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GrindoSonic
THE IMPULSE EXCITATION TECHNIQUE

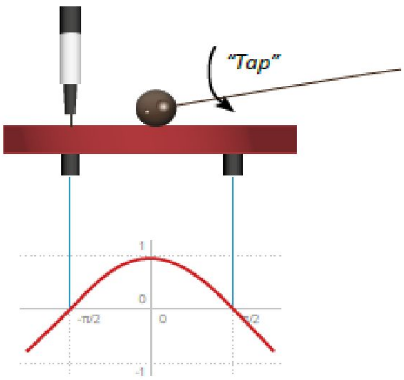
Dynamic Material Testing

The basics of the Impulse Excitation Technique

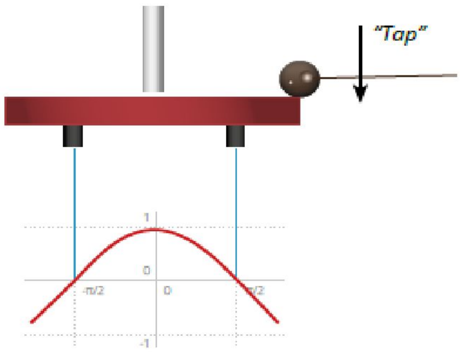
The principal, vibration modes and practical issues for testing at room temperature

The Principal

Using usually a beam test-piece with uniform cross-section (round, square or rectangular), the characteristic vibration frequencies are determined either by continuously driving the vibration and sweeping the frequency to detect resonances, or by striking it, allowing it to ‘ring’, and then deconvoluting the recorded sound spectrum. The latter method is often termed the ‘natural frequency’ method or the Impulse Excitation Technique (IET). The same set of equations is employed for both methods in order to relate frequency to elastic modulus. The methods and key issues are described in the following sections, with more detailed information relevant to particular set-ups or measurements – such as more detailed background on the equations and geometric correction factors, advice on selection of bar dimensions appropriate for different test modes, disc specimens and single crystal materials.



Piezo-electric probe



Accoustic vibration measurement

There are a number of standards which specify the formulae to be used for calculating elastic moduli. The formulae generally have a long and well-developed pedigree, and generally have their simplest form for long thin beams. With shorter aspect ratios there are usually correction functions with increasing complexity. Within the accuracy of determining the dimensions or density of the test-piece, these more complex forms can generally be avoided or truncated without significant error provided that the beam test-pieces are ‘thin’, i.e. length to thickness ratio typically > 20. The formulae are well described in the ***ASTM* E 1876**

Vibration modes of beams

The first step in understanding these methods is to appreciate the modes of vibration of the test-piece. These will depend to some extent on how and where the beam is supported, and the location at which the vibration is driven or excited by striking. For a prismatic beam, there are basically four vibration modes of interest:

- Out-of-plane flexural
- In-plane flexural
- Torsional
- Longitudinal

The ***flexural vibration frequencies*** are controlled primarily by the Young's modulus (E) of the test-piece in the longitudinal direction, essentially independently of any material anisotropy.

The ***torsional mode vibrations*** for an isotropic material are controlled primarily by the shear modulus (G) of the test-piece. If the material is anisotropic, things become more complicated, and it is best if the principal axes of the test-piece are parallel to the axes of anisotropy. The vibrations are controlled by a mix of shear stiffness in the principal planes of the test-piece containing the longitudinal direction.

The ***longitudinal mode vibrations*** are controlled primarily by the Young's modulus (E) of the test-piece and the Poisson's ratio in the longitudinal direction.

1. *Out-of plane flexural*

There is a series of nodes (minimum vibration) and antinodes (maximum vibration) along the length of the freely supported beam (a 'free-free' beam in mathematical terminology). At the lowest resonant or natural frequency (the fundamental mode), the nodes are 0.223 of the length of the beam from each end, with antinodes at each end and in the centre. For overtones or harmonics, there are more closely spaced nodes and antinodes, and these are shown schematically in Figure 20.

