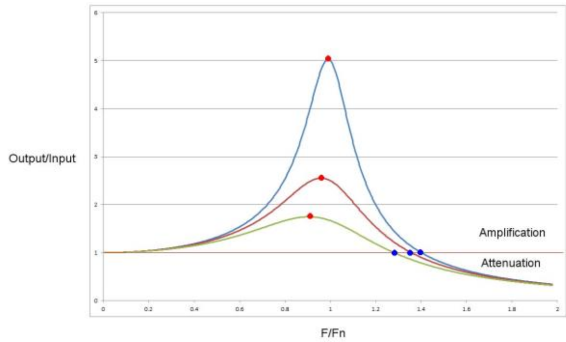
Transmissibility of spring-mass system



Increasing damping reduces natural frequency and introduces a phase shift

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It’s 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.

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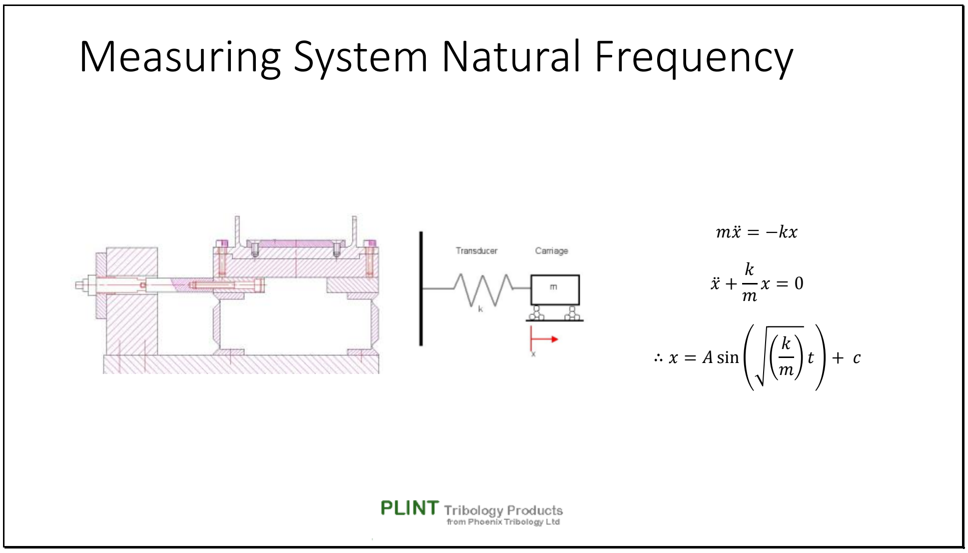

This is a diagram of a typical transducer and stationary specimen carrier (as used on the TE 77 High Frequency Friction Machine).

Because the flexures have low stiffness in the horizontal plane compared with the force transducer, the effective horizontal restraint on the specimen carrier is the force transducer itself. This can be modelled as a vertically restricted mass on a spring.

The spring is the piezo transducer. A +/-500 N piezo transducer typically has a stiffness of about 40 MN/m. The mass is the specimen carrier, with a typical weight of approximately 1 kg. For an un-damped, unforced spring the equations of motion are as shown.

A single spring system has a natural frequency of 1/2π √ (stiffness/mass) Hz. The TE 77 transducer/carriage assembly has a calculated natural frequency of approximately 830 Hz. We typically use this measuring assembly in conjunction with a low pass filter, with a cut-off frequency of 300 Hz.

Now, we could, if required, increase the signal bandwidth of the system, but as with most engineering systems, this is a matter of compromise, hence choice.

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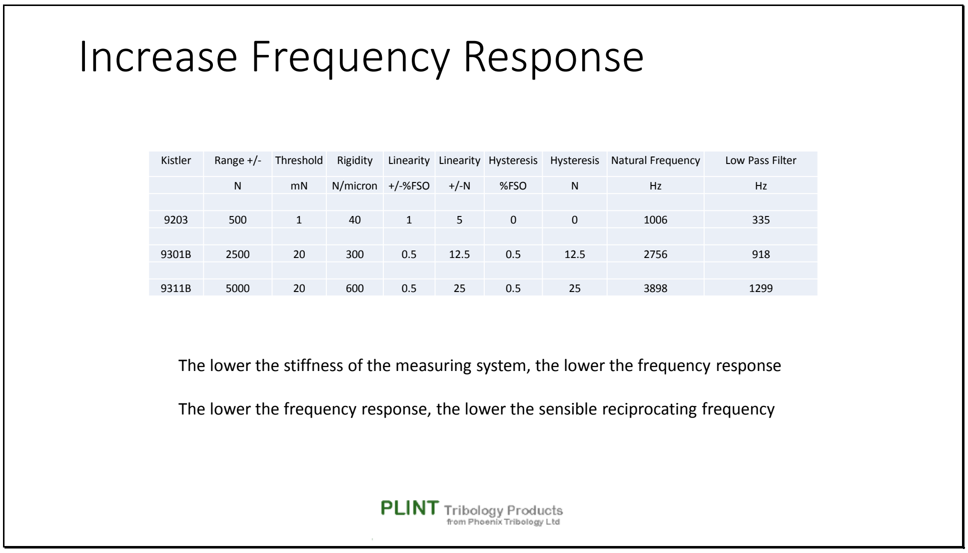

Kistler	Range +/-	Threshold	Rigidity	Linearity	Linearity	Hysteresis	Hysteresis	Natural Frequency	Low Pass Filter
	N	mN	N/micron	+/-%FSO	+/-N	%FSO	N	Hz	Hz
9203	500	1	40	1	5	0	0	1006	335
9301B	2500	20	300	0.5	12.5	0.5	12.5	2756	918
9311B	5000	20	600	0.5	25	0.5	25	3898	1299

There are two ways to increase the frequency response of the measuring system, either by reducing the mass of the assembly/tooling/specimen or by increasing the stiffness of the measuring transducer.

Assuming that there is limited scope for reducing the mass of the test assembly, without reducing the size of the fixed specimen, hence the achievable reciprocating stroke of the test, what could be achieved by using a stiffer transducer?

It will be noted that although there are significant gains to be made in terms of increased natural frequency from using a stiffer and hence higher load range transducer, there are corresponding losses in terms of sensitivity, linearity and hysteresis.

In essence, the lower the force we may wish to measure, the less stiff the measuring system and the lower the frequency response, hence the lower the frequency of variation in force that can be detected. That is the basic choice we have to make.

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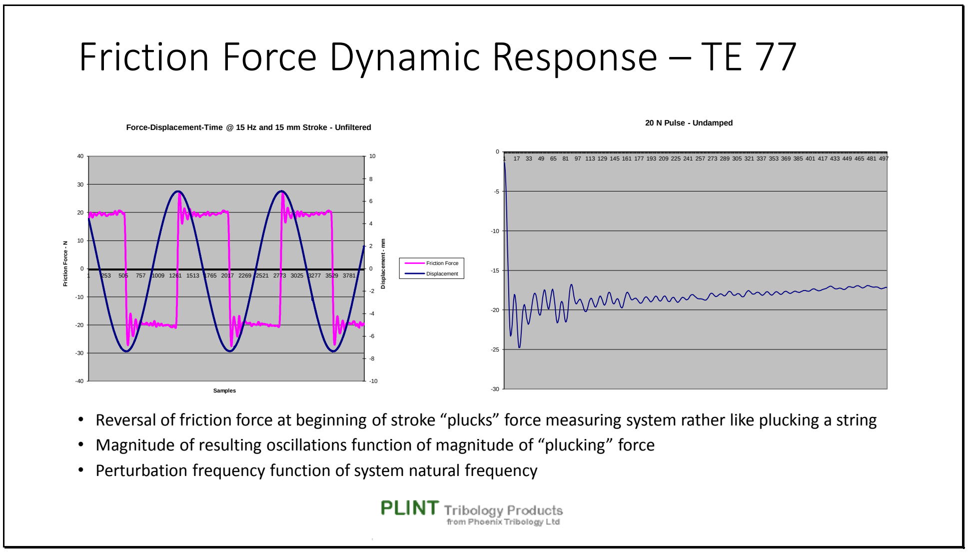

The graph shows an unfiltered friction force signal from a standard TE 77 High Frequency Friction Machine, running at 15 Hz and 15 mm stroke.

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 superimposed on a quasi-steady state signal, giving rise to under-damped oscillation.

This is a common feature of all reciprocating tribometer friction force measurements: reversal of the friction force 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 (sample, bath, tooling etc) and 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 contact.

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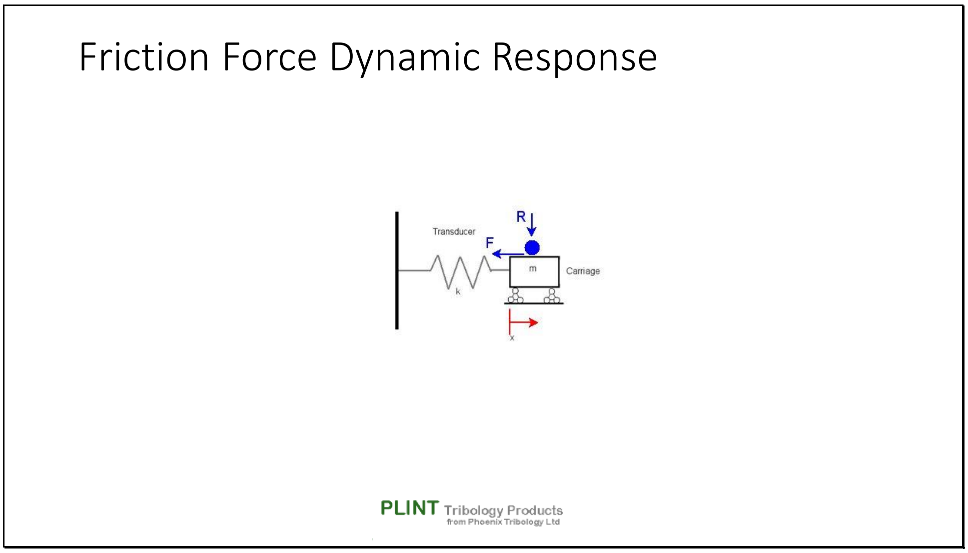

The higher the friction in the contact, the higher the damping coefficient; the time for the signal to decay is fixed for the damping coefficient.

It follows that the higher the reciprocating frequency, the shorter the period and the greater the percentage of the force signal that includes superimposed resonance. The lower the reciprocating frequency, the better the signal (friction force) to noise (harmonic vibration) ratio. Of course, appropriate filtering will improve matters.

It is also worth noting that reciprocating stroke length does not have any bearing on the system response.

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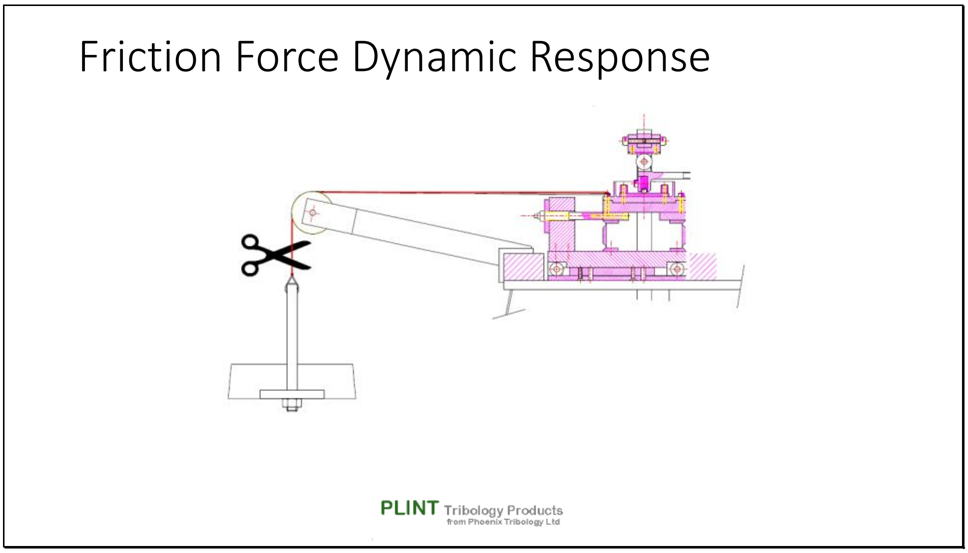

Confirmation of this resonance effect and measurement of the test assembly's natural frequency can by demonstrated by a simple experiment: the system response to a step change input. A mass is attached to the transducer assembly by means of a string and pulley arrangement. Cutting the string "plucks"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

Measurement of the period of oscillation from both the friction signal and the "plucking" experiments indicates a resonant frequency, without friction damping, of approximately 830 Hz, in other words, so close to the calculated value to be beyond doubt.

