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SHAKER E MARTELLO STRUMENTATO

3.0 INTRODUCTION

In this chapter we shall be concerned with the measurement techniques which are used for modal testing. First, it is appropriate to consider vibration measurement methods in general in order to view the context of those used for our particular interest here. Basically, there are two types of vibration measurement:

(i) those in which just one parameter is measured (usually a response level), and

(ii) those in which both input and response output are measured.

Recalling the basic relationship:

$$\boxed{\text{RESPONSE}} = \boxed{\text{PROPERTIES}} \times \boxed{\text{INPUT}}$$

we can see that only when two of the three terms in this equation have been measured can we define completely what is going on in the vibration of the test object. If we measure only the response, then we are unable to say whether a particularly large response level is due to a strong excitation or to a resonance of the structure. Nevertheless, both types of measurement have their applications and much of the equipment and instrumentation used is the same in both cases.

We shall be concerned here with the second type of measurement, where both excitation and response are measured simultaneously so that the basic equation can be used to deduce the system properties directly from the measured data. Within this category there are a number of different approaches which can be adopted but we shall concentrate heavily on one which we refer to as the 'single-point excitation' method. All the others involve simultaneous excitation at several points on the structure and although we shall discuss these briefly at the end of this chapter, our interest will be focused on the more straightforward approach (at least from the viewpoint of the experimenter) where the excitation is applied at a single point (although in the course of a test, this point may be varied around the structure). This type of measurement is often referred to as 'mobility measurement', and that is the name we shall use throughout this work.

3.1 BASIC MEASUREMENT SYSTEM

The experimental setup used for mobility measurement is basically quite simple although there exist a great many different variants on it. In terms of the specific items used. There are three major items:

(i) an excitation mechanism,
(ii) a transduction system (to measure the various parameters of interest), and
(iii) an analyser, to extract the desired information (in the presence of the inevitable imperfections which will accumulate on the measured signals).

Figure 3.1 shows a typical layout for the measurement system, detailing some of the 'standard' items which are usually found. An additional component has been included in this illustration in the form of a Controller. This is now a common feature in many if not most modern measurement 'chains' and can be provided by a desktop, mini- or micro-computer. As many of the detailed procedures in mobility measurements are repetitive and tedious, some form of automation is highly desirable and, if provided by a computer, this can also serve to process the measured data as required for the modal analysis stage, later in the overall process.

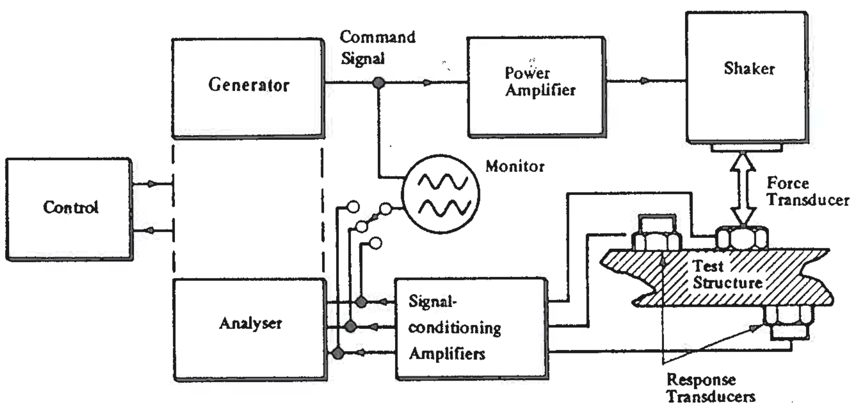


Fig 3.1 *General Layout of Mobility Measurement System*

The main items in the measurement chain are then:

(a) a source for the excitation signal. This will depend on the type of test being undertaken and can be any of the following:
- sinusoidal (from an oscillator)
- random (from a noise generator)
- transient (from a special pulse generating device, or by applying an impact with a hammer)
- periodic (from a special signal generator capable of producing a specific frequency content).

(b) Power Amplifier. This component will be necessary in order to drive the actual device used to vibrate the structure which, in turn, will take one of a number of different forms, as discussed below. The power amplifier will necessarily be selected to match the excitation device.

(c) Exciter. The structure can be excited into vibration in several ways, although the two most commonly (and successfully) used are by an attached shaker or by a hammer blow. Other possibilities exist by step relaxation (releasing from a deflected position) and by ambient excitation (such as wave, wind or roadway excitations), but these are relatively special cases which are only used when the more conventional methods are not possible.

(d) Transducers. Here again, there are a great many different possibilities for the devices available to measure the excitation forces and the various responses of interest. For the most part, piezoelectric transducers are widely used for both types of parameter although strain gauges are often found to be convenient because of their minimal interference with the test object.

(e) Conditioning Amplifiers. The choice of amplifier depends heavily on the type of transducer used and should, in effect, be regarded as part of it. In all cases, its role is to strengthen the (usually) small signals generated by the transducers so that they can be fed to the analyser for measurement.

(f) Analyser. The function of this item is simply to measure the various signals developed by the transducers in order to ascertain the magnitudes of the excitation force(s) and responses. In essence, it is a voltmeter but in practice it is a very sophisticated one. There are different types of analyser available and the choice will depend on the type of excitation which has been used: sinusoidal, random, transient, periodic. The two most common devices are Spectrum (Fourier) Analysers and Frequency Response Analysers although the same functions as provided by these can be performed by a tunable narrow-band filter, a voltmeter and a phase meter plus a great deal of time and patience!

3.2 STRUCTURE PREPARATION

3.2.1 Free and Grounded Supports

One important preliminary to the whole process of mobility measurement is the preparation of the test structure itself. This is often not given the attention it deserves and the consequences which accrue can cause an unnecessary degradation of the whole test.

The first decision which has to be taken is whether the structure is to be tested in a 'free' condition or 'grounded'. By 'free' is meant that the test object is not attached to ground at any of its coordinates and is, in effect, freely suspended in space. In this condition, the structure will exhibit rigid body modes which are determined solely by its mass and inertia properties and in which there is no bending or flexing at all. Theoretically, any structure will possess 6 rigid-body modes and each of these has a natural frequency of 0 Hz. By testing a structure in this free condition, we are able to determine the rigid-body modes and thus the mass and inertia properties which can themselves be very useful data.

In practice, of course, it is not feasible to provide a truly free support - the structure must be held in some way - but it is generally feasible to provide a suspension system which closely approximates to this condition. This can be achieved by supporting the testpiece on very soft 'springs', such as might be provided by light elastic bands, so that the rigid body modes, while no longer having zero natural frequencies, have values which are very low in relation to those of the bending modes. ('Very low' in this context means that the highest rigid-body mode frequency is less than 10-20% of that for the lowest bending mode.) If we achieve a suspension system of this type, then we can still derive the rigid body (inertia) properties from the very low frequency behaviour of the structure without having any significant influence on the flexural modes that are the object of the test. (In fact, there are several instances where a test of this type may be carried out only to examine the rigid body modes as this is an effective way of determining the full inertia properties of a complex structure.) One added precaution which can be taken to ensure minimum interference by the suspension on the lowest bending mode of the structure - the one most vulnerable - is to attach the suspension as close as possible to nodal points of the mode in question. Lastly, particular attention should be paid to the possibility of the suspension adding significant damping to otherwise lightly-damped testpieces.