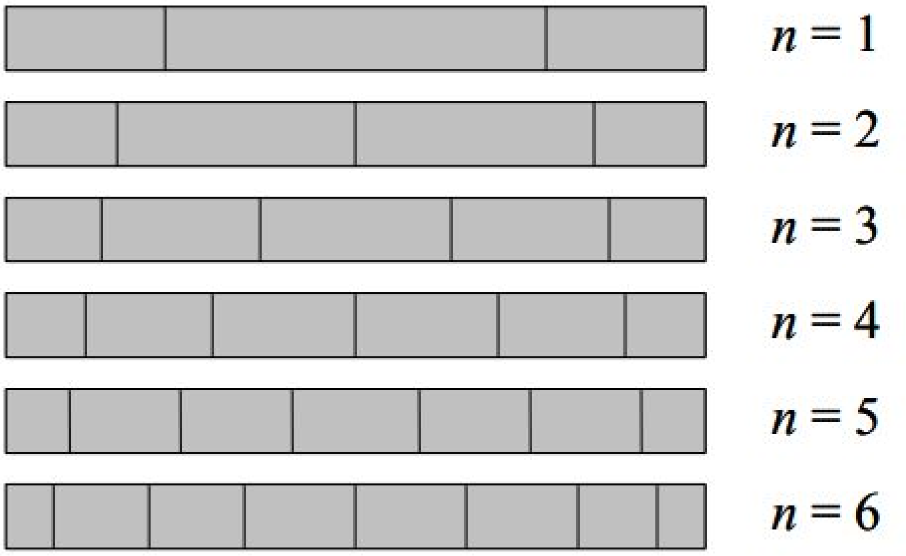


***Figure 20:** Out-of-plane vibration modes of a 'free-free' beam with the nodal lines indicated*

From Figure 20, it can be seen that the fundamental mode can be driven or arise during impact if the beam is supported at the nodes so as not to damp out the vibration, and excited elsewhere, especially midway between nodes or at either end. It can also be deduced that driving or striking it in the beam centre will not result in harmonics n = 2, 4, 6, etc., because these also have a node in the centre, so only alternating modes will be detected. All modes can be excited if the beam is driven or impacted near the end.

Since in most cases the interest is in identifying the fundamental (n = 1) mode, it is general practice to support the test-piece at or close to the fundamental flexural node positions and to drive or impact the test-piece in the centre. If a suspension method (e.g. a loop of thread or wire) is used both to support and to drive the test-piece in resonance, then the usual practice is for the loops to be positioned just outside the nodal positions.

However, it is also of interest, but sometimes a necessity, to be able to identify the series of frequencies to ensure that the fundamental has been detected and correctly identified. This often arises with long thin test-pieces where higher modes are more easily excited than the fundamental. For flexure, these higher mode frequencies are not in a convenient simple relationship to the fundamental mode as a result of beam end inertia effects, and for a long thin beam the series is in the ratio (see Annex 2):

1 : 2.757 : 5.404 : 8.933 : 13.344 : 18.638 : .....

The ratios become a little smaller as the length to thickness ratio of the beam is reduced. By comparing the measured ratios with these predicted values, a clear

identification of the series can be made and, importantly, the flexural modes can be distinguished from other modes that may be excited.

2. *In-plane flexural*

Turning the test-piece by 90° about an axis parallel to its length produces another fundamental and series of harmonics that relate to the geometry of the test-piece. The pattern will be the same as in Figure 20.

