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TECHNOLOGY UTILIZATION

SHOCK AND VIBRATION TECHNOLOGY WITH APPLICATIONS TO ELECTRICAL SYSTEMS

A SURVEY

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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Foreword

The Technology Utilization Office of the National Aeronautics and Space Administration has established a program for the rapid dissemination of information on technological developments that may have potential utility outside the aerospace community. These technological developments, when assessed realistically, should improve everyday life and contribute to the prosperity of the nation.

This survey includes NASA's contributions to shock and vibration technology. The first part of the survey is design oriented without reference to specific applications. As a part of shock technology, analytical, design, and experimental techniques, as well as new concepts for shock isolation and absorption, are reviewed. Vibration environments and analytical solutions to design problems are discussed with respect to existing problems. Vibration absorber and isolator concepts are surveyed. The latter part of the document is concerned with the application of NASA's technology to specific classes and subclasses of electrical systems. Shock and vibration associated with electronic components and subassemblies, electronic equipment, and rotating machines of electrical systems are reviewed.

This survey, along with an extensive bibliography, should stimulate the use of NASA's new techniques and applications for the design of systems that are influenced by or those that produce shock and vibration environments. Although the survey is oriented toward electrical systems, it is so organized that it should be useful in other disciplines.

Director
Technology Utilization Office

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Contents

Introduction

The era of high-speed transportation, automation, and industrialization has produced an abundance of consumer goods and services, some with disturbing side effects. People, machines, and structures are affected adversely. Shock and vibration disturbances are problems—social and economic—in our fast-moving age because of the speed of massive and complex mechanical systems. Control or elimination of shock and vibration phenomena, through fundamental knowledge of the underlying physical causes, has been the subject of one of the fastest growing technologies in the physical sciences. In order to control shock and vibration phenomena the designer of hardware (electrical, mechanical, optical, chemical, or medical) must consider shock and vibration as environmental or internal disturbance phenomena that affect the function, performance, and integrity of his product.

Personnel of the NASA space program have had to cope with the reality of devastating shock and vibration environments. Fortunately this fact was recognized early in the program, and a considerable effort has been expended in understanding and applying shock and vibration technology. The acoustical loading of launch vehicle structures is an example of a vibration disturbance. The impact of the Lunar Module landing on the moon is an example of shock loading. Although these shock and vibration environments were obvious to the public, the solution of other, more critical problems in less obvious areas was necessary for the success of the space program.

This survey provides a medium for the transfer of NASA-developed shock and vibration technology from aerospace to nonaerospace use. The increasing use of electrical components and systems in shock and vibration environments dictates a need for applying recent technological advances in order to provide improved isolation techniques. For this reason, this survey is concerned with the design of electrical systems. Specific attention is paid to electrical components and subassemblies, equipment structures, and rotating machines. The necessity of using mechanical and structural components in electrical systems, whether they are simple closures, rotating armatures, or recorders, renders the electrical system vulnerable to mechanical vibrations. Mechanical vibrations, caused by shock or vibration environments, are sustained by the interaction of the system mass and stiffness properties. The system damping provides dissipation of the environmental energy and thereby provides a control of the mechanical performance of the system in the presence of a shock or vibration environment. The shock environment is caused by large motions or forces composed of multiple-frequency components applied to the system in a short time interval. The vibration environment is caused by the application of a periodic multiple-frequency force or motion over a long time interval. Often, shock loading causes an instant failure, while the vibration loading hampers good system performance or causes fatigue failure. The goal of the systems designer is to provide an isolation component or system that will attenuate these environmental hazards to obtain acceptable component life and system performance.

The survey is organized in a parallel format. Shock and vibration technology developed by NASA are presented in chapters 2 and 3. These chapters are design oriented without reference to specific applications and, therefore, could apply equally well to equipment generated in various disciplines. Chapters 4, 5 and 6 are concerned with the application of NASA's shock and vibration technology to specific classes and subclasses of electrical systems. The equipment-oriented user can study new developments or design innovations in these applications chapters. He will be referred to the appropriate new technology in the two design chapters. For the reader in other disciplines, the material on shock and vibration technology provides an organized survey of design technology easily applicable to specific problems.

NASA has generated an abundance of fundamental shock and vibration technology. In "Shock Technology" (chapter 2) the shock environment problem is discussed. New analytical techniques such as modeling, computer simulation, damping, and response analysis are surveyed. Design techniques based on the use of an analog computer, shock spectra, optimization, and nonlinear isolation materials are discussed. The section on experimental techniques contains NASA's contribution to experimental investigations in the context of classical experimental procedure. New ideas are given in test specifications, procedures, specimens, instrumentation, and data reduction. Finally new concepts in shock isolation and absorption are examined.

"Vibration Technology" (chapter 3) is concerned with vibration environments; their characterization is essential to the solution of design problems. The analytical solution of design problems is discussed with respect to damping, natural frequencies and mode shapes, and response. New NASA approaches to vibration experimentation are related to existing problems. The large number of vibration-absorber and isolator concepts, including mass absorbers, elastomers, pneumatic devices, and elasto-plasto-viscous dampers, are surveyed.

Chapter 4 is oriented toward the isolation of electronic components and subassemblies. The mechanisms of structural failure and mounting failure, in addition to temporary lack of performance of components due to shock and vibration environments, are discussed. Subassembly packaging of printed circuits and potted assemblies, as well as encapsulation innovations, are reviewed. Shock and vibration mounting of heavy components (e.g., transformers and solenoids), of fragile components (e.g., tubes and capacitors), and of components subject to relative motion problems (e.g., instruments and switches) are discussed in detail.

The design and isolation of electronic-equipment housings from shock and vibration phenomena are covered in chapter 5. Included are component mounting panels and chassis, cabinet design, vibration suspension of equipment, and shock suspension of equipment.

Chapter 6 contains NASA's vast technology on rotating machinery for use by the designer of high-performance electrical rotating machines. Applications of this technology should be valuable in improving the design of generators, motors, and recorders. Also, new technology on rotor-bearing hardware, including criteria and environments, fluid-film bearings, and rotors is presented. Shock mounting of rotors for performance and survival, and vibration isolation techniques, are reviewed. Control of vibration transmitted from the newly designed machine and techniques in rotating-machine testing are reported.

It must be recognized that this is a survey on shock and vibration technology; it is not intended to serve as a design manual. The intention is to stimulate the reader to use NASA technical literature in the successful design and development of electrical systems.

Shock Technology

Sudden loading of objects occurs frequently in the course of activities such as transportation and manufacturing. Commonly called shock, impact, or impulse, sudden loadings can result from dropping a package, cutting steel, or breaking concrete. This chapter examines shock as an environmental short-time-duration loading of packages, components, machines, and structures. Shock environments affect the working life, the reliability, and the effectiveness of electrical components and information is available in current technical literature concerned with shock technology.

Included in this chapter are: (1) analytical techniques for analyzing a physical situation in order to determine the extent of isolation needed, (2) design and experimental techniques for selecting or developing the isolator or isolation system, and (3) energy absorption concepts for possible application to isolation of electrical systems from a shock environment. Although classical methods may be introduced for purposes of reference and comparison, this chapter and the ones that follow are not a documentary on classical methods; they are meant to serve as an introduction to new ideas, i.e., techniques that have evolved over the last 10 years as a result of NASA's space program.

Because the following three terms are used frequently when discussing shock isolation, it is important that they be defined clearly:

A *Shock Isolator* (fig. 1) is a resilient support that isolates a system from a shock loading. Such isolators are characterized by their long stroke, which allows dissipation of energy over a long period. In the figure, the mass constitutes the isolated item, while the isolator is composed of stiffness and damping. A *Shock Absorber* (fig. 2) is a device that dissipates energy in order to modify the response of a mechanical system to applied shock. The strategy in using this system is isolation of the fixed wall from the excitation with an absorber. A *Shock Pulse* (fig. 3) is a substantial disturbance characterized by rise and decay of acceleration in a short period. The shock pulse normally is displayed graphically: a curve of acceleration, force, displacement, or velocity is plotted as a function of time. The shock environmental condition (fig. 3) experienced in a workshop from a drop hammer shows loading over a period of milliseconds with sudden removal.

The technology to attenuate, if not eliminate, the effects of shock has grown tremendously in the last 10 years. The space program has provided the motivation for much of this fundamental work because shock loading reduces system reliability and/or causes system failure. Shock technology is reviewed in

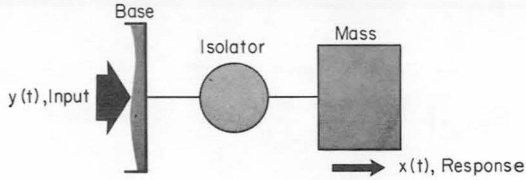

Figure 1.–Shock and vibration isolator.

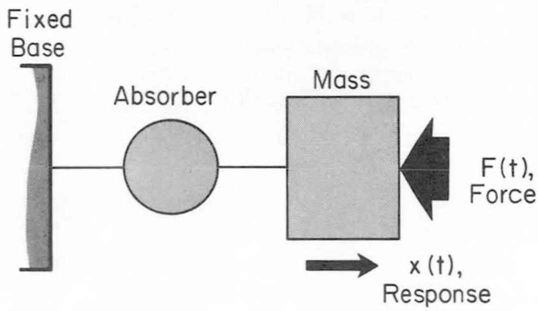

Figure 2.–Shock and vibration absorber.

the context of the engineering design process. In this chapter the following major subjects are considered:

- Shock environments for transportation, package handling, manufacturing, and explosives.
- The many new analytical techniques that evolve yearly.
- The literature showing the importance of mathematical modeling of analytical problems.
- Damping mechanisms and representations with respect to analytical response prediction.
- The classical analytical methods along with new digital computer simulation techniques.
- System response calculation examples concerned with structures, rotor dynamics, and transportation.
- The analytical generation of design shock spectra.
- The analog computer and its role in design.
- Experimental techniques and the use of scale models which have become important because full-scale testing to failure is often not practical. Although scaling of damping is difficult because the damping mechanism may not be known, work on damping has been documented along with experimental system response. Experimental techniques and instrumentation as applied to damping measurement and response determination are outlined.

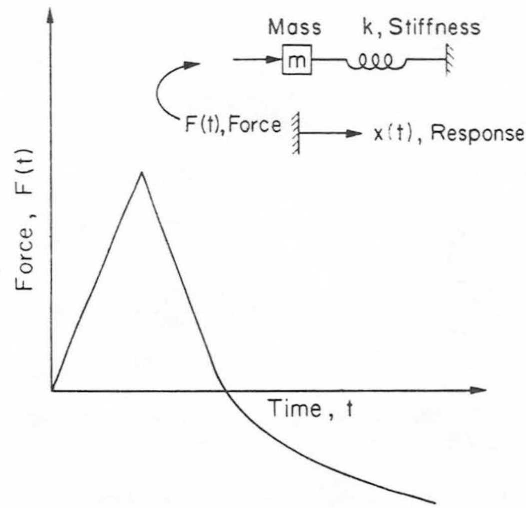

Figure 3.–Shock pulse.

- Design techniques from a problem-oriented viewpoint; e.g., nonlinear springs and damping materials, including the state of development and the usefulness to the designer of optimum isolator design methods and the use of design synthesis charts and shock spectra charts.
- Shock absorption and isolation devices that have evolved for use in shock isolation design. These devices rely on the deformation of elastic or plastic materials and/or the dissipation of energy through friction. The plastic absorbers are extruded and/or crushable structures. Pneumatic springs, foams, and linear springs depend on elastic material energy storage. The coulomb, viscous, or material damping devices dissipate the energy to the surroundings in the form of heat.

ENVIRONMENTS

The shock environment is described as a disturbance–displacement, velocity, acceleration, or force–whose duration is short relative to the characteristic period of the system. Equipment and systems are subject to many types of shock environments. Attempts have been made to classify shock environments for use in frequently occurring design situations. The areas of common concern are

- Transportation
- Package handling
- Manufacturing
- Explosives (military).

In a survey by Ostrem and Rumerman (ref. 1), information and test data

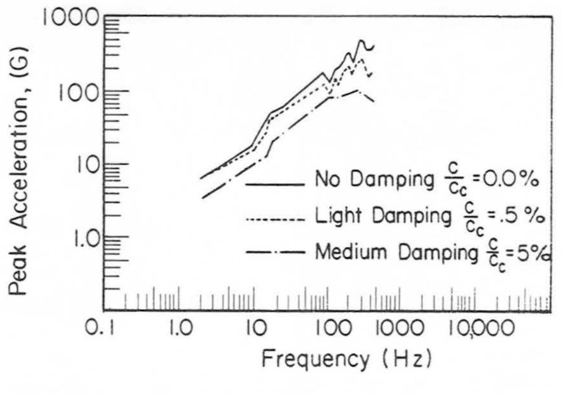

Figure 4.–Railroad-coupling shock spectrum (8.0 mph) standard-draft gear.

describing shock and vibration environments in the four major modes of transportation (railroad, aircraft, ship, and truck) were collated, analyzed, and combined to provide a unified reference source of transportation environments. The shock and vibration data were later updated by Ostrem and Rumerman (ref. 2). From these data, envelopes of acceleration as a function of frequency were constructed for each of the four major transportation modes. These envelopes (fig. 4) allow comparison of acceleration bounds for varied system damping when presented as shock spectra.[1]

Coupling of railroad cars provides the majority of the railroad shock-environment data. Spectra are given for various speeds of impact. Figure 5 summarizes the distribution of coupling speeds based on 3369 measured impacts from independent investigations.

