THE DYNAMIC TESTING OF ELASTOMERS

by R. C. Frampton

(Technical Director Davenport-Nene Testing Systems)

1 Introduction

Elastomers are finding wider and wider applications throughout industry. For example, the automotive industry is continually attempting to improve fuel efficiency. The resulting reduction in body and engine weights put increasing pressure on suspension and sound insulation designers to meet the car buyers demands for improved ride quality and quiet cars. Designers are no longer satisfied with simple static performance data but need data that assists the design of complex systems involving elastomeric components.

A difficulty often encountered lies in the communication problems between the materials technologist, the component designer and the test equipment manufacturer. This paper attempts to try to clarify some of the confusing terms in dynamic testing and point out some to the potential problems, which can be encountered by those unfamiliar with dynamic testing.

2 What is Dynamic Testing?

The term 'Dynamic' implies movement, however, in reality any form of mechanical testing will involve some form of movement. For example, a simple tensile modulus test will involve stretching the material at some defined speed.

The distinction between static testing and dynamic testing is not simply the speeds involved. There are often cases where a 'static' test will require a higher test speed than a 'dynamic' test. The distinction is actually in the nature of the information we need to obtain from the test.
When related to elastomers, the information from a conventional static test is usually concerned with material quality and tells us very little about the properties that affect the application.

An example of this is the anti-vibration mount: The designer selecting a material for an anti-vibration mount needs sufficient information to ensure he will obtain a suitable resonant frequency to give adequate vibration isolation and provide sufficient damping to prevent magnification of the vibration at the resonant frequency of the system. Therefore, the designer has little need for conventional static parameters such as extension at break and 100% modulus. The designer needs information such as: 'dynamic modulus' and 'tan-delta'. Furthermore, after designing and manufacturing such a mount, it will become necessary to validate the design by testing.

In simple terms the dynamic test is intended to provide:

a) Data for engineering design purposes.
b) Performance validation of sample products.

It may seem surprising but it is usually possible to generate this information using a simple tensile testing machine providing the operator is adequately skilled. The principal difficulty is in the time taken to interpret the data. There is a second difficulty in that the test does not directly simulate operational conditions, therefore the end user of the information, or product, may not have confidence in the results.

We can conclude that an important reason for dynamic testing is to simulate as far as possible real operational conditions before carrying out field trials, and to feed back quantitative information to the component/systems designer.

3 Dynamic Properties

Elastomers exhibit a time dependent behaviour often referred to as 'Viscoelasticity': If we rapidly compress a piece of rubber and remove the force, the rubber will take time to recover to its original dimensions. Furthermore the force required to compress the material will depend upon the speed of the compression. A very simple model of viscoelastic behaviour is the combination of a spring and a dashpot (Fig. 1).

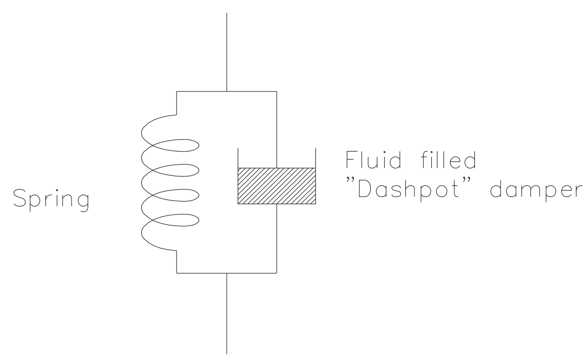


FIG.1 Spring and Dashpot model of a Viscoelastic Material

The spring will obey Hooke's Law and give us a force directly proportional to displacement, but without any speed or other time dependent effects (Fig. 2).

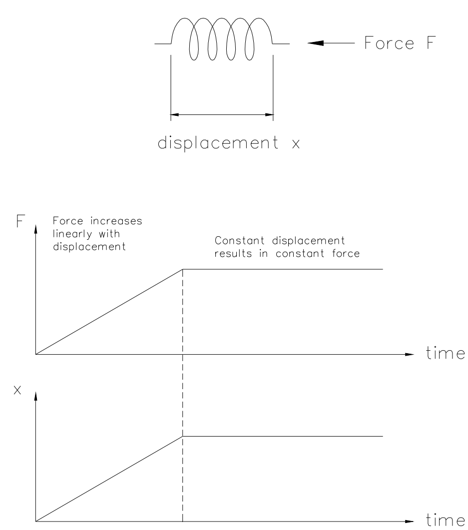


FIG.2 Relationship between force and displacement for a spring.