As a parting comment on this type of suspension, it is necessary to note that any rigid body will possess no less than 6 modes and it is necessary to check that the natural frequencies of all of these are sufficiently low before being satisfied that the suspension system used is sufficiently soft. To this end, suspension wires etc. should generally be normal to the primary direction of vibration, as in Figure 3.6b rather than the case shown in Figure 3.6a.

The other type of support is referred to as 'grounded' because it attempts to fix selected points on the structure to ground. While this condition is

extremely easy to apply in a theoretical analysis, simply by deleting the appropriate coordinates, it is much more difficult to implement in the practical case. The reason for this is that it is very difficult to provide a base or foundation on which to attach the test structure which is sufficiently rigid to provide the necessary grounding. All structures have a finite impedance (or a non-zero mobility) and thus cannot be regarded as truly rigid but whereas we are able to approximate the free condition by a soft suspension, it is less easy to approximate the grounded condition without taking extraordinary precautions when designing the support structure. Perhaps the safest procedure to follow is to measure the mobility of the base structure itself over the frequency range for the test and to establish that this is a much lower mobility than the corresponding levels for the test structure at the point of attachment. If this condition can be satisfied for all the coordinates to be grounded then the base structure can reasonably be assumed to be grounded. However, as a word of caution, it should be noted that the coordinates involved will often include rotations and these are notoriously difficult to measure.

From the above comments, it might be concluded that we should always test structures in a freely-supported condition. Ideally, this is so but there are numerous practical situations where this approach is simply not feasible and again others where it is not the most appropriate. For example, very large testpieces, such as parts of power generating stations or civil engineering structures, could not be tested in a freely-supported state. Further, in just the same way that low frequency properties of a freely supported structure can provide information on its mass and inertia characteristics, so also can the corresponding parts of the mobility curves for a grounded structure yield information on its static stiffness. Another consideration to be made when deciding on the format of the test is the environment in which the structure is to operate. For example, if we consider a turbine blade it is clear that in its operating condition the vibration modes of interest will be much closer to those of a cantilevered root fixing than to those of a completely free blade. Whereas it is possible to test and to analyse a single blade as a free structure, the modes and frequencies which will then form the basis of the test/analysis comparison will be quite different from those which obtain under running conditions. Of course, theoretically, we can validate or obtain a model of the blade using its free properties and expect this to be equally applicable when the root is grounded, but in the real world, where we are dealing with approximations and less-than-perfect data, there is additional comfort to be gained from a comparison made using modes which are close to those of the functioning structure, i.e. with a grounded root. A compromise procedure can be applied in some cases in which the test object (such as the blade) is connected at certain coordinates to another simple component of known mobility, such as a known mass. This modified testpiece is then studied experimentally and the effects of the added component 'removed' analytically.

In the above paragraphs, we have presented a number of considerations which must be made in deciding what is the best way to support the test structure for mobility measurements. There is no universal method: each test must be considered individually and the above points taken into account. Perhaps as a final comment for those cases in which a decision is difficult we should observe that, at least from a theoretical standpoint, it is always possible to determine the grounded structure's properties from those in a free condition while it is not possible to go in the opposite direction. (This characteristic comes from the fact that the free support involves more degrees of freedom, some of which can later be deleted, while it is not possible - without the addition of new data - to convert the more limited model of a grounded structure to one with greater freedom as would be necessary to describe a freely-supported structure.)

Examples of both types of test configuration are shown in Figure 3.2.

3.2.2 Local stiffening

If it is decided to ground the structure, care must be taken to ensure that no local stiffening or other distortion is introduced by the attachment, other than that which is an integral part of the structure itself. In fact, great care must be paid to the area of the attachment if a realistic and reliable test configuration is to be obtained and it is advisable to perform some simple checks to ensure that the whole assembly gives repeatable results when dismantled and reassembled again. Such attention to detail will be repaid by confidence in the eventual results.

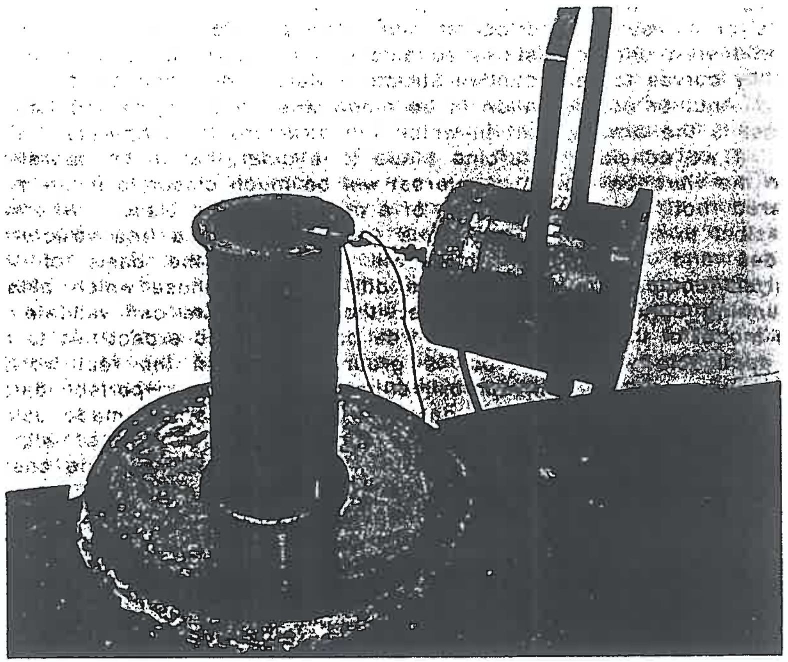
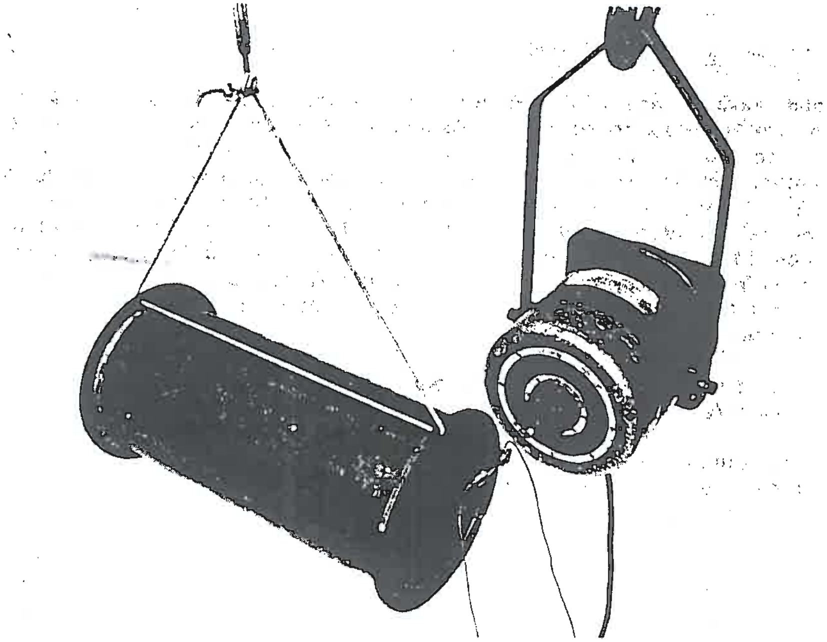