3. *Torsion*

Unlike the flexural modes, torsional vibration modes for a long thin isotropic 'free-free' beam are in a convenient ratio sequence of:

1 : 2 : 3 : 4 : 5 : 6 : ..... as shown in Figure 21

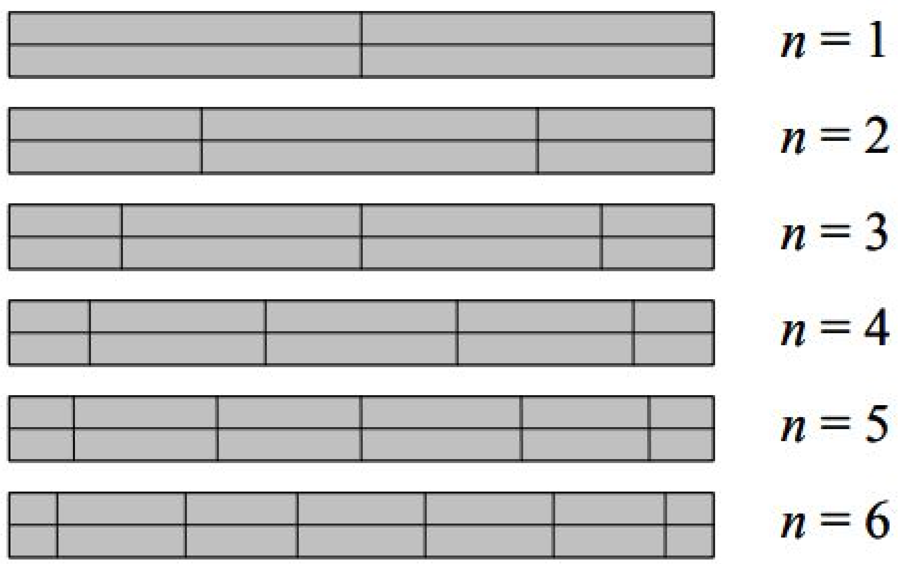


Figure 21: *Torsional vibration modes of a 'free-free' beam with the nodal lines indicated.*

In order to obtain the fundamental mode, the beam therefore has to be supported at its centre, and vibrated at or near its ends by driving or impacting at a location off the centre line, so that most of the energy goes towards twisting the test-piece rather than flexing it. This is more difficult to achieve than the simple support or suspension that can be used for the flexural mode. This aspect will be dealt with later. An alternative approach that has been tried at NPL is to target the n = 2 mode. By comparing Figures 20 and 21 it can be seen that the nodal positions for fundamental flexure and second mode torsion have similar geometries. With these support positions, if a vibration driver or a strike is applied near the beam centre, but to one side of the centre line, both n = 1 flexure and n = 2 torsion will be

preferentially excited. If the system employed can detect both modes simultaneously, a second geometry set-up is not needed to obtain the two key parameters of Young's modulus and shear modulus.

In order to identify whether a torsional mode interferes with a flexural mode excited at the same time, the information given in Annex 2 can be used. The dimensions of the test-piece, typically the width to thickness ratio, can be selected to ensure separation of flexural and torsional frequencies to make identification clear.

To calculate Poisson's ratio (v) for an isotropic material, both Young's modulus (E) and shear modulus (G) need to be determined. Then the following formula can be used:

$$\mathrm{v} = \frac{E}{2G} - 1$$

Accurate values of both E and G are needed for an accurate value of Poisson's ratio. A 1% error in either E or G results in typically a 4% error in v when v ≈ 0.33!

4. *Longitudinal*

Exciting longitudinal mode vibrations is commonly undertaken, but care has to be taken to ensure that rather lower frequencies due to simultaneous flexure or torsion are not also identified as the fundamental longitudinal mode. Typically, the fundamental longitudinal mode frequency is five to ten times greater than that of the fundamental flexural mode, and so should be easy to identify. Higher longitudinal harmonics are seldom encountered, but for a long thin test-piece they can be assumed to be an integral multiplier of the fundamental mode. This assumption becomes invalid when the ratio of the cross-sectional dimension to wavelength exceeds about 0.2.

Vibration modes of discs

An alternative approach that has been found to work well for isotropic materials is to use disc-shaped test-pieces and to excite the out-of-plane vibration. There are two series of vibration modes, one of which is symmetrical about the disc axis with concentric ring nodes (termed 'flexural' or 'diaphragm mode' here), and the other has a series of radial nodes and antinodes (termed 'torsional', or 'saddle' mode). Figure 22 shows the lowest order modes of each type. The ratio of the frequencies of the two lowest modes gives Poisson's ratio through a look-up table. Either of the two frequencies with Poisson's ratio gives Young's modulus. Shear modulus can then be computed from Young's modulus and Poisson's ratio. See Annex 3 for further details.

