CHAPTER 25
VIBRATION TESTING MACHINES

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INTRODUCTION

This chapter describes some of the more common types of vibration testing machines which are used for developmental, simulation, production, or exploratory vibration tests for the purpose of studying the effects of vibration or of evaluating physical properties of materials or structures. A summary of the prominent features of each machine is given. These features should be kept in mind when selecting a vibration testing machine for a specific application. Digital control systems for vibration testing are described in Chap. 27. Applications of vibration testing machines are described in other chapters.

A vibration testing machine (sometimes called a *shake table* or *shaker* and referred to here as a *vibration machine*) is distinguished from a vibration exciter in that it is complete with a mounting table which includes provisions for bolting the test article directly to it. A *vibration exciter,* also called a *vibration generator,* may be part of a vibration machine or it may be a device suitable for transmitting a vibratory force to a structure. A *constant-displacement* vibration machine attempts to maintain constant-displacement amplitude while the frequency is varied. Similarly, a *constant-acceleration* vibration machine attempts to maintain a constant-acceleration amplitude as the frequency is changed.

The *load* of a vibration machine includes the item under test and the supporting structures that are not normally a part of the vibration machine. In the case of equipment mounted on a vibration table, the load is the material supported by the table. In the case of objects separately supported, the load includes the test item and all fixtures partaking of the vibration. The load is frequently expressed as the weight of the material. The *test load* refers specifically to the item under test exclusive of supporting fixtures. A *dead-weight load* is a rigid load with rigid attachments. For nonrigid loads the reaction of the load on the vibration machine is a function of frequency. The vector force exerted by the load, per unit of acceleration amplitude expressed in units of gravity of the driven point at any given frequency, is the *effective load* for that frequency. The term *load capacity,* which is descriptive of the performance of reaction and direct-drive types of mechanical vibration machines, is the maximum dead-weight load that can be vibrated at the maximum acceleration rating of the vibration machine. The *load couple* for a dead-weight load is equal to the product of the force exerted on the load and the distance of the center-of-mass from the line-of-action of the force or from some arbitrarily selected location (such as a table surface). The static and dynamic load couples are generally different for nonrigid loads.

The term *force capacity,* which is descriptive of the performance of electrodynamic shakers, is defined as the maximum rated force generated by the machine. This force is usually specified, for continuous rating, as the maximum vector amplitude of a sinusoid that can be generated throughout a usable frequency range. A corresponding maximum rated acceleration, in units of gravity, can be calculated as the quotient of the force capacity divided by the total weight of the coil table assembly and the attached dead-weight loads. The *effective force* exerted by the load is equal to the effective load multiplied by the (dimensionless) ratio g, which represents the number of units of gravity acceleration of the driven point [see Eq. (25.1)].

DIRECT-DRIVE MECHANICAL VIBRATION MACHINES

The direct-drive vibration machine consists of a rotating eccentric or cam driving a positive linkage connection which forces a displacement between the base and table of the machine. Except for the bearing clearances and strain in the load-carrying members, the machine tends to develop a displacement between the base and the table which is independent of the forces exerted by the load against the table. If the base is held in a fixed position, the table tends to generate a vibratory displacement of constant amplitude, independent of the operating rpm. Figure 25.1 shows the direct-drive mechanical machine in its simplest forms. This type of machine is sometimes referred to as a *brute force machine* since it will develop any force necessary to produce the table motion corresponding to the crank or cam offset, short of breaking the load-carrying members or stalling the driving shaft.

The simplest direct-drive mechanical vibration machine is driven by a constant-speed motor in conjunction with a belt-driven speed changer and a frequency-indicating tachometer. Table displacement is set during shutoff and is assumed to hold during operation. An auxiliary motor driving a cam may be included to provide frequency cycling between adjustable limits. More elaborate systems employ

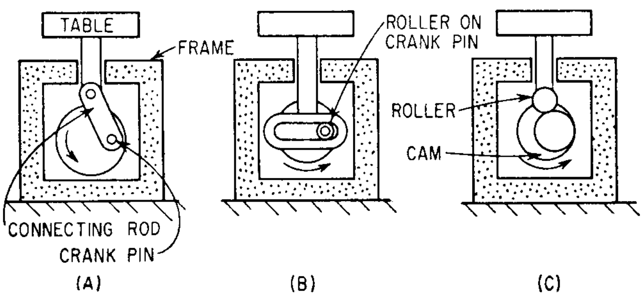

FIGURE 25.1 Elementary direct-drive mechanical vibration machines: (*A*) Eccentric and connecting link. (*B*) Scotch yoke. (*C*) Cam and follower.

a direct-coupled variable-speed motor with electronic speed control, as well as amplitude adjustment from a control station. Machines have been developed which provide rectilinear, circular, and three-dimensional table movements—the latter giving complete, independent adjustment of magnitude and phase in the three directions.

Many types of mechanisms are used to adjust the displacement amplitude and frequency of the mounting table. For example, the displacement amplitude can be adjusted by means of eccentric cams and cylinders.

PROMINENT FEATURES

- Low operating frequencies and large displacements can be provided conveniently.
- Theoretically, the machine maintains constant displacement regardless of the mechanical impedance of the table-mounted test item within force and frequency limits of the machine. However, in practice, the departure from this theoretical ideal is considerable, due to the elastic deformation of the load-carrying members with change in output force. The output force changes in proportion to the square of the operating frequency and in proportion to the increased displacement resulting therefrom. Because the load-carrying members cannot be made infinitely stiff, the machines do not hold constant displacement with increasing frequency with a bare table. This characteristic is further emphasized with heavy table mass loads. Accordingly, some of the larger-capacity machines which operate up to 60 Hz include automatic adjustment of the crank offset as a function of operating frequency in order to hold displacement more nearly constant throughout the full operating range of frequency.
- The machine must be designed to provide a stiff connection between the ground or floor support and the table. If accelerations greater than 1g are contemplated, the vibratory forces generated between the table and ground will be greater than the weight of the test item. Hence, all mass loads within the rating of the machine can be directly attached to the table without recourse to external supports.
- The allowable range of operating frequencies is small in order to remain within bearing load ratings. Therefore, the direct-drive mechanical vibration machine can be designed to have all mechanical resonances removed from the operating frequency range. In addition, relatively heavy tables can be used in comparison to the weight of the test item. Consequently, misplacing the center-of-gravity of the test item relative to the table center for vibration normal to the table surface and the generation of moments by the test item (due to internal resonances) usually have less influence on the table motions for this type of machine than would other types which are designed for wide operational frequency bands.
- Simultaneous rectilinear motion normal to the table surface and parallel to the table surface in two principal directions is practical to achieve. It may be obtained with complete independent control of magnitude and phase in each of the three directions.
- Displacement of the table is generated directly by a positive drive rather than by a generated force acting on the mechanical impedance of the table and load. Consequently, impact loads in the bearings, due to the necessary presence of some bearing clearance, result in the generation of relatively high impact forces which are rich in harmonics. Accordingly, although the waveform of displacement might be tolerated as such, the waveform of acceleration is normally sufficiently dis-

torted to preclude recognition of the fundamental driven frequency, when displayed on a time base.