Fig 3.2a Example of Grounded Structure



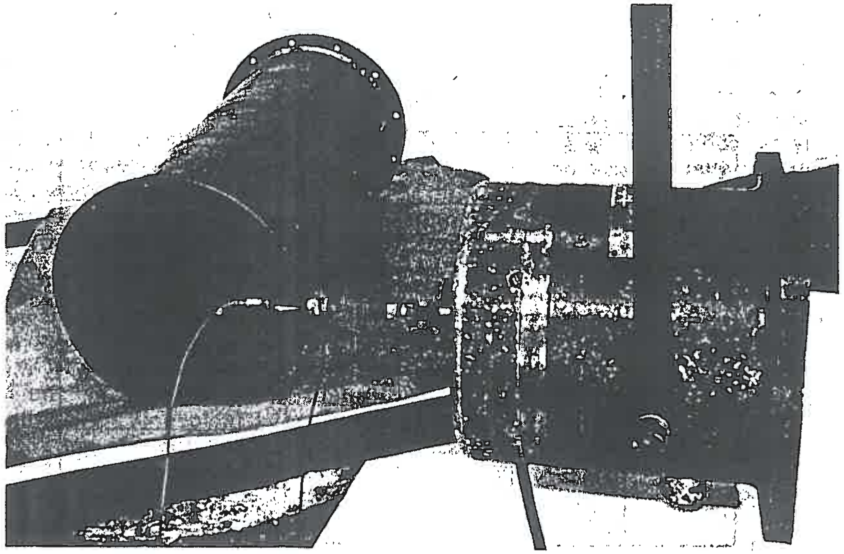


Fig 3.2b Examples of Freely-Supported Structures

3.3 EXCITATION OF THE STRUCTURE

3.3.1 General

Various devices are available for exciting the structure and several of these are in widespread use. Basically, they can be divided into two types: contacting and non-contacting. The first of these involves the connection of an exciter of some form which remains attached to the structure throughout the test, whether the excitation type is continuous (sinusoidal, random etc.) or transient (pulse, chirp). The second type includes devices which are either out of contact throughout the vibration (such as provided by a non-contacting electromagnet) or which are only in contact for a short period, while the excitation is being applied (such as a hammer blow).

We shall discuss first the various types of vibrator, or shaker, of which there are three in use:

mechanical (out-of-balance rotating masses),
electromagnetic (moving coil in magnetic field),
electrohydraulic.

Each has its advantages and disadvantages - which we shall attempt to summarise below - and each is most effective within a particular operating range, as illustrated by some typical data shown in Figure 3.3. It should be noted that exciters are often limited at very low frequencies by the stroke (displacement) rather than by the force generated.

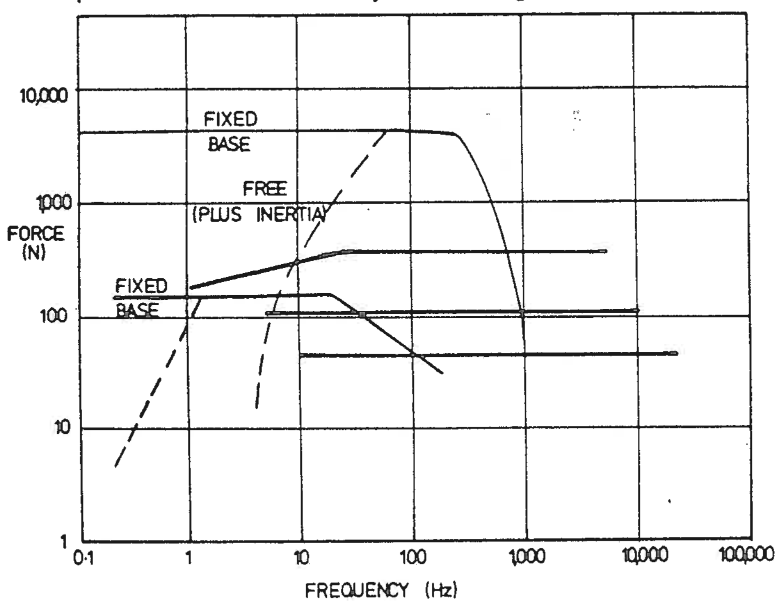


Fig 3.3 Typical Exciter Characteristics

3.3.2 Mechanical Exciters

The mechanical exciter is capable of generating a prescribed force at a variable frequency although there is relatively little flexibility or control in its use. The magnitude of the force is restricted by the out-of-balance and is only variable by making adjustments to this quantity - not something which can be done while the vibration is continuing. Also, this type of excitation mechanism is relatively ineffective at low frequencies because of the speed-squared dependence. However, unless the amplitude of vibration caused by the exciter becomes large relative to the orbit of the out-of-balance masses, the magnitude and phase of the excitation force is known quite accurately and does not need further measurement, as is the case for the other types of exciter.

3.3.3 Electromagnetic Exciters

Perhaps the most common type of exciter is the electromagnetic (or 'electrodynamic') shaker in which the supplied input signal is converted to an alternating magnetic field in which is placed a coil which is attached to the drive part of the device, and to the structure. In this case, the frequency and amplitude of excitation are controlled independently of each other, giving more operational flexibility - especially useful as it is generally found that it is better to vary the level of the excitation as resonances are passed through. However, it must be noted that the electrical impedance of these devices varies with the amplitude of motion of the moving coil and so it is not possible to deduce the excitation force from a measurement of the voltage applied to the shaker. Nor, in fact, is it usually appropriate to deduce the excitation force by measuring the current passing through the shaker because this measures the force applied not to the structure itself, but to the assembly of structure and shaker drive. Although it may appear that the difference between this force (generated within the shaker) and that applied to the structure is likely to be small, it must be noted that just near resonance very little force is required to produce a large response and what usually happens is that without altering the settings on the power amplifier or signal generator, there is a marked reduction in the force level at frequencies adjacent to the structure's natural frequencies. As a result, the true force applied to the structure becomes the (small) difference between the force generated in the exciter and the inertia force required to move the drive rod and shaker table and is, in fact, much smaller than either. See Figure 3.4a.

As this is an important feature of most attached-shaker tests using continuous sinusoidal, random or periodic excitation, it is worth illustrating the point by the following example.