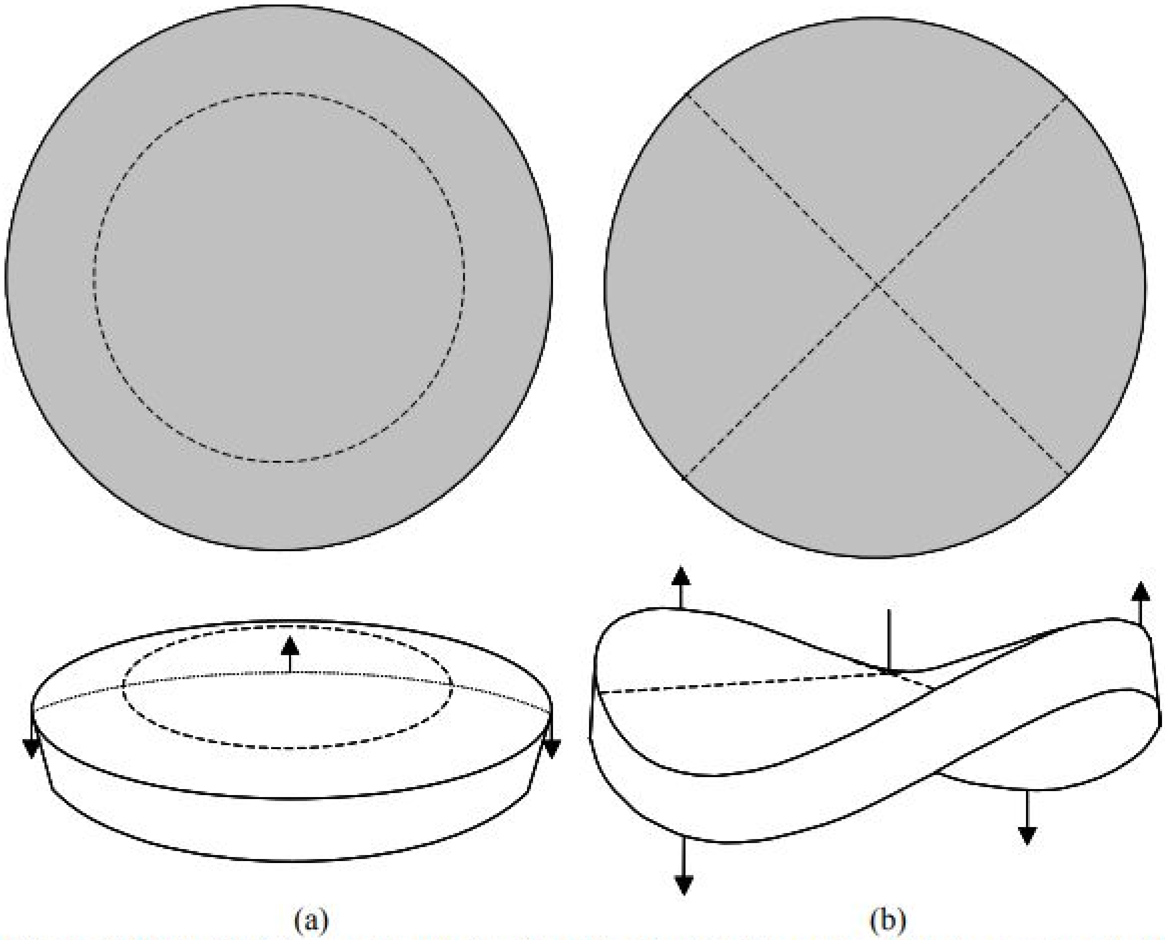


Figure 22: *The first two modes of vibration of a disc, (a) flexural or 'diaphragm' mode, (b) torsional or 'saddle' mode. The dashed lines indicate the nodal lines and the arrows the direction of motion.*

Practical issues

Test-piece dimensions and surface finish

It is clearly best if the test-pieces are specially prepared for this measurement, and have dimensions and dimensional tolerances that are appropriate. In particular, there are three main issues to consider:

- To ensure that the frequencies at which the test-pieces vibrate in the anticipated modes are within the capability of the measurement system. If appropriate, use an estimate of Young's modulus to calculate the frequency that is to be expected.