Aircraft, ship, and truck environments are longer-time-duration phenomena, which are classified here as vibration. The one major ship shock environment, except for surface and underwater explosions, occurs from slamming and emergency maneuvers.

Cargo or package handling is an important source of shock excitation. Ostrem and Rumerman (ref. 2) conducted a comprehensive literature survey and search for data and information applicable to cargo-handling environment. This environment was defined to include those motions resulting from operations such as physical handling, loading and unloading, and related physical movements in the dock or storage area. It was found that, in general, the shocks received by packages and equipment during handling operations are greater than those experienced on a vehicle, ship, or aircraft in transit. This information is summarized to show the distribution of drop heights for

[1] A shock spectrum is defined as the maximum response (acceleration, velocity, or displacement) of a series of damped or undamped single-degree-of-freedom systems resulting from a specific shock excitation. Therefore, an independent mass-spring-damper system is associated with each frequency in figure 4.

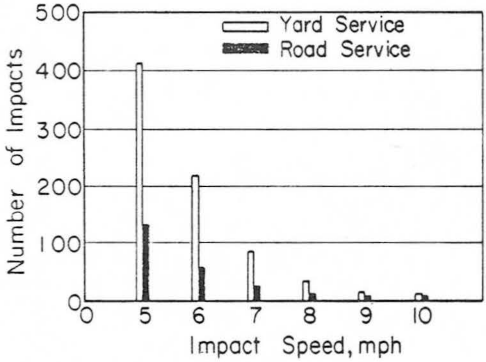

Figure 5.–Distribution of impact speed of railroad cars.

particular packages, distribution systems, and handling operations. Handling environments are described in terms of drop heights. Drop heights and package orientation are related to package size. The summary of this work contains the following: (1) number of drops received per package per trip; (2) distribution of the drop over faces, edges, and corners of the package; (3) effects of package size and weight; and (4) effects of labels and handholds.

Manufacturing environments have to be characterized individually because of the variety of equipment that can cause shock environments for electrical systems. Machines such as punch presses, forging equipment, and any heavy machinery provide shocks to adjacently mounted equipment.

Military vehicles and systems are subject to shock due to surface air blasts and underwater explosions. Shock spectra (fig. 6) change for different levels in a ship for a typical external loading.

ANALYTICAL TECHNIQUES

Analytical techniques, essential to good shock-isolation system design, are valuable in selecting isolators that meet the required space and force constraints. The fundamental analysis of elastic-damped systems is well documented (refs. 3, 4, and 5). Major improvements in analysis have resulted from the evolution of both efficient numerical methods and rapid digital computers. Computers make it possible to model systems in fine networks and to provide detailed system response analysis. Computer simulation allows the formation of a simulation model of a basic concept, with parameter variation available at the turn of a knob. Within the simulated concept, the design parameters of the system can be varied to obtain the best design through a parameter-variation study.

System response for the specialized technologies concerned with electrical systems is described in this section. Shock responses of rotating machines,

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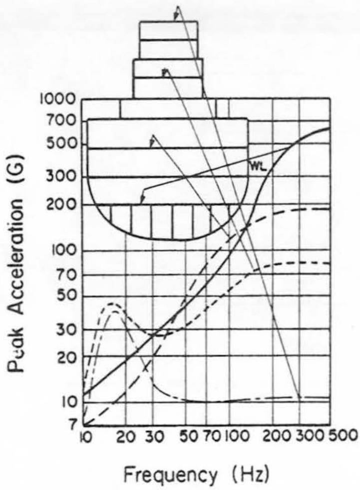

Figure 6.–Shock spectra for ships subject to surface-air blasts and underwater explosions.

structures, and packaging are examples of NASA work that can be easily translated to industrial use.

Modeling

The most important aspect of the solution of a shock and vibration problem is the selection of a mathematical model for the physical system. The system must be separated into components (such as gears, shafts, and bearings) that can be mathematically described. The modeling of a small electric motor and a box structure are shown in figure 7. The motor is modeled as a single-degree-of-freedom system,[2] the box structure as a multidegree-of-freedom system.

The two basic types of error that occur in the solution of a shock and vibration problem are numerical error and modeling error. The best mathematical technique will not overcome modeling error, and the use of an erroneous model with a good mathematical technique will yield an undesirable solution. It is important, therefore, to obtain the best model available; then, if an approximate solution is desired, the model can be reworked to facilitate an easier solution. This method of problem solving provides the analyst with some indication of whether his answer is high or low. Methods of solution are always the final indicator in the modeling process because the mathematical model must be amenable to solution.

The solution of any mathematically modeled system entails calculating the system's response to a given excitation. The response of the isolated system to shock input pulses with small durations relative to the fundamental natural

[2] Degree of freedom: the minimum number of independent coordinates required to define completely the positions of all parts of a system at any instant of time.

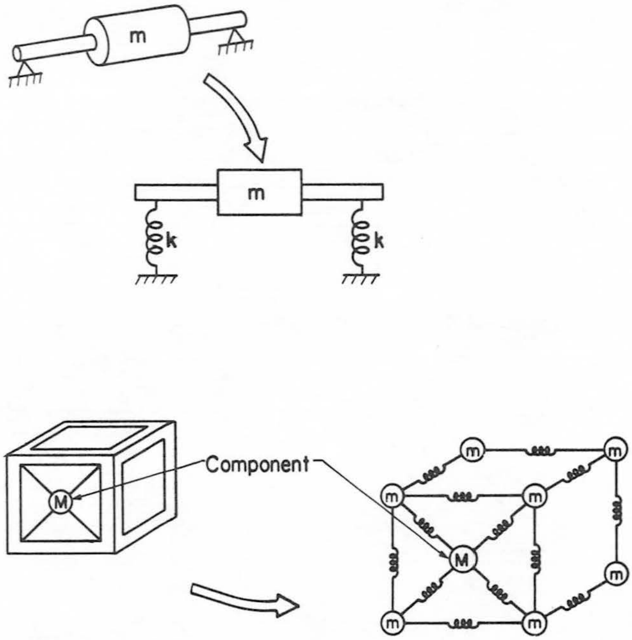

Figure 7.–Lumped-parameter modeling of structures and rotors.

period[3] of the system is desired. The mathematical system can be modeled by one of two basic techniques or their combination: lumped-parameter model or continuous-parameter model.

A lumped-parameter model of a system is composed of an arrangement of lumped masses (no elasticity) and lumped springs (no mass) that simulate its dynamic response. Figure 7 shows typical examples of lumped-parameter models of structural and mechanical systems. The behavior of systems modeled with lumped parameters have only time as an independent variable and, therefore, give no information on its space distribution. Mathematical representation of a lumped-parameter model is:

$$m \frac{d^2x}{dt^2} + C \frac{dx}{dt} + kx = q(t) \text{ . . . ordinary differential equation.}$$

A continuous-parameter model is composed of an arrangement of elements in which mass and elasticity are distributed in space (fig. 8). Systems modeled with continuous parameters are described mathematically by partial dif-

[3] Natural period: a physical characteristic of a system, dependent on the magnitude and arrangement of the system mass and elasticity; the period is observed physically by disturbing the system and noting the time required for it to complete one cycle of vibration.

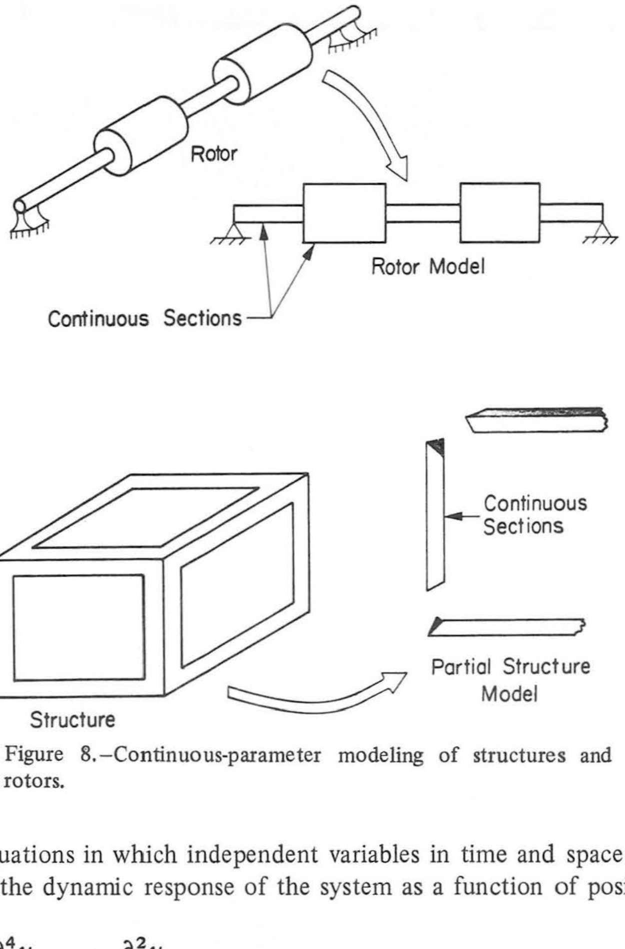

Figure 8.–Continuous-parameter modeling of structures and rotors.

ferential equations in which independent variables in time and space are used to describe the dynamic response of the system as a function of position and time:

$$\frac{\partial^4 u}{\partial t^2} + a^2 \frac{\partial^2 u}{\partial t^2} = f(s, t) \ldots \text{partial differential equation.}$$

At the microscopic level, physical systems can be represented best as having continuous mass and elastic properties. When the solution of the continuous-parameter system is not possible or practical (because partial differential equations are utilized to find the dynamic solution to continuously represented system properties), the lumped-parameter system, characterized by ordinary differential equations, is used. Lumped-parameter systems often yield excellent approximations for fundamental natural frequency[4] and calculated response. A

[4] Fundamental natural frequency is the reciprocal of the fundamental natural period. It is labeled fundamental because massive elastic systems have a large number of observable natural frequencies. Generally these natural frequencies can be observed physically by applying a vibratory disturbance of frequency equal to a natural frequency of the system and observing the large system response.

lumped-parameter model is adequate in the frequency range of the fundamental natural frequency. Although natural frequencies are obtained in a lumped-parameter system, the error of higher natural frequencies increases as the frequency increases. When a smooth mass distribution is obtained, it is easier to solve the partial differential equation than the set of N ordinary differential equations needed to represent the system. The derivation of the partial differential equation and its solution also yields a more complete understanding of the mechanics of the problem than can be obtained through the use of lumped-parameter analyses.

The basic problem in modeling is representing the mass and stiffness properties of a system so that the desired solution will be obtained. Essentially the optimum mathematical model of a physical system is represented by a combination of lumped- and continuous-parameters. The absolute value of the system's various stiffnesses have no meaning in the modeling process, the relative stiffness of the various parts being of greater importance. For example, if a large mass is located on a shaft, the stiffness of the mass is so large that it will exhibit little deflection in comparison to the shaft. From this point of view, it is more practical to represent this lump as having infinite stiffness. The mass of the "natural" lump can be represented several different ways, depending on its configuration. In a rotating system, the gyroscopic moments become important for large thickness-to-diameter ratio disks. The configuration of the mass is important, therefore, and can affect a good or bad solution.

Damping

The analytical representation of stiffness and mass can be accomplished with ease for most structures; and natural frequency, normal-mode calculations[5] can be made with the confidence that they will correlate with experimental data. Because emphasis has been placed on natural frequency calculations, analytical representation of damping has lagged far behind that of stiffness and mass. A second reason is the difficulty of representing damping analytically. Because damping is usually nonlinear, the solution of the equations of motion of a structure is difficult. With the advent of large complicated structures that cannot be tested either for natural frequencies or, even more important, for dynamic response, realistically scaled models must be used to obtain this information. It must be concluded, therefore, that if scale models are to be used effectively to calculate dynamic response and elastic stability of a structure, damping must be properly represented both analytically and with scaled models.

A thorough analytical representation of damping for a single-degree-of-freedom system has been done by Reed (ref. 6). His mathematical forms

[5]Normal-mode: the normal mode is the space configuration the system assumes when it is vibrating at a natural frequency.

identify different damping mechanisms with common structural applications. Of course, combinations of the energy-dissipative mechanisms are found in each structure. Five forms of damping described by Reed are

- Viscous
- Material-displacement
- Velocity-squared
- Material-viscoelastic
- General.

Viscous and velocity-squared damping are damping forces that are proportional to the relative velocity of the vibrating body. In the more general case, the velocity damping force $\alpha(V)^{\gamma}$ is proportional to the velocity of a noninteger exponent (γ) that depends on the relative velocity of the vibrating body.

Viscous damping, $\gamma = 1$, is usually a low-speed phenomenon found in viscous fluid films of bearings and joints, while V^2 and V^{γ} are high-speed environmental phenomenon (e.g., the aerodynamic effect of drag). Linear elastic materials exhibit damping properties such that the damping force is proportional to the stress history of the material. In displacement damping (where the force is proportional to a coulomb friction coefficient) the displacement, which results from the relative movement of unlubricated structural joints, is known as slip damping.