REACTION-TYPE MECHANICAL VIBRATION MACHINE

A vibration machine using a rotating shaft carrying a mass whose center-of-mass is displaced from the center-of-rotation of the shaft for the generation of vibration, is called a *reaction-type vibration machine.* The product of the mass and the distance of its center from the axis of rotation is referred to as the *mass unbalance,* the *rotating unbalance,* or simply the *unbalance.* The force resulting from the rotation of this unbalance is referred to as the *unbalance force.*

The *reaction-type vibration machine* consists of at least one rotating-mass unbalance directly attached to the vibrating table. The table and rotating unbalance are suspended from a base or frame by soft springs which isolate most of the vibration forces from the supporting base and floor. The rotating unbalance generates an oscillating force which drives the table. The unbalance consists of a weight on an arm which is relatively long by comparison to the desired table displacement. The unbalance force is transmitted through bearings directly to the table mass, causing a vibratory motion without reaction of the force against the base. A vibration machine employing this principle is referred to as a reaction machine since the reaction to the unbalance force is supplied by the table itself rather than through a connection to the floor or ground.

CIRCULAR-MOTION MACHINE

The reaction-type machine, in its simplest form, uses a single rotating-mass unbalance which produces a force directed along the line connecting the center-of-rotation and the center-of-mass of the displaced mass. Referred to stationary coordinates, this force appears normal to the axis of rotation of the driven shaft, rotating about this axis at the rotational speed of the shaft. The transmission of this force to the vibration-machine table causes the table to execute a circular motion in a plane normal to the axis of the rotating shaft.

Figure 25.2 shows, schematically, a machine employing a single unbalance producing circular motion in the plane of the vibration-table surface. The unbalance is driven at various rotational speeds, causing the table and test item to execute circular motion at various frequencies. The counterbalance weight is adjusted to equal the test item mass moment calculated from *d,* the plane of the unbalance force, thereby keeping the combined center-of-gravity coincident with the generated force. Keeping the generated force acting through the combined center-of-gravity of the spring-mounted assembly eliminates vibratory moments which, in turn, would generate unwanted rotary motions in addition to the motion parallel to the test mounting surface. The vibration isolator supports the vibrating parts with minimum transmission of the vibration to the supporting floor.

For a fixed amount of unbalance and for the case of the table and test item acting as a rigid mass, the displacement of motion tends to remain constant if there are no resonances in or near the operating frequency range. If balance force must remain constant, requiring the amount of unbalance to change with shaft speed.

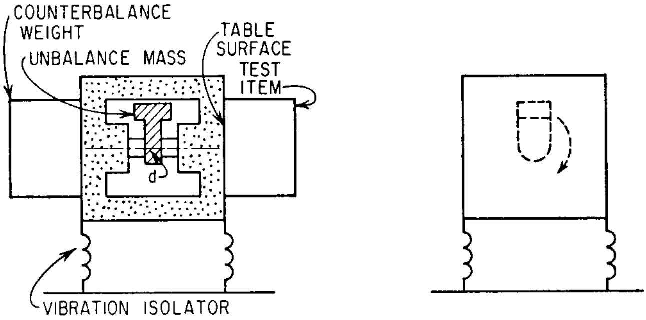

FIGURE 25.2 Circular-motion reaction-type mechanical vibration machine.

RECTILINEAR-MOTION MACHINE

Rectilinear motion rather than circular motion can be generated by means of a reciprocating mass. Rectilinear motions can be produced with a single rotating unbalance by constraining the table to move in one direction.

Two Rotating Unbalances. The most common rectilinear reaction-type vibration machine consists of two rotating unbalances, turning in opposite directions and phased so that the unbalance forces add in the desired direction and cancel in other directions. Figure 25.3 shows schematically how rectilinear motion perpendicular and parallel to the vibration table is generated. The effective generated force from the two rotating unbalances is midway between the two axes of rotation and is normal to a line connecting the two. In the case of motion perpendicular to the surface of the table, simply locating the center-of-gravity of the test item over the center of the table gives a proper load orientation. Tables are designed so that the resultant force always passes through this point. This results in collinear-

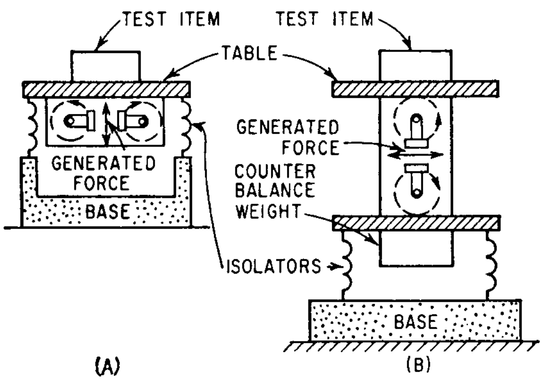

FIGURE 25.3 Rectilinear-motion reaction-type mechanical vibration machine using two rotating unbalances: (*A*) Vibration perpendicular to table surface. (*B*) Vibration parallel to table surface.

ity of generated forces and inertia forces, thereby avoiding the generation of moments which would otherwise rock the table. In the case of motion parallel to the table surface, no simple orientation of the test item will achieve collinearity of the generated force and inertia force of the table and test item. Various methods are used to make the generated force pass through the combined center-of-gravity of the table and test item.

Three Rotating Unbalances. If a machine is desired which can be adjusted to give vibratory motion either normal to the plane of the table or parallel to the plane of the table, a minimum of three rotating unbalances is required. Inspection of Fig. 25.4 shows how rotating the two smaller mass unbalances relative to the single larger unbalance results in the addition of forces in any desired direction, with cancellation of forces and force couples at 90° to this direction. Although parallel shafts are usually used as illustrated, occasionally the three unbalances may be mounted on collinear shafts, the two smaller unbalances being placed on either side of the single larger unbalance to conserve space and to eliminate the bending moments and shear forces imposed on the structure connecting the individual shafts.

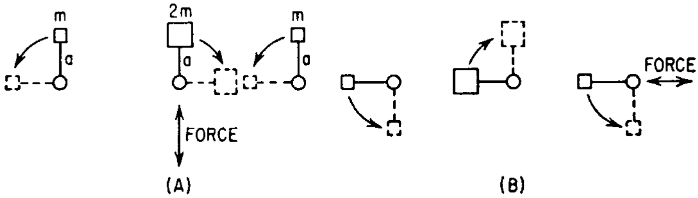

FIGURE 25.4 Adjustment of direction of generated force in a reaction-type mechanical vibration exciter: (*A*) Vertical force. (*B*) Horizontal force.

PROMINENT FEATURES

- The forces generated by the rotating unbalances are transmitted directly to the table without dependence upon a reactionary force against a heavy base or rigid ground connection.
- Because the length of the arm which supports the unbalance mass can be large, relative to reasonable bearing clearances and the generation of a force which does not reverse its direction relative to the rotating unbalance arm, the generated waveform of motion imparted to the vibration machine table is superior to that attainable in the direct-drive type of vibration machine.
- The generated vibratory force can be made to pass through the combined center-of-gravity of the table and test item in both the normal and parallel directions relative to the table surface, thereby minimizing vibratory moments giving rise to table rocking modes.
- The attainable rpm and load ratings on bearings currently limit performance to a frequency of approximately 60 Hz and a generated force of 300,000 lb (1.3 MN), respectively, although in special cases frequencies up to 120 Hz and higher can be obtained for smaller machines.

ELECTRODYNAMIC VIBRATION MACHINE

GENERAL DESCRIPTION

A complete electrodynamic vibration test system is comprised of an electrodynamic vibration machine, electrical power equipment which drives the vibration machine, and electrical controls and vibration monitoring equipment.