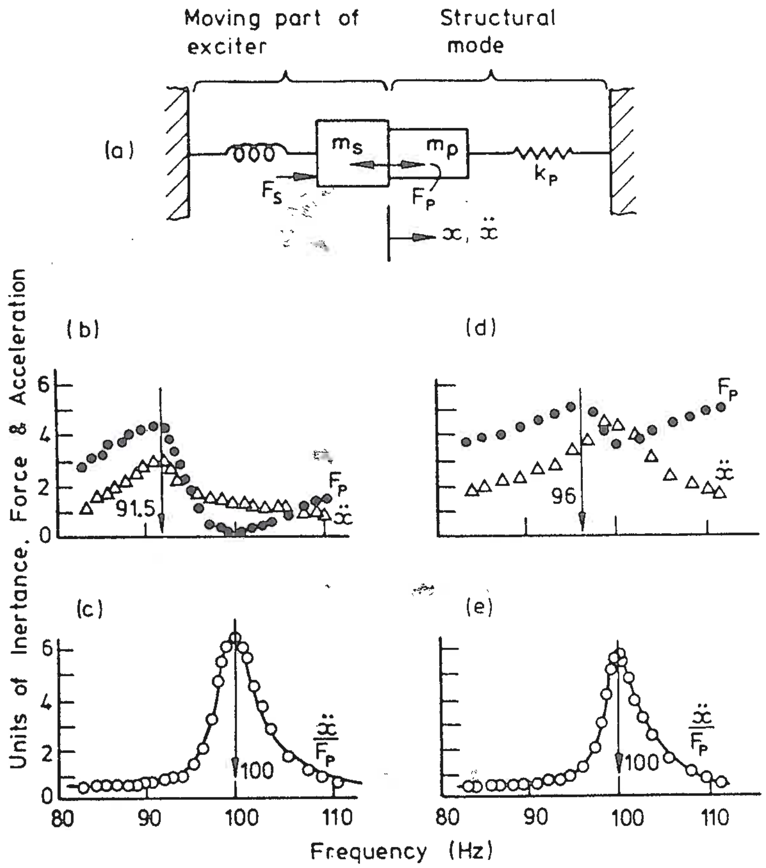


Fig 3.4 *Parameter Variations Around Resonance*
(a) Shaker/Structure Model
(b) Measured Data at Point 1
(c) Inertance Measured at Point 1
(d) Measured Data at Point 2
(e) Inertance Measured at Point 2

Suppose we are testing a plate and are trying to determine the properties of one of its modes. In one measurement, where the excitation and response are measured at the same point (a point mobility) and in the immediate vicinity of a natural frequency, the plate behaves very similarly to a single-degree-of-freedom oscillator with an apparent mass of m_{p1} and an apparent stiffness of k_{p1}. (Note that the natural frequency of this mode is given by $(k_{p1}/m_{p1})^{1/2}$.) Suppose also that the mass of the moving part of the shaker and its connection to the structure (which is not part of the structure proper) is m_s. Now, let the force generated in the shaker be F_s and the force actually applied to the structure (the one we want to measure) be F_p. If the acceleration of the structure is denoted by $\ddot{x}$, and we consider the vibration test to be conducted at various sinusoidal frequencies ω, then we may write the simple relationship:

$$F_p = F_s - m_s \ddot{x}$$

Taking some typical data, we show in Figure 3.4b the magnitudes of the various quantities which are, or which could be, measured. Also shown in Figure 3.4c is the curve for the mobility quantity of interest, in this case the inertance $\ddot{x}/F_p$, and it is particularly interesting to see how the true natural frequency (indicated when the inertance reaches a maximum) is considerably displaced from that suggested by the apparent resonance when the response alone reaches a maximum.

We now move to a different point on the structure which, for the same mode, will have different values for the apparent mass and stiffness, m_{p2} and k_{p2}, although these two quantities will necessarily stay in the same ratio (i.e. $k_{p1}/m_{p1} = k_{p2}/m_{p2} = \omega_0^2$). Another plot of the various quantities in this case is shown in Figure 3.4d and 3.4e from which it is clear that although the mobility parameter shows the natural frequency to be at the same value as before, the system resonance is now at a different frequency to that encountered in the first measurement simply because of the different balance between the structure's apparent properties (which vary from point to point) and those of the shaker (which remain the same throughout).

This example serves to illustrate the need for a direct measurement of the force applied to the structure as close to the surface as possible in order to obtain a reliable and accurate indication of the excitation level, and hence the mobility properties. It also illustrates a characteristic which gives rise to some difficulties in making such measurements: namely, that the (true) applied excitation force becomes very small in the vicinity of the resonant frequency with the consequence that it is particularly vulnerable to noise or distortion, see Figure 3.4b.

Generally, the larger the shaker, the greater the force which may be generated for exciting the structure. However, besides the obvious penalty of expense incurred by using too large an exciter, there is a limitation imposed on the working frequency range. The above discussion, which shows how the force generated in the exciter itself finds its way out to the structure, applies only as long as the moving parts of the shaker remain a rigid mass. Once the frequency of vibration approaches and passes the first natural frequency of the shaker coil and drive platform then there is a severe attenuation of the force which is available for driving the test object and although some excitation is possible above this critical frequency, it does impose a natural limit on the useful working range of the device. Not surprisingly, this frequency is lower for the larger shakers. Figure 3.3 shows, approximately, the relationship between maximum force level and upper frequency limit for a typical range of shakers of this type.

3.3.4 Electrohydraulic Exciters

The next type of exciter to be considered is the hydraulic (or electrohydraulic, to be precise). In this device, the power amplification to generate substantial forces is achieved through the use of hydraulics and although more costly and complex than their electromagnetic counterparts, these exciters do have one potentially significant advantage. That is their ability to apply simultaneously a static load as well as the dynamic vibratory load and this can be extremely useful when testing structures or materials whose normal vibration environment is combined with a major static load which may well change its dynamic properties or even its geometry. Without the facility of applying both static and dynamic loads simultaneously, it is necessary to make elaborate arrangements to provide the necessary static forces and so in these cases hydraulic shakers have a distinct advantage.

Another advantage which they may afford is the possibility of providing a relatively long stroke, thereby permitting the excitation of structures at large amplitudes - a facility not available on the comparably-sized electromagnetic shakers. On the other hand, hydraulic exciters tend to be limited in operational frequency range and only very specialised ones permit measurements in the range above 1 kHz, whereas electromagnetic exciters can operate well into the 30-50 kHz region, depending on their size. Also, as mentioned earlier, hydraulic shakers are more complex and expensive, although they are generally compact and lightweight compared with electromagnetic devices.

The comments made above concerning the need to measure force at the point of application to the structure also apply to this type of exciter, although the relative magnitudes of the various parameters involved will probably be quite different.

3.3.5 Attachment to the structure

For the above excitation devices, it is necessary to connect the driving platform of the shaker to the structure, usually incorporating a force transducer. There are one or two precautions which must be taken at this stage in order to avoid the introduction of unwanted excitations or the inadvertent modification of the structure. The first of these is perhaps the most important because it is the least visible. If we return

to our definition of a single mobility or frequency response parameter, Y_{jk}, we note that this is the ratio between the harmonic response at point or coordinate j caused by a single harmonic force applied in coordinate k. There is also a stipulation in the definition that this single force must be the only excitation of the structure and it is this condition that we must be at pains to satisfy in our test. Although it may seem that the exciter is capable of applying a force in one direction only - it is essentially a uni-directional device - there exists a problem on most practical structures whose motion is generally complex and multidirectional. The problem is that when pushed in one direction - say, along the x axis - the structure responds not only in that same direction but also in others, such as along the y and z axes and also in the three rotation directions. Such motion is perfectly in order and expected but it is possible that it can give rise to a secondary form of excitation if the shaker is incorrectly attached to the structure. It is usual for the moving part of the shaker to be very mobile along the axis of its drive but for it to be quite the reverse (i.e. very stiff) in the other directions. Thus, if the structure wishes to respond in, say, a lateral direction as well as in the line of action of the exciter, then the stiffness of the exciter will cause resisting forces or moments to be generated which are, in effect, exerted on the structure in the form of a secondary excitation. The response transducers know nothing of this and they pick up the total response which is that caused not only by the driving force (which is known) but also by the secondary and unknown forces.