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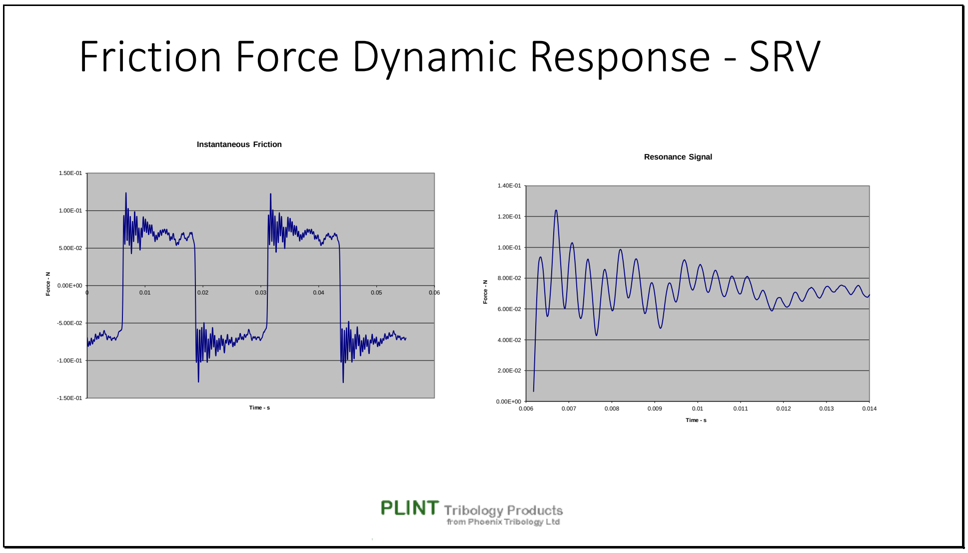

We can see similar behaviour with an SRV, in this case, a pre-SRV-4 version of the machine.

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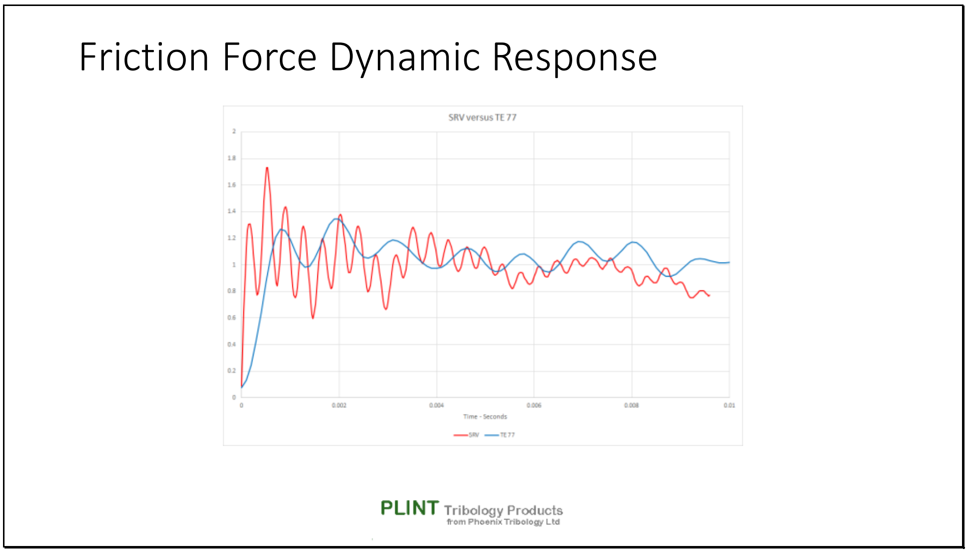

Comparing the resonant frequency of the SRV and the TE 77 test assemblies, we can see that the SRV has a higher resonant frequency, hence signal bandwidth, than the TE 77, which is as one would expect; the machine has a much smaller and lower mass fixed specimen assembly, consistent with a shorter stroke tribometer, and a much stiffer piezo transducer, consistent with a higher load range device. So, in both cases, rational engineering choices have been made.

It is worth noting that Optimol have sensibly avoided any issues with operating at different frequencies, by specifying the same reciprocating frequency, 50 Hz, for all their ASTM test standard.

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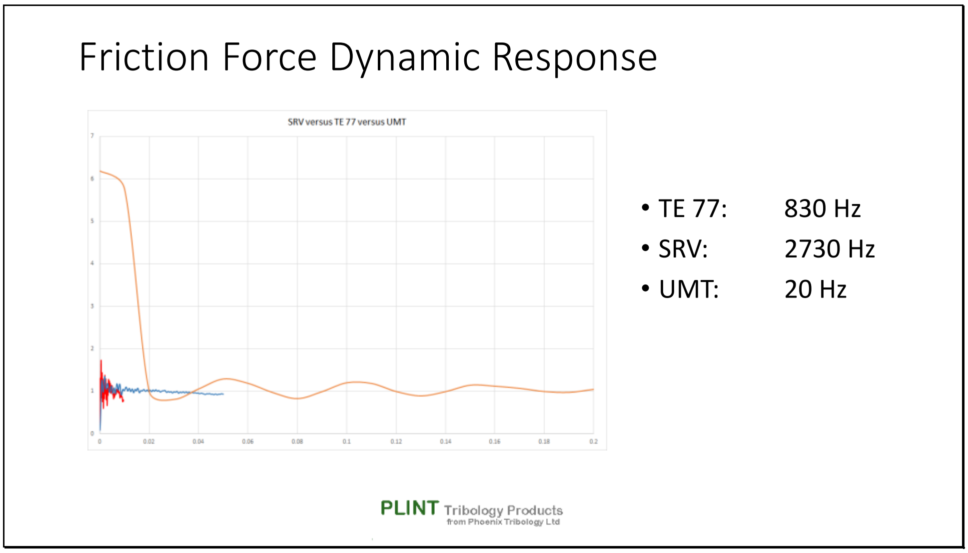

Repeating the “plucking” test with a UMT machine, we see that it has a much lower signal bandwidth than either the TE 77 or the SRV. This is because the device has a relatively high mass specimen assembly, cantilevered off a relatively low stiffness force transducer.

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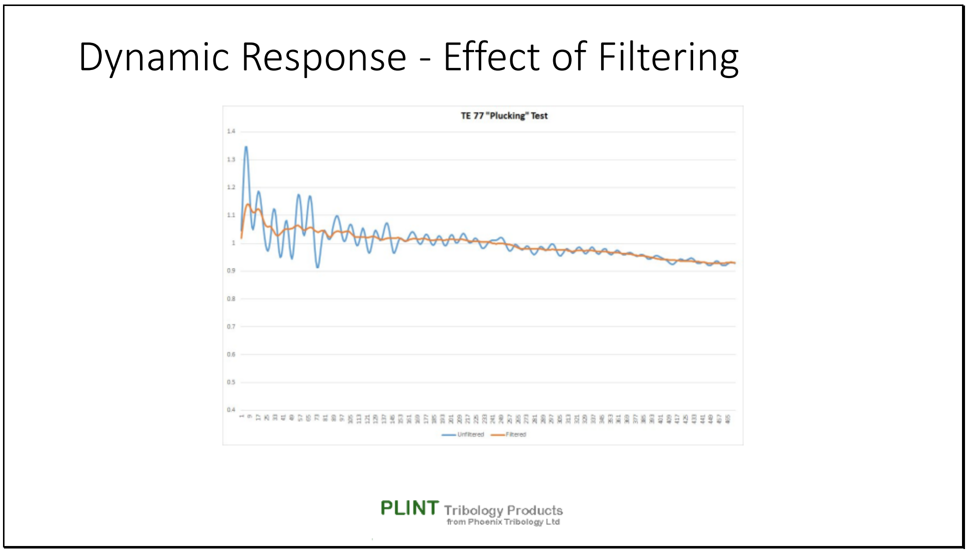

We can of course tidy up our friction signal with a suitable filter. In this example, we have used a filter that removes the resonance, but we have to bear in mind that it will also remove any true friction effects at the same frequency.

So, let’s now move on to consider filtering.

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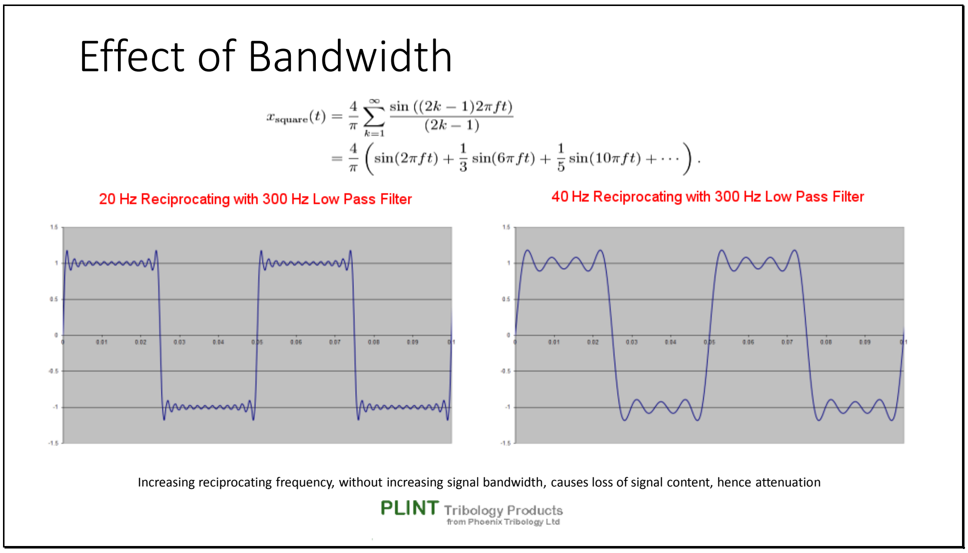

In dry or boundary lubricated reciprocating tribometers, we would normally expect the friction force to be approximately independent of sliding velocity, hence generating an approximately square wave friction signal. To demonstrate the effect on the interaction of reciprocating frequency and the frequency response of measuring system, we could perhaps start by using a mathematically generated square wave as a model. The Fourier equation for a square wave shows that it can be represented by the sum of odd harmonics according to the formula given, where "f" is the fundamental frequency. In a reciprocating tribometer this is of course the reciprocating frequency.

Hence, if we reciprocate at 20 Hz, but operate with a low pass filter of 300 Hz, the resulting signal will comprise a fundamental sine wave at 20 Hz and odd harmonics at 60, 100, 140 Hz etc, up to the cut-off frequency. All signal harmonic content above the cut-off frequency will be lost and the resulting signal will be attenuated. We can model this by summing the series up to the highest harmonic below the chosen cut-off frequency.

Increasing the reciprocating frequency, without increasing the signal bandwidth, causes further loss of signal content, hence further attenuation.

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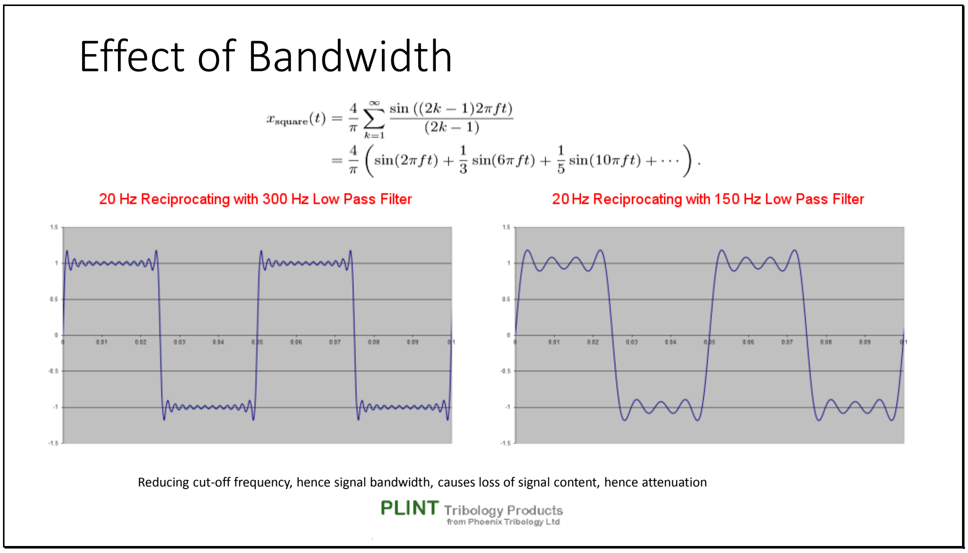

Similarly, reducing the cut-off frequency, which is effectively reducing the signal bandwidth, causes a similar loss of signal content.