Crandall (ref. 7) has made analyses of similarity to provide scaling laws; they indicate the effects of amplitude, frequency, and material properties on resonant damping encountered by linear structural elements when damping is due to internal material properties. Two nonlinear damping relationships are considered. The first concerns frequency-independent hysteresis damping in which the material damping coefficient c satisfies a relationship of the form

$$c = \left(\frac{\text{stress amplitude}}{S_0} \right)^{n} ,$$

where S_0 and n are material constants. The second concerns nonlinear damping for which

$$c = \left(\frac{\text{stress amplitude}}{S_0} \right)^{n} \frac{\omega\tau}{1 + \omega^2\tau^2} ,$$

with the relaxation time τ an additional material constant equal to the system's vibration frequency. Correlations provided by the scaling laws were achieved on data from a large number of tests with steel, brass, and aluminum cantilever beams. The steel and brass data were correlated reasonably well on the basis of the first expression, while the aluminum data were better correlated on the basis of the latter expression.

Nonlinear rubber-like materials exhibit viscoelastic behavior, and the damping force is proportional to the deformation and its time derivative. This damping mechanism is obtained in rubber-like material coatings, viscoelastic structures, adhesives in joints, and the like. The general damping mechanism can be used to describe mechanisms not listed if the force history of the mechanism is obtained experimentally.

Ruzicka, Derby, et al. (ref. 8) have predicted the damping properties of structural composites incorporating distributed viscoelastic shear-damping mechanisms for the case in which the structural composites consist of two or more elastic elements of arbitrary material and size with an intervening viscoelastic damping layer. For geometrical configurations incorporating a thin layer of viscoelastic damping material that is soft relative to the stiffness of structural materials employed in the structural composite, the structure loss factor η may be expressed in terms of three parameters—β, a loss factor of the viscoelastic material, X, a shear parameter, and Y a geometrical parameter:

$$\eta = \eta(\beta, X, Y)$$

Physically the damping material loss factor β is the ratio of the imaginary and real components of the complex shear modulus $G^* = G' + jG''$, where G'' and G' are the loss modulus and storage modulus of the viscoelastic material, respectively. The shear parameter X depends on the parameters of storage modulus and amount of viscoelastic material employed, weight loading on the structural member, flexural rigidity, geometry of the cross section, and frequency of vibration. The geometrical parameter Y, which is a function only of the geometry of the cross section and the modulus of elasticity of the elastic elements comprising the structural composite, may be expressed:

$$Y = \frac{(EI)_\infty}{(EI)_0} - I ,$$

where $(EI)_0$ is the flexural rigidity of the structural composite when its elastic members are uncoupled and $(EI)_\infty$ is the flexural rigidity of the structural composite when its elastic members are completely coupled.

Ruzicka's (ref. 8) investigation has resulted in (1) the generation of extensive design data for the geometrical properties of viscoelastic shear-damped structural composites, (2) the development of manual and automated design procedures for predicting the loss factor of viscoelastic shear-damped structural composites, and (3) the performance of laboratory experiments that have confirmed the adequacy of existing theory and design procedures developed.

In any structure, combinations of many mechanisms are obtained and must be modeled. Even in a structural joint, three forms of energy dissipation may occur as a result of cyclic strain at an interface: (1) dry sliding (displacement coulomb), (2) lubricated sliding (viscous), and (3) cyclic strain in the separated adhesive (viscoelastic layer damping between mating surfaces). Combinations of viscous, structural, and velocity-squared damping exist in almost every structure. The problem encountered in structures is allocation of the proper magnitude of damping to each mechanism. Care must be exercised where damping is to be modeled because it has been found by Hanks and Stephens (ref. 9) that geometric scaling does not allow good scaling of the damping mechanism. In the physical and mathematical modeling of a structure, environmental effects that influence the damping mechanism of both model and prototype must be described, so that a model that simulates the dynamic behavior of the prototype can be scaled and constructed. In many shock and vibration systems, damping is sufficient to attenuate the response of the system at a natural frequency, and vibration is not a problem. When it is recognized, however, that a physical forcing function exists at a natural frequency, the design must be modified either to remove the forcing function or to change the natural frequency.

CLASSICAL ANALYTICAL METHODS

After a system has been modeled for mathematical analysis, a technique suitable for determining the system's response to a given excitation is selected. Although a detailed account of classical analytical methods is not presented here, a summary is instructive for comparison with recently developed techniques used to determine the response of structures, rotors, and packages. Analytical response solutions to a shock-pulse excitation of a single-degree-of-freedom system (fig. 9), therefore, are considered.

When a shock response problem is modeled linearly, superposition of solutions is valid. That is, if the response to a number of waveform excitations is known, then these solutions can be used in combination to find the solution to the particlar shock excitation in question. For example, if the simple absorber has a shock-pulse-loading function (fig. 9), then the solutions for the waveforms (fig. 10) can be superposed to obtain the complete system response. The system response $x(t)$ is a function of the shock pulse $F(t)$ and τ_3, the period of the shock pulse.

Duhamel's Integral, a well-known approach to the solution of transient vibration problems in linear systems, is based on the superposition of the

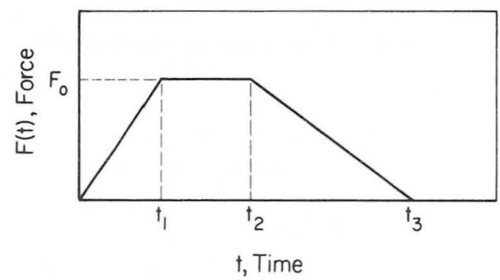

Figure 9.–Typical shock pulse.

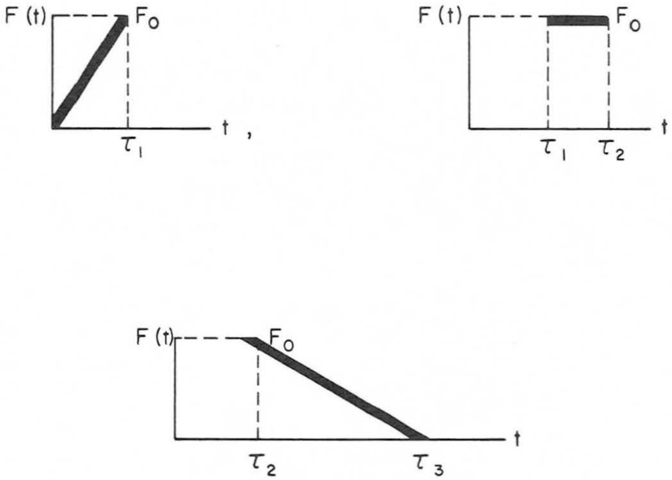

Figure 10.–Simple shock waveforms.

responses of a system to a sequence of impulses (ref. 10). In general, the response $x(t)$ to a unit impulse excitation is used to establish the response to an arbitrary force $F(t)$. The arbitrary pulse $F(t)$ is broken into a series of pulses of time duration $\Delta\widetilde{t}$ as shown in figure 11a.

Figure 11b shows that $g(t)$ is the system response to the unit step function $F(\widetilde{t})$, $\Delta\widetilde{t}$. Since the system is linear, the principle of superposition holds; therefore, the series of unit excitation functions is summed to obtain the total response:

$$x(t) = \int_0^t F(t)\, g\,(t - \widetilde{t})\, dt\,.$$

A shock is a disturbance or excitation pulse of displacement, velocity, acceleration, or forces whose duration is short relative to the characteristic period of the system. The application of this shock pulse to a single-degree-of-freedom system (oscillator) results in a time response of the oscillator. The maximum value of the time response, for a given shock pulse, depends on the natural frequency and damping of the oscillator. The plot of the maximum response of the oscillator against the natural frequency of the oscillator is the shock spectrum of the disturbance.

A mathematical method such as the Laplace transform (ref. 11) is a powerful tool for the solution of linear differential equations. In this method a transformation integral

$$F(s) = \int_0^\infty e^{-st}\, F(t) dt$$

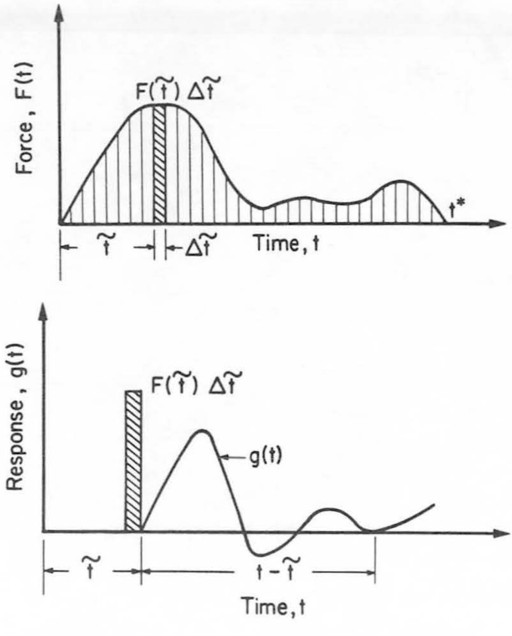

Figure 11(a).–Shock pulse divided into a series of pulses of duration $\Delta\tilde{t}$. (b) system response to a shock pulse of magnitude $F(\tilde{t})$, and duration $\Delta\tilde{t}$.

is defined, and the differential equation is shifted from a time plane to a complex s-plane through the use of the initial conditions and the differential equation of motion. This yields an algebraic equation, which is manipulated into a form in which inverse transformations that return the response from the complex s-plane to the physical t-plane can be made.

Computer Simulation

Two types of electronic computers, the analog and the digital, are in general use for scientific investigation and data processing. The digital computer seems to be the more popular of the two, possibly because the communication language between man and the digital computer now closely resembles existing scientific language.

Both computers can solve engineering problems, but each has specific advantages. Since they operate sequentially with discrete numbers, digital computers probably are most effective for handling problems that involve large quantities of tabulated data in discrete numbers. On the other hand, problems that concern integration of continuous data in a fixed system model are solved more readily on the analog computer. Another advantage of the analog computer is that it can perform a variation of parameters on a given model to obtain a desired solution; however, the digital computer is inherently precise and can handle large amounts of tabulated data on varied system models. Consequently it is useful to incorporate the advantages of the two types by a simulation language.

Digital Simulation Languages (DSL) allow the digital computer to infringe on the domain of the analog computer by their duplication of the performance of integrators, summers, and limiters through the use of subroutines within the simulator program. The common starting point for continuous systems simulation is the conventional analog block diagram, and the common approach is the breakdown of the mathematical system model into its component parts or functional blocks. These blocks have close to a one-to-one correspondence with analog computing elements (e.g., integrators, summers, and limiters) and usually appear as subroutines within the simulator program. When a simulation package is used, "programming" usually involves interconnecting the functional blocks by a sequence of statements prepared according to the rules of the input language.

A variety of simulation languages, each with advantages and disadvantages for a given problem, has been developed by various companies and universities over the years in order to satisfy a particular need or to reflect a current philosophy on the subject. These languages are available and include IBM's DSL, COBLOC, DAS, DES-1, DYNASAR, DYANA, and MIDAS, and the standardized Continuous System Simulation Language (CSSL) (ref. 12). The continuous change models usually are represented mathematically by differential or difference equations that describe rate of change of the variable over time.

A computer language that enables the engineer to communicate his problem, defined in engineering terms, directly to the computer is commonly called a problem-oriented language (POL). Problem-oriented languages can be used by persons not trained in procedural programming languages (e.g., FORTRAN) or in advanced mathematical techniques. Using problem-oriented languages, a problem can be solved immediately, and system parameters can be varied or optimized to obtain a better design. It must be recognized, however, that these languages are useful only for specific problems or classes of problems. The following are among POL's, that have been developed recently or are being developed currently:

- Integrated Civil Engineering System (ICES) (ref. 13)
- Computer-Oriented Mechanical Design (COMMEND) (ref. 14)
- The Stress Postprocessor (ref. 15).

Insofar as engineering problem-oriented languages are concerned, the most significant effort has been in civil engineering.

The concept of the *Macro* is used in the simulation language. The Macro consists of a set of prototype statements that are inserted into the program each time the Macro is referenced by name. Macros are referenced exactly as if they were operators of the language. The user also can create his own Macro library, which will appear to the system as an extension or replacement for the system library.

There are many digital computer codes for solving general and specific mathematical problems. Most of these codes were developed by industry and government in conjunction with specific research programs. Unfortunately the only common denominator in their construction is their procedural language, FORTRAN.

The SHARE library is a source of general computer subroutines oriented toward mathematical operations such as integration, matrix manipulation, and function evaluation. These subroutines form the body of problem-oriented computer codes. The simulation languages are included in the SHARE library.

The Computer Software Management and Information Center (COSMIC) (ref. 16) has readily available over 300 codes. These codes encompass all disciplines and some can be adapted to solve shock-isolation problems.

The SAMIS system (ref. 17), developed at the Jet Propulsion Laboratory, (JPL) is a large-capacity program for the analysis of frame and shell-type structures. Plate and shell elements can be idealized by an appropriate assemblage of triangular finite elements (FACET). Bars, elementary beams, and shear beams can be approximated by line elements in a similar manner. A suitable collection of FACET and line element card data can be processed by the SAMIS system to generate the stiffness and mass matrices of a structure. All processing and matrix operations are user-controlled by a set of pseudo-instructions that perform the desired operations. Twenty pseudo-instructions are available to the user. Operations performed by SAMIS pseudo-instructions include:

- Matrix addition, subtraction, and multiplication
- Element-stiffness matrix generation from card data input and the assemblage of the complete structure-stiffness matrix incorporating boundary conditions.