The electrodynamic vibration machine derives its name from the method of force generation. The force which causes motion of the table is produced electrodynamically by the interaction between a current flow in the armature coil and the intense magnetic dc field which passes through the coil, as illustrated in Fig. 25.5. The table is structurally attached to a force-generating coil which is concentrically located (with radial clearances) in the annular air gap of the dc magnet circuit. The assembly of the armature coil and the table is usually referred to as the *driver coil-table* or *armature.* The magnetic circuit is made from soft iron which also forms the *body* of the vibration machine. The body is magnetically energized, usually by two field coils as shown in Fig. 25.5*C,* generating a radially directed field in the air gap, which is perpendicular to the direction of current flow in the armature coil. Alternatively, in small shakers, the magnetic field is generated by permanent magnets. The generated force in the armature coil is in the direction of the axis of the coil, perpendicular to the table surface. The direction of the force is also perpendicular to the armature-current direction and to the air-gap field direction.

The table and armature coil assembly is supported by elastic means from the machine body, permitting rectilinear motion of the table perpendicular to its surface, corresponding in direction to the axis of the armature coil. Motion of the table in all other directions is resisted by stiff restraints. Table motion results when an ac current passes through the armature coil. The body of the machine is usually supported by a base with a trunnion shaft centerline passing horizontally through the center-of-gravity of the body assembly, permitting the body to be rotated about its center, thereby giving a vertical or horizontal orientation to the machine table. The base usually includes an elastic support of the body, providing vibration isolation between the body and the supporting floor.

Where a very small magnetic field is required at the vibration machine table due to the effect of the magnetic field on the item under test, *degaussing* may be pro-

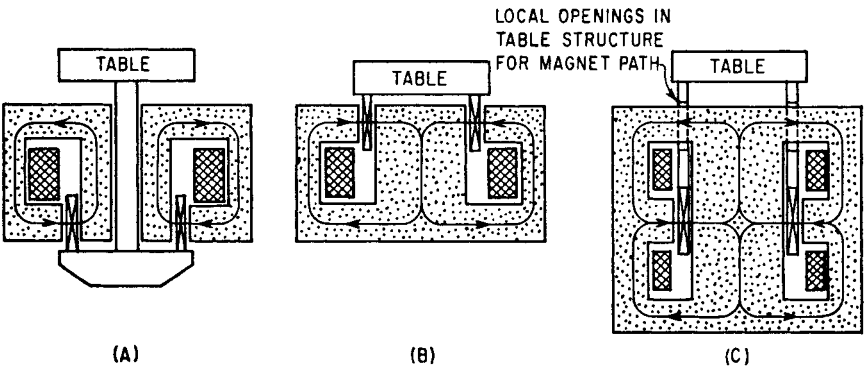

FIGURE 25.5 Three main magnet circuit configurations.

vided. Magnetic fields of 5 to 30 gauss several inches above the table are normal for modern machines with double-ended, center air-gap magnet designs, Fig. 25.5*C*, without degaussing accessories; in contrast, with degaussing accessories, magnetic fields of 2 to 5 gauss can be achieved.

Because of copper and iron losses in the electrodynamic unit, provision must be made to carry off the dissipated heat. Cooling by convection air currents, compressed air, or a motor-driven blower is used and, in some cases, a recirculating fluid is used in conjunction with a heat exchanger. Fluid cooling is particularly useful under extremes of hot or cold environments or altitude conditions where little air pressure is available.

MAGNET CIRCUIT CONFIGURATIONS

Three magnet circuit configurations that are used in the electrodynamic machines are shown schematically in Fig. 25.5. In Fig. 25.5*A*, the table and driver coil are located at opposite ends of the magnet circuit. The advantage of this configuration is that the location of the annular air gap, the region of high magnetic leakage flux, is spaced from the table and the body itself acts as a magnetic shield, resulting in lower magnetic flux density at the table. The disadvantage lies in the loss of rigidity in the connecting structure between the driver coil and the table because of its length. This configuration is usually cooled by convection air currents or by forced air from a motor-driven blower.

In Fig. 25.5*B*, the table is connected directly to the driver coil. This eliminates the length of structure passing through the magnet structure, thereby increasing the rigidity of the driver coil-table assembly and allowing higher operating frequencies. The leakage magnetic field in the vicinity of the table is high in this configuration. It is therefore difficult, if not impossible, to reduce the leakage to acceptable levels without adding extra length to the driver coil assembly, elevating the table above the air gap. The configuration in Fig. 25.5*C* has a complete magnet circuit above and below the annular air gap, thereby reducing the external leakage magnetic field to a minimum. This configuration also increases the total magnetic flux in the air gap by a factor of almost 2 for the same diameter driver coil, giving greater force generation and a more symmetrical magnetic flux density along the axis of the coil. Hence a more uniform force generation results when the driver coil is moved axially throughout its total stroke. All high-efficiency and high-performance electrodynamic vibration machines use the configuration shown in Fig. 25.5*C*. Configurations *B* and *C* of Fig. 25.5 may use air cooling throughout or an air-cooled driver coil and liquid-cooled field coil(s) or total liquid cooling.

The main magnetic circuit uses dc field coils for generating the high-intensity magnetic flux in the annular gap in all of the larger and most of the smaller units. Permanent magnet excitation is used in small portable units and in some general-purpose units up to about 500-lb (2-kN) generated force.

INDUCTION-TYPE SHAKER

In the induction-type electrodynamic shaker, a stator coil is fixed in the shaker body (see Fig. 25.6). The varying current from the power source is passed trough the stator coil. The armature coil is a cylinder of conductive material (usually aluminum). The stator current is coupled inductively to the armature coil. The stator coil (many turns) acts as the primary in a transformer. The armature coil (a single shorted turn)

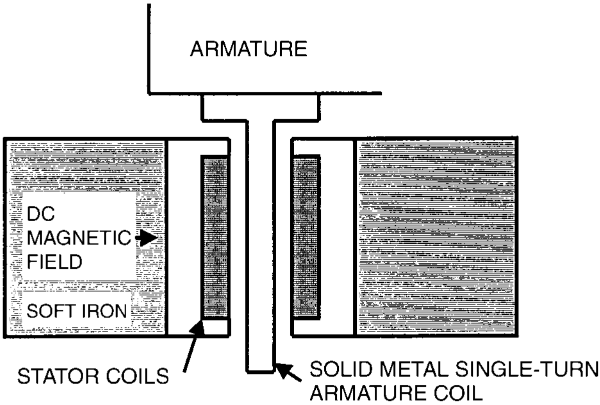

FIGURE 25.6 Cross section in the vicinity of the armature of an induction-type shaker.

acts as the secondary in the transformer. The stator current inductively generates a current in the single turn shorted armature coil. In Fig. 25.6, the dc magnetic field is across the paper, the armature current is into the paper, and the generated force is vertical. The advantages are a rugged armature design, and an armature that is electrically isolated from the rest of the shaker. The disadvantages include a decrease in performance at low frequencies due to inductive coupling losses and a slight problem cooling the armature. Because the induction losses are a function of scale, this design is usually found in the larger electrodynamic shakers.