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Effect of Bandwidth

Model Compromised:

- Inherent problem with Fourier equation
- Difference between theoretical and practical filters
- Ignores resonance of real dynamic force measuring systems

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This model is not bad as a fairly crude illustration of the effect of varying input frequency with a fixed signal bandwidth system, but it is by no means a perfect model of a real dynamic force measurement system. There are three reasons for this: the first is to do with the Fourier equation itself, the second to do with the difference between a theoretical and a practical filter, and the third to do with the resonance of a real dynamic force measuring system.

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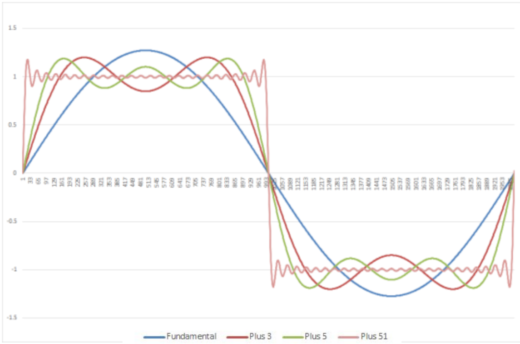
Fourier Fundamental versus Real Signal



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The problem with the Fourier formula is that as the higher frequency harmonics are progressively removed, as the cut-off frequency is reduced, it becomes clear that the amplitudes of the lower order harmonics and the fundamental exceed the amplitude of the modelled square wave.

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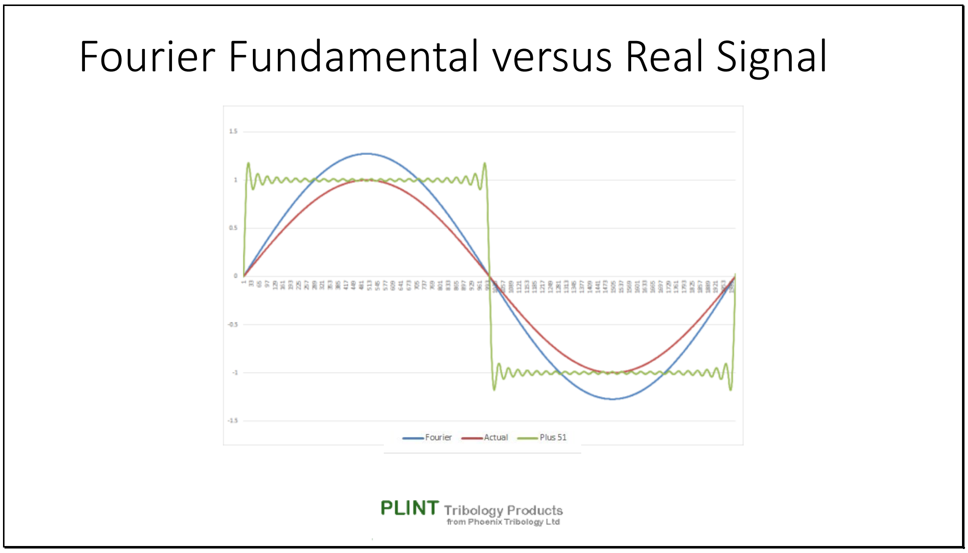

To work properly as a model of a real friction signal, the amplitude of the fundamental sine wave cannot exceed the mid-stroke maximum value of the square wave.

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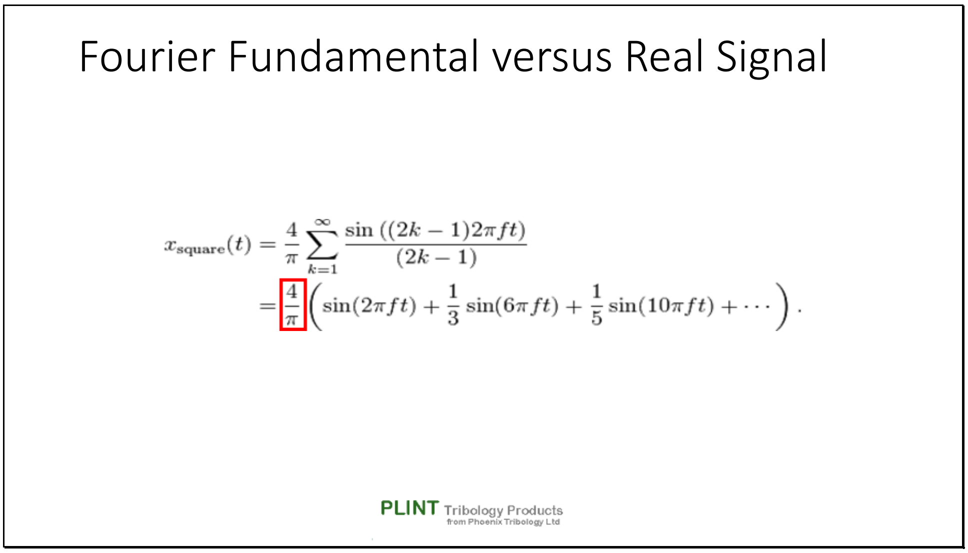

The problem is of course the 4/Pi term in the equation; the model works well when there are a large number of harmonic components, but not when these are limited to just a few. The result is that the apparent level of signal attenuation is underestimated.

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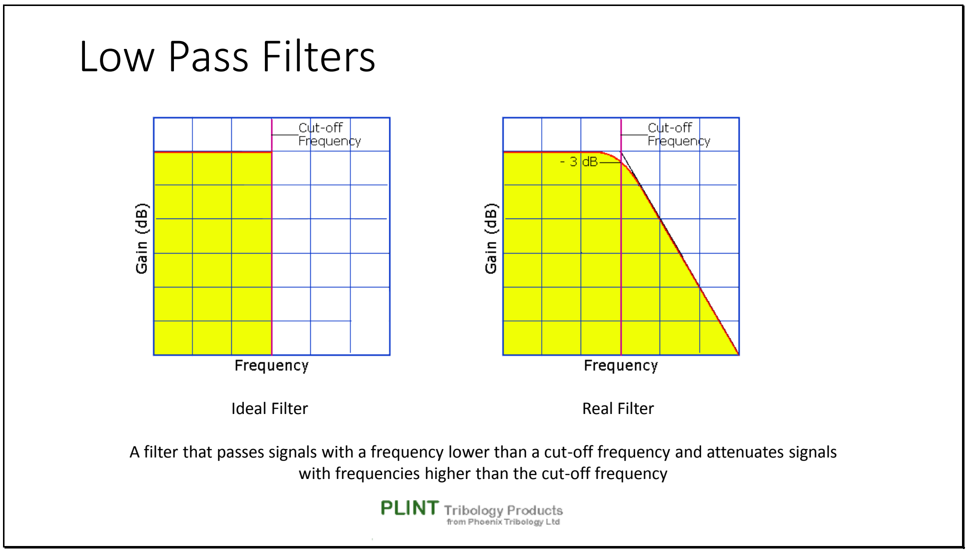

The second reason why the Fourier model is not ideal is that it assumes a sharp cut-off frequency, which is quite unlike the response of a real filter.

In order to overcome the limitations of the mathematical model, the simplest solution is to replace it with a physical model. In the following series of experiments, we used a square wave signal generator and a low pass filter, which we use to represent the bandwidth of the system. This allows us to observe the effect of varying input frequency with different cut-off frequency filters, each representing a model of a different fixed signal bandwidth system.

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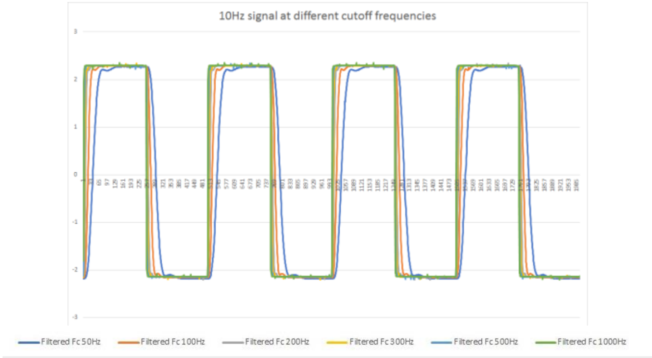
5^{th} Order Bessel Filter



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This graph shows the output from a 5^{th} Order Bessel filter, with different cut-off frequencies set, in response to a 10 Hz square wave input. In this example, very little attenuation is observed.

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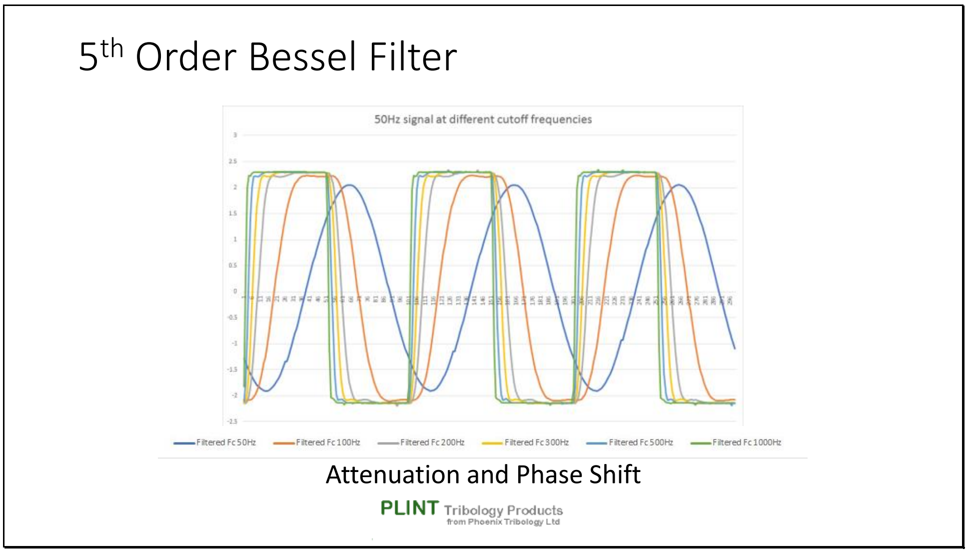
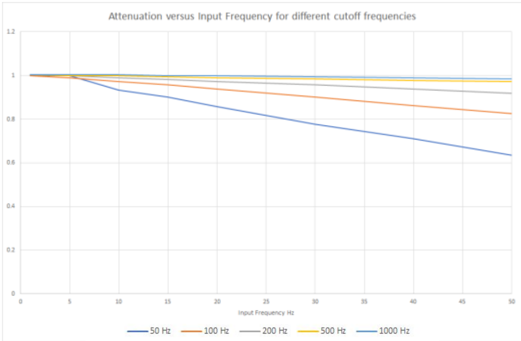


In this graph, the input frequency is increased to 50 Hz, with the same filter cut-off frequencies. As we would expect, the higher the ratio of excitation frequency to cut-off frequency, the greater the loss of information content, plus the greater the phase lag between output and input.

When the input frequency equals the cut-off frequency, in this case both equal to 50 Hz, the output is effectively a sine wave with an amplitude slightly less than the unfiltered square wave amplitude; there is no 4/Pi factor in this real-life experiment!

Slide 27

5th Order Bessel Filter



RMS Attenuation

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Generating an r.m.s. value of the instantaneous output signal, as might be applied to a reciprocating friction signal, shows the level of attenuation against input frequency. The level of attenuation increases with increasing input frequency, but this depends on the signal bandwidth of the measuring system; this is the same effect as the friction apparently going down, as the reciprocating frequency goes up, a problem which can only be substantially avoided if the friction measuring system has an appropriately high signal bandwidth.