A similar computer program called NASA Structural Analysis (NASTRAN) (ref. 18) was developed to perform static and dynamic analyses of large complicated systems. It utilizes recently developed techniques in finite element modeling and numerical calculation.

System Response

This section reviews analytical techniques as they relate to determining the response of systems that include railroad cars, structural packages, and rotating machines. The technique for generating design shock spectra also is discussed.

A method for estimating the response of a container and its contents has been developed by Scialdone (ref. 19). The specific problem involved impacting a railroad car, on which containers were mounted, into a group of braked cars standing in the railroad yard. Equations were derived for container-car-equivalent impact velocity, container-car acceleration, car-body transmissibility, and container transmissibility. By applying data characteristics of the car and its coupler, the natural frequency of the car was calculated. The container-car-equivalent impact velocity was obtained from the actual impact speed and the car weights. Input accelerations of the car and shipping container were calculated over a range of 1 to 14 mph of input velocities for bottoming and nonbottoming of the coupler during impact. The container transmissibility then was determined from the following data: (1) the inner-container-design natural frequency, (2) its damping characteristics, and (3) the forcing frequencies resulting from the impact at the car coupler. The accelerations resulting from the impact at the inner container were calculated for zero and 10-percent coupler viscous damping ratios and compared with the upper bound values of accelerations obtained from impact tests. The results indicated that, if a 10-percent value for equivalent viscous damping of the car coupler was used for car impact speeds above 7.5 mph, accelerations at the inner container and at the car bed adjacent to the outer container exceeded 1.8 and 20 g, respectively.

Ostrem and Rumerman (ref. 2) used the approach of Mindlin (ref. 20) to determine analytically the response of packages to shock excitation. This approach not only represents load-deflection characteristics of a given cushioning material by a relatively simple analytical expression but also provides closed-form expressions for maximum acceleration and displacement resulting from a given velocity step (e.g., sudden drops, bumps, and machine movements). The mechanical adequacy of cushioning materials is based on experimentally determined stress-strain (load-deflection) characteristics.

The analytical solution of the dynamic response of a complex structure (fig. 12) to a deterministic transient input by the normal mode method is outlined by Hasselman and Hwang (ref. 21). The utility of the normal mode method for computing the dynamic shock response of base-excited cabinet structures is illustrated. The detailed dynamic loads obtained eliminate many of the uncertainties encountered in the load factor approach to a design problem; in addition, a structure can be "tested analytically" before the hardware is available, thus allowing a more efficient developmental process. New concepts are particularly suited for the treatment. Several sonar system cabinets, which averaged 65 in. in height and weighed approximately 300 to 1200 lb, were analyzed. Excitation consisted of an acceleration pulse recorded during prior testing of similar equipment on the Navy medium-weight shock-testing machine using the 25- to 30-Hz (natural frequency) simulated deck.

Insight into a specific exitation-system-response relationship is obtained by examining plotted mode shapes and comparing their corresponding frequencies with the frequency content of the exitation. Isometric representation and orthographic projection of overall maximum displacements serve to focus attention on critical areas and to indicate the ways by which certain load conditions arise. The results of this analysis depend on the validity of various

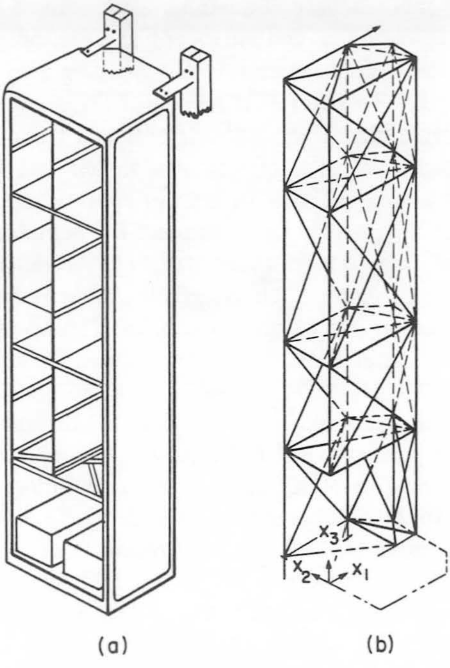

Figure 12.–Modeling of actual cabinet structure (a) cabinet (b) Analytic Model.

assumptions made and, therefore, should be justified by actual testing of the equipment.

The response of a rigid-bearing, simple rotor system subjected to an impact load applied to the rotor or pedestal support was analyzed by Malanoski (ref. 22). Before impact loading, the rotor-bearing system is under steady-state pressure. The typical journal-bearing system (fig. 13) can be modeled mathematically. A linearized "ph" (p, pressure; and h, film thickness) solution represents the model for the finite, plain journal bearing; a linearized "p" solution represents the model for the finite, spiral journal bearing. The perturbed equations of motion are derived for shock loads applied to either the pedestal or the rotor. The shock loadings are characterized by either impulse, unit step, or finite-duration pulse forcing functions. Results of frequency response (steady-state vibration response at varied forcing frequencies) and transient response (shock-excitation response) are shown for varied effects of the dimensionless parameters L/D, ϵ, Λ, and Ω. These responses are presented and discussed for plain journal and spiral-grooved journal bearings. The dimensionless parameters L/D (length to diameter of ratio of the journal), ϵ (eccentricity), Λ (compressibility number) and Ω (mass parameter which represents the normalized rotor mass per bearing) are used also to obtain

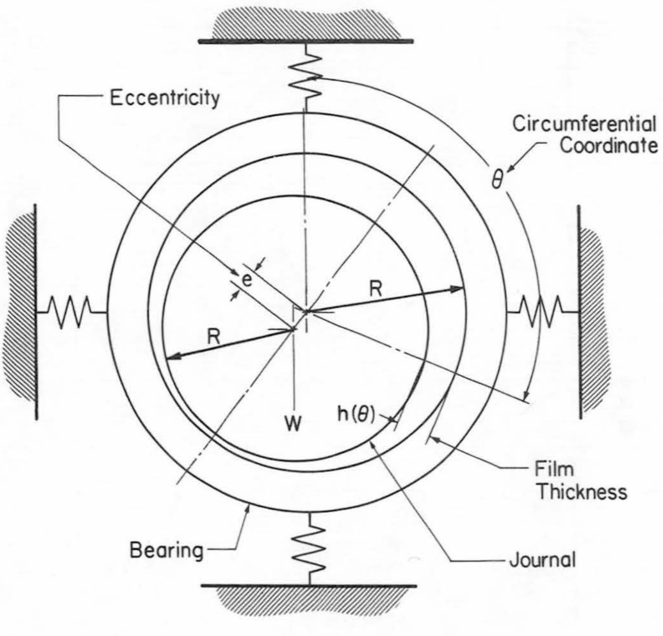

Figure 13.–Fluid-film bearing geometry.

general results in bearing analysis. The prescribed procedures can be used to generate data for a shock response characterized by either an impulse, unit step, or finite pulse input utilizing the preceding frequency response data (fig. 14). Data (fig. 15) for a particular finite plain bearing undergoing a shock loading are given. Because the rotor mass cannot be separated from the problem, preparation of sufficient data for use in design charts would require an enormous amount of computer time.

Gunter's (ref. 23) excellent work on the dynamic stability of rotor-bearing systems should be consulted for additional background information about rotor-bearing systems. The application of rotor-bearing technology to rotating systems (e.g., electric motors and drum recorders) is discussed in Chapter 6.

The response of a component or system to a shock environment usually is expressed as the history of a parameter describing the system motion. For simple systems, the magnitudes of response peaks can be summarized as functions of the system's natural frequency at various fractions of critical damping. This type of graphical plot, termed a shock spectrum or, more properly, a shock-response spectrum, can be plotted in two or three dimensions (figs. 16 and 17). The response (fig. 16) of a given structure varies with shock environment. The three-dimensional shock spectrum (fig. 17) provides the response peaks through their history.

The shock spectrum can be used as a display for experimental data or as a

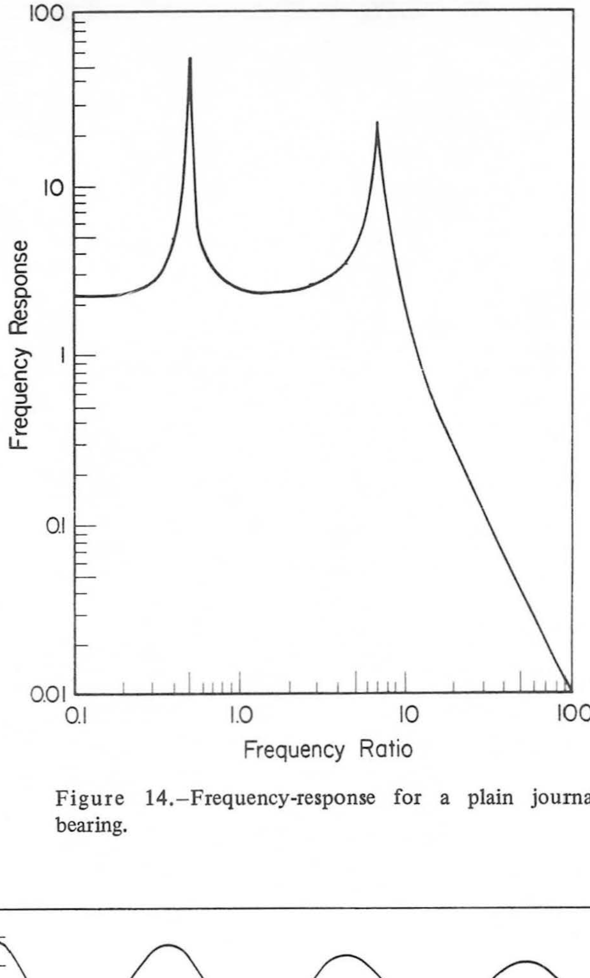

Figure 14.–Frequency-response for a plain journal bearing.

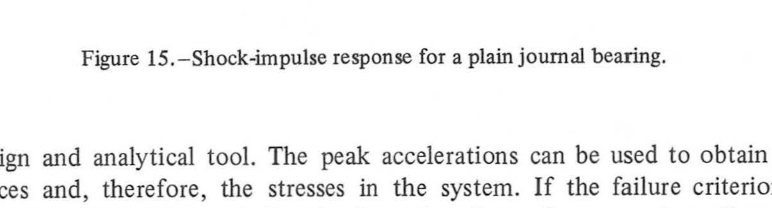

Figure 15.–Shock-impulse response for a plain journal bearing.

design and analytical tool. The peak accelerations can be used to obtain the forces and, therefore, the stresses in the system. If the failure criterion is deflection, it can be read directly from the plot and compared to allowable deflections.

The uncoupling of response at various modes severely limits this approach

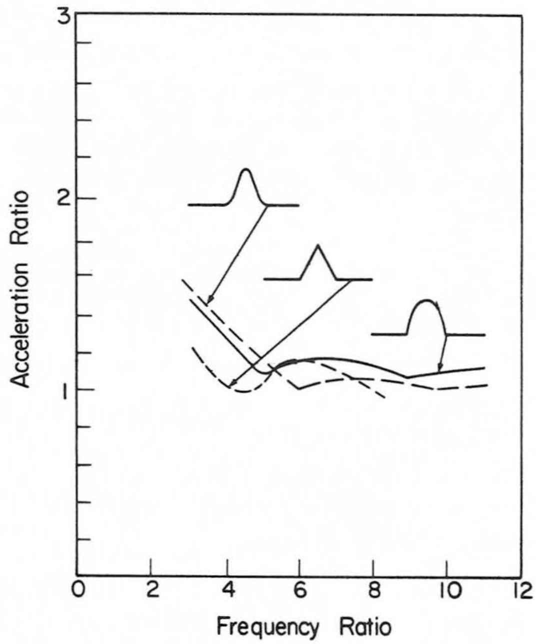

Figure 16.–Two-dimensional shock-response spectrum with varied input-shock waveforms.

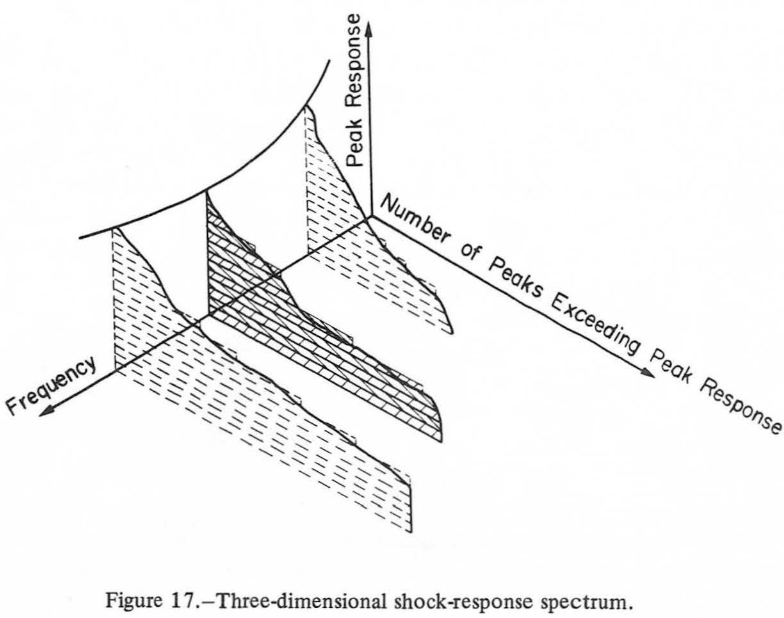

Figure 17.–Three-dimensional shock-response spectrum.

for complex systems where a multidegree-of-freedom model is required to describe the response of the system. For simple systems, however, it is a valuable tool. An example of its use is given by Davis (ref. 24) in the generation of torsional shock spectra. An analog computer was used to determine the history of acceleration response of the mass to an acceleration input applied to the foundation of a single-degree-of-freedom model of the system.