FREQUENCY RESPONSE CONSIDERATIONS

Testing procedures which call for sinusoidal motion (see Chap. 20) of a vibration-machine table can be performed even though the frequency response curve of the electrodynamic vibration machine is far from flat. For a test at a fixed frequency, the driving voltage is adjusted until the table motion is equal in amplitude to that required by the test specifications. If the procedure calls for cycling the frequency between two frequency limits while keeping a constant displacement or acceleration, a control system or servo control adjusts the driver-coil voltage as required to maintain the desired vibration machine table motion independent of the frequency of operation. This control system provides a correction at any frequency of operation within the testing frequency limits, but it can correct for only one operating frequency at any instant of time. The closer the frequency response is to the desired variation in acceleration with frequency, the smaller the corrections in driver-coil voltage will be from the control system—thereby improving the attainable accuracy of the control.

Similarly for test procedures that call for a random vibration source, the autospectrum of the source must be adjusted, because of test requirements and the frequency response of the test system. A shaker with a more constant response will allow for a greater range of spectral values than can be controlled.

Test procedures can also call for the reproduction of a transient. This test method is called *waveform control* or *waveform reproduction.* For this test method, the frequency response function between the power amplifier input voltage and the control accelerometer is measured with the test item in place. This information is used to construct an input voltage time-history that will reproduce the desired test time-history at the control point.

In the past, analog control systems were used, but with the advent of relatively inexpensive computers, digital control is now almost exclusively used. Digital vibration control systems are discussed in Chap. 27.

CHARACTERIZATION OF AN ELECTRODYNAMIC SHAKER AS A TWO-PORT NETWORK

An electrodynamic shaker can be modeled as a mixed electrical/mechanical two-port network[1,2] (see Chap. 10). This characterization can give good insight about the performance capabilities of a shaker and/or a shaker/power supply combination. In matrix form, this characterization can be written as

$$\begin{Bmatrix} E \\ A \end{Bmatrix} = \begin{bmatrix} Z_{11} & Z_{12} \\ Z_{21} & Z_{22} \end{bmatrix} \begin{Bmatrix} I \\ F \end{Bmatrix} \tag{25.1}$$

where E = the voltage required to drive the shaker
I = the current required to drive the shaker
A = the acceleration observed at the shaker/load interface
F = the force at the shaker/load interface

All the variables are complex functions of frequency as described in Chap. 22. The terms in the impedance matrix are frequency response functions defined as

$$\begin{aligned} Z_{11} &= \frac{E}{I}\bigg|_{F=0} \quad Z_{12} = \frac{E}{F}\bigg|_{I=0} \\ Z_{21} &= \frac{A}{I}\bigg|_{F=0} \quad Z_{22} = \frac{A}{F}\bigg|_{I=0} \end{aligned} \tag{25.2}$$

Two of the terms are easily measured. Z_{11} is the unloaded table (no mechanical load on the shaker) electrical impedance of the shaker, and Z_{21} is the ratio of the unloaded acceleration to input current of the shaker. Z_{22} is the accelerance (ratio of acceleration to force) looking into the shaker with the shaker electrical input open (zero current, but with the field on). Z_{12} is the ratio of voltage, generated at the open electrical shaker input, to a driving force applied at the armature. The direct measurement of Z_{12} and Z_{22} would require that an external force be applied to the shaker and the resulting open circuit voltage and acceleration be measured, a difficult feat in practice. But the terms in the impedance matrix can be measured experimentally by performing experiments with two or more known loads attached to the shaker. The general case is given by a system of equations for n measured load conditions, where the subscripts indicate the different loading conditions.

$$\begin{bmatrix} E_1 & E_2 & \cdots & E_n \\ A_1 & A_2 & \cdots & A_n \end{bmatrix} = \begin{bmatrix} Z_{11} & Z_{12} \\ Z_{21} & Z_{22} \end{bmatrix} \begin{bmatrix} I_1 & I_2 & \cdots & I_n \\ F_1 & F_2 & \cdots & F_n \end{bmatrix} \tag{25.3}$$

Each test requires the measurement of the input voltage and current and the output acceleration and force. If the test item is a rigid mass, the force can be estimated from $F = ma$. In short hand, Eq. (25.3) will be written as

$$\mathbf{E} = \mathbf{ZI} \tag{25.4}$$

The impedance matrix can then be found using a Moore-Penrose pseudoinverse[3]

$$\mathbf{Z} = \mathbf{E}\mathbf{I}^{-1} \tag{25.5}$$

If the number of test conditions is greater than two, the solution is in a least-squares sense. This assumes the inverse exists. The equation is typically solved at a finite set of discrete frequencies using techniques described in Chap. 22. Other forms of the impedance matrix can be defined which give frequency response functions that may be more useful in a particular application. The admittance matrix is defined as

$$\begin{Bmatrix} I \\ F \end{Bmatrix} = \begin{bmatrix} Y_{11} & Y_{12} \\ Y_{21} & Y_{22} \end{bmatrix} \begin{Bmatrix} E \\ A \end{Bmatrix} \tag{25.6}$$

The transmission matrix is defined as

$$\begin{Bmatrix} E \\ I \end{Bmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{Bmatrix} A \\ F \end{Bmatrix} \tag{25.7}$$

The reciprocal transmission matrix is defined as

$$\begin{Bmatrix} A \\ F \end{Bmatrix} = \begin{bmatrix} R_{11} & R_{12} \\ R_{21} & R_{22} \end{bmatrix} \begin{Bmatrix} E \\ I \end{Bmatrix} \tag{25.8}$$

These matrices are all related by the equations

$$\begin{gathered} \mathbf{Y} = \mathbf{Z}^{-1} \qquad \mathbf{R} = \mathbf{T}^{-1} \\ \mathbf{T} = \frac{1}{Z_{21}} \begin{bmatrix} Z_{11} & Z_{12}Z_{21} - Z_{11}Z_{22} \\ 1 & -Z_{22} \end{bmatrix} \\ \mathbf{R} = \frac{1}{Z_{12}} \begin{bmatrix} Z_{22} & Z_{12}Z_{21} - Z_{11}Z_{22} \\ 1 & -Z_{11} \end{bmatrix} \end{gathered} \tag{25.9}$$

For example, for a sine test, the voltage and current required for a particular load acceleration are easily determined by substituting

$$F = Z_m A \tag{25.10}$$

into Eq. (25.7) to give

$$\begin{Bmatrix} E \\ I \end{Bmatrix} = A \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \begin{Bmatrix} 1 \\ Z_m \end{Bmatrix} \tag{25.11}$$

Z_m is the driving point (the interface at the shaker) free effective mass[4] (the ratio of force to acceleration) of the load (test item and fixtures). The free effective mass is related to the mechanical impedance, Z (the ratio of force to velocity), defined in Chap. 10, by the relationship, $Z_m = j\omega Z$. In general, Z_m is a frequency response function. If the load and fixtures are a rigid mass, Z_m is a constant equal to the mass of the test item and fixtures.

Similarly, for a given shaker power supply with known characteristics (the maximum output voltage and current capability), the shaker performance capabilities (the achievable acceleration) for a given load are easily determined from Eq.

(25.11). The maximum acceleration that can be achieved for a given voltage limit is

$$A_{E\text{lim}} = |E_{\text{lim}}/(T_{11} + T_{12}Z_m)|$$

The maximum acceleration that can be achieved for a given current limit is

$$A_{I\text{lim}} = |I_{\text{lim}}/(T_{21} + T_{22}Z_m)|$$

The maximum acceleration that can be reached before either limit is reached is the smaller of these two numbers.

$$A_{\text{max}} = \min(A_{E\text{lim}}, A_{I\text{lim}})$$

The development is easily generalized for random and transient testing using the techniques in Chap. 22. The development can be generalized for the multiple shaker system driving a single test item.[5]

A useful review of electrodynamic shakers is given in Ref. 6.