The illustrated tests were run with the input frequency always less than or equal to the cut-off frequency. So what happens if the input frequency exceeds the cut-off frequency, with a real filter?

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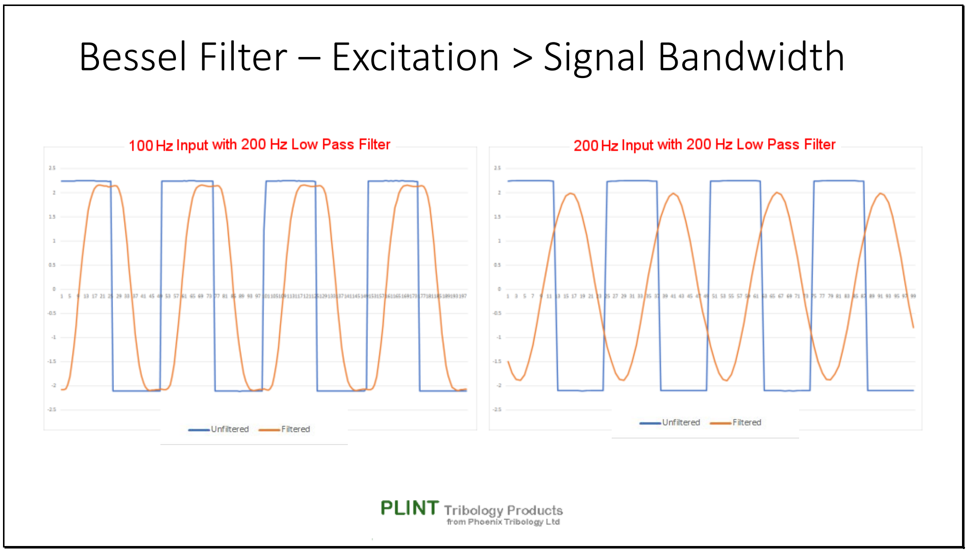

These graphs show the output from the Bessel filter set to 200 Hz with a square wave input at 100 Hz and at 200 Hz. As was observed before, when the input frequency is equal to the filter cut-off frequency, the output signal is an approximate sine wave, with an amplitude slightly less than the input square wave.

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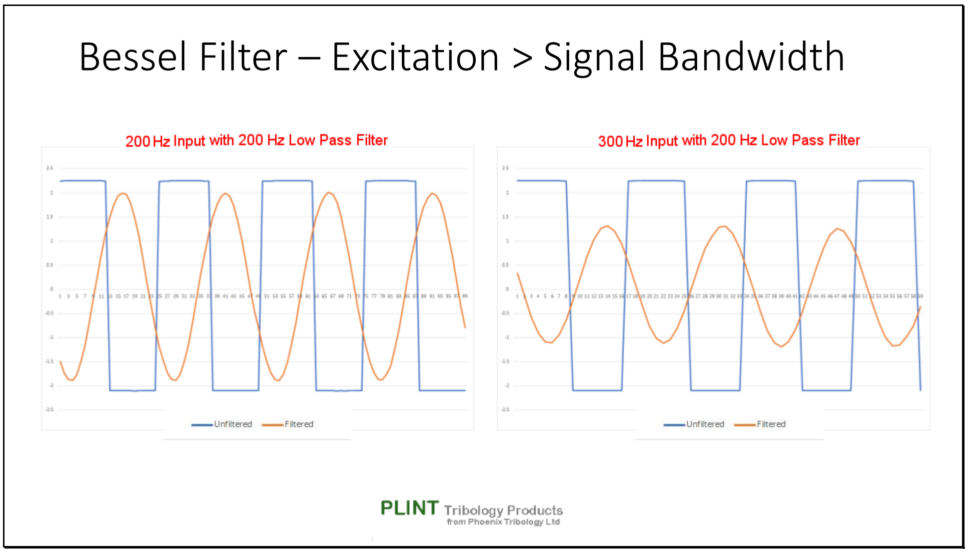

Increasing the input frequency above the cut-off frequency increases the phase shift and reduces the amplitude of the output signal. This is because we are using a real filter, which does not have a step cut-off.

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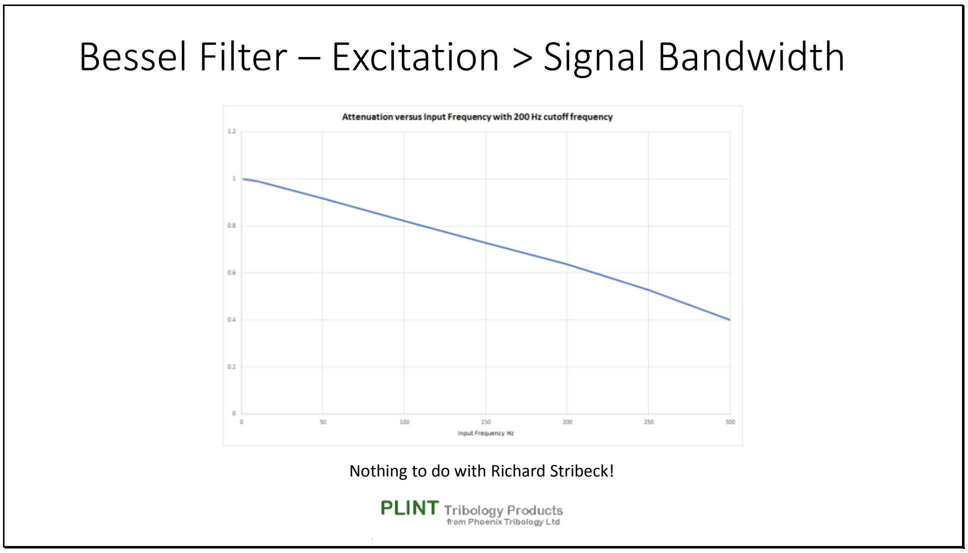

In this experiment, the combination of 300 Hz input signal and 200 Hz low pass filter results in an r.m.s. output signal that is attenuated by 60%. Because the input frequency exceeds the signal bandwidth, the output has been substantially attenuated; if instead of a signal generator we had a reciprocating tribometer and a friction force measuring system with similar limited bandwidth, the friction would appear to go down as the reciprocating frequency went up, but this would not have anything to do with Richard Stribeck!

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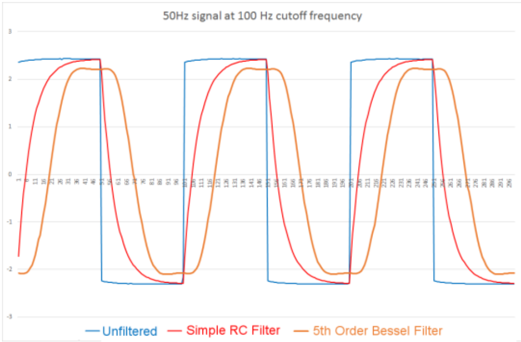
Effect of Filter Type



Different types of filter, with same cut-off frequency, give different output signals and different phase shift

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It would perhaps be an omission to conclude this section on filtering and signal bandwidth without stating a preference for a particular type of filter for reciprocating tribometer applications. The best filter for time domain applications is a Bessel filter, a type of filter frequently used for cleaning up digital signals, which are perhaps a good analogue for a dry or boundary lubricated friction signal. It provides minimum distortion of rapid slope changes, a uniform group delay and the lowest wideband noise. A Bessel filter is less well suited to frequency domain applications, as it has a drooping amplitude response and a gentler cut-off frequency than other filter types. But how does a Bessel filter compare with, say, a simple Resistance-Capacitor or RC filter?

So there we have it, for the same nominal cut-off frequency, we get a different output signal and a different phase shift, depending on the type of filter.

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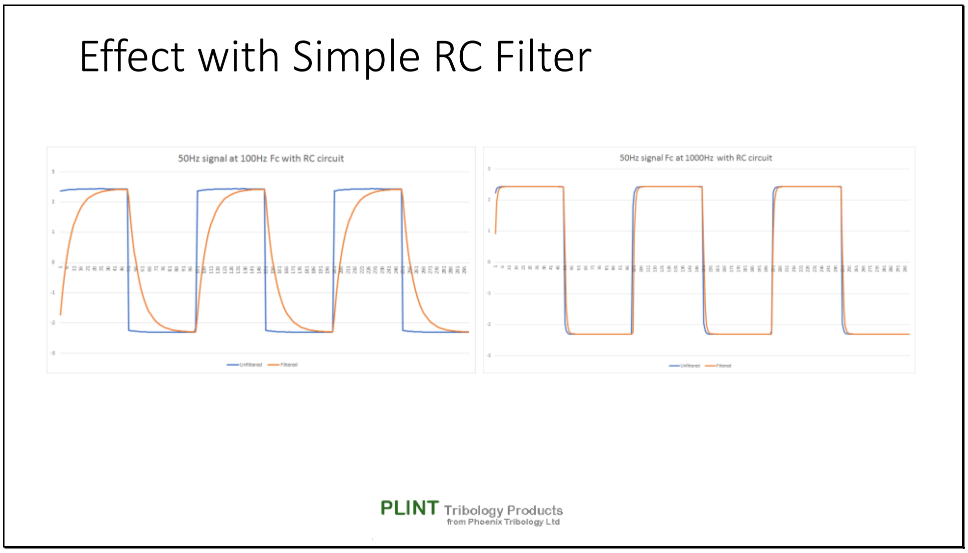

It’s worth noting that if we only need a very limited amount of filtering, a simple RC filter works fine. However, with a dynamic force measuring system, we usually need much more than a very limited amount of filtering.

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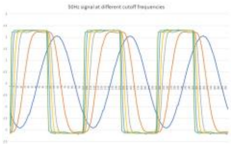
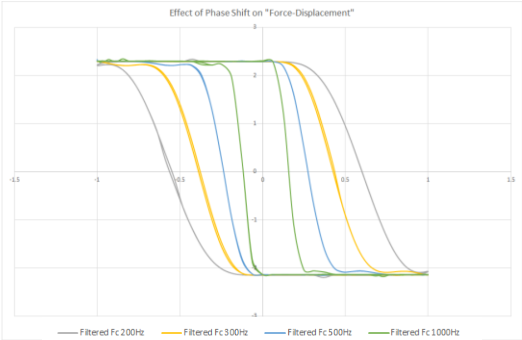
Force-Displacement Loop





x-axis unfiltered resulting in phase shift between x and filtered y-axis

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We normally apply some filtering on all measuring channels in order to eliminate higher frequency signal noise and aliasing.

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 is not subject to differing time delays.

This graph shows what happens to the equivalent of a force-displacement curve, if there is a phase lag between the two inputs; in this example, a sine wave, representing displacement, has been synchronised with the 1000 Hz filtered signal, with the lower filter frequency signals subject to the indicated measurement lag.

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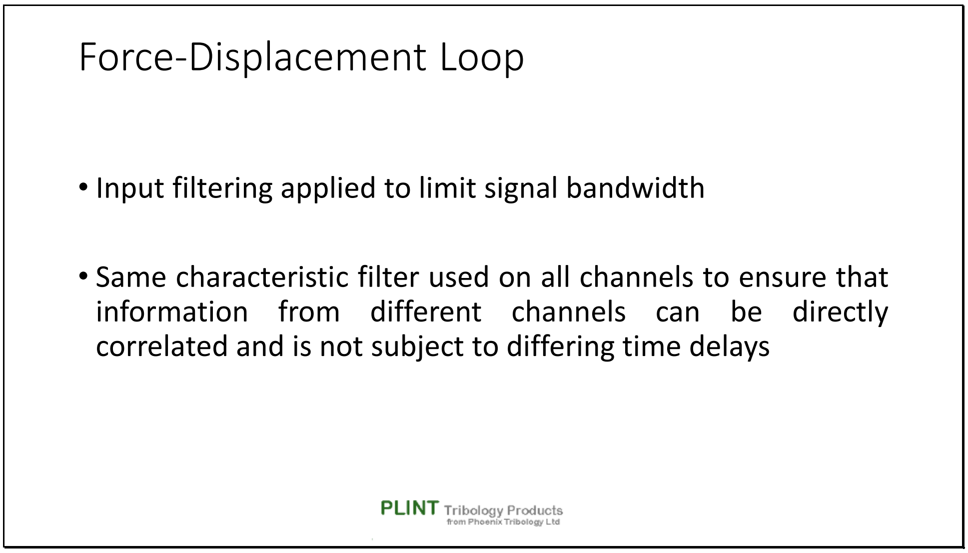

From this we can see how profoundly the size and shape of the curves are affected by different time delays, indicating the importance of applying the same characteristic filters to the two input channels. The implications for calculating frictional energy dissipation from force-displacement loops are equally profound; curiously, there is no mention of this issue in ASTM G203 Standard Guide for Determining Friction Energy Dissipation in Reciprocating Tribosystems!