The energy used to compress the spring is stored by the spring and can be recovered at any time. The dashpot will give us a force proportional to velocity but as it produces no force when not moving it cannot store and return energy (Fig. 3).

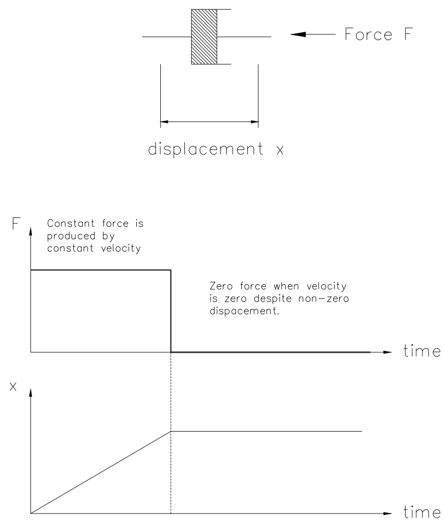


FIG.3 Relationship between Force and Displacement for a Dashpot.

Furthermore the energy used to move the force against the viscous drag of the dashpot is lost as heat in the dashpot oil. This simplified model demonstrates the two fundamental properties of elastomers:

a) Energy storage in an elastic medium.
b) Energy loss in a viscous medium.

This combination also results in the observed time dependent behaviour (Fig. 4).

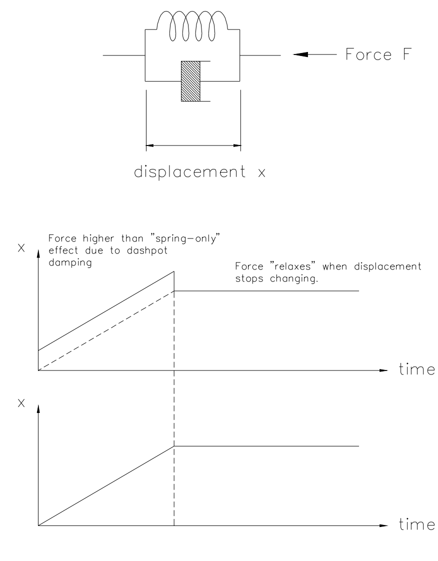


FIG.4 Time dependent effects of a spring/dashpot combination.

If we now examine how the two components of this model behave when subject to sine wave excitation we can begin to see how the various parameters in common use have been developed.

3.1 Elastic Stiffness

When compressing a spring with a sine wave displacement the resultant force will follow the displacement waveform (Fig. 5) therefore at zero displacement we will observe zero force. As the displacement increases, the force will increase, and at maximum displacement we will observe maximum force. Similarly in tension we will observe maximum force at maximum displacement. This is a direct consequence of Hooke's Law and we can see that the force is exactly in phase with the applied displacement.

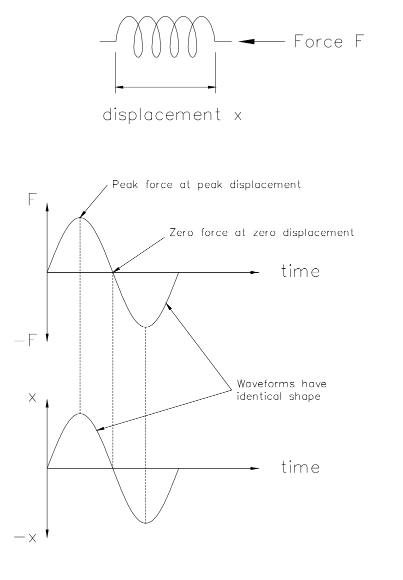


FIG.5 Effect of a sinewave on a spring.

3.2 Loss Stiffness

The viscous effects are quite different: The forces generated are totally independent of displacement and are proportional to velocity. If we examine a sine wave (Fig. 6) we will see that maximum velocity occurs at zero displacement and minimum velocity (zero velocity) occurs at the turning points (peaks). It is easy to see that peak viscous forces are generated at zero displacement and minimum viscous forces are generated at the displacement peaks.