The differential equation describing the response of the system is

$$\ddot{q} + 2c\,\omega_n\dot{q} + \omega_n{}^2 q = -A(t)\,,$$

where

q	is the system motion with respect to the foundation, (x-y),
c	is the damping factor,
k	is the stiffness,
ω_n	is the natural frequency, and
$q + A(t)$	is the acceleration response of mass, in absolute coordinates, relative to a fixed reference.

The results of the parameter variation on c and ω_n yield a shock spectrum and variation of $A(t)$; the input acceleration environmental description yields a series of shock spectra.

DESIGN TECHNIQUES

The following paragraphs, intended to serve as an introduction to the philosophy behind shock-isolator design, summarize methods used and factors that must be considered in approaching specific design problems.

Shock-isolation mounts and cushioning devices are designed by various techniques. At one extreme, a package design may evolve by trial-and-error using paper or cardboard packing. At the other extreme, a concept may be stated and modeled mathematically. In this case a parameter-variation or synthesis technique is used to determine the shock isolation obtained from the given package environment. Each procedural extreme is useful; but, if a problem is to be solved realistically, several factors must be considered before a specific technique is chosen. These factors include:

- Generality of solution
- Fragility of system
- Environment
- Size and complexity of system
- Cost

Generality of solution, one consideration in choosing a design technique, may be achieved either by geometrically scaled or dynamically scaled variations of analytical or experimental models. The experimental technique can be used in designing the initial system; other results then can be scaled from this initial design. Since geometrical scaling may lead to a nonvalid design, however, some designs must be scaled dynamically. In such cases analysis is a valid and inexpensive technique; the formulation of a nondimensional model of the concept allows generation of design charts that are functional for all geometric variations evolved for one analysis.

Every component or system has a measure of failure or working strength of either long- or short-term duration. This measure of strength is commonly called the fragility level. Fragility, important in choosing a design technique, is used as a quantitative index of the strength of equipment subjected to shock and vibration. The quantitative index for fragility is expressed in terms of a system's acceleration response, $\ddot{x}$, and is referred to in terms of g, gravitational acceleration. The system being designed is modeled (fig. 18), and its acceleration response to a given shock environment determined through experiment or by mathematical analysis.

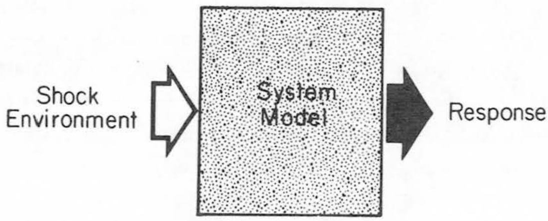

Figure 18.–Schematic model of a shock-excited system.

Although not as exact a technique as experimentation, mathematical analysis determines the shock and vibration response for a class of components. If the response exceeds the strength (i.e., fragility level) of a component or system, then there is a real danger of failure. In a mathematical analysis, the shock environment must be expressed in mathematical terms. The second method for determining whether a component exceeds its fragility level is by experiment, in which the component is subjected to a simulated shock environment, and a simple decision on failure or nonfailure is made. This method may pose a problem, however, because the test establishes the fragility level only for the component or system being tested.

Environment is another important criterion that must be considered in choosing a specific technique for solving a design problem; environment must be simulated. If environment causes a complex waveform in a shock-isolation device, for example, an experimental design may not be a valid one.

Size and geometrical complexity also are important criteria in choosing a method of shock-isolation design. Large isolation devices are not readily adaptable to experimental design because experimental equipment is not available. Scaling from large to smaller test devices is possible but must be approached with caution, especially if a design involves nonlinear materials, large size variations, or both.

Cost is probably the most important consideration in choosing a design method. The selection of a design process must be based on cost of the item to be isolated, reliability of the isolator, cost of the isolation design, and cost of isolation construction.

Because shock isolation is needed by both industry and the military, and because shock-isolation problems can be modeled and solved mathematically, new design techniques evolve continuously. Applications of the analog computer are described below. Shock spectra, optimum design techniques, design synthesis, and nonlinear material use are relatively new techniques that are discussed also with respect to shock-isolation design.

Analog Computer

Analog computers can be used effectively in the design of systems for shock and vibration isolation protection. The electrical simulation of a design concept for an isolation system can be programmed on the analog computer (fig. 19); the isolation system is then designed by varying the parameters in the circuit that represent stiffness, damping, and mass. In modeling a system concept for the analog computer, the principles and the simulation languages used in modeling problems for the digital computer are applied.

The analog computer relates the response of an electric circuit (current) and that of a mechanical or structural system (velocity) to some excitation. The circuits may be active or passive. Passive circuits consist of assemblies of inductances, capacitances, and resistances. The active components in active circuits, are dc amplifiers that electrically perform the mathematical operations required to solve the differential equations of motion for a mechanical system. The dynamic response of electric circuits is a model of, and is analogous to, the response of a mechanical system. The dynamic analogy for simulating the dynamic behavior of shock- and vibration-isolation systems is based on the observation that Newton's laws of motion for a mass correspond to the form of Kirchhoff's laws for electric circuits. The analog computer is ideally suited to the solution of ordinary differential equations and can solve nonlinear problems for which analytical techniques are nonexistent or too complex to be practical.

Shock Spectra

The generation of shock spectra for use in the design of shock-isolation systems was described under analytical techniques. Shock spectra are useful in design if the technique is recognized as a problem-oriented one. The maximum response of a single-degree-of-freedom system is plotted against its natural frequency for a particular environmental disturbance. An example of how a

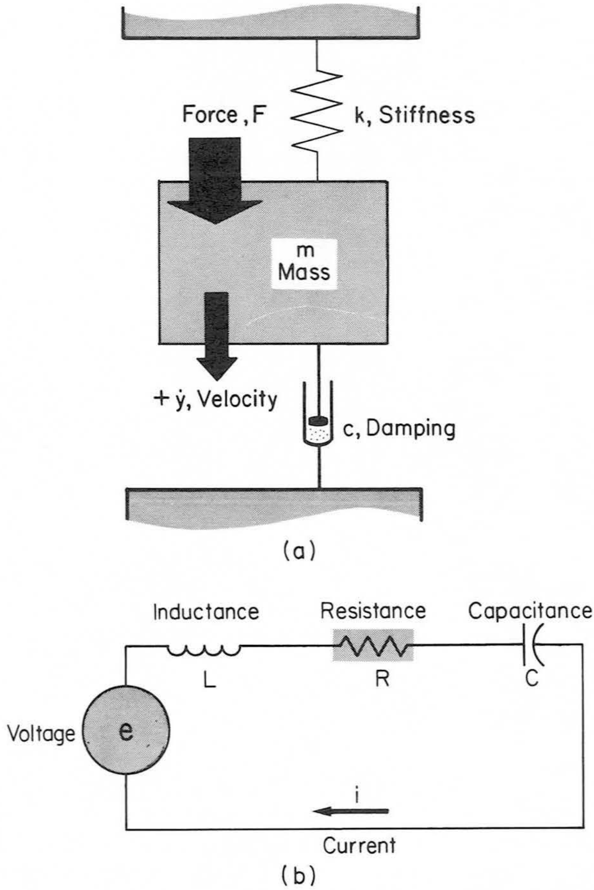

Figure 19.–Analog representation of a mechanical model. (a) Mechanical system; (b) electrical simulation.

simple packaging problem is handled with shock spectra is shown in figure 20 where a model of a package with contents of mass m (weight=ω), isolated by a material of stiffness k, is subjected to specified allowable drop heights to obtain displacement peaks. In designing the shock isolator of stiffness k, the displacement spectra are used; the maximum amplitudes, for the specified drop height and the package natural frequency are selected. It should be noted that the frequency is a function of the spring stiffness k. The maximum amplitude obtained multiplied by the spring constant yields the force on the isolated package, and, since

$$F = ks = ma ,$$

and the mass of the object is known, the g-level (a/g) on the isolated object is

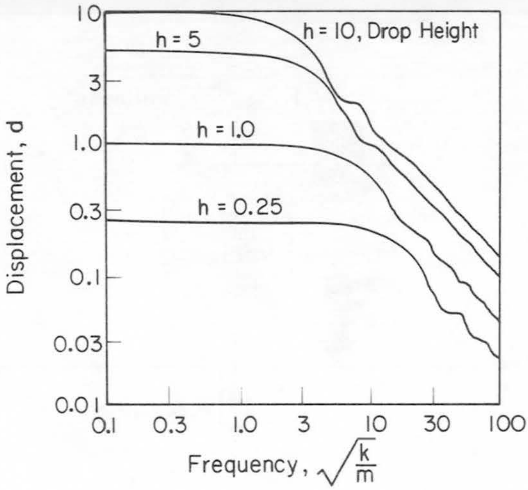

Figure 20.–Displacement shock spectra for package drop.

ks/w. If this value does not exceed the fragility level (damage criterion), then the selected spring is acceptable.

A method of design with shock spectra for nonlinear systems, proposed by Young (ref. 25), uses a nonlinear spring to limit peak displacement and acceleration to specified magnitudes. He states that it is sometimes advantageous to use nonlinear springs. For example, it may be beneficial to consider crushable materials, plastic yielding metals, or nonlinear softening springs; or it may be desirable to use a spring that is hardening under moderate loads but yielding under large loads for protection against strong shocks. The design procedure for applying nonlinear-system shock spectra is an extension of the linear procedure described previously. Detailed procedures for the design of nonlinear shock isolators using shock spectra are described by Young.

Optimum Isolator Design

The capability for designing the mathematical model of simple shock-isolation systems, combined with the motivation for optimum design within problem constraints and performance index, can yield fundamental information on shock-isolator synthesis. Eubanks et al. (ref. 26) reported techniques for optimum shock-isolation and absorber designs. The work of Liber and Sevin (ref. 27) was concerned with single-degree-of-freedom systems where the relative isolator displacement was minimized for a prescribed level of input acceleration attenuation. Both dynamic programming and linear programming were utilized to provide a discrete parameter version of the problem. It was found that the optimum shock isolator is active and has "bang-bang" control (i.e., constant force on or off). Results of optimum isolator performance were obtained in four types of input shock pulses. The conceptual design of shock isolators can be compared with the performance of any given candidate isolator or to the performance of the optimum shock isolator for a given environment and concept. The results of the study, expressed as a relationship between limiting values of relative isolator displacement and mass acceleration for the given input pulse [a maximum displacement (Dy) and/or a maximum acceleration (Ay)] shown in figure 21, permit direct determination of the improvement margin that exists between the optimum and any specified candidate isolator. The dependence of optimum isolator performance on the detail of isolator function and input shock pulse is discussed in reference 27. Results for the given input pulse are shown in shock-spectra form in figure 22.

Schmit and Rybicki (ref. 28) have studied simple shock-isolator synthesis with bilinear stiffness and variable damping. A one-degree-of-freedom system with a single package of mass m is protected from a multiplicity of shock pulses. Two common shock-isolator problems were considered. In one case a design was obtained that minimizes the absolute acceleration of a package subject to relative displacement limitations. In the second case, a solution was sought that would minimize relative displacement, subject to limitation on the absolute acceleration of a package. These design variables were utilized to

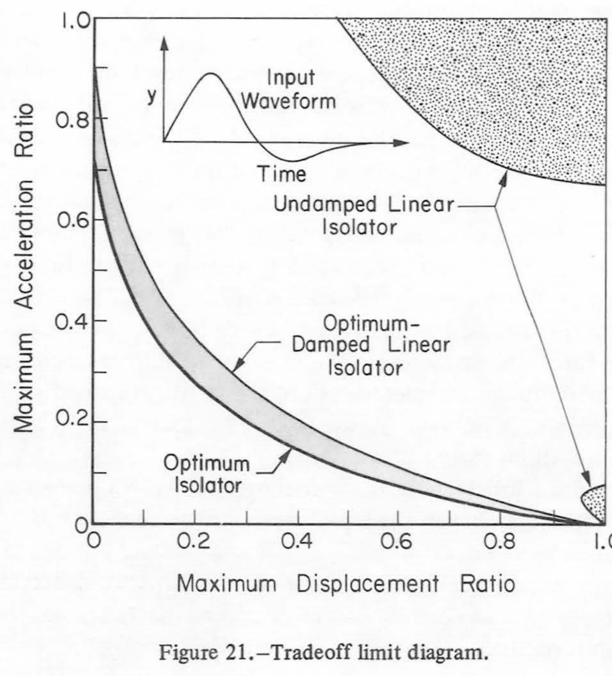

Figure 21.–Tradeoff limit diagram.