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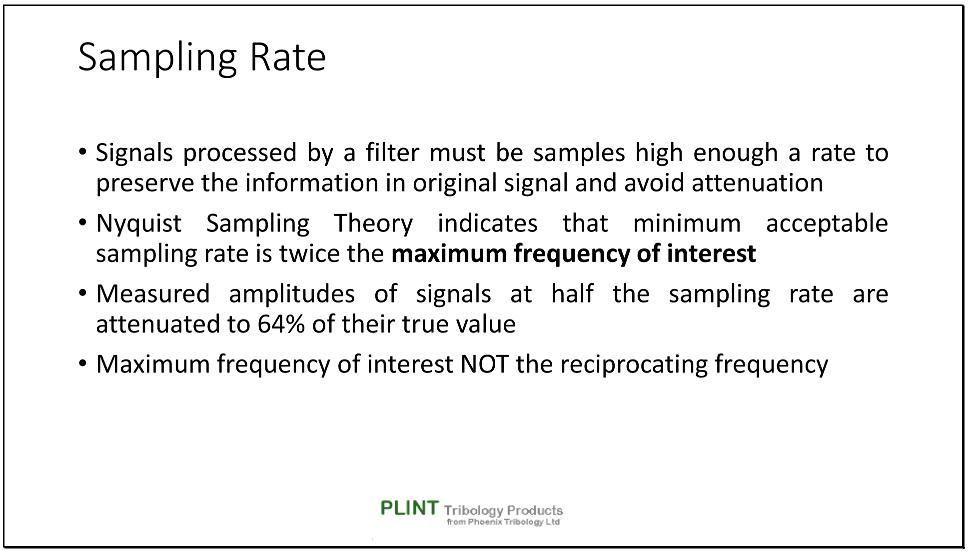
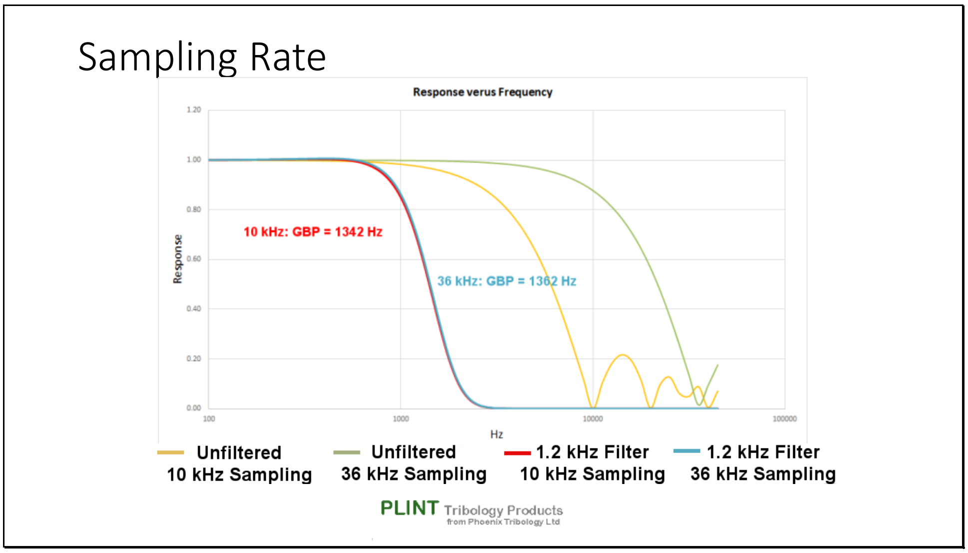


When sampling the signals processed by a filter it is important to sample at high enough a rate to preserve the information in the original signal. The Nyquist Sampling Theory indicates that the minimum acceptable sampling rate is twice the **maximum frequency of interest**. This is obviously much higher than the tribometer reciprocating frequency!

Slide 36



The measured amplitudes of signals sampled at half the Nyquist sampling rate are attenuated to 64% of their true value. It is good practice to sample at higher rates than the Nyquist rate, within reason.

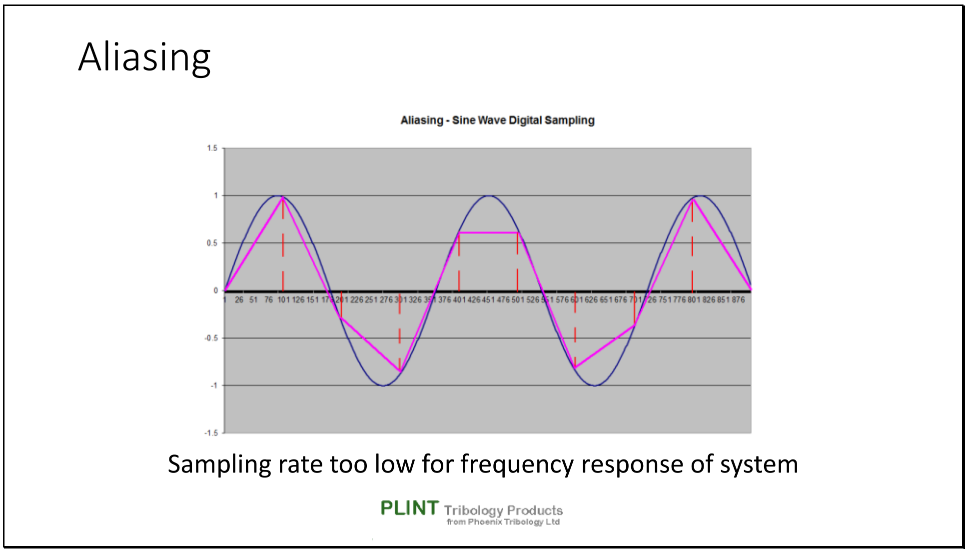
With a system sampled at 10 kHz, a 1 kHz signal, which would have suffered 36% attenuation with 2 kHz sampling rate, only suffers 2% attenuation.

The graph shows the differences between the combined effects of a 1.2 kHz input filter and the attenuation effects of 10 kHz sampling and 36 kHz sampling. The differences are barely noticeable.

The Gain Bandwidth Product or GBP is, as its name implies, the product of the Theoretical Gain and the Theoretical Bandwidth of the system. It provides an estimate of the information carrying ability of the system. In this example we see that the 36 kHz sampled system is only 1% better than the 10 kHz system. In other words sampling faster than necessary may produce more data but not more information! We could obtain the same amount of data by using a lower sampling rate and simply interpolating between the samples.

To conclude, the Sampling rate of the system should be well matched to its signal bandwidth in order to preserve information content. Furthermore the signal bandwidth should be well matched to the measuring transducer systems.

Slide 37



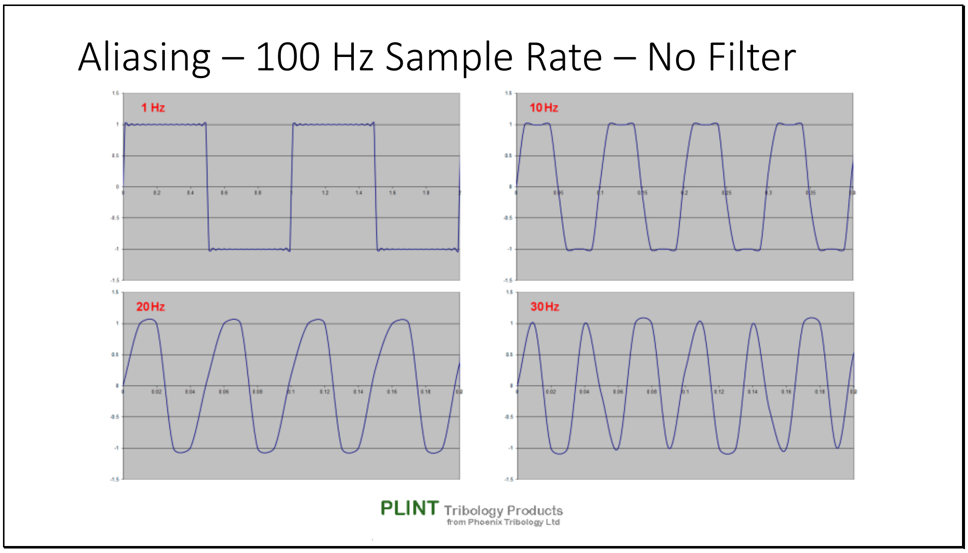
Aliasing occurs when the sampling rate is too low for the frequency response of the system. As a simple example of aliasing, consider sampling the levels of illumination by looking out of the window, at the same time, just once per day. If the observations were always made at midnight, the collected 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 taking three samples per day, at eight hours interval. If sampled at 1200, 2000 and 0400, the observations will indicate twice as much dark as light. If sampled at 0000, 0800 and 1600, the observations will indicate twice as much light as dark.

As the signal is varying twice per day (from day to night) the absolute minimum sampling rate in order to avoid aliasing would be four equi-spaced observations per day.

The above graph shows the effect aliasing when sampling a sine wave every 100 degrees, which is equivalent to sampling a 360 Hz signal with a 100 Hz sampling rate.

Slide 38

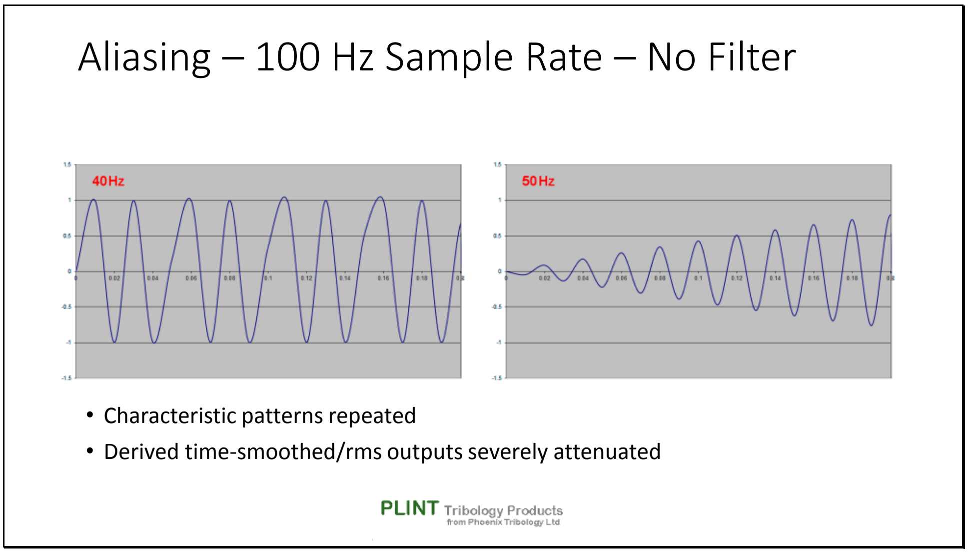


In these graphs, I have used the Fourier square wave to demonstrate the effect of using a fixed sampling rate, with a variable input frequency, so this is the equivalent of running a model of a reciprocating test at different frequencies, without adjusting the sampling rate. To avoid confusion, no filtering has been applied, maximising the bandwidth.

At 1 Hz reciprocating, with 100 Hz sampling rate, much of the detail of the square wave is preserved. At 10 Hz reciprocating and 100 Hz sampling, we only have 10 data-points per cycles, so quite a lot of information has already been lost. As the reciprocating frequency increases, more and more information is lost.

If you consider what a rectified and time smoothed or r.m.s. value of these signals would be, it is easy to understand just how severely attenuated successive values would be. So here we have another mechanism for generating a false Stribeck curve.