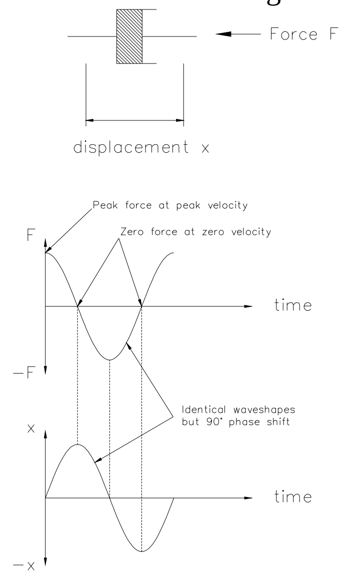


FIG.6 Effect of a sinewave on a dashpot.

A slightly more rigorous approach is to say that: since the viscous force is proportional to velocity, we can determine the applied velocity by differentiating the displacement waveform (the differential of a sine wave is a cosine wave) and this is 90^o out of phase with the displacement waveform.

It is easy to conclude that a real material which is excited at a frequency and amplitude where the peak elastic force and the peak viscous forces are equal will exhibit a net combined force which is developed at a mid point between the zero displacement point of the viscous effect, and the peak displacement point of the elastic effect (Fig. 7). Hence we have 45^o phase shift.

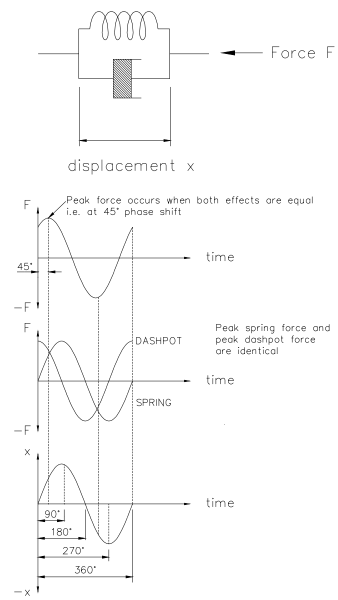


FIG.7 Effect of a sinewave on a spring/dashpot combination

Furthermore materials, which have a low viscous effect, will exhibit a small phase angle between force and displacement whereas materials with a high viscous effect will exhibit a high phase angle. The maximum theoretical phase angle being 90^{o}.

In this discussion we have introduced the concept of:

i) Elastic stiffness - which in simple terms could be regarded as the 'static' stiffness.

ii) Viscous stiffness - which could be regarded as a measure of damping.

Although our hypothetical design engineer could make use of these parameters, they are not directly measurable by classical methods, and should be regarded as theoretical concepts to help our understanding of the behaviour of the material. The two fundamental parameters, which are normally measured in practise, are:

a) Dynamic stiffness (this being the combined effect of elastic and viscous effects).

b) Phase difference between force and displacement. (Damping)

If we can measure these two parameters we can then calculate the elastic and viscous effects using simple vector arithmetic (Fig. 8):

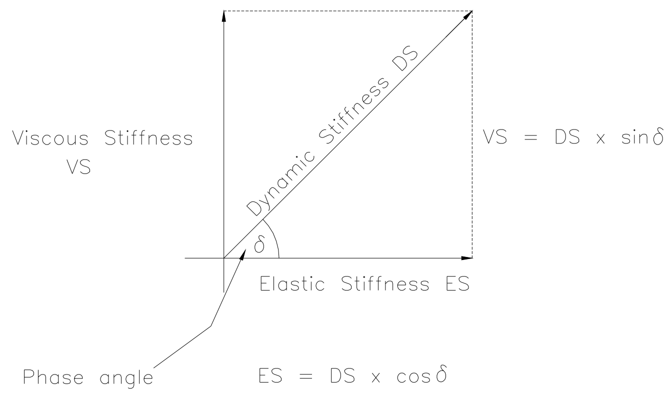


FIG.8 Relationship between Dynamic, Viscous and Elastic stiffness

Viscous stiffness = DS x sin δ
Elastic stiffness = DS x cos δ
Where DS = the measured dynamic stiffness.
δ = the phase difference between the force and displacement waveforms

Since we have used vector arithmetic to relate the three stiffness values, it is more correct to refer to 'Dynamic Stiffness' as 'Complex Stiffness'; however, both terms are in common use.