The solution is to attach the shaker to the structure through a drive rod or similar connector which has the characteristic of being stiff in one direction (that of the intended excitation) while at the same time being relatively flexible in the other five directions. One such device is illustrated in Figure 3.5a. Care must be taken not to over-compensate:

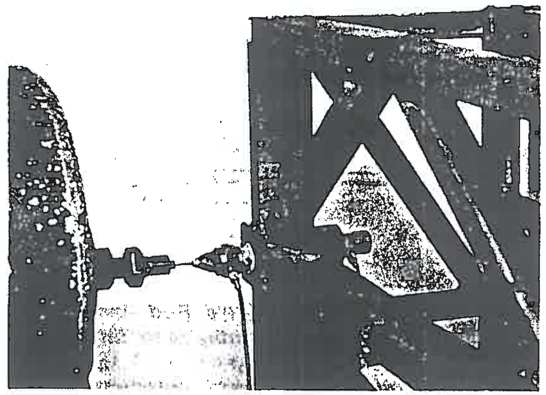


Fig 3.5(a) Exciter Attachment and Drive Rod Assemblies - Practical Assembly

If the drive rod or 'stinger' is made too long, or too flexible, then it begins to introduce the effects of its own resonances into the measurements and these can be very difficult to extricate from the genuine data. For most general structures, an exposed length of some 5-10 mm of 1 mm dia. wire is found to be satisfactory, although by experience rather than by detailed analysis. Various alternative arrangements are sometimes found, as illustrated in Figure 3.5b,c,d and e. Of these, b is unsatisfactory while c and d are acceptable, if not ideal, configurations. It is always necessary to check for the existence of an internal resonance of the drive rod - either axially or in flexure - as this can introduce spurious effects on the measured mobility properties. Futhermore, in the case of an axial resonance, it will be found that very little excitation force will be delivered to the test structure at frequencies above the first axial mode. (This should also be noted as it applies to cases where a non-flexible extension rod is used to overcome problems of access, Figure 3.5e).

Another consideration which concerns the shaker is the question of how it should be supported, or mounted, in relation to the test structure. Of the many possibilities, some of which are illustrated in Figure 3.6, two are generally acceptable while others range from 'possible-with-care' to unsatisfactory. The setup shown in Figure 3.6a presents the most satisfactory arrangement in which the shaker is fixed to ground while the test structure is supported by a soft suspension. Figure 3.6b shows an alternative configuration in which the shaker itself is resiliently supported.

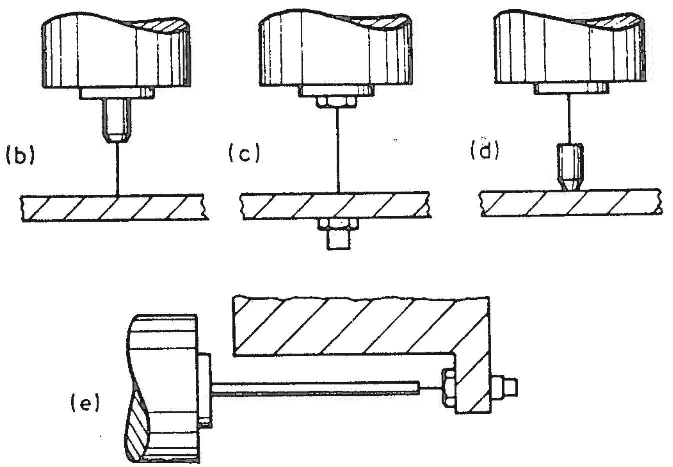


Fig 3.5 *Exciter Attachment and Drive Rod Assembly*
(b) Unsatisfactory Assembly with Impedance Head
(c) Acceptable Assembly
(d) Acceptable Assembly
(e) Use of Extension Rod

In this arrangement, the structure can be grounded or ungrounded, but it may be necessary to add an additional inertia mass to the shaker in order to generate sufficient excitation forces at low frequencies. The particular problem which arises here is that the reaction force causes a movement of the shaker body which, at low frequencies, can be of large displacement. This, in turn, causes a reduction in the force generation by the shaker so that its effectiveness at driving the test structure is diminished.

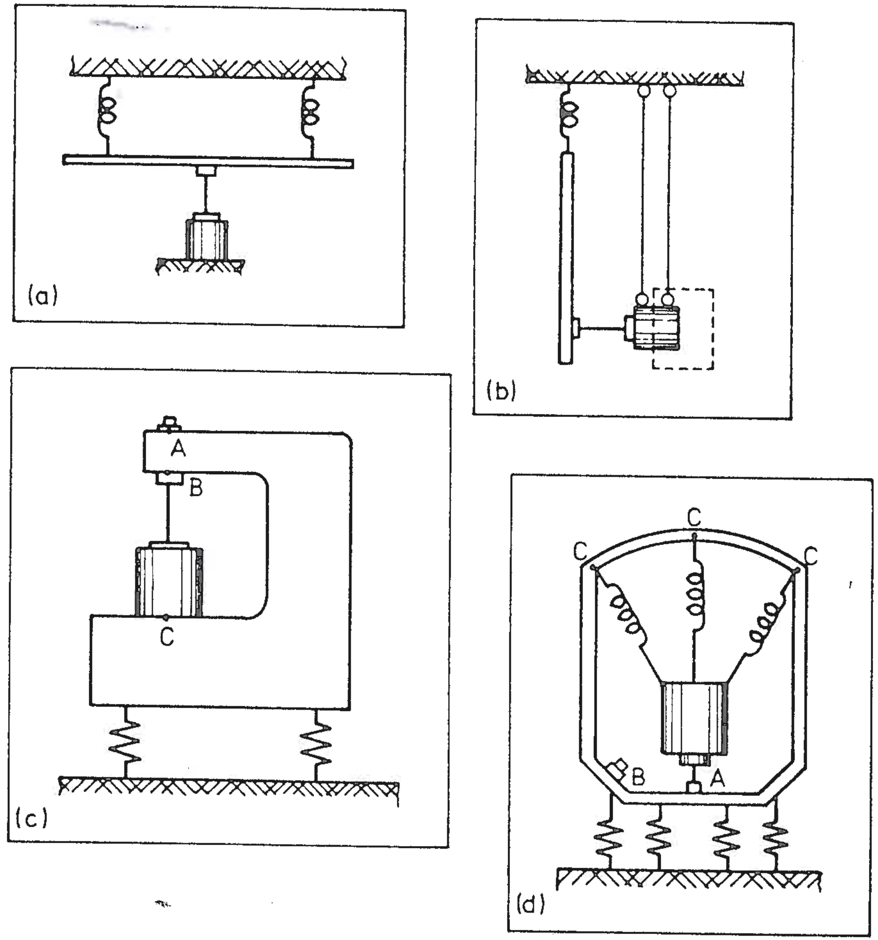


Fig 3.6 *Various Mounting Arrangments for Exciter*
(a) Ideal Configuration
(b) Suspended Exciter Plus Inertia Mass
(c) Unsatisfactory Configuration
(d) Compromise Configuration

In both cases 3.6a and 3.6b above, we have sought to ensure that the reaction force imposed on the shaker (equal and opposite to that applied to the drive rod) is not transmitted to the test structure. Figure 3.6c shows a set-up which does not meet that requirement with the result that an invalid mobility measurement would be obtained because the response measured at A would not be due solely to the force applied at B (which has been measured), but would, in part, be caused by the (unmeasured) force applied at C.