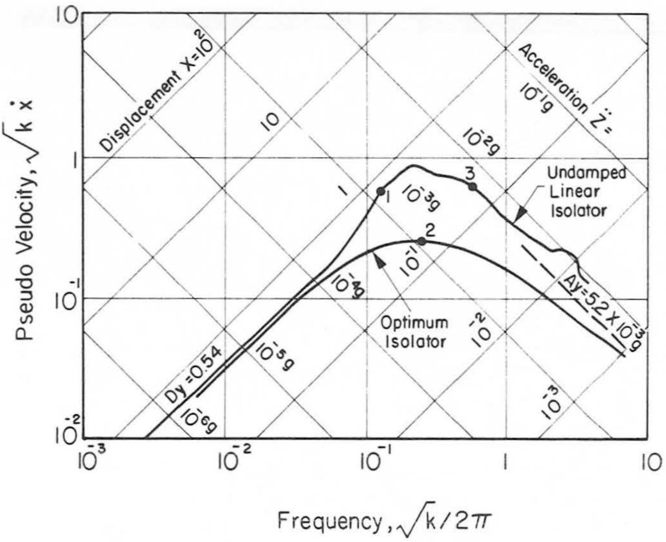

Figure 22.–Optimum- and undamped-linear isolator shock spectra.

characterize the bilinear spring; six additional design variables were used to represent a linearly variable damping coefficient.

In later, more general work on synthesis of dynamic systems, Sevin, Pilkey, and Kalinowski (ref. 29) utilized solution techniques based on linear, nonlinear, and dynamic programming to optimize a system of interconnected mechanical elements subjected to a time-dependent input shock pulse. This is a design-oriented computational scheme wherein some unspecified system parameters or input pulses may be determined. If the system parameters are specified fully, then the input pulse is specified only by a class description. If the input pulses are specified, then certain system elements or design parameters are characterized numerically. Finally, in some cases neither the input pulses nor the system elements are fully specified. In the problem solution, a single function of the state variables (i.e., description of the system's motion) is selected as an index of the system's performance, and a set of constraint functions, involving equality or inequality relationships among the state variables, design parameters, or both, may be prescribed.

From procedures described above (ref. 29), several criteria concerning shock-isolation systems can be determined:

- Extreme-disturbance analysis for finding bounds to a performance index for a given system when the input shock pulses are described as a class of unspecified waveforms.
- Optimum system performance for bounding a performance index for a class of inputs when certain of the system elements are unspecified and constraints on the system response are imposed.

- Optimum system design for identifying parameters that uniquely specify a system so that a performance index is minimized for the "worst disturbance" among a class of admissible pulses.

Nonlinear Isolator Design

Charts based on analytical studies for packaging design using linear isolation materials have been prepared by Harris and Crede (ref. 30). They have determined graphically the relation (fig. 23) between rise time and maximum acceleration at constant drop heights for a packaged item subjected to a half sinusoidal pulse. In addition, their charts provide information about the properties of cushioning materials. Included also are design techniques that utilize their charts.

Some materials available for use in packaging and discussed by Harris and Crede (ref. 30) are

- Latex hair
- Urethane foam
- Fibrous glass
- Polyethylene foam
- Creped cellulose wadding
- Reclaimed latex foam
- Preexpanded polystyrene
- Laminated corrugated fiberboard
- Pneumatic pockets in plastic

Mustin (ref. 31) has published dynamic shock isolation data for 16 nonlinear cushioning materials. He showed that shock isolation information needed by the designer could be reduced to two curves for each material plus a general

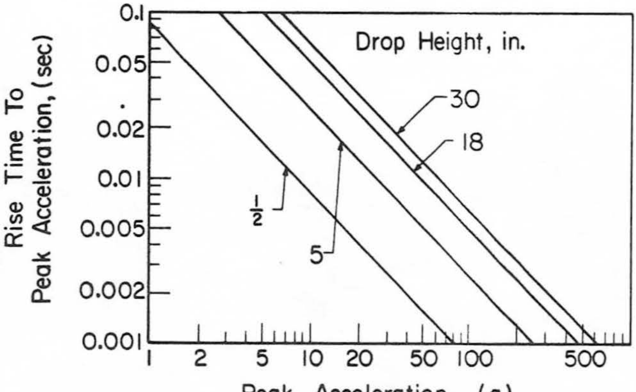

Figure 23.–Relation between rise time and maximum acceleration for a packaged system subjected to drop.

envelope applicable to all materials. This is an improvement over current use of an acceleration–static-stress curve for each thickness and drop height. Mustin's recent monograph (ref. 32) on the theory and practice of cushion design is a comprehensive guide to package design.

Ostrem and Rumerman (ref. 1) proposed an alternate approach to cushioning design. Regardless of whether a designer has access to statistical height of drop data, he must choose a material and determine the quantity required and the application method for a single drop height. Maximum acceleration-static-bearing stress curves provide the simplest method with general applicability. Figure 24 shows that these curves give the maximum acceleration of a package for a specified material and a specified height of drop as a function of the static bearing stress exerted on the cushioning [i.e., weight/bearing area (W/A)] and the thickness of the cushioning. The dynamic properties of most materials are virtually unaffected by temperature reductions down to some critical point; below this temperature, however, maximum accelerations greatly increase. For polyester urethane foam this critical temperature is 14° F.

The required thickness and the method of application must be determined concurrently for a given isolation material because the application method affects the bearing area, and therefore, the static stress. The three most commonly used application methods are complete encapsulation, side pads, and corner pads. These are illustrated in figure 25, which shows side views of cubical items in their outer containers. Side pads may allow the designer to use a smaller volume of material than would be necessary for complete

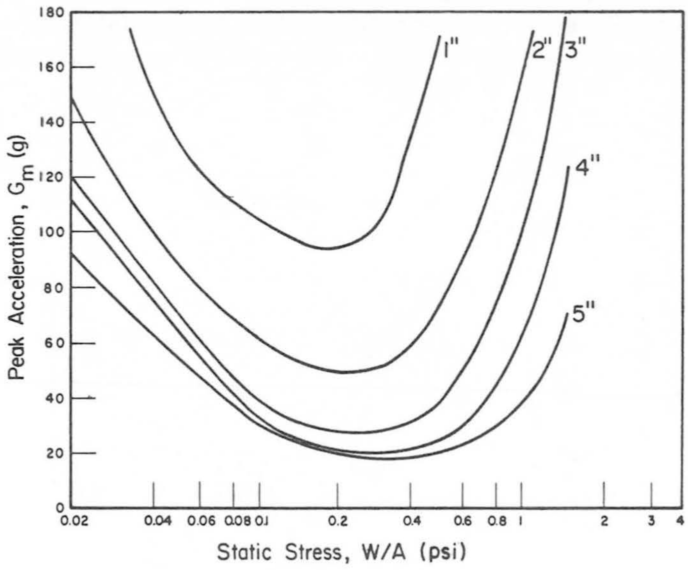

Figure 24.–Urethane foam polyester, thirty-inch drop height.

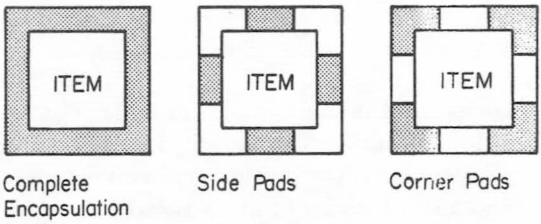

Figure 25.–Package design.

encapsulation. Care should be taken, however, to ensure that the pads are not so slender that they act as columns and buckle. If the pads do buckle, the item might rotate within the outer container and bump sharply against the interior walls. It has been shown that buckling will not occur if $A > 16d^2/3$, where A is the cross-sectional area of the side pad and d is its initial thickness.

Young (ref. 25) has shown how shock spectra for nonlinear spring-mass systems can be applied to design. He has established a method for presenting response spectra for a nonlinear system and indicated applications for analysis and design problems. Typical results have been given for nonlinear systems with cubic-softening and bilinear spring characteristics. Young does not deal with the methods (which may be analytical, numerical, or experimental) used to determine response spectra; it is assumed that the pertinent response data have been obtained. His discussion is concerned with methods for presenting the response in spectrum form to show both the effect of the system's parameters on the dynamic behavior of the system and the way this behavior differs from that of a linear system. He has considered also the use of spectra in practical problems of analysis and design (e.g., selecting a nonlinear spring to limit peak displacement and acceleration to specified magnitudes).

EXPERIMENTAL TECHNIQUES

The importance of experimental work in the design of components, equipment, and systems must not be underestimated. The purpose of any testing or experimentation should be defined clearly before any procedures are begun; the value of a series of tests is dependent upon good planning. The following procedures must be ordered and/or defined before testing is begun because many of them depend upon the results of others:

- Test objectives
- Test specifications
- Test procedures
- Test equipment selection
- Test instrumentation
- Calibration

- Test specimens
- Data reduction
- Data analysis

In a series of complicated procedures such as those given above, each step must be considered individually. This survey does not provide procedural details involved in experimental investigation; rather it outlines the procedures listed above as they apply to testing of shock isolation systems. New developments generated in NASA centers or by contractors are given.

Test Objectives

The reasons for performing a test should be understood clearly before it is carried out, whether it is for qualification, development, quality control, or trouble shooting. Since the objective of each test is different, the procedures involved in performing tests vary also. Thus, the objective of a test should be defined so that the correct data can be obtained. Most shock-isolation tests are designed to determine one or several of the following: (1) performance of a shock isolator (attenuation of an input pulse), (2) fragility of equipment or a component, and (3) response of equipment to a given environment. Trouble-shooting tests are more informal and cannot be so well documented as, for example, quality control tests.

Test Specifications

There are two basic methods for specifying a shock test: (1) the input shock pulse to which the item under test is to be subjected must be known or (2) the shock-machine selection, the mounting specification of the test item, and the procedure for operating the machine must be known. Although the first specification is preferred, the shock pulse can be used only when it can be defined simply. In addition care must be exercised in the simulation of mounting conditions. The second method is used largely when the shock pulse cannot be specified; then, special testing machines usually must be developed. Qualification tests and quality control tests usually fall into this category.

An example of an interesting torsional shock qualification test was specified by Davis (ref. 24). The implications of torsional shock on design of the Advanced Orbiting Solar Observatory (AOSO) were found by Davis in a design study. As a result, a qualification-level shock consisting of two separate applications of a transient pulse having several cycles at a discrete frequency was established. The pictorial representation of the torsional shock (fig. 26) has a maximum amplitude of 96.6 rad/sec^2; the sine wave frequency was specified as the resonant frequency of the spacecraft in the 60- to 70-Hz band. If the torsional-resonant frequency as determined by the vibration test was 69.9 Hz, then this frequency had to be used for the high-frequency wave of the torsion shock test.

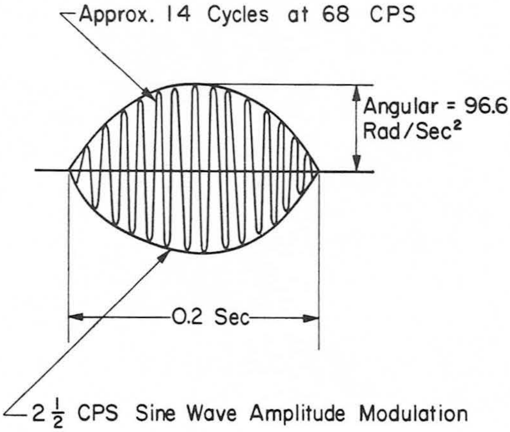

Figure 26.—Torsional-shock-qualification test pulse.

Two separate pulses were applied with a discrete time interval between them. Each pulse consisted of a high-frequency wave modulated by a 2.5-Hz sine wave whose angular amplitude was 96.6 rad/sec^2; the length of the pulse was one-half the period of the modulating sine wave. If a major torsional resonance of the observatory structure during the sweep test was indicated in the 60- to 75-Hz band, this resonant frequency was used as the high-frequency wave. If no resonance was indicated in this band, 68 Hz was used as the high-frequency wave.

Test Procedures

A shock test can be defined either in the frequency (independent variable) domain or in the response domain. The response domain is preferred since a transient response is obtained and limited cycles of response are utilized.

Accumulation of damage to equipment should be considered in the design of a system; the number of stress cycles in the life of a piece of equipment is equal to the number of cycles of vibration obtained from a single shock multiplied by the number of shocks per equipment life. This calculation is made under the assumption that the response of the previous shock has attenuated before the next loading occurs. An analysis must be made if the system is shock-loaded and the response of one shock pulse adds to the previous pulse.

Laboratory support conditions have an important effect on the simulation of environmental conditions. If the mechanical impedance (a measure of the structure's resistance to motion) of an actual mounting structure is high relative to that of the equipment mounted on the structure, then the equipment has negligible influence on the structure response. These environmental conditions can be duplicated in the test laboratory by hard mounting. Components or equipment mounted on a resilient foundation also must be simulated in the laboratory if valid test results are to be obtained. A heavy structure with the same mass but different stiffness changes the velocity shock spectra (fig. 27) of a high-impact shock machine. The solid and dotted lines represent different test machine mounting-table impedance (ref. 33).

Several recent additions to the test literature include Reed's (ref. 6) work on basic determinations of system damping. He used five representations of damping force in terms of their basic working mechanisms:

- Viscous
- Material-displacement
- Velocity-squared
- Material-viscoelastic
- General.