As before we still have a measure of component stiffness and a measure of viscous drag or to use the more common term 'damping'.

These parameters are just as meaningful to our design engineer, but are more directly measurable (Fig. 9):

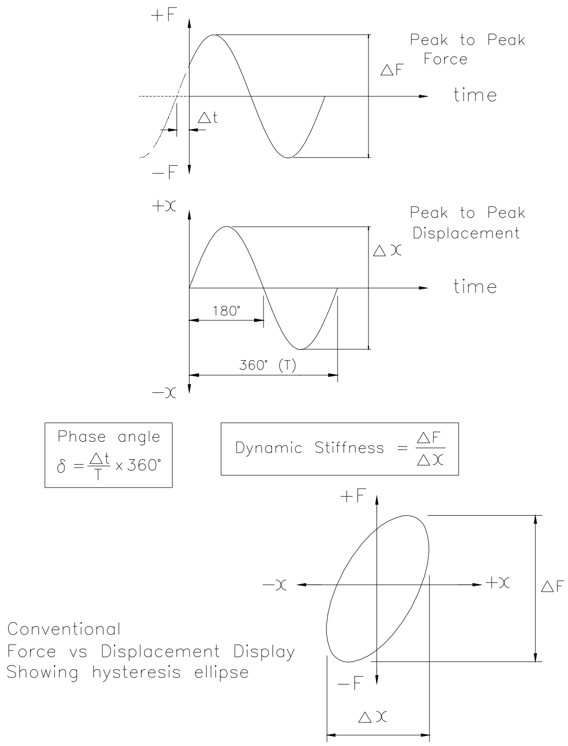


FIG.9 Measurement of Dynamic Stiffness and phase angle

We can measure peak to peak displacement = $\boldsymbol{\Delta x}$
We can measure peak to peak force = $\boldsymbol{\Delta F}$

Therefore complex (dynamic) stiffness = $\frac{\Delta F}{\Delta x}$

and phase angle δ is simply the phase shift between the two waveforms, which can be found by measuring the time difference between the zero points of the two waveforms $\Delta \boldsymbol{t}$ and providing we know the period of the waveform $\boldsymbol{T}$ then:

$$\delta = \frac{\Delta t}{T} \cdot 360^{o}$$

All other parameters, currently in common use, are either derivations of these or are synonyms of these parameters.

For example:

i) tan delta (Fig. 10) is useful as it is the ratio of loss stiffness to elastic stiffness and is simply found by calculating the tangent of the force/displacement phase shift.

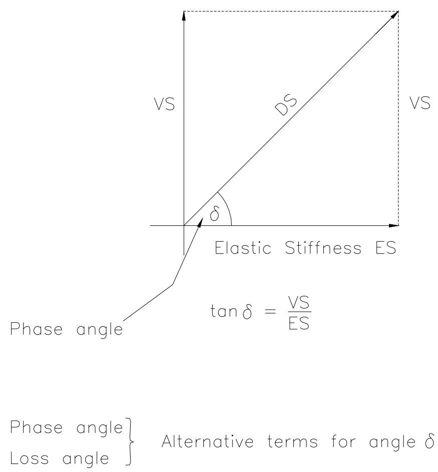


FIG.10 Definition of Tan–Delta

ii) Synonyms for the force/displacement phase shift
Loss angle
Phase angle

iii) Synonyms for our fundamental elastic stiffness (Fig. 11)

	Elastic Stiffness **Spring Stiffness** **Storage Stiffness** **In-Phase Stiffness**
Fig 11	Common Terms for Elastic Stiffness

iv) Synonyms for our fundamental loss stiffness parameter (Fig. 12)

	Loss Stiffness **Out-of-Phase Stiffness** **Inelastic Stiffness** **Damping Stiffness** **Viscous Stiffness**
Fig 12	Common Terms for Loss Stiffness

v) Synonyms for Viscoelastic stiffness
(Fig. 13)