The final example, Figure 3.6d, shows a compromise which is sometimes necessary for practical reasons. In this case, it is essential to check that the measured response at A is caused primarily by the directly applied force at B and that it is not significantly influenced by the transmission of the reaction on the shaker through its suspension at C. This is achieved by ensuring that the frequency range for the measurements is well above the suspension resonance of the shaker: then, the reaction forces will be effectively attenuated by normal vibration isolation principles.

3.3.6 Hammer or Impactor Excitation

Another popular method of excitation is through use of an impactor or hammer. Although this type of test places greater demands on the analysis phase of the measurement process, it is a relatively simple means of exciting the structure into vibration. The equipment consists of no more than an impactor, usually with a set of different tips and heads which serve to extend the frequency and force level ranges for testing a variety of different structures. The useful range may also be extended by using different sizes of impactor. Integral with the impactor there is usually a load cell, or force transducer, which detects the magnitude of the force felt by the impactor, and which is assumed to be equal and opposite to that experienced by the structure. When applied by hand, the impactor incorporates a handle – to form a hammer (Figure 3.7a). Otherwise, it can be applied with a suspension arrangement, such as is shown in Figure 3.7b.

Basically, the magnitude of the impact is determined by the mass of the hammer head and the velocity with which it is moving when it hits the structure. Often, the operator will control the velocity rather than the force level itself, and so an appropriate way of adjusting the order of the force level is by varying the mass of the hammer head.

The frequency range which is effectively excited by this type of device is controlled by the stiffness of the contacting surfaces and the mass of the impactor head: there is a system resonance at a frequency given by (contact stiffness/impactor mass)$^{1/2}$ above which it is difficult to deliver energy into the test structure. When the hammer tip impacts the test structure, this will experience a force pulse which is substantially that of a half-sine shape, as shown in Figure 3.8a. A pulse of this type can be shown to have a frequency content of the form illustrated in Figure 3.8b which is essentially flat up to a certain frequency (f_c) and then of diminished and uncertain strength thereafter. Clearly, a pulse of this

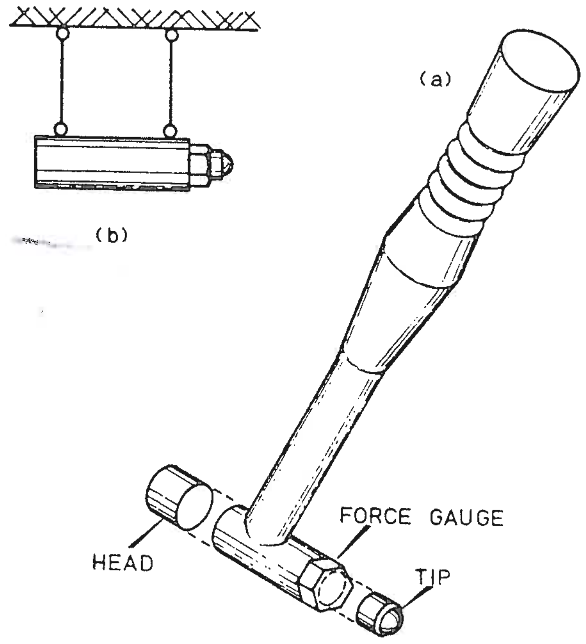


Fig 3.7 Impactor and Hammer Details

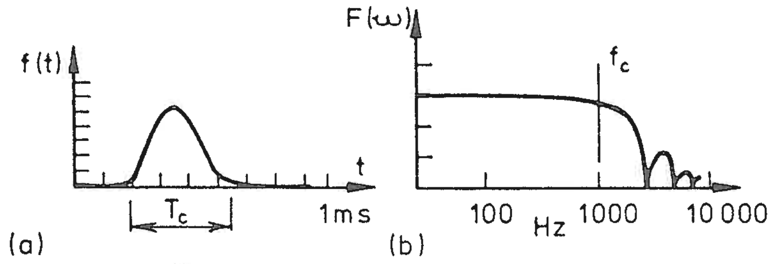


Fig 3.8 Typical Impact Force Pulse and Spectrum
(a) Time History
(b) Frequency Spectrum

type would be relatively ineffective at exciting vibrations in the frequency range above f_c and so we need to have some control over this parameter. It can be shown that there is a direct relationship between the first cut-off frequency f_c and the duration of the pulse, T_c, and that in order to raise the frequency range it is necessary to induce a shorter pulse length. This in turn, can be seen to be related to the stiffness (not the hardness) of the contacting surfaces and the mass of the impactor head. The stiffer the materials, the shorter will be the duration of the pulse and the higher will be the frequency range covered by the impact. Similarly, the lighter the impactor mass the higher the effective frequency range. It is for this purpose that a set of different hammer tips and heads are used to permit the regulation of the frequency range to be encompassed. Generally, as soft a tip as possible will be used in order to inject all the input energy into the frequency range of interest: using a stiffer tip than necessary will result in energy being input to vibrations outside the range of interest at the expense of those inside that range.

On a different aspect, one of the difficulties of applying excitation using a hammer is ensuring that each impact is essentially the same as the previous ones, not so much in magnitude (as that is accommodated in the force and response measurement process) as in position and orientation relative to the normal to the surface. At the same time, multiple impacts or 'hammer bounce' must be avoided as these create difficulties in the signal processing stage.

Yet another problem to be considered when using the hammer type of excitation derives from the essentially transient nature of the vibrations under which the measurements are being made. We shall return to this characteristic later but here it is appropriate to mention the possibility of inflicting an overload during the excitation pulse, forcing the structure outside its elastic or linear range.

3.4 TRANSDUCERS AND AMPLIFIERS

3.4.1 General

The piezoelectric type of transducer is by far the most popular and widely-used means of measuring the parameters of interest in modal tests. Only in special circumstances are alternative types used and thus we shall confine our discussion of transducers to these piezoelectric devices.

Three types of piezoelectric transducer are available for mobility measurements - force gauges, accelerometers and impedance heads (although these last are simply a combination of force- and acceleration-sensitive elements in a single unit). The basic principle of operation makes use of the fact that an element of piezoelectric material (either a natural or synthetic crystal) generates an electrical charge across its end faces when subjected to a mechanical stress. By suitable design, such a crystal may be incorporated into a device which induces in it a stress proportional to the physical quantity to be measured (i.e. force or acceleration).

3.4.2 Force Transducers

The force transducer is the simplest type of piezoelectric transducer. The transmitted force F (see Figure 3.9), or a known fraction of it, is applied directly across the crystal which thus generates a corresponding charge, q, proportional to F. It is usual for the sensitive crystals to be used in pairs, arranged so that the negative sides of both are attached to the case, and the positive sides are in mutual contact at their interface. This arrangement obviates the need to insulate one end of the case from the other electrically. One important feature in the design of force gauges is the relative stiffness (in the axial direction) of the crystals and of the case. The fraction of F which is transmitted through the crystals depends directly upon this ratio. In addition, there exists the undesirable possibility of a cross sensitivity - i.e. an electrical output when there is zero force F but, say, a transverse or shear loading - and this is also influenced by the casing.