SYSTEM RATINGS

The electrodynamic vibration machine system is rated: (1) in terms of the *peak value* of the sinusoidal generated force for *sinusoidal* vibration testing and (2) in terms of the *rms* and *instantaneous* values of the maximum force generated under *random* vibration testing. In order to determine the acceleration rating of the system with a test load on the vibration table, the weight of the test load, assumed to be effective at all frequencies, must be known and used in the following expressions:

$$\mathfrak{g} = \frac{F}{W_L + W_T} \qquad \mathfrak{g}_{\text{rms}} = \frac{F_{\text{rms}}}{W_L + W_T} \tag{25.12}$$

where $\mathfrak{g} = a/g$, a dimensionless number expressing the ratio of the peak sinusoidal acceleration to the acceleration due to gravity (i.e., the peak sinuosidal acceleration in g's)
$\mathfrak{g}_{\text{rms}} = a_{\text{rms}}/g$, a number expressing the ratio of the rms value of random acceleration to the acceleration due to gravity
W_L = weight of load
W_T = equivalent weight of table driver-coil assembly and associated moving parts
F = rated peak value of sinusoidal generated force
F_{rms} = rated rms value of random generated force

The *force rating* of an electrodynamic vibration machine is the value of force which can be used to calculate attainable accelerations for any rigid-mass table load equal to (or greater than) the driver coil weight. It is not necessarily the force generated by the driver coil. These two forces are identical only if the operating frequencies are sufficiently below the axial resonance frequency of the armature assembly, where it acts as a rigid body. As the axial resonance frequency is approached, a mechanical magnification of the force generated electrically by the driver coil results. The design of the driving power supply takes into account the possible reduction in driver-coil current at frequencies approaching the armature axial resonance frequency, since full current in this range cannot be used without exceeding the rated value of transmitted force at the table, possibly causing structural damage.

In those cases where the test load dissipates energy mechanically, the system performance should be analyzed for each specific load since normal ratings are based on a dead-mass, nondissipative type of load. This consideration is particularly significant in resonance-type fatigue tests at high stress levels.

PROMINENT FEATURES

- A wide range of operating frequencies is possible, with a properly selected electric power source, from 0 to above 30,000 Hz. Small, special-purpose machines have been made with the first axial/resonance mode above 26,000 Hz, giving inherently a resonance-free, flat response to 10,000 Hz.
- Frequency and displacement amplitude are easily controlled by adjusting the power-supply frequency and voltage.
- Pure sinusoidal table motion can be generated at all frequencies and amplitudes. Inherently, the table acceleration is the result of a generated force proportional to the driving current. If the electric power supply generates pure sinusoidal voltages and currents, the waveform of the acceleration of the table will be sinusoidal, and background noise will not be present. Operation with table acceleration waveform distortion of less than 10 percent through a displacement range of 10,000-to-1 is common, even in the largest machines. Velocity and displacement waveforms obtained by the single and double integration of acceleration, respectively, will have even less distortion.
- Random vibration, as well as sinusoidal vibration, or a combination of both, can be generated by supplying an appropriate input voltage.
- A unit occupying a small volume, and powered from a remote source, can be used to generate small vibratory forces. A properly designed unit adds little mass at the point of attachment and can have high mobility without mechanical damping.
- Leakage magnetic flux is present around the main magnet circuit. This leakage flux can be minimized by proper design and the use of degaussing coil techniques.

SPECIFICATIONS

Design Factors

Force Output. The maximum vector-force output for sinusoidal excitation shall be given for continuous duty and may additionally be given for intermittent duty. When nonsinusoidal motions are involved, the force may additionally be given in terms of an rms value together with a maximum instantaneous value. The latter value is especially significant when a random type of excitation is required.

In some cases of wide-frequency-band operation of the electrodynamic vibration machine, the upper frequencies are sufficiently near the axial mechanical resonance frequency of the coil-table assembly to provide some amplification of the generated force. Most system designs account for this magnification, when present, by reducing the capacity of the electrical driving power accordingly.

The peak values of the input electrical signal, for random excitation, may extend to indefinitely large values. In order that the armature coil voltage and generated force may be limited to reasonable values, the peak values of the excitation are clipped so that no maxima shall exceed a given multiple of the rms value. The magnitude of the maximum clipped output shall be specified preferably as a multiple of the rms value. If adjustments are possible, the range of magnitudes shall be given.

Weight of Vibrating Assembly. The weight of the vibration coil-table assembly shall be given. It shall include all parts which move with the table and an appropriate percentage of the weight of those parts connecting the moving and stationary parts giving an effective over-all weight.

Vibration Direction. The directions of vibration shall be specified with respect to the surface of the vibration table and with respect to the horizontal or vertical direction. Provisions for changing the direction of vibration shall be stated.

Unsupported Load. The maximum allowable weight of a load not requiring external supports shall be given for horizontal and vertical orientations of the vibration table. This load in no way relates to dyanmic performance but is a design limitation, the basis of which may be stated by the manufacturer.

Static Moments and Torques. Static moments and torques may be applied to the coil-table assembly of a vibration machine by the tightening of bolts and by the overhang of the center-of-gravity of an unsupported load during horizontal vibration. The maximum permissible values of these moments and torques shall be specified. These loads in no way relate to the dynamic performance but are design limitations, the basis for which may be stated by the manufacturer.

Total Excursion Limit. The maximum table motion between mechanical stops shall be given together with the maximum vibrational excursion permissible with no load and with maximum load supportable by the table.

Acceleration Limit. The maximum allowable table acceleration shall be given. (These large maxima may be involved in the drive of resonant systems.)

Stiffness of Coil-Table Assembly Suspension System

AXIAL STIFFNESS: The stiffness of the suspension system for axial deflections of the coil-table assembly shall be given in terms of pounds per inch of deflection. The natural frequency of the unloaded vibrating assembly may also be given. Provisions, if any, to adjust the table position to compensate for position changes caused by different loads shall be described.

SUSPENSION RESONANCES: Resonances of the suspension system should be described together with means for their adjustment where applicable.

Axial Coil-Table Resonance. The resonance frequency of the lowest axial mode of vibration of the coil-table assembly shall be given for no load and for an added dead-weight load equal to 1 and to 3 times the coil-table assembly weight. If this resonance frequency is not obvious from measurements of the table amplitude vs. frequency, it may be taken to be approximately equal to the lowest frequency, above the rigid-body resonance of the table-coil assembly on its suspension system, at which the phase difference between the armature coil current and the acceleration of the center of the table is 90°.

Impedance Characteristics. When an exciter or vibration machine is considered independent of its power supply, information concerning the electrical impedance characteristics of the machine shall be given in sufficient detail to permit matching of the power-supply output to the vibration-machine input. It is suggested that consideration be given to providing schematic circuit diagrams (electrical and mechanical or equivalent electrical) together with corresponding equations that contain the principal features of the machine.

Environmental Extremes. When it is anticipated that the vibration machine will be used under conditions of abnormal pressure and temperature, the following information shall be supplied as may be applicable: maximum simulated altitude (or minimum pressure) under which full performance ratings can be applied; maximum simulated altitude under which reduced performance ratings can be applied; maximum ambient temperature for rated output; low-temperature limitations; humidity limitations.