Slide 39



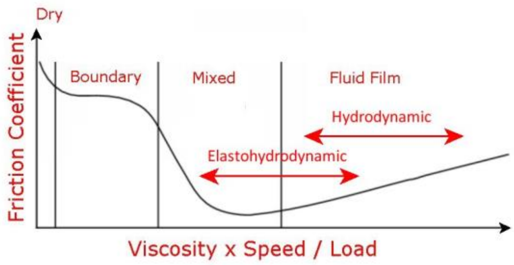
As we increase the reciprocating frequency still further, without adjusting the sampling rate, we get to the point where the output is obviously completely wrong!

If you have a trace with strange repeated patterns, or peaks and troughs, that come in and out of phase, or that appear to tilt in the direction of the increasing time axis, check you haven’t got aliasing.

As a simple rule, if you are determined to run reciprocating tests at varying frequencies, if you choose a sampling rate that is appropriate for the maximum test frequency, it is guaranteed to be more than enough at lower reciprocating frequencies.

Slide 40

Stribeck Curve



X-axis is normally logarithmic scale

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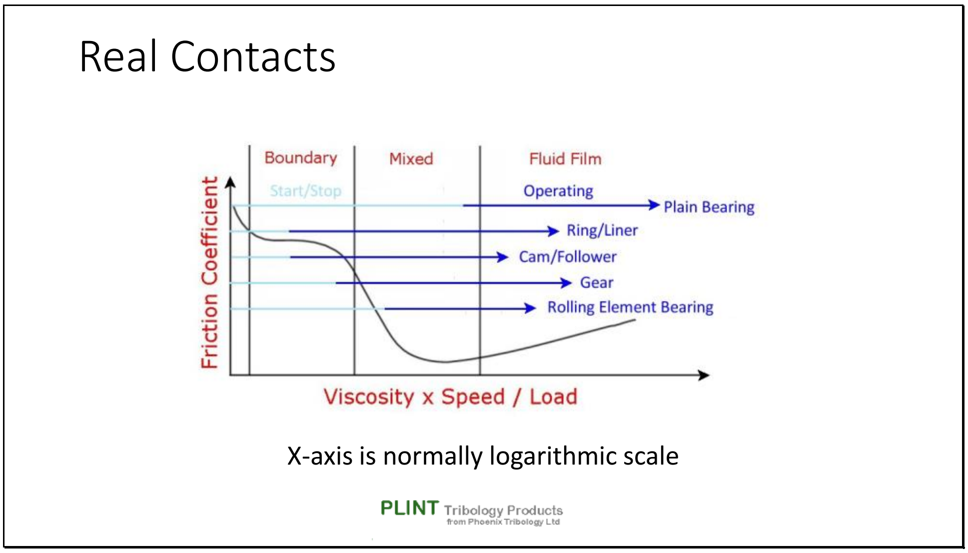
So, we have now explored the complexities of measuring a dynamic force, the kind of errors that can occur and what we need to do to mitigate the problems, by optimising the measuring system frequency response and the use of appropriate signal filtering and sampling rate. We have seen how failure to get these things right can result in the generation of false Stribeck curves, but how can we prove that they are indeed in error?

Slide 41

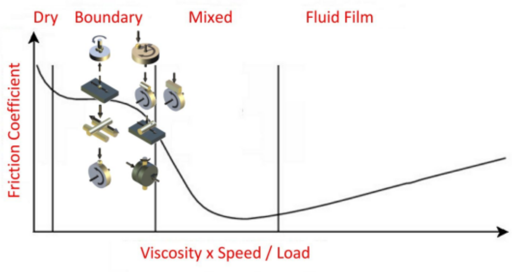


We can show where various devices operate on the Stribeck curve, noting that in all cases, the systems have to start and stop at some time, causing the contact to transit through the regimes.

It is important to note that some devises will have a single operating point for a given speed, for example a plain or rolling element bearing, whereas other devices may operate across a range of speeds and regimes, because of cyclically varying speeds, for example, the ring-liner contact between top or bottom dead-centre and mid-stroke and the gear contact between pitch point and tip.

Slide 42

Sliding Hertzian Contacts



X-axis is normally logarithmic scale

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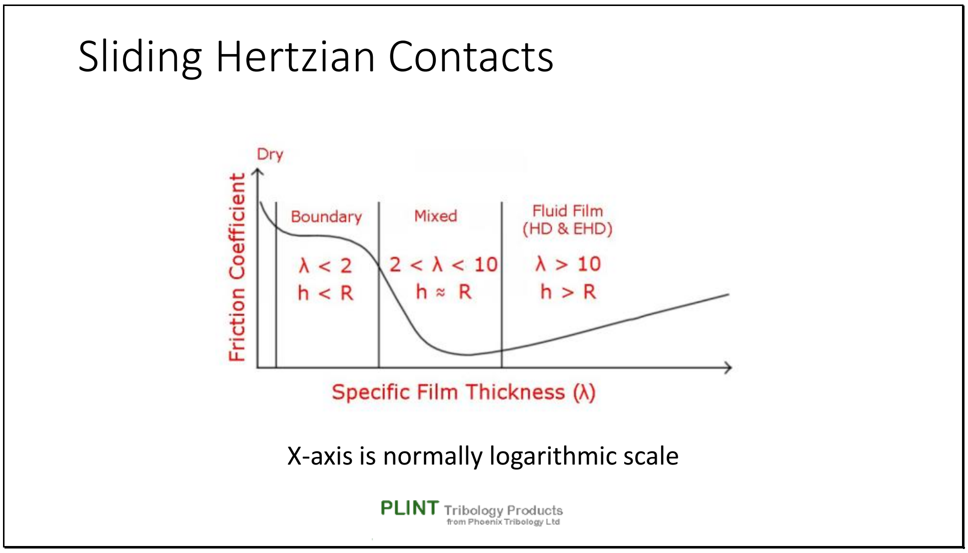
An important point to note is that sliding hertzian point and line contacts, despite claims to the contrary by certain parties, always operate under boundary, or at best, mixed lubrication regimes. Even without calculating the lubricant film thickness, this should be abundantly clear, if one considers practical applications, for example, those involving sliding/rolling hertzian line contacts.

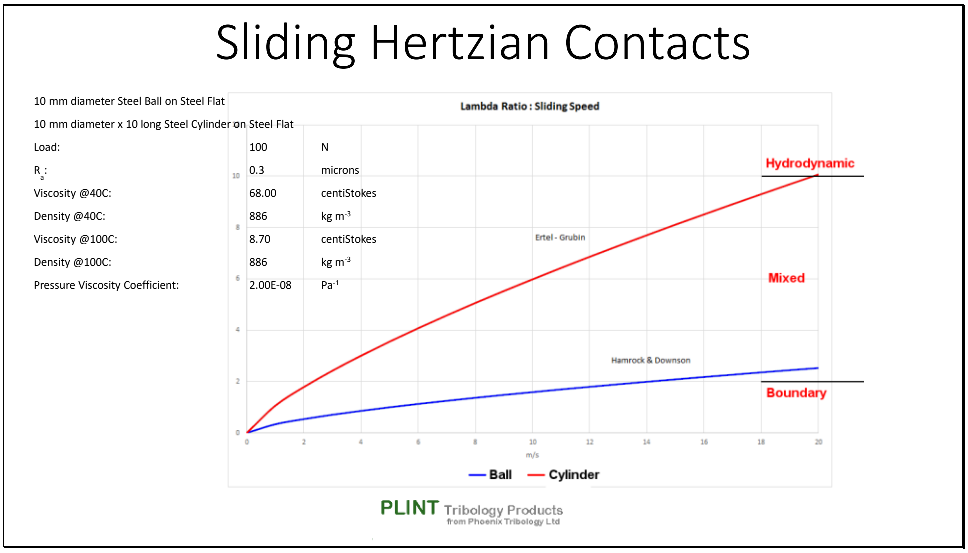
We know that gears and cams and followers operate under mixed lubrication regimes, which is why we need additive chemistry to prevent scuffing. How could we possibly expect a sliding hertzian point contact, in a reciprocating tribometer, going from zero at stroke ends to some modest maximum velocity at mid-stroke, to achieve what cannot be achieved with a real system sliding/rolling line contact?

Slide 43



An alternative form of Stribeck curve can be generated by plotting friction against specific film thickness. It is not difficult to estimate the lubricant film thickness in our test system and make some reasonable assumptions as to where our system might be operating, on the Stribeck curve.

Slide 44



Doing the sums, using some typical values and assuming a surface roughness of 0.3 microns, I arrive at the following for a 10 mm diameter steel ball on flat and a 10 mm diameter by 10 mm long steel roller on flat. For the ball on flat, I have used Hamrock and Dowson's equation and for the line contact, Ertel-Grubin. There are other equations to choose from, all semi-empirical, but all giving similar order of magnitude results.

These film thickness curves of course assume that with increasing sliding velocity, the temperature in the contact does not go up, with a corresponding reduction in lubricant viscosity and film thickness.

Not to put too fine a point on it, with a ball sliding on a flat, even at fairly massive sliding speeds, is unlikely to be operating anywhere other than in the boundary regime. Even with the line contact in this example, a sliding speed in excess of 2 m per second is required, just to make the transition from boundary to mixed regimes. There are not too many reciprocating tribometers that can achieve that sort of sliding speed!

Slide 45

Sliding Hertzian Contacts

If the test, whatever the tribometer and whatever the specimen configuration, produces wear, we can say with a high degree of certainty that the surfaces have not been separated by a hydrodynamic lubricant film

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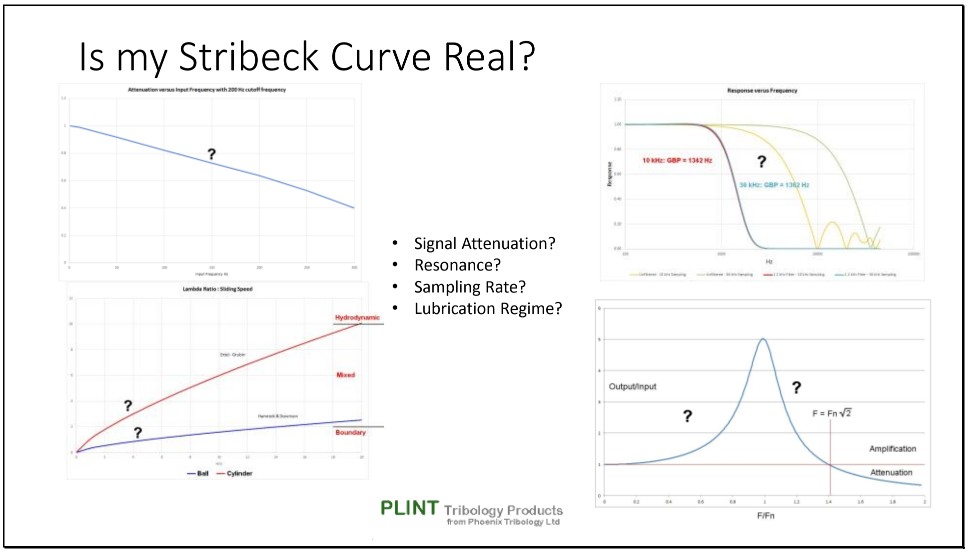
Of course, we don’t really need to go to the bother of doing any calculations, if we consider one very obvious point: if the test, whatever the tribometer and whatever the specimen configuration, produces wear, we can say with a high degree of certainty that the surfaces have not been separated by a hydrodynamic lubricant film.