- Test-piece dimensions should be scaled to ensure that the appropriate frequencies can be obtained, and are not confused by overlapping values for different modes.

- The test-pieces should be prepared with accurate geometry and dimensions. The principal factor in flexural measurement is usually the thickness of the beam or disc, which enters into the equations as the third power. Ensure that the faces of the test-pieces are accurately parallel to better than 0.3% (i.e. parallelism better than 0.001 mm for a 3 mm thick test-piece). The test-piece ends, often ignored for other types of testing, also need to be square to the length and machined flat.

Test-pieces should also have a good surface finish. Thickness measurements are generally made on the raised parts of the surface, and not on the centre-line of the roughness. It can be imagined that the stiffness of the test-piece is determined by the thickness of the bulk material excluding the rough surface. Smooth, preferably lapped finishes are desirable, otherwise overestimation of the structural thickness of the test-piece will occur, leading to an underestimate of moduli.

Support methods

Support methods for test-pieces depend on the arrangements made for excitation and detection. In some respects, there is no ideal solution. The key issue is to minimise the interaction between the test-piece and its environment, but at the same time keep it in position during the testing. This is particularly critical for high-temperature testing where readjustment of the position during the tests is usually not possible

For the flexural resonance method, the test-piece should be supported at its nodes (normally at the fundamental mode flexure nodes) as presented in Figure 23a.

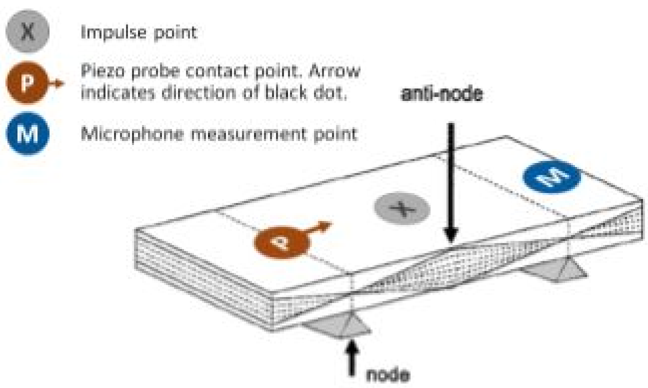


***Figure 23a**: supporting and exciting principal flexural resonance mode*

For the torsional resonance method, the best option is presented in Figure 23b. This ensures that torsion is excited preferentially.

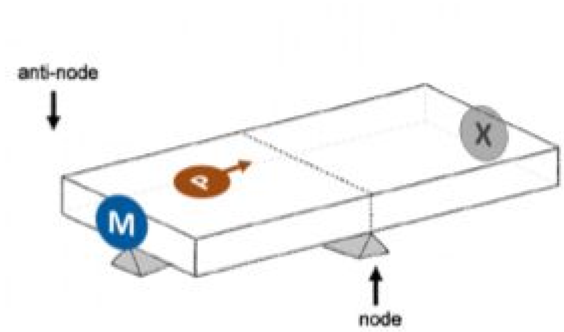


***Figure 23b**: Supporting and exciting principal torsional resonance mode*

For the longitudinal resonance method, the test-piece can be rested on a compliant support such as a foam layer, such that the driver contacts one end and the detector contacts the other.

In the case of disc shapes or grinding wheels, the supporting and exciting principle is illustrated in Figure 23c.