Performance. The performance relates in part to the combined operation of the vibration generator and its power supply.

Amplitude-Frequency Relations. Data on sinusoidal operation shall be given as a series of curves for several table loads, including zero load, and for a load at least 3 times the weight of the coil-table assembly. Maximum loads corresponding to 20g and 10g table acceleration under full-rated force output would be preferred. These curves should give amplitudes of table displacement, velocity, or acceleration, whichever is limiting, throughout the complete range of operating frequencies corresponding to maximum continuous ratings of the system. Additionally, the maximum rated force should be given. If this force is frequency-dependent, it should be presented as a curve with the ordinate representing the force and the abscissa the frequency.

If the system is for broad-band use, necessarily employing an electronic power amplifier, the exciting voltage signal applied to the input of the system shall be held constant and the output acceleration shall be plotted as a function of frequency with and without filters or other compensating devices for the loads and accelerations indicated above. If the vibrator is used only for sinusoidal vibrations, and employs servo amplitude control, the curves should be obtained under automatic frequency sweeping conditions with the control system included.

Waveform. Total rms distortion of the acceleration waveform at the center of the vibration table, or at the center on top of the added test weight, shall be furnished to show at least the frequencies of worst waveform under the test conditions specified under the above paragraph. The pickup type, and frequency range, shall be given together with the frequency range of associated equipment. It is desirable to have the over-all frequency range at least 10 times the frequency of the fundamental being recorded. Tabular data on harmonic analysis may alternatively or additionally be given.

Magnetic Fields. The maximum values of constant and alternating magnetic fields, due to the vibration exciter, in the region over the surface of the vibration table should be indicated. If degaussing coils are furnished, these values should be given with and without the use of the degaussing coils.

Frequency Range. The over-all frequency range shall be given. A group of frequency ranges shall also be given for electronic power supplies if they require changes of their output impedance for the different ranges.

Frequency Drift. The probable drift of a set frequency shall be stated, together with factors that contribute to the drift. This shall apply for nonresonant loads.

Signal Generator. A vibration pickup, if built into the vibration machine, shall have calibrations furnished over a specified frequency and amplitude range.

Installation Requirements. Recommendations shall be given as to suitable methods for installing the vibration machine and auxiliary equipment. Electrical and other miscellaneous requirements shall be stated.

HYDRAULIC VIBRATION MACHINE

The *hydraulic vibration machine* is a device which transforms power in the form of a high-pressure flow of fluid from a pump to a reciprocating motion of the table of the vibration machine. A schematic diagram of a typical machine is shown in Fig. 25.7. In this example, a two-stage electrohydraulic valve is used to deliver high-pressure fluid, first to one side of the piston in the actuator and then to the other side, forcing the actuator to move with a reciprocating motion. This valve consists of a pilot stage and power stage, the former being driven with a reciprocating motion by the electrodynamic driver. At the time the actuator moves under the force of high-pressure fluid on one side of the piston, the fluid on the other side of the piston is forced back through the valve at reduced pressure and is returned to the pump.

The electrohydraulic valve is usually mounted directly on the side of the actuator cylinder, forming a close-coupled assembly of massive steel parts. The close proximity of the valve and cylinder is desirable to reduce the volume and length of the connecting fluid paths between the several spools and the actuator, thereby minimizing the effects of the compliance of the fluid and the friction to its flow. Many types of electrohydraulic valves exist, all of which fail to meet the requirement of sufficient flow at high frequencies to give vibration machine performance equivalent to existing electrodynamic machine performance at 2000 Hz.

OPERATING PRINCIPLE

In Fig. 25.7, the *pilot* and *power spools* of a hydraulic vibration machine are shown in the "middle" or "balanced" position, blocking both the pump high-pressure flow *P* and

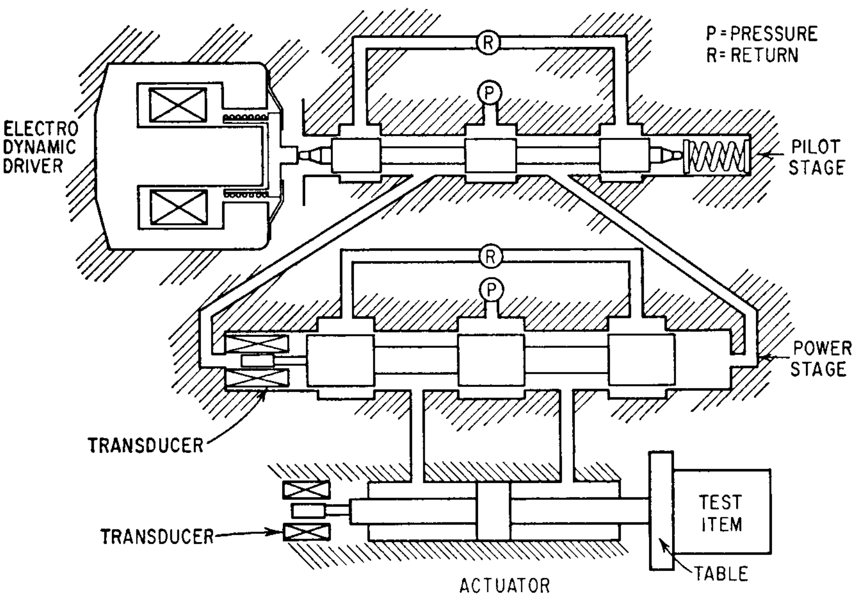

FIGURE 25.7 Schematic diagram of a typical hydraulic vibration machine.

the return low-pressure flow *R*. Correspondingly, the piston of the actuator must be stationary since there can be no fluid flow either to or from the actuator cylinder. If the pilot spool is displaced to the right of center by a force from the electrodynamic driver, then high-pressure fluid *P* will flow through the passage from the pilot spool to the left end of the power spool, causing it to move to the right also. This movement forces the trapped fluid from the right-hand end of the power spool through the connecting passage, back to the pilot stage, and then through the opening caused by the displacement of the pilot spool to the right, to the chamber *R* connected to the return to the pump. Correspondingly, if the pilot spool moves to the left, the flow to and from the power spool is reversed, causing it to move to the left. For a given displacement of the pilot spool, a flow results which causes a corresponding velocity of the power spool. A displacement of the power spool to the right allows the flow of high-pressure fluid *P* from the pump to the left side of the piston in the actuator, causing it to move to the right and forcing the trapped fluid on the right of the piston to be expelled through the connecting passage to the power spool and out past the right-hand restrictions to the return fluid chamber *R*. The transducers shown on the power spool and the actuator shaft are of the differential transformer type and are used in the feedback circuit to improve system operation and provide electrical control of the average (i.e., stationary) position of the actuator shaft relative to the actuator cylinder.