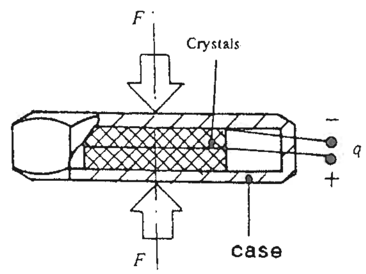


Fig 3.9 Force Transducer

The force indicated by the charge output of the crystals will always be slightly different from the force applied by the shaker, and also from that transmitted to the structure. This is because a fraction of the force detected by the crystals will be used to move the small amount of material between the crystals and the structure. The implications of this effect are discussed later in a section on mass cancellation (Section 3.9), but suffice it to say here that for each force gauge, one end will have a smaller mass than the other, and it is this (lighter) end which should be connected to the structure under test.

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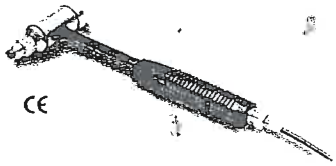
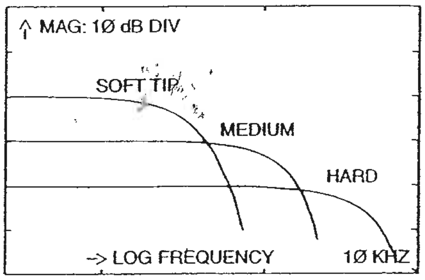
Modally Tuned ICP® Impact Hammers

Model 086C05 - tests medium to heavy structures such as machine tools, light trucks, at low to medium frequencies.

- 5 kHz frequency range
- 5000 lb amplitude range
- 1 mV/lb sensitivity
- 1 lb hammer mass
- 1 inch head diameter

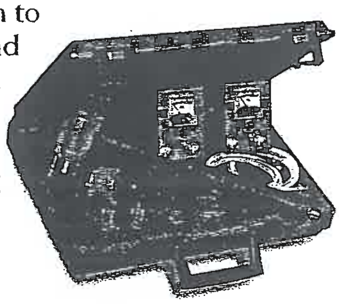


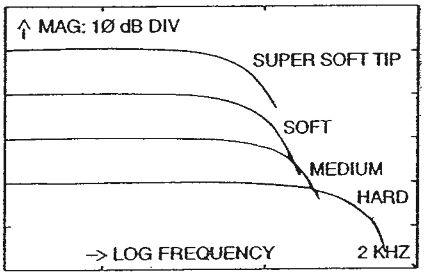


Model 086C05
(shown with cable attached)

Model 086C20 - small sledge, tests medium to heavy structures such as tool foundations and storage tanks at low to medium frequencies.

- 1 kHz frequency range
- 5000 lb amplitude range
- 1 mV/lb sensitivity
- 3 lb hammer mass
- 2 inch head diameter

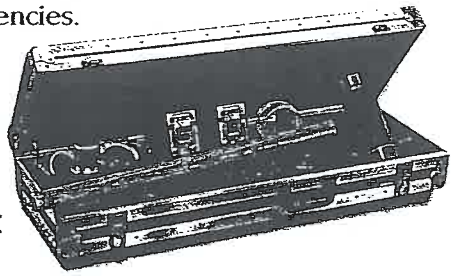


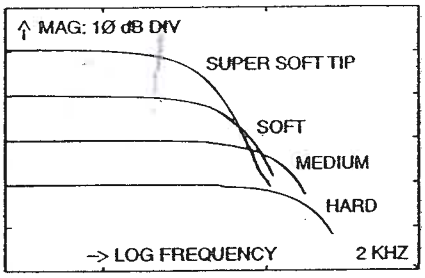


Model 086C20
(shown in Model GK291D20 kit)

Model 086C50 - large sledge, tests very heavy structures such as buildings, locomotives, ships, and foundations at low to very low frequencies.

- 500 Hz frequency range
- 5000 lb amplitude range
- 1 mV/lb sensitivity
- 12 lb hammer mass
- 3 inch head diameter

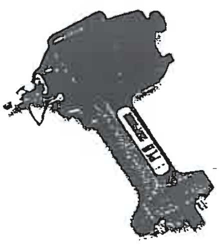


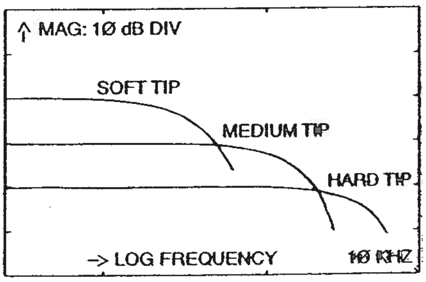


Model 086C50
(shown in Model GK291D50 kit)

Model 086C09 - electric solenoid actuated, for general purpose use, when controlled, repeatable impulse force is required such as with production testing.

- 8 kHz frequency range
- 1000 lb amplitude range
- 10 mV/lb sensitivity
- 0.6 inch head diameter
- local and remote trigger





Model 086C09

Mini-Shaker

type 4810

FEATURES:

- Force rating 10 Newton (2,25 lbf) Sine Peak
- Frequency range DC to 18 kHz
- First axial resonance above 18 kHz
- Max. bare table acceleration 550 ms^{-2} (56 g)
- Rugged construction

USES:

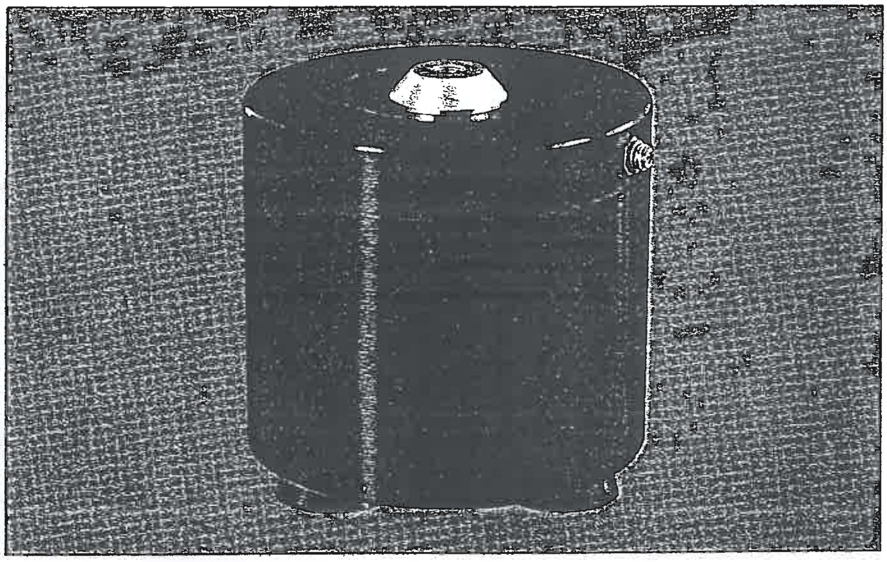
- Calibration of accelerometers
- Vibration testing of small objects
- Educational demonstrations
- Mechanical impedance measurements

The Mini-Shaker Type 4810 is a small machine for the dynamic excitation of lighter objects, it is manufactured from quality materials to a high degree of precision and has proved to be a reliable and versatile tool in dynamic testing.