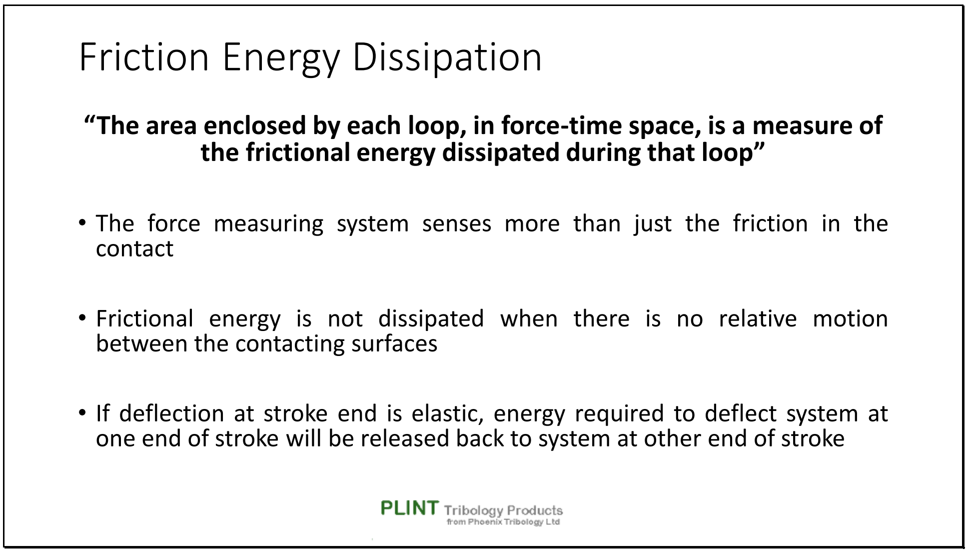
Slide 46



So let us now summarise what we need to be clear about, before we claim to have produced a real Stribeck curve with a reciprocating tribometer. Firstly, am I sure that the signal I am measuring is friction, substantially free of measuring system resonance? Secondly, am I sure that my signal has not been attenuated by poorly tuned filtering or an inappropriate sampling rate? Thirdly, do I believe that my experiment could actually produce anything other than a boundary lubricated regime?

Now let’s think a bit about what effect all this might have on frictional energy dissipation.

Slide 47



ASTM 203 Standard Guide for Determining Friction Energy Dissipation in Reciprocating Tribosystems states:

“The area enclosed by each loop, in force-time space, is a measure of the frictional energy dissipated during that loop”

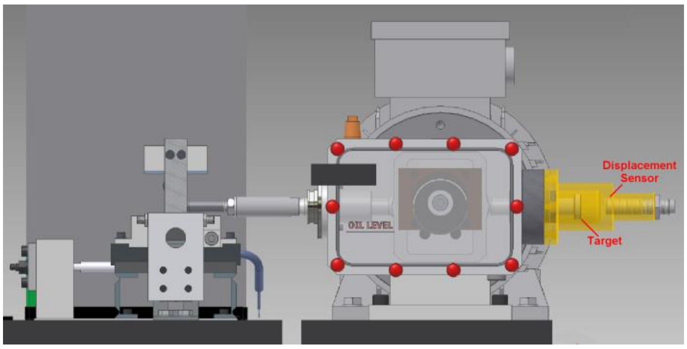
This statement and the subsequent analysis do not stand up to too much scrutiny and are indeed obviously not true for fretting contacts.

The reasons are as follows:

1. The force measuring system, as previously discussed, senses more than just the friction in the contact, of which vibration and resonance are potentially significant components, depending of the dynamics of the measuring system.
2. Frictional energy is clearly not dissipated when there is no relative sliding motion between the contacting surfaces. It follows that if there is a dwell period, at stroke reversal, caused by deflection of the test system itself, this should not be included as contributing to frictional energy dissipated.
3. Assuming the deflection at stroke end is elastic, then the energy required to deflect the system at one end of the stroke will be released back to the system at the other end of the stroke. In this case, the area enclosed by the loop is not a measure of the energy dissipated in the system.

Slide 48

Friction Energy Dissipation – Bulk Sliding



Stroke sensor remote from the contact

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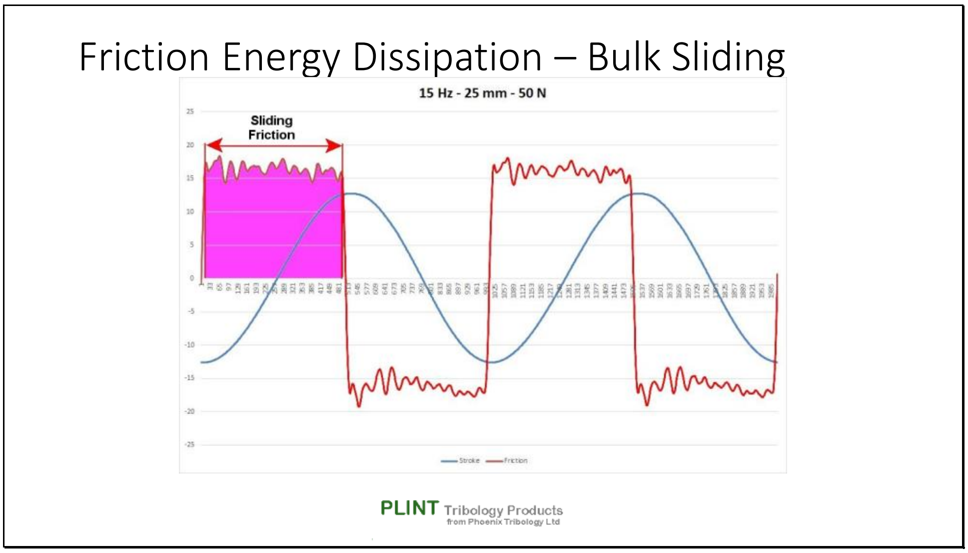
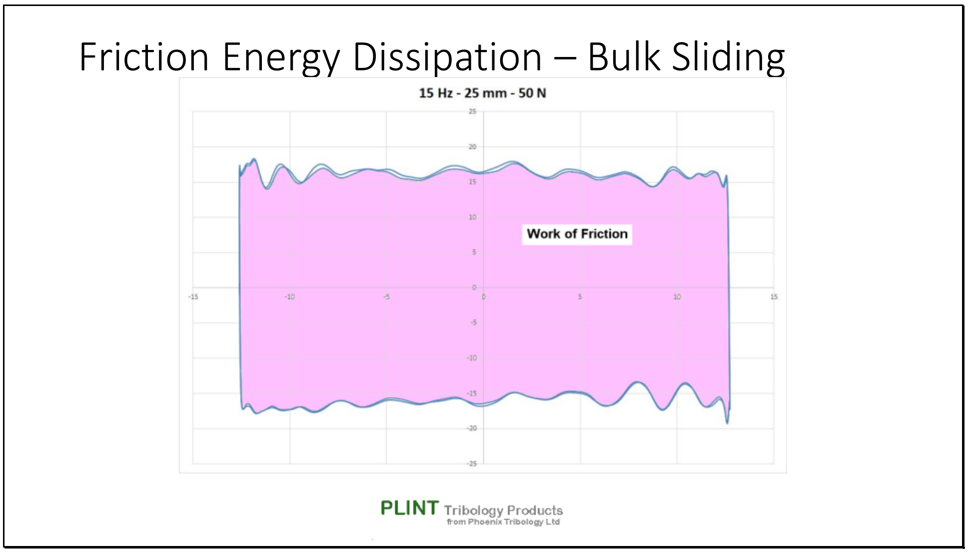
The first thing to consider is where the displacement is being measured. In the case of the TE 77, we measure the reciprocating motion on the drive system, not by measuring the motion of the moving specimen referenced to the fixed specimen. This works, because the machine and its driving system are relatively stiff and the deflection small compared with the typical stroke length.

Slide 49



For bulk sliding, if the test system is relatively stiff, the sliding distance, during which frictional energy is dissipated, will closely match the mechanical displacement of the driving system. Provided the measuring system has the necessary signal bandwidth, the friction trace at the beginning of the stroke will rise sharply to the point at which sliding commences, indicating minimal elastic deflection and adequate signal bandwidth.

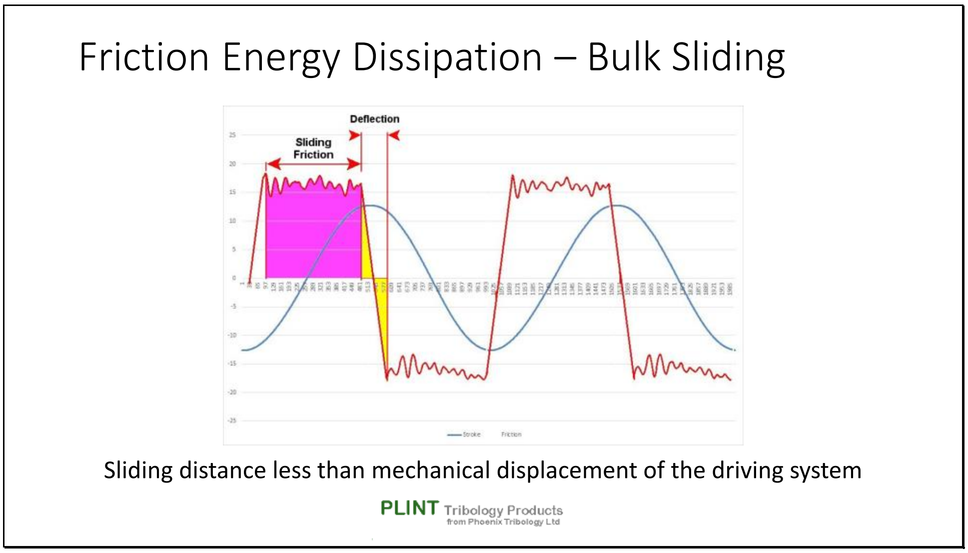
Slide 50



The corresponding force-displacement loop will have straight and near vertical transitions from negative to positive and vice versa, at stroke ends.

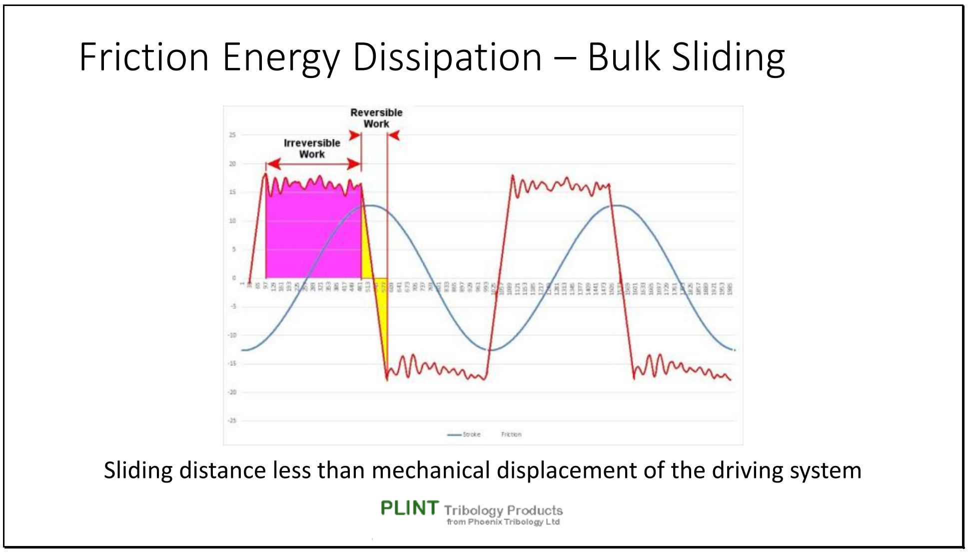
If there are horizontal wobbles in these vertical transitions, in other words, perturbations or oscillations on the displacement axis, we know there must be mechanical vibration in that plane.

Slide 51



For a less stiff system, the sliding distance will be less than the mechanical displacement of the driving system, so the frictional energy dissipation will be lower than for a stiffer system.

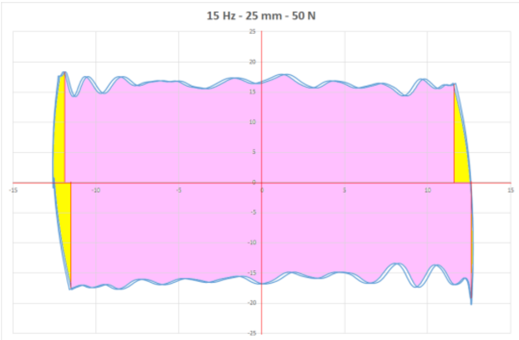
Slide 52



The area within the loop (or under the curve) includes both the irreversible work and the reversible work associated with deflection of the system. Assuming that the test system deflection is elastic, the work done deflecting the system at the beginning of the stroke, to the point at which sliding starts, will be recovered at the end of the stroke, as the direction of motion reverses and the deflection forces are released. In this case, the area enclosed by each loop is not a measure the energy dissipated during that loop, it is a measure of the energy dissipated, plus the stored energy.