Dynamic Stiffness **Complex Stiffness** **Viscoelastic Stiffness** **Dynamic Rate** **Dynamic Spring Constant**
Fig 13 Common Terms for Dynamic Stiffness

NOTE: dynamic stiffness is traditionally defined as the ratio of the peak-to-peak force and the peak-to-peak displacement:

$$DS = \frac{\Delta F}{\Delta x}$$

NOTE: Wherever the expression 'stiffness' has been used this refers to a component characteristic. When making a material assessment it is more usual to refer to 'modulus'. To convert from stiffness values to modulus values we simply take the component dimensions into account and normalise the results for standard unit dimensions. Therefore wherever a stiffness parameter is defined we may assume that an equivalent modulus parameter is also defined.

vi) Energy loss (per cycle)

This is another form of our generalised damping parameter. Although it is not as directly useful to our long-suffering design engineer, he can still derive the information he needs from it. As we saw earlier the energy used to overcome the viscous properties of the material is lost as heat in the material and can be expressed quantitatively by loss angle. This energy loss can be expressed directly in energy terms. Its use has probably grown through it simplicity of measurement by non-sophisticated measurement equipment:

Display on a graph (or oscilloscope) the force - displacement curve and the shape will usually be a closed hysteresis loop (ellipse) (Fig. 14). The area enclosed by the ellipse is the energy loss per cycle - simply count the squares on the graph paper!

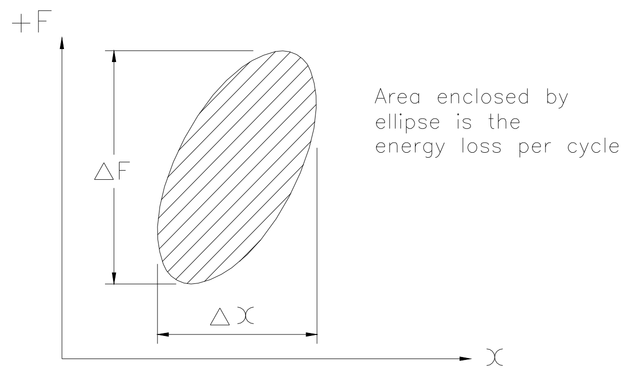


FIG.14 Energy loss per cycle

The relationship between loss angle and energy loss per cycle E can be shown to be (Fig. 15):

Energy loss per cycle $$E = \frac{\pi}{4} \cdot \Delta F \cdot \Delta x \cdot Sin\delta$$

Where ΔF = peak to peak force
and Δx = peak to peak displacement

This relationship is to be found in most national standards such as:
DIN 53513
and BS 903

Fig 15 Relationship between Energy Loss and Loss Angle

Practical Considerations

When considering the application of the preceding theory to modern test equipment design, three practical factors have to be taken into account.

i) Linearity. Real materials and, in particular, components are never linear in behaviour (Fig. 16) and therefore caution should be used when calculating the interrelationships between parameters.

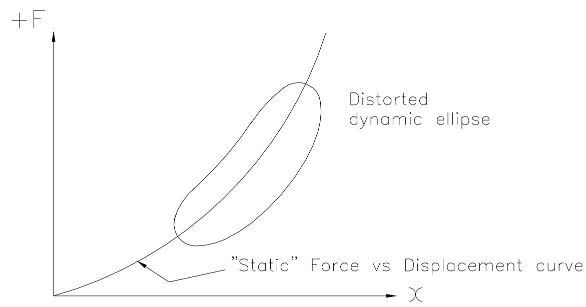


FIG.16 Force vs Displacement curve
due to non–linear material

As long as everyone uses the same definitions and test conditions, no problems should be encountered with customer and supplier discussions, but our design engineer may experience some difficulties in relating measured performance to design predictions. With non-linear components care should be taken to always use consistent pre-loads or pre-strains, dynamic amplitudes and test frequency.

ii) Signal noise. This is probably the most significant problem in real measurement systems. In dynamic testing we normally have to deal with small signals and high frequencies.