Reed calculated the energy loss per cycle of vibration to obtain a measure of system damping and refined this method to study transient damping. He determined the effects of response decay and of a constant, nonzero steady-state amplitude. Reed found that, for different damping forces, the energy loss per cycle has different powers of amplitude and frequency. Transient energy losses were considerably different from steady-state losses when the vibration decay was large and the frequency small. For damping forces dependent on amplitude, the mean response amplitude affected the energy loss per cycle.

Different damping forces yield different hysteresis loops. With an equation consisting of a series of damping terms it should be possible to approximate

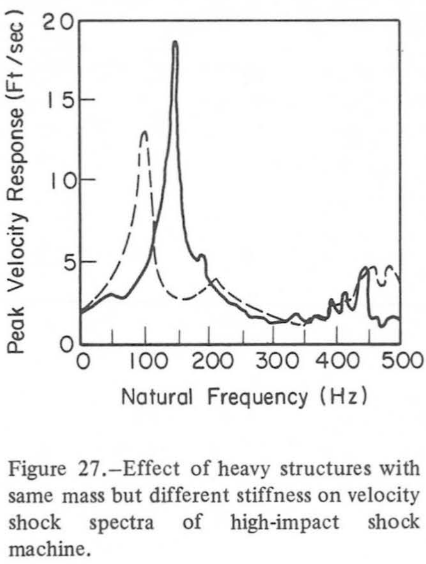

Figure 27.–Effect of heavy structures with same mass but different stiffness on velocity shock spectra of high-impact shock machine.

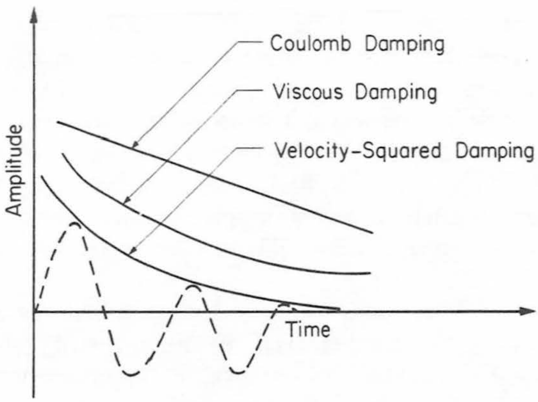

Figure 28.–System damping envelopes.

most experimentally determined hysteresis loops, some of which are dependent on mean-vibration amplitude.

Amplitude as a function of time-curve envelopes (fig. 28) shows the dependence of damping on displacement and velocity. In some cases, the differences in shape are very pronounced. A three-term exponential series solution of the system's nonlinear differential equation of motion can be used to obtain these curves and their envelopes.

It appears that a single, experimental amplitude-time envelope cannot be attributed to one form of damping without further investigation. In other words, the magnitude of the damping coefficient and the damping function may be simultaneously changed to yield curves that are very similar.

Reed describes procedures for determining the analytical representation of damping force by one term or a series of terms. Energy losses are described as a power series in amplitude and frequency, with coefficients determined experimentally. Finally the actual shape of the transient amplitude-response envelope is correlated with those generated by different analytical representations of damping.

Ruzicka et al. (ref. 8) describe the decay-rate method for determining the loss factor of viscoelastic shear-damped beam specimens. Repeated measurements of vibration decay on a structural member can be made in rapid sequence under the same conditions. An average of the data provides an accurate measurement of damping.

The experimental procedure for measurement of the structure-loss factor is based on a vibration test. The structural specimen is excited by harmonic vibration and, when a resonant frequency is located, allowed to attain a steady-state vibration. The cut-off frequency of the high-pass filter is set approximately at the resonant frequency. As part of the electronic switching function performed by the decay-rate meter, the excitation vibration is removed abruptly from the structure, and the ensuing vibration decay is sensed by the accelerometer. The accelerometer signal is processed through the cathode-follower amplifier, high-pass filter (which filters out high frequency components), and decay-rate meter. The decay-rate meter processes the signal through a logarithmic amplifier and generates a separate calibrated logarithmic decay signal. The structure-vibration-decay signal and the calibrated decay signal are alternately displayed on the oscilloscope on a repetitive basis, thereby allowing adjustment of the calibrated decay signal to match the vibration.

The method of determining damping by logarithmic decay curves was employed by Hanks and Stephens (ref. 9) in their study of damping in a practical structural joint. The total damping of the system was measured at atmospheric pressure for a range of joint clamping pressures (by varying bolt-tightening torque) and beam-tip amplitudes. For a particular clamping pressure, the beam was deflected manually and released to oscillate in the first cantilever-vibration mode. Oscillations of the beam were sensed by an electrical strain gage attached to one side of the beam. The strain gage was coupled, through an amplifier, to an electronic damping meter. The damping meter counted the number of cycles during amplitude decay between preset limits; this provided a measure of a system's damping.

Since the damping was measured over a displacement of the decay envelope, the logarithmic decrement was specified at the average amplitude. Measurements were made at several amplitude levels for each bolt torque by varying the triggering voltage of the damping meter. In all tests, sufficient initial deflection was given to the beam to allow transients to die out before the triggering amplitude of the damping meter was attained.

Test Equipment and Instrumentation

The nature of a shock test machine depends on the test objective and the environment that is to be simulated. The damage potential of a shock pulse simulated by a shock test machine is dependent on the nature of the equipment subjected to shock, as well as the shape and intensity of the shock pulse. A comparative measure of the damage potential of shock pulses simulated by a shock testing machine can be obtained by comparing the response of the testing machine to that of a test standard; thus shock spectra of the shock testing machine must be examined. Shock pulses applied to the test machine must be reproducible, and calibration of the shock test machine for a given test must be possible. Harris and Crede (ref. 34) report various shock machines and their characteristics. Figure 29 is a schematic diagram of an electrohydraulic closed-loop system that can provide a simulated environment to an item being tested. The input shock pulse must be described when this machine is used to test equipment. The Navy's high-impact shock machine (fig. 30) is an example

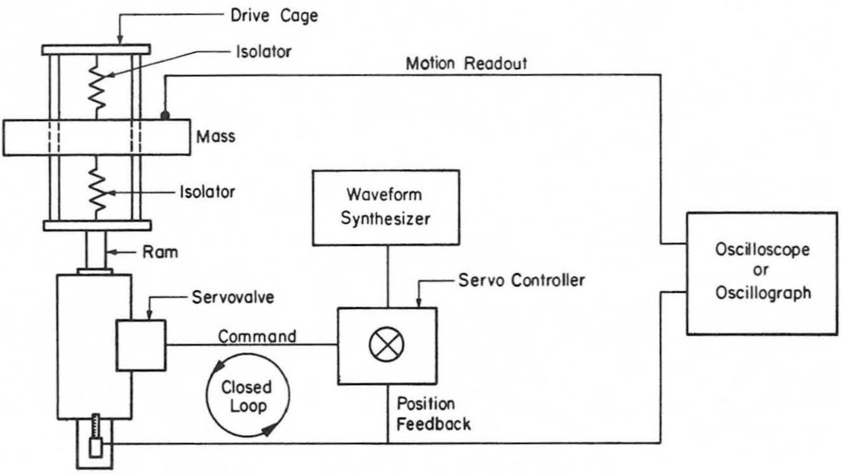

Figure 29.–Block diagram for closed-loop isolated-mass test system.

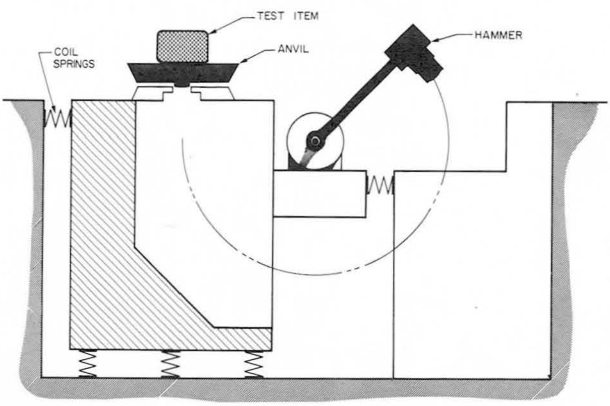

Figure 30.–Schematic of high-impact shock machine for medium weight equipment.

of a semicontrolled shock test machine. It is assumed that the test specimen will survive an actual shock environment if it survives simulated shock environments. Such experimental results, establishing a correlation between simulated environment (by machine test) and actual environment, provide information about the fragility of the test item.

Experimental instrumentation has been well developed and consists of three stages. At the first level, a transducer converts mechanical motion into electrical signals. Transducers consist of strain gages and capacitance pickups to measure displacement, coils in a magnetic field to measure velocity, or piezoelectric transducers to measure acceleration. At the second stage, the signal from the transducer is prepared by secondary or modifying instruments for the third stage, the display of the signal on a cathode ray oscilloscope paper, a recorder, or magnetic tape. Modifying instruments amplify, attenuate, integrate, and condition the signal. A typical instrumentation setup is shown schematically in figure 31.

An example of an experimental system for measuring the loss factor of viscoelastic shear-damped beam specimens (ref. 8) is shown in figure 32. The structural specimen is supported vertically by a string suspension. A small driver coil cemented to the specimen is placed so that it adds a minimum amount of stiffness or weight and allows centering of the driver coil within the magnetic housing of the electrodynamic exciter. The electrodynamic exciter, which provides a linear magnetic field for the driver coil, is driven by the harmonic oscillator through a power amplifier, and can deliver 25 W of power to a beam specimen for extended periods of time at a maximum linear peak-to-peak displacement of 1/2 in.

The response of the beam specimen is detected by an accelerometer mounted near the end of the beam; a counterweight of equal magnitude is

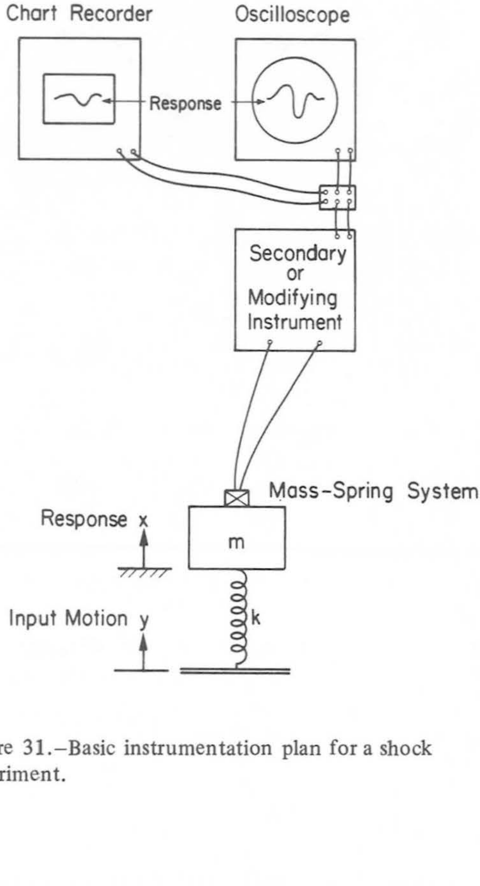

Figure 31.–Basic instrumentation plan for a shock experiment.

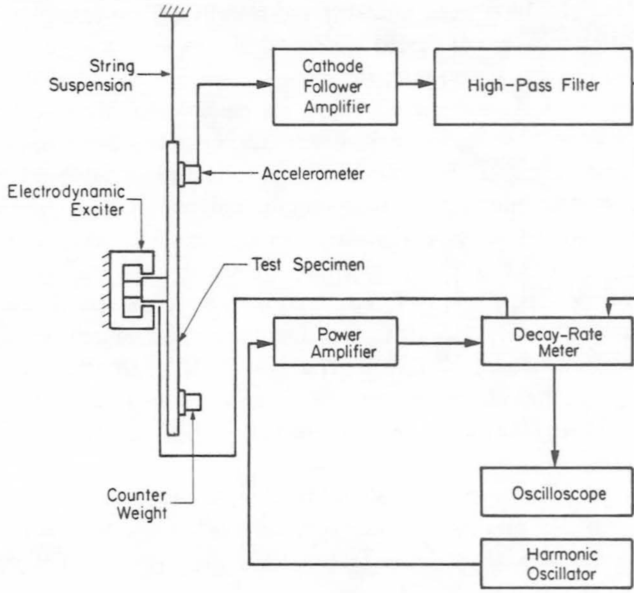

Figure 32.–Experimental system for measuring the loss factor of viscoelastic shear-damped composite structural beam specimens.

mounted on the opposite end of the beam for balance. A high-pass filter is used to reject all frequencies less than the resonant frequency at which the loss factor is being measured. The decay-rate meter provides an electronic switch between alternating functions: (1) the processing of the signal from the high-pass filter through a logarithmic amplifier and (2) the generation of a calibrated logarithmic decay signal representing the beam vibration and the calibrated logarithmic decay signal.

Test Specimens

When possible, an actual component or piece of equipment should be tested. If the test is a destructive one, a replica of the actual device should be tested. If the available test machine is too small to allow a full-scale test, then a subscale model of the prototype should be tested. Although it need not have the same appearance as the prototype, a scale model must behave in a manner similar to that of the prototype. If this behavioral relationship is not relatively simple, it will be more economical to determine the behavior of the prototype by analytical study rather than by scale-model testing.

The use of scale models to determine dynamic behavior of large prototypes is a challenge. NASA has conducted many experimental investigations in which large-scale vehicles have been modeled successfully. These cases, usually found in the vibration area, are discussed in chapter 3.