Slide 53

Friction Energy Dissipation – Bulk Sliding



Sliding distance less than mechanical displacement of the driving system

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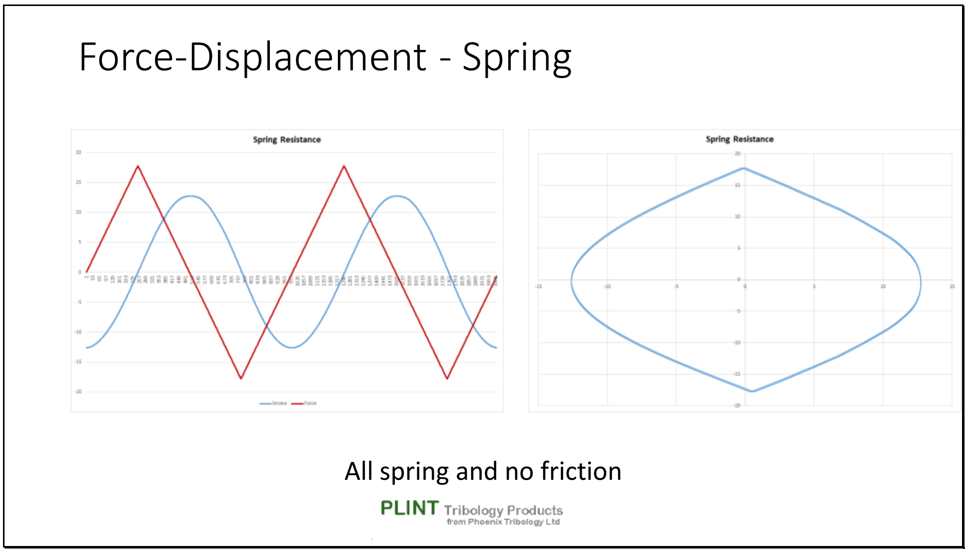
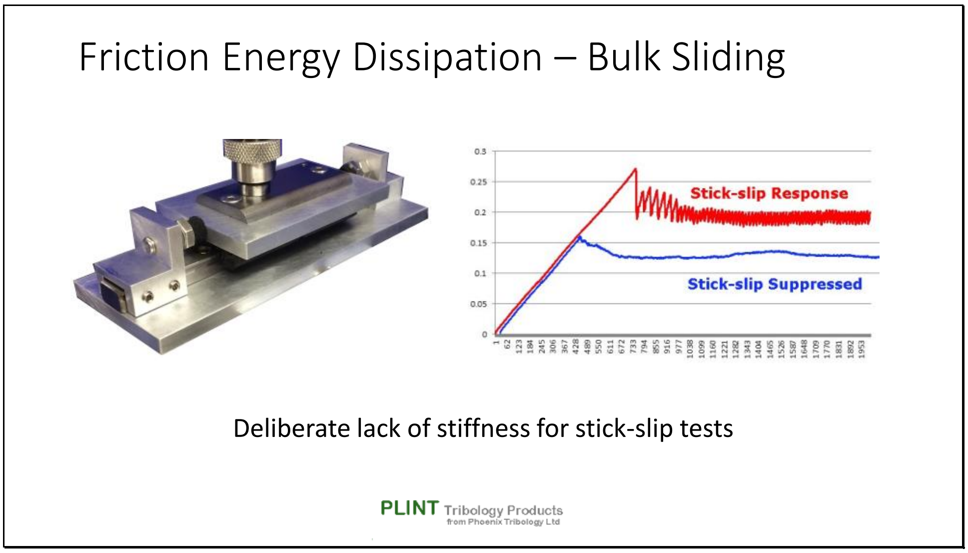
When we plot the force-displacement loop, the transitions become less straight. If we can’t make our test system stiffer, we can, of course, correct our force-displacement calculation to include only the parts of the loop during which sliding actually occurs.

Slide 54



It is perhaps instructive to see what happens if instead of friction resisting the driving motion, we drive against a linear spring; in other words, we have a system where we have all elastic deflection and no sliding. In essence, for bulk reciprocating sliding, the less like a vertically sided rectangle the force-displacement loop becomes, the more of an issue we have with deflection of the tribometer.

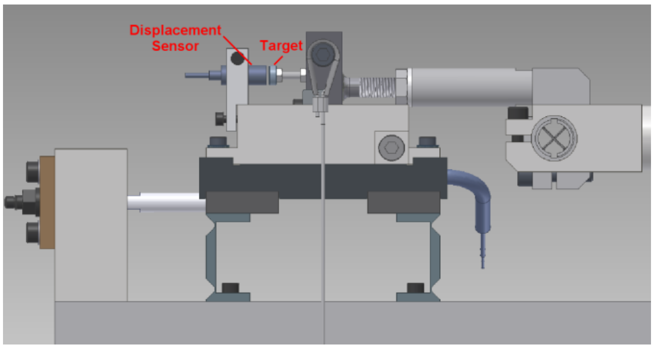
Slide 55



It is worth noting that sometimes we deliberately reduce the stiffness of our measuring system, in order to increase the dwell time and to promote stick-slip. Because of the deliberately low stiffness of the system, it will be obvious that we have to run this type of experiment at low reciprocating frequencies.

Slide 56

Friction Energy Dissipation - Fretting



Stroke sensor measuring between fixed and moving specimens

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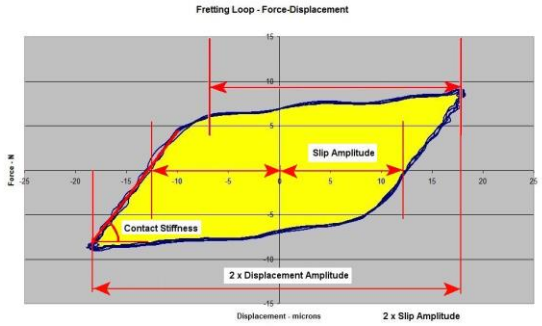
For fretting tests, where we are dealing with very small relative movements, we have to measure the reciprocating motion of the moving specimen, relative to the fixed specimen. This avoids any issues with flexing of the machine itself, which may be of the same order of magnitude as the required fretting motion.

Slide 57

Friction Energy Dissipation - Fretting



In order to achieve satisfactory results, test system must be substantially stiffer than fretting tribo-contact

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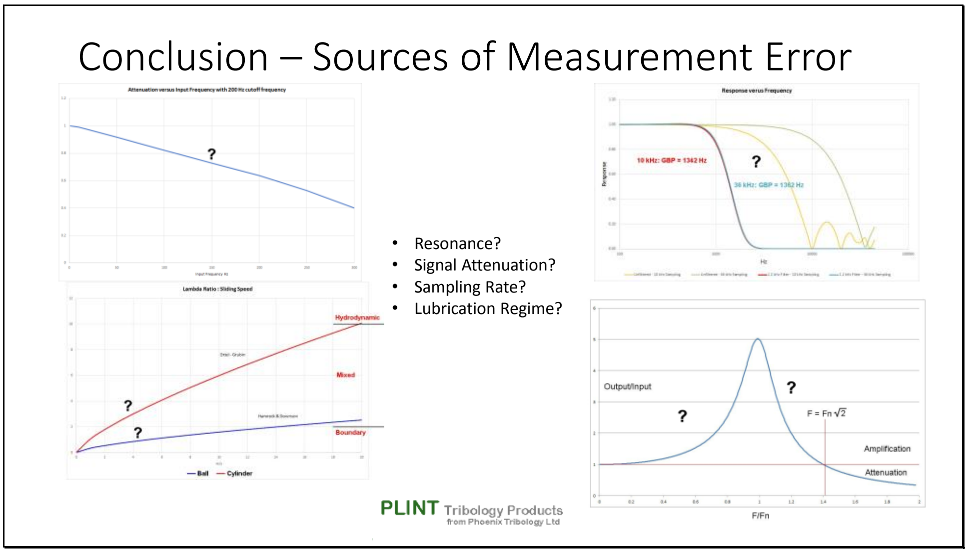
A fretting loop is somewhat more complicated than a bulk sliding loop, because the contact itself involves deflection. However, in this case, the deflection is not elastic, as with test system deflection; the work done in deflection of the contact drives plastic deformation and fracture, so is irreversible.

It follows that the area enclosed by each loop is a measure of the total energy dissipated during that loop, but this is not the same as the frictional energy dissipated.

The slip amplitude is the actual slip in the contact. Because of compliance within the contact, it is less than the driving amplitude of the system.

It will be apparent that in order to achieve satisfactory results, the test system itself must be substantially stiffer than the fretting tribo-contact.

Slide 58



To conclude, to understand the indicated value from a reciprocating tribometer friction measurement we need to know the operating point in terms of both friction force measuring system resonance and the overall measuring system signal bandwidth. The latter of course includes any filtering. We also need to use an appropriate sampling rate, to avoid aliasing.

Ideally the maximum reciprocating frequency should be no more than a third of the natural frequency of the measuring system and orders of magnitude less than the signal bandwidth.

If we want to plot force-displacement loops to determine energy dissipation, we must ensure that the force and displacement measurements are not subject to different time delays.

Before claiming that results from a reciprocating test run at different frequencies indicates something to do with Richard Stribeck, at least make some attempt to calculate the lubricant film thickness, to verify whether it is remotely possible.

We need to be careful not to confuse energy dissipation with frictional energy dissipation and, for systems of limited stiffness, we need to be aware that some energy might be stored and released, not dissipated.

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Conclusion - Need to Know

For Friction Force Measurement:
- Measuring system signal bandwidth
- Measuring system resonance
- Type and cut-off frequency of any filtering
- Sampling rate

For Energy Dissipation Loops:
- Force and displacement signals not subject to differing time delays
- Stiffness of tribo-tester to determine stroke end deflection

For Stribeck Curves:
- Estimate of lubricant film thickness

For Fretting Loops:
- Slip amplitude versus driving amplitude

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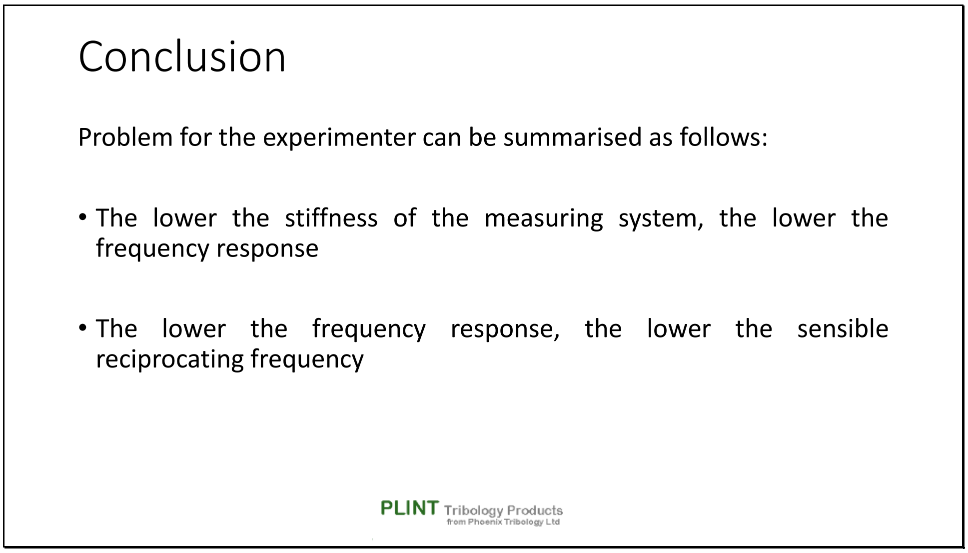
And finally, any test standard involving dynamic force measurement should include the requirement to specify:

- The signal bandwidth of the measuring system
- The resonant frequency of the measuring system
- The type and cut-off frequency of any filtering
- The sampling rate

Or perhaps we should just ask everybody to equipment themselves with some string, a pulley, a weight and a pair of scissors.

Slide 60



To conclude, overall, the problem for the experimenter can be summarised as the following:

> The lower the stiffness of the measuring system, the lower the frequency response.
>
> The lower the frequency response, the lower the sensible reciprocating frequency.

Slide 61

Acknowledgements and Thanks

For their comments, guidance, assistance and suggestions:

Professor John Williams	University of Cambridge
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