These are not the most favourable conditions for good signal to noise ratio, therefore the measurement and analysis techniques must employ noise rejection techniques, which have little influence on the results.

iii) Speed. In today's world answers are always required immediately if not sooner! The techniques used to speed up test methods are usually directly incompatible with those used to combat noise and to some extent incompatible with non-linear components.

For example, when applying a pre-load it is good practice to allow a delay before commencing the test, to allow the part to settle (remember the viscoelastic recovery!) at this point any error (in the pre-load) should be corrected before commencing the test (Fig. 17). If the part is linear then the pre-load error will normally not be significant and no delay and correction would be necessary.

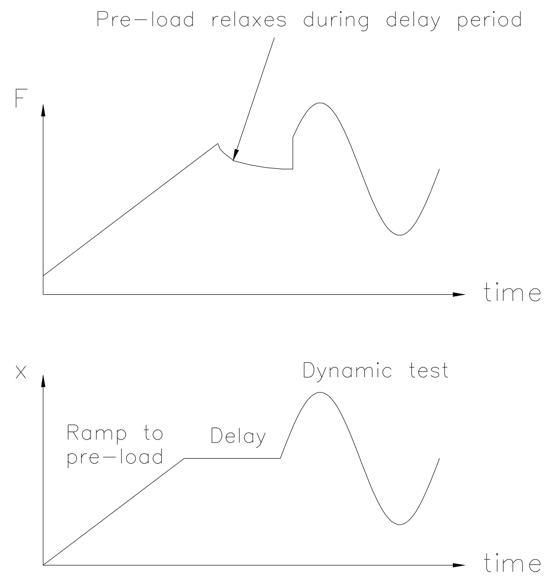


FIG.17 Preload Relaxation

In order to combat noise, apart from good design practise, only one technique is available to the equipment designer - Integration. I.E. averaging results. If we take as an example the definitions of dynamic stiffness and loss angle described in the national standards, where dynamic stiffness is defined as peak-to-peak force divided by peak-to-peak displacement we can see that the result must be very sensitive to signal noise. We can reduce the effect of noise by averaging over a number of cycles. The effect of the noise error will be reduced by the factor $\sqrt{n}$ where n is the number of cycles evaluated (Fig. 18):

$$\Delta DS = \frac{\Delta DS'}{\sqrt{n}}$$

Where $\Delta DS'$ = typical dynamic stiffness standard deviation
ΔDS = standard deviation after averaging
n = number of cycles

Fig 18 Improvement in Dynamic Stiffness variations by Averaging

Similarly the loss angle evaluated from the hysteresis loop area can be calculated over a number of cycles to reduce the effects of noise.

An alternative technique is to apply a very narrow band filter to the signals.

For example consider a testing machine, which is capable of evaluating components up to a maximum frequency of 100Hz. We could carry out a test at say 50Hz with a filter bandwidth of

1Hz, which would reject any signal (and noise) which does not lie between 49.5Hz and 50.5Hz. The net result (Fig. 19) is that the effect of the noise is reduced by the factor of:

$$\sqrt{\frac{100Hz}{1Hz}} = 10$$

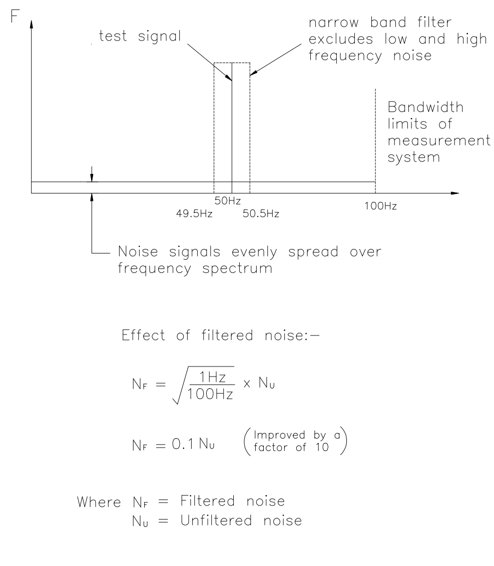


FIG.19 Effect of narrow band filter on noise

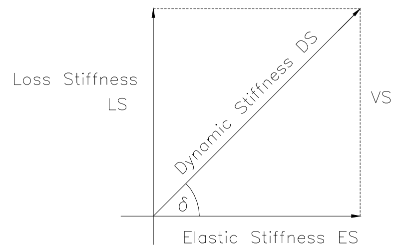
With modern instrument techniques this is relatively easy to achieve. The most common form of this filter is a mathematical calculation known as the Fourier Transform.