In scaling for shock response, the dynamic characteristics of the prototype must be preserved. Two types of scaling are possible, i.e., dimensional analysis or nondimensional mathematical system formulation. Dimensional analysis allows selection of pertinent system variables and, through algebraic procedures using fundamental units (time, mass, length, and temperature), determination of nondimensional scaling relationships. In the second scaling technique the dynamic behavior of a shock-isolation system is described by differential equations of motion. These differential equations are converted to a form that is independent of time, size, mass, and temperature (nondimensional) by using defined nondimensional variables. The equations of motion are arranged algebraically so that the nondimensional scaling parameters can be selected from them. These techniques are documented by Murphy (ref. 35) and Harris and Crede (ref. 36).

Regier (ref. 37) reports the use of scaled dynamic models in the design of aerospace vehicles and shows the aerothermoelastic interactions (fig. 33) involved in model scaling. Scale factors with their physical descriptions are given by Regier.

Data Reduction and Analysis

Test data may be produced in various forms, depending on the display or recording instrument used. The oscilloscope provides a visual display, and data can be recorded immediately (with a camera) or rejected. Oscilloscopes are best

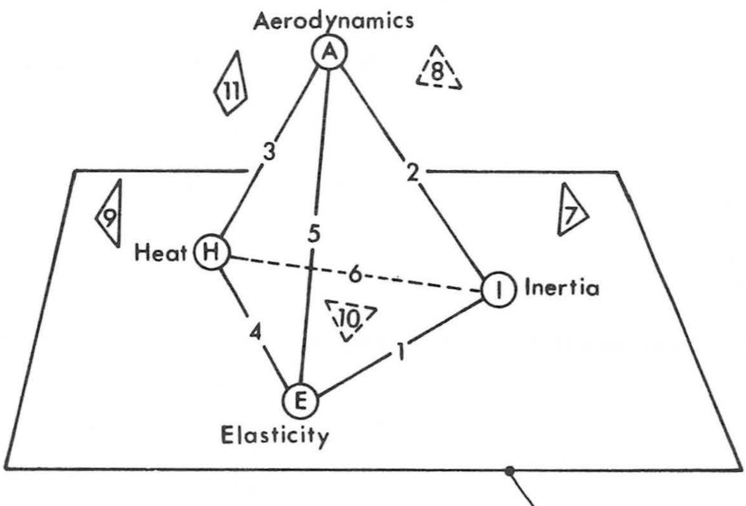

Figure 33.–Tetrahedron illustrating aerothermoelastic interactions.

for cases that involve scanning a large number of test parameters. Data can be recorded directly on a strip recorder or on magnetic tape. Regardless of the method used to obtain it, shock-response data usually must be reduced and analyzed, either by sight, mechanical means, or electronic filtering.

The objectives sought in data reduction are the removal of all nonessential information, the retention of essential data, and the abstraction of the history of some parameter (usually acceleration) in a concise and useful form. The characteristics of the function that best describes the data should be determined, and the reduced data should be manipulated so that they can be correlated with other significant parameters. The form of the reduced data can be displayed either in the frequency- or response-domain figure.

Data reduction in the frequency domain is one of many sophisticated methods for harmonic analysis of data waveforms. Harmonic analysis is carried out by frequency separation either mechanically, by computer, or electronically; such data reduction in the frequency domain yields a shock spectrum. Data reduction in the response domain is confined to identification of response peak magnitudes and response wave shapes for given environmental shocks. Specific effects resulting from an input pulse can be found by varying the system's properties systematically. In linear systems, response-domain data can be converted to frequency-domain data, and vice versa with some loss in detail of the response-domain data.

SHOCK ABSORBERS AND ISOLATORS

The shock absorber and shock isolator are vital to the successful design of any shock-isolation system; this section is devoted to new concepts, many of which have been reviewed and developed as a result of NASA's Lunar Module. The utility of a shock absorber depends upon its specific energy absorption (SEA), i.e., the energy absorbed divided by the weight of the material deformed and its geometrical configuration. A measure of the value of a shock isolator is its attenuation of a shock pulse (force attenuation). The isolator that meets this requirement must have a long stroke so that the short-time-duration high-amplitude shock pulse can be attenuated over a long period. A shock isolator should not be confused with a short-stroke vibration isolator or vice versa. A shock isolator transforms little energy into heat; rather, it stores energy temporarily.

Recent advances and new concepts have been primarily in the shock absorber field. The conventional classes of shock isolators (e.g., helical springs, pneumatic springs, liquid springs, and elastomeric mounts) are not discussed here because they now have universal acceptance in industry. Shock absorbers can be classified conceptually. The general classes are discussed below, and more detailed discussions are presented on several recent shock absorber designs. Kiker (ref. 38) has studied shock absorbers with respect to their application to manned spacecraft.

Materials Deformation

Many absorbers have been designed to take advantage of the deformation of material. Some, like the crushing of honeycomb and balsa, function almost entirely on the principle of plastic deformation of material. Foams usually are not loaded sufficiently to destroy them; in addition they may have a multi-cycle capability. Liber and Epstein (ref. 39) have shown analytically and experimentally that foam is an ideal shock isolator. They have generated a basic mathematical model that includes viscoelastic and air effects and have shown how basic foam properties can be characterized experimentally. Other material deformation absorbers (e.g., frangible tube extrusion, frangible tube fragmentation, crushable metal tube, wire drawing, and metal cutting) are combinations of other basic mechanisms. All these absorbers depend on friction in addition to material deformation.

Fluid Compression

Pneumatic springs, and liquid springs, in which the flexibility of the isolator depends on the compressibility of the liquid in the spring, are commercially available. The oleo gear, which is used by the aircraft industry, absorbs energy by fluid flow through an orifice and is velocity sensitive. The flexibility of the liquid allows this absorber to be as functional as desired. Large air bags are a stable, simple absorbing mechanism if space is available.

Mass Acceleration

Controlled acceleration of a massive object provides a mechanism for shock isolation. This type of isolation usually is designed exclusively for a given task and requires design ingenuity. In addition, an analysis of the process is a requirement in sizing the parts.

Friction

Any absorber that depends on friction is adequate as long as it gives reproducible, predictable results and is reliable. Many material deformation concepts depend on friction. Coulomb friction devices that have brake shoes are efficient absorbers; however, if the shock stroke is too long, the temperature rise in the shoe may be prohibitive. In addition, the return stroke of the absorber must be ensured.

Pneumatic Bag

The pneumatic bag used successfully in the Mercury Space Program (fig. 34) is practical and compact even though it is inefficient. These nylon or glass-

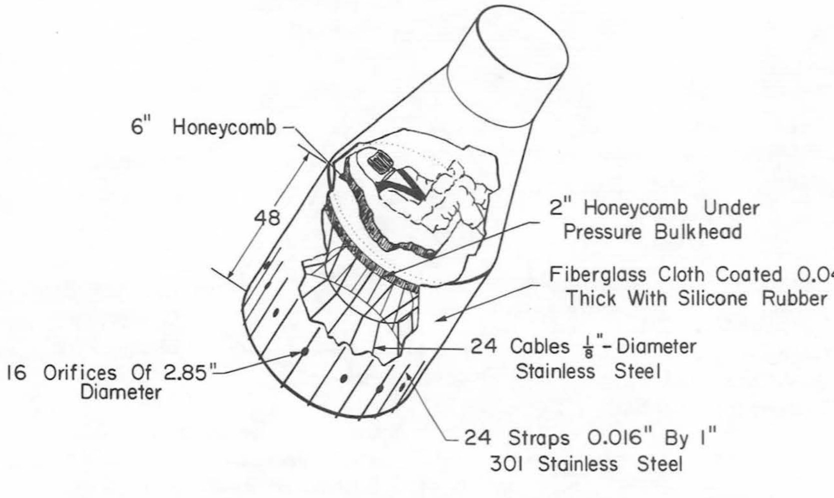

Figure 34.–Mercury impact-attenuation system.

fabric coated rubber bags are fitted with an orifice for exhausting air during the compression stroke. The bag, suspended beneath the load, begins to build pressure upon contact with the surface, and air is exhausted through the orifices. The orifices may be fitted with rupturable diaphragms that break at a predetermined pressure. The bag acts like a nonlinear spring during the first part of the stroke.

Honeycomb Shock Struts

Honeycomb shock struts were used in the Apollo capsule to protect the crew (fig. 35). These struts provide excellent shock isolation because of the large strokes (14 in. in one direction). Model and full-scale tests were conducted to determine impact dynamics, including onset rates, maximum accelerations, and vehicle turnover characteristics.

Filament-Wound Toroids

A study was conducted by MacNeal and Loisch (ref. 40) on the use of filament-wound toroids as pneumatic shock absorbers. The pneumatic shock absorber has a toroidal rubber bladder covered by high-strength structural filaments; the load is applied parallel to the polar axis. The concept (fig. 36) reveals that structural fibers carry meridional tension, while rings on the polar circles carry hoop tension. In other concepts the structural fibers carry both meridional and hoop tension by virtue of the helix angle used in the construction. For large motions, a stack of filament-wound toroids provides a long

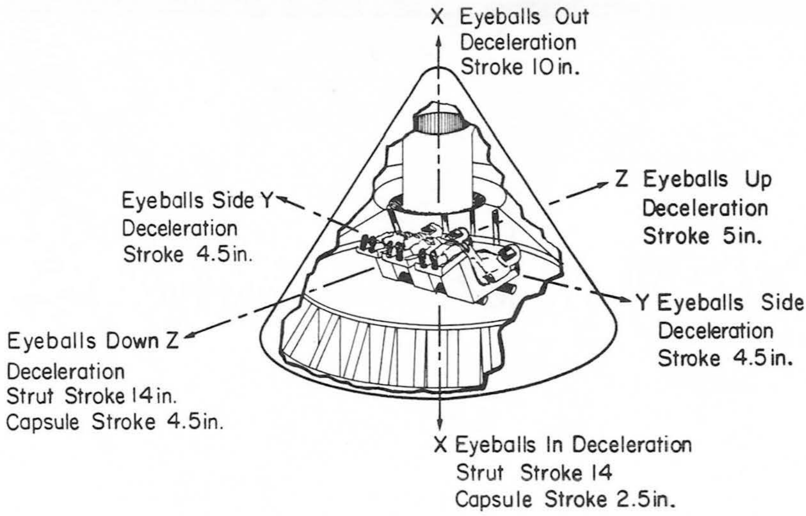

Figure 35.–Apollo attenuating system.

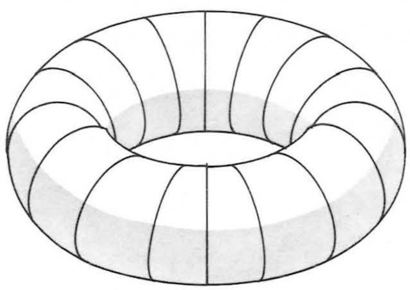

Figure 36.–Filament-wound toroid.

stroke. In this study, expressions for specific energy absorption under isobaric and adiabatic conditions were derived and compared with like systems. It was found that the specific energy absorption of the absorber is large compared with competing systems. Specific energy absorption can be increased substantially by using nylon fibers filled with light gases, particularly helium. Deflections due to lateral load can be accurately predicted. Mathematical expressions and design charts yield the necessary information to design filament-wound toroidal shock absorbers.

Reversing Aluminum Tube

The concept and the design equations for the reversing aluminum tube (fig. 37) were developed by Warner and Marble (ref. 41). In this shock absorber, energy is absorbed by plastic bending, plastic compression, plastic shearing, and

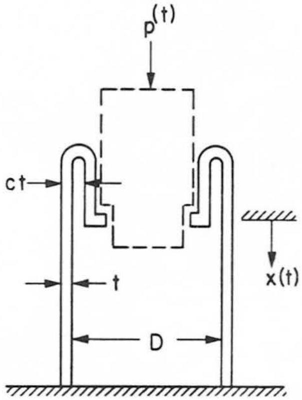

Figure 37.–Reversing of aluminum tube.

friction. The reversing aluminum tube is a one-cycle absorber since plastic material deformation is the energy-absorbing mechanism.

The total energy for deformation of the tube is used to determine the specific energy absorption of the absorber. It is a function of (1) effective yield stress, (2) material density, (3) wall thickness, and (4) tube inside diameter. Analytical results for 3003-H14 aluminum tube are compared to experimental ones in this report. These results are then compared with hexagonal honeycomb for various wall thicknesses and honeycomb cell sizes.

Hexagonal Honeycomb

The hexagonal honeycomb was investigated by Warner and Marble (ref. 41) as an energy-absorption mechanism for the Lunar Module. The theoretical relationship for the crushing stress of hexagonal cell structures to axial loading were developed by McFarland (ref. 42). A series of experiments was carried out to verify the analysis that provides good upper and lower bounds on the crushing stress. In figure 38 the partially crushed hexagonal cell structure shows shear failure. Another aluminum energy absorber called Trussgrid has been developed by Lewallen and Ripperger (ref. 43) for a single-cycle cushioning material.

Expansion Tube Absorber

Figure 39 shows the expansion-tube shock absorber developed by Hodgins (ref. 44) which is a tube through which a mandrel is pulled, expanding the tube. Energy is absorbed through material deformation by expansion of the tube and by friction between the mandrel and tube. The objective was to design