A block diagram of the complete hydraulic vibration machine system is shown in Fig. 25.8. The pump, in conjunction with accumulators in the pressure and return lines at the hydraulic valve, should be capable of variable flow while maintaining a fixed pressure. Most systems to date have required an operating pump pressure of 3000 lb/in.^{2} (20 MPa). The upper limit of efficiency of the hydraulic valve is approximately 60 percent, the losses being dissipated in the form of heat. Mechanical loads are seldom capable of dissipating appreciable power; most of the power in the pump

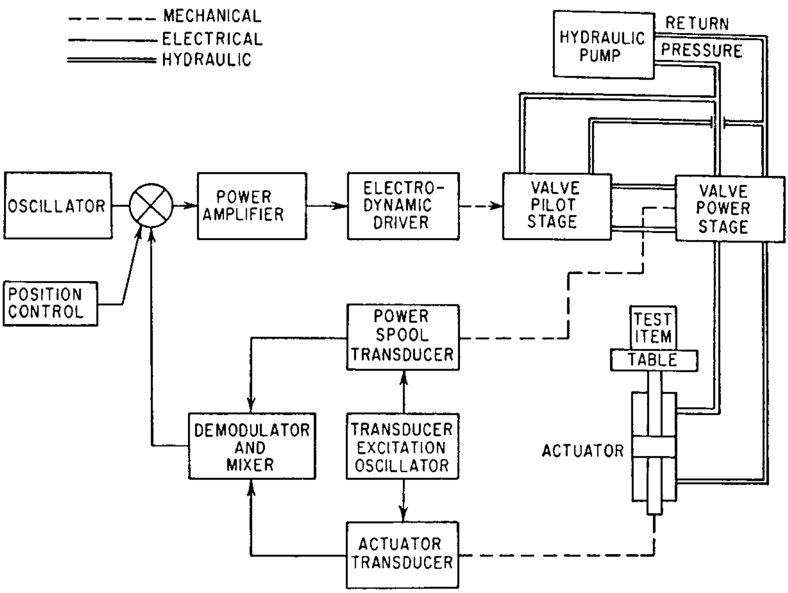

FIGURE 25.8 Block diagram—hydraulic vibration machine system.

discharge is converted to a temperature rise in the fluid. Therefore a heat exchanger limiting the fluid temperature must be included as part of the system.

PROMINENT FEATURES

- Large generated forces or large strokes can be provided relatively easily. Large forces and large velocities of motion, made possible with a large stroke, determine the power capacity of the system. For example, one hydraulic vibration machine has a peak output power of 450,000 lb-in./sec (approximately 34 hp or 25 kW) with a single electrohydraulic valve. This power can be increased by the installation of several valves on a single actuator. Appreciable increases in valve flow can be realized by sacrificing high-frequency performance. Hence, the hydraulic vibration machine excels at low frequencies where large force, stroke, and power capacity are required.
- The hydraulic machine is small in weight, relative to the forces attainable; therefore, a rigid connection to firm ground or a large massive base is necessary to anchor the machine in place and to attenuate the vibration transmitted to the surrounding area.
- The main power source is hydraulic, which is essentially dc in character from available pumps. The electrical driving power for controlling the valve is small. Therefore, the operating frequency range can be extended down to zero Hz.
- The magnetic leakage flux in the region of the table is insignificant by comparison with the electrodynamic-type vibration machine.
- The machine, with little modification, is suitable for use in high- and low-temperature, humidity, and altitude environments.
- The machine is inherently nonlinear with amplitude in terms of electrical input and output flow or velocity.

PIEZOELECTRIC VIBRATION EXCITERS

A piezoelectric material (see Chap. 12) can be used to generate motion and act as a *piezoelectric vibration exciter.* Typically a piezoelectric exciter employs a number of disks of piezoelectric material as illustrated in Fig. 25.9; this arrangement increases the ratio of the displacement output to voltage input sensitivity of the exciter. The strain is proportional to the charge, and the charge is increased by increasing the voltage gradients across the piezoelectric material. The voltage gradient is increased by using many thin layers of piezoelectric material, separated with a conducting material, with alternating polarity on the conducting separators. This arrangement of alternating layers of piezoelectric material and conducting material is called a *piezoelectric stack.* Because the piezoelectric stack has little tensile strength, the stack must be preloaded. The stiffness of the preloading mechanism must be much less than the stiffness of the piezoelectric stack so that preloading will not influence the mechanical output significantly. The combination of the piezoelectric stack (acting like a displacement actuator) and a reaction mass forms a reaction-type vibration exciter as described above. The reaction mass of the piezoelectric exciter can be the armature mass of a small electrodynamic exciter. This effectively places an electrodynamic and a piezoelectric exciter in series, producing a machine with a usable output over a wide frequency range.

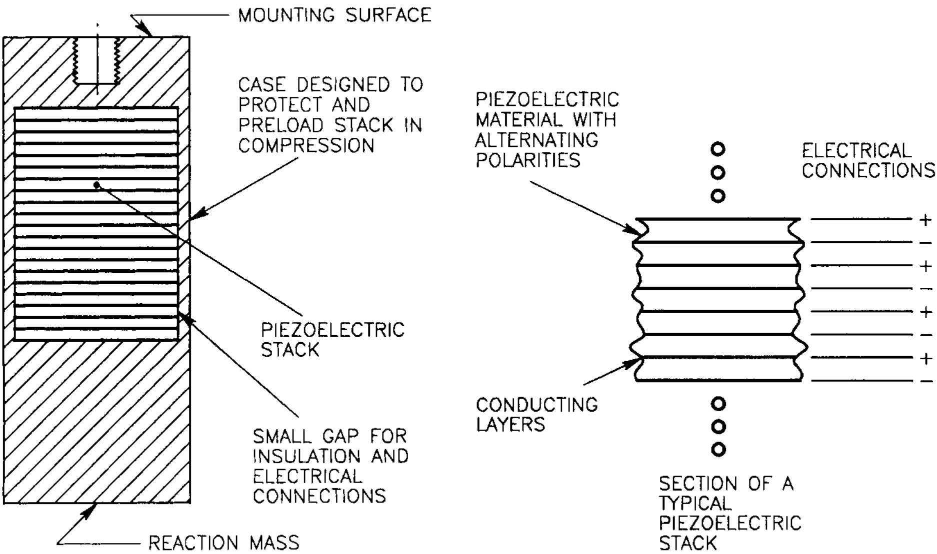

FIGURE 25.9 Simplified cross section of a piezoelectric vibration exciter. A compressed piezoelectric stack is excited with an oscillating voltage. An electrical voltage applied to the electrical connections causes the piezoelectric stack to elongate and contract, producing a relative displacement between the mounting surface and the reaction mass. The inertia of the reaction mass results in a force being applied to an item mounted on the mounting surface.

PROMINENT FEATURES

- The exciters can have a usable frequency range from 0 to 60 kHz.
- The low-frequency output is severely limited by the displacement limits of the piezoelectric stack, usually a few thousandths of an inch (a few hundredths of a millimeter).
- The high-frequency output is limited by internal resonances of the vibration exciter.
- The force output of the exciter is limited by the displacement limit of the piezoelectric stack and by the mass of the reaction mass.
- The power supply for a piezoelectric exciter requires high voltages (typically about 1000 volts) and sufficient current to drive the capacitance (typically 10 to 1000 nanofarads) of the device.