Nene Instruments have developed a special high-speed form of the Fourier transform analysis algorithm that is processed by the main control computer.

Unlike the standard (area method) of evaluation the Fourier transform will produce as its primary parameters:

Elastic stiffness (ES) and loss stiffness (LS)

From these we calculate, as before, by Vector arithmetic (Fig. 20):



$$DS = \sqrt{ES^2 + LS^2}$$

$$\text{Loss angle } \delta = \arctan \frac{LS}{ES}$$

FIG.20 Derivation of loss angle and Dynamic Stiffness from Fourier Transform Results

The two approaches are fundamentally compatible. The Fourier transform giving more consistent results due to its ability to filter more efficiently.

There is, as always, a potential problem. If the component or material is non-linear the Fourier transform will filter out some of the non-linear distortion effects along with the noise. The result being that we should expect that Fourier transform and 'standard' method will give different answers. This is in fact the case, but in practice the differences are small and Nene Instruments would always recommend the use of Fourier transform methods wherever possible.

It should be remembered that *'standard'* methods do not give 'true' measurements of dynamic stiffness and loss angle for non-linear components any more than Fourier transform methods do. The method only gives a *'conventional'* approximation. It is true to say that the hysteresis loop area does give a true measure of energy loss but the conversion to loss angle still involves linear theory assumptions.

The end result is that Dynamic stiffness and loss angle evaluated by either method for non-linear materials or components involves some theoretical assumptions. Therefore the method chosen cannot be made on the basis of which is correct! The decision may have to be made according to which method your customer uses. The good news is that the two methods usually give results that agree closely and Nene Instruments offer both methods in their standard analysis package.

It is probably worth mentioning at this point, that when evaluating materials (as opposed to components) tests conducted in shear mode will tend to behave more linearly than when tested in compression.

Another, effect often overlooked, is that an elastomer will never give test results that are as repeatable as a metal spring. This is thought to be due to the filler structure which allows a number of stable states when un-preloaded. The effect of this is to increase the apparent noise in the load signal.

Test Machines

Machines capable of applying the forces over an appropriate range of frequencies are usually of the servo-hydraulic type and most modern machines are software driven to improve ease of use and to produce the answers quickly.

However, no amount of software sophistication can compensate for poor mechanical arrangements at the 'front-end'. All dynamic machines will have a 'closed-loop' straining frame similar to any simple tensile tester. The difference is that the rigidity of the frame must be high and that there must be no structural resonances within the operating frequency range of the machine. Ideally there should be no structural resonance at frequencies below at least twice the maximum operating frequency! A problem often overlooked by the user is that of design of jigs and fixtures. There is little point in having a high stiffness frame and a low stiffness sample holder (or grips). Remember unless an extensometer is used (which may be difficult at high frequencies) it is usually impossible to know whether the sample, or the sample holder is moving, and the test may be measuring the characteristics of the sample holder. The complement to this problem is that high mass is just as undesirable as low stiffness.

High mass attached to the load cell converts it into an accelerometer and any movement of the machine frame will be detected as an error in the load signal which cannot be distinguished from a real signal.

The designer of a good machine will help to minimise the problem by ensuring that the frame is as heavy as possible thus reducing the amount of movement which can be transmitted to the load cell and its attached mass. A good machine will also have a low mass piston to minimise the inertial reaction forces transmitted to the frame. Good jig design therefore is also essential to avoid adding to the piston mass. Many users are aware that load cell mass should be minimised but do not realise that minimising mass attached to the actuator piston is just as important.