Type 4810 is well suited as the motive force generator in mechanical impedance measurements where only smaller forces are required. It can also be used in the calibration of vibration transducers, both to determine their sensitivity by comparison with a standard accelerometer, and to determine their frequency response up to 18 kHz.

The Mini-Shaker is of the electrodynamic type with a permanent field magnet. A coil, which is an integral part of the table structure, is flexibly suspended in one plane in the field of the permanent magnet. An alternating current signal, provided by an external oscillator is passed through the coil to produce a vibratory motion at the table. A sectional drawing illustrating the method of construction is shown in Fig. 1.

The suspension system consists of radial flexure springs which restrict the moving element to almost perfectly rectilinear motion. Laminated flexure springs provide a high degree of damping to minimize distortion due to flexure resonances. The frequency response curves shown in Fig.2 show the highly damped flexure resonance around 50 to 60 Hz.



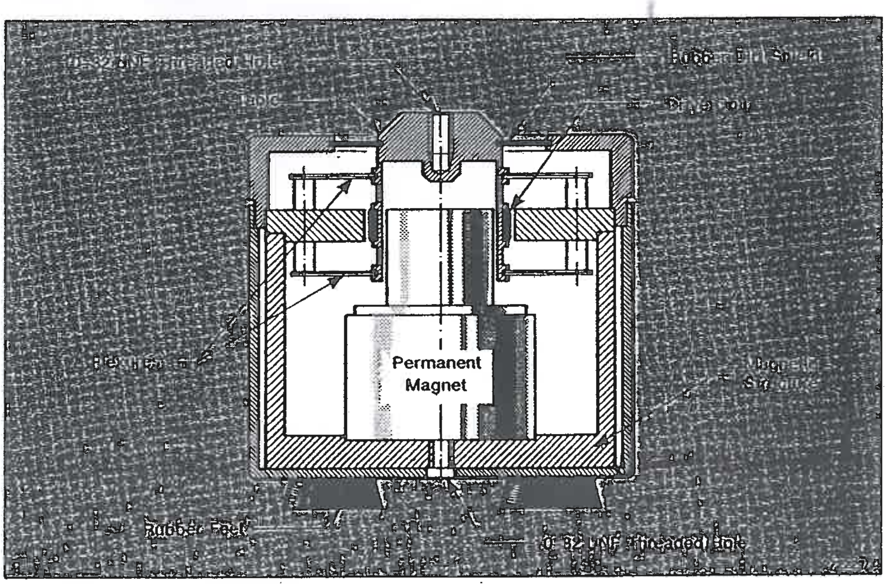


Fig. 1. Sectional drawing of the Mini-Shaker Type 4810

The object to be vibrated is attached to the table by means of a 10 – 32 UNF screw; the thread size commonly used for mounting accelerometers. Performance limits which are defined by the maximum displacement (6 mm), maximum force (10 N or 7 N depending on frequency), and the first axial resonance of the moving element (above 18 kHz), are graphically shown in Fig.3.

Within these limits, the attainable acceleration can be determined by the expression.

$$a = \frac{F}{W}$$

where a = acceleration in ms^{-2} ($1\,ms^{-2} = 0{,}102\,g$)
F = shaker rated force in Newtons
W = exciter moving element weight + test object weight in kg

Examples of maximum test object weight for accelerations of 20 g and 5 g are drawn in on the curve.

In order to attain full rated output force from the 4810 it should be driven by Power Amplifier Type 2706. This is a power amplifier specially designed to drive small vibration exciters and has a current limiter to prevent overdriving the 4810.

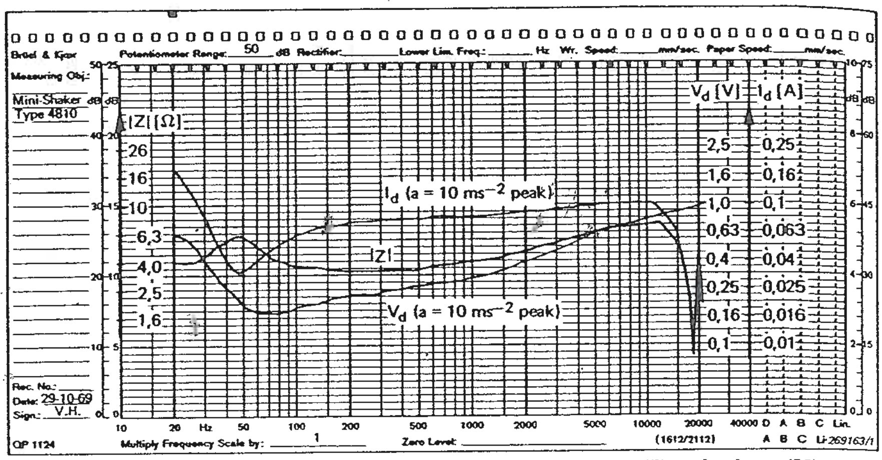


Fig. 2. Frequency response of the 4810 for Impedance (z), current (I) and voltage (V)

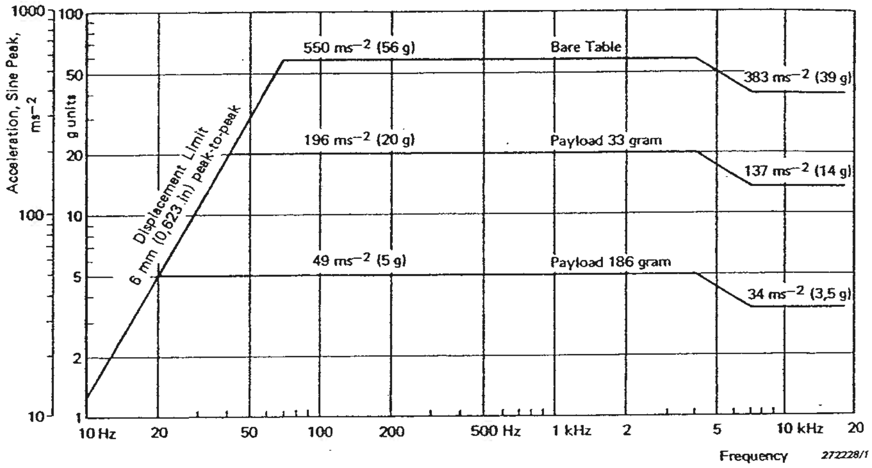


Fig. 3. Sine performance curves for the 4810

Specifications 4810

FREQUENCY RANGE
[illegible]

FIRST MAJOR ARMATURE RESONANCE
[illegible]

FORCE RATING (PEAK)
[illegible]

MAX. BARE TABLE ACCELERATION (PEAK)
[illegible]

MAX. DISPLACEMENT PEAK-TO-PEAK
[illegible]

DYNAMIC FLEXURE STIFFNESS
[illegible]

DYNAMIC WEIGHT OF THE MOVING SYSTEM
[illegible]

MAGNET FIELD
Permanent magnet

MAX. INPUT CURRENT
[illegible]

COIL IMPEDANCE
[illegible]

CONNECTIONS
[illegible]

TABLE SIZE
[illegible]

FASTENING THREAD
[illegible]

WEIGHT
[illegible]

DIMENSIONS
Diameter: [illegible]
Height: [illegible]

ACCESSORIES INCLUDED
[illegible]

ACCESSORIES AVAILABLE
[illegible]