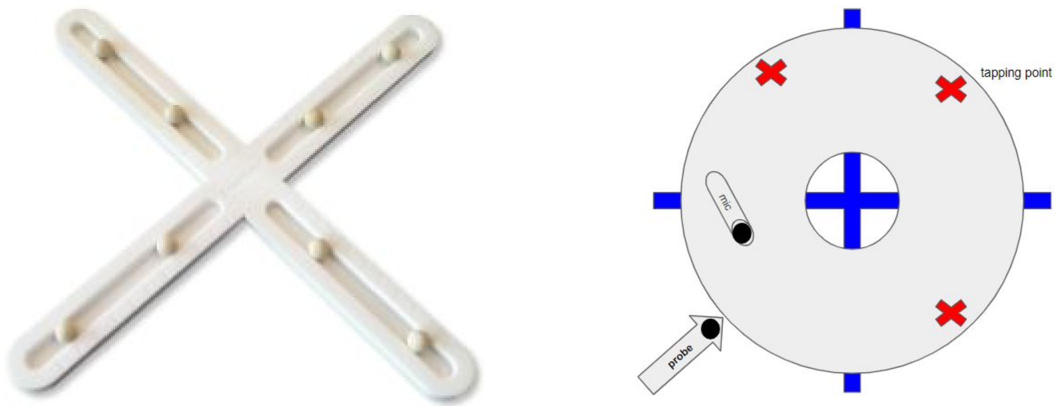


***Figure 23c:** Support and excitation of grinding wheel*

Resonance excitation

The usual method with the resonance techniques is to employ a device which vibrates at a controllable level, and where the driving frequency can be swept over a wide range (typically 100 Hz to 200,000 Hz). The GrindoSonic MK7 includes a set of excitation hammers of various sizes and materials. Also included is a ceramic reference bar to verify the proper functioning of the device. See Figure 24.

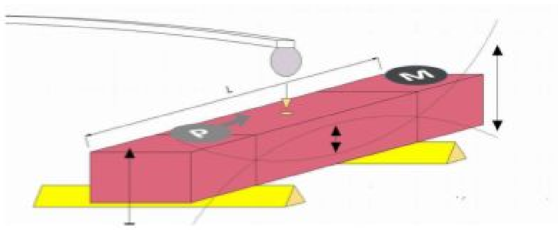


Figure 24: *excitation tools*

Automated impactors are also suitable, especially for high-temperature use. An electro-magnetically driven pulser with a captive flier attached is typical. The flier strikes the test-piece at the appropriate position and falls back into the pulser. It is probably true to say that more consistent and controllable impacts can be made using a pulser than can be achieved manually or by the ball-drop method.

Vibration detection

In resonance, the traditional method is to use a record player pick-up, but nowadays if this is not available a sensitive piezo-sensor is needed. A piezo-sensor can also be used for the impact excitation method. The tip of the sensor can be placed in direct contact with the test-piece, although this does risk affecting the dynamic mass of the test-piece. It is often better to lightly contact the test-piece close to, but not at, a node, using a thin extension probe to the sensor, so that there is minimum interference with the vibration developed. See also in Fig. 23 a,b,c

As an alternative method, a microphone can be used, but it is important to ensure that it is capable of detecting the ultrasonic frequencies that can be developed. It may be necessary to place it very close to the surface of the test-piece if the sound intensity is low.

Damping

In addition to the identification of resonant frequencies, the dynamic methods can also be used to measure damping, or at least to undertake comparative measures of damping. To do this, the damping of vibration arising from the suspension system must be minimised, and this requires accurate positioning of the suspension or support system at the test-piece nodes. Damping has the effect of broadening the shape of the resonant peaks, and in most materials will usually increase markedly at temperatures at which the materials start to become plastic or where one or more of the phases begins to melt.

Sources :

1. *User Manual GrindoSonic MK7 vn.4.1.*
2. *National Physical Laboratory (NPL) : A National Measurement Good Practice Guide N° 98 - Elastic Modulus Measurement by J D Lord and R Morrell*