The effect of these masses can be seen in the following diagram (Fig. 21):

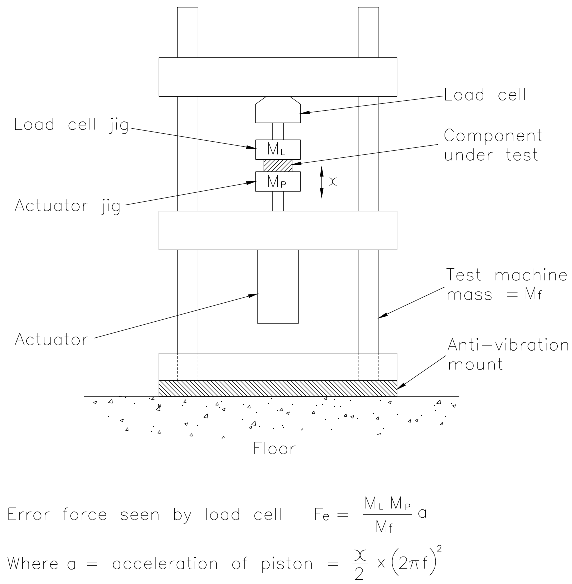


FIG.21 Effect of test jig masses on load cell

$$Fe = \frac{Ml\,Mp}{Mf}\,a$$

Where *Fe* = force error measured by load cell

Ml = mass attached to load cell

Mp = mass attached to piston (including piston mass)

Mf = mass of test machine frame

a = acceleration of piston at test conditions

= amplitude x $(2\pi f)^2$

f = test frequency

It is also important to realise that mounting the machine rigidly on a solid floor helps to reduce mass coupling to the load cell. Suspending the machine on anti-vibration mounts deliberately allows the machine to move, and this movement will be sensed by any mass attached to the load cell. The policy within Nene Instruments is to use a rigid support which stands on the floor without any isolation.

Fatigue Testing

This is a specialised branch of general dynamic testing. The purpose, in simple terms, is to find out how long a component (or material) will survive when subjected to a particular set of conditions.

For example. An engine mount may be subject to a static compressive preload of 2kN with a ±1.5kN dynamic load superimposed. The test frequency may be 15Hz and the waveform sinusoidal. The object being to find out how many cycles the component will endure before failure. A second requirement is to determine the mode of failure to allow the designer to make improvements to the design.

The description of this typical test has two important implications.

a) The test conditions imply a constant maximum and minimum load over the life of test. Most characterisation tests are carried out with a controlled displacement and the resultant force is measured. Fatigue life is normally highly dependent on peak forces. Depending on the material, life may be inversely proportional to the third or fourth power of force. Therefore, it is very important to accurately control the peak values of the force. The highly non-linear relationship between force and life means that the shape of the force waveform is not usually relevant, just the magnitude of the peaks.

A system that will allow accurate evaluation of fatigue life must maintain accurate peak loads despite reducing component stiffness. This criterion is generally not achieved particularly well, especially with highly non-linear components, by conventional force feedback control loops traditionally used by servo-hydraulic machines. Good control will normally only be achieved by modern adaptive control loops such as the software based iterative load control loop employed by Nene Instruments. This can accommodate highly non-linear components whose characteristics change with time and yet control load peaks to within better than 1%.

b) The second point for consideration is how do we define the end point of the test: when do we declare that the part has failed?

The traditional method was to apply a simple displacement limit to the control system
For example our hypothetical engine mount may require an input displacement of ±5mm in order to achieve ±1.5kN force. The designer might decide that failure should be considered to have occurred when the machine needs to supply ±8mm in order to achieve the required force.

The difficulty with a simple 'over displacement limit' is that catastrophic failure has probably occurred by this point. It is probably more realistic to define failure by a performance degradation limit: for example define failure when the loss angle has fallen by 20%. Traditional testing systems would only allow this to be done by stopping the fatigue test, for example every 6 hours and carrying out a dynamic characterisation test.

Modern digital, software based, systems can now perform this task automatically. Allowing a more frequent assessment of performance, thereby allowing the test to be stopped prior to catastrophic failure with the resultant opportunity to carry out detailed visual examination.

Nene Instruments have taken this automated fatigue evaluation a step further with 'real-time calculations' this allows the continuous monitoring of the evolution of any measured or calculated parameter with the opportunity to use these parameters to define the end of test condition.

Conclusions

This paper has attempted to clarify some of the confusion surrounding dynamic testing and at the same time show how the evolution of testing methods is providing the manufacturers and users of elastomeric components and materials with more powerful measurement and analytical tools.

R C Frampton