IMPACT EXCITERS

A limited amount of vibration testing, such as some modal testing and some stress screening, require a broad frequency bandwidth of relatively uncontrolled vibration. A class of exciters broadly known as *impact exciters* (and also called *repetitive shock machines*) is sometimes used for the above applications. These devices depend on the property that a short impact generates a broad bandwidth of vibration energy. Each impact is a short transient, for example see Fig. 26.1, but repeated impacts result in a quasi-steady-state vibration having a wide frequency bandwidth. If the impacts are periodic, the spectrum is composed of the fundamental frequency of the impacts and many harmonics of this fundamental frequency, i.e., the excitation is essentially a periodic function. However, the impacts are often varied randomly in magnitude and spacing to produce a time-averaged spectrum that is smoother, much like random vibration. Nevertheless, the instantaneous spectrum or Wigner distribution (see Chap. 22) for the excitation will still reveal an instantaneous periodic function with a time-varying magnitude and fundamental frequency. The probability distribution can vary significantly from a Gaussian distribution. The vibration characteristics are strongly influenced by the dynamics of the structure on which they are mounted. The impact exciters can be mounted directly to the test specimen, or the exciters can excite a table on which the test item is mounted. The latter can be classed as a vibration testing machine.

PROMINENT FEATURES

- The design is usually simple, compact, and rugged.
- The maximum attainable displacement is usually small.
- The vibration is relatively uncontrolled. The user has little control over the spectrum of the resulting vibration.

MULTIPLE SHAKERS DRIVING A SINGLE TEST ITEM

It is sometimes desirable to have more than one shaker driving a test item. Some of the reasons include:

Desire to excite many modes. This is the motivation for multiple input modal tests. A single input may not be capable of exciting all the modes, but multiple input tests have a better chance.

Desire to provide more representative boundary conditions. Many test items are not mounted in service on rigid foundations. Single-axis testing on rigid fixtures is often a poor simulation of the boundary conditions of service environments. Multiple input tests can sometimes provide more realistic boundary conditions. The vibration input in the field environment is often not through a single point.

Large test items. Large test items are difficult to drive with a single shaker. Examples include complete airplanes or space launch systems, seismic simulations, automobiles, and other large transportation systems. The size and/or force requirements to test these items are often beyond the capabilities of a single shaker.

Desire to provide excitation in more than one direction. Most conventional shakers excite the test item in one rectilinear direction. Most environments include vibration in several directions (both rectilinear and rotation) simultaneously. In an effort to provide more realistic testing, shaker systems with inputs in several directions at the same time are desirable.

Multiple exciters driving a single test item have been used extensively in modal testing (see Chap. 21). This is relatively easy because control of the vibration input is not usually necessary. Multiple input tests with controlled inputs are more difficult because of cross-coupling effects. Cross-coupling is where the input at one point causes response at the control point of another input. Control of systems with cross-coupling requires a careful mechanical design and a carefully designed control system (see Chap. 27). The shaker, the fixture, and the control system form three legs of a triad. They must all work together; a weakness in any of the three can result in the system failure. The mechanical design must minimize cross-coupling effects and the control system must compensate for the remaining cross-coupling.

Systems with two inputs typically controlling one translation and one rotation degree of freedom are not very difficult to design. An example would be a horizontal beam-like structure with the vertical translation controlled independently at each end. Isolation of the rotation from the shakers can usually be accomplished with fixtures that are stiff axially but soft in bending.

The mechanical design of systems with more than two degrees of freedom is more difficult. The shaker providing the input can usually move in only one direction. If the test item is to move in more than one direction and/or rotate, the mechanical design of the system must isolate all the motion except in one direction from the shakers. It is also difficult to restrain other degrees-of-freedom, for example, rotations. Restraint of unwanted motion is usually accomplished with passive restraints (for example, hydrostatic bearings) or with active restraints using the exciters and the control system. Undesired motion, compromising the test, will result if the uncontrolled degrees of freedom are not restrained.

A system using three electrodynamic shakers controlling three orthogonal translations, with the three rotations passively restrained, has been built.[7] This system has a usable bandwidth of almost 2 kHz. Electrodynamic systems with six degrees-of-freedom have also been built with varying degrees of success. Electrohydraulic shaker systems with six rigid-body degrees-of-freedom (three translations and three rotations) have been built.[8] These systems have a usable bandwidth of about 500 Hz. Larger electrohydraulic systems with two to six degrees-of-freedom have been built for seismic simulation with a bandwidth of about 50 Hz (see Chap. 24). Other electrohydraulic systems with as many as 18 hydraulic actuators with a bandwidth of about 50 Hz are used as road simulators in the automotive industry. One of these systems is illustrated in Fig. 25.10. An advantage of electrohydraulic shakers for multiple input applications is that their mechanical input impedance is relatively high, reducing the cross-coupling effects. Their disadvantage is that they are all inherently nonlinear, which makes control more difficult. All of these systems, both electrodynamic and electrohydraulic, are capable, with appropriate control systems, of performing sine, random, and transient tests.

VIBRATION FIXTURES

Test items are usually attached to a shaker with a fixture. Seldom will the test item mount directly on the shaker. These fixtures are usually designed to be rigid in the frequency band of interest and lightweight. Rigidity is required because the vibration test is typically controlled at a single point. The assumption is that the motion of the control point is representative of the input to the test item. If the fixture is not rigid, this assumption is obviously not true. Also, flexible fixtures typically have one or more frequencies where the operating shape at the control point is near zero. This will result in large, unrealistic responses of the test item. The fixtures need to be

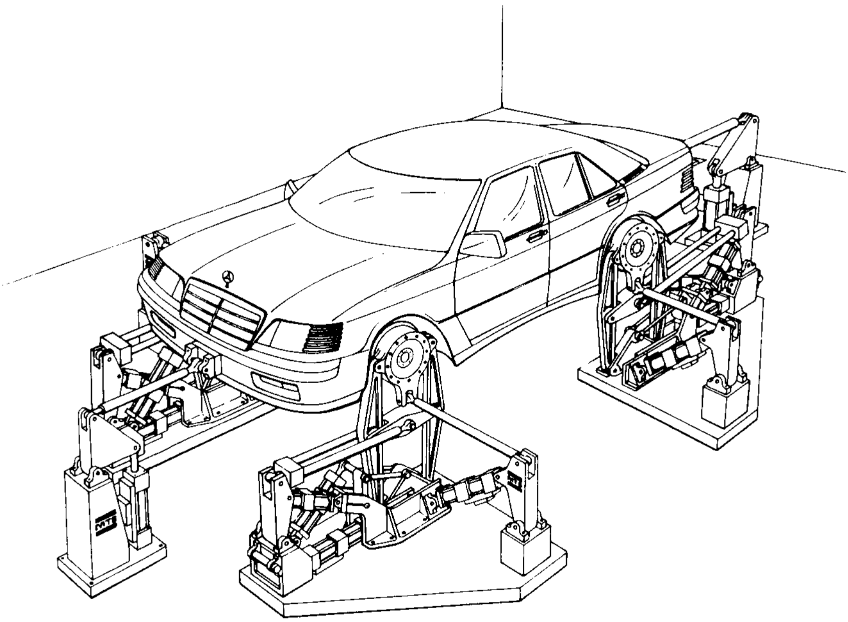

FIGURE 25.10 A road simulator which uses a cross-coupled multiple-drive/multiple-control-point predetermined waveform control system. The predetermined waveforms (with a bandwidth of about 1 to 50 Hz) are measured on the vehicle while driving on a road. The predetermined waveforms are reproduced on the vehicle during the simulation on the road simulator. Four hydraulic actuators drive each wheel hub, and two hydraulic actuators drive the vehicle fore and aft at the bumpers. (*MTS Corp.*)

lightweight to maximize the force available to drive the test item. Light weight and rigidity are contradictory requirements. Design of satisfactory vibration fixtures is a combination of experience, analysis, and compromise. Vibration fixtures are discussed in Chap. 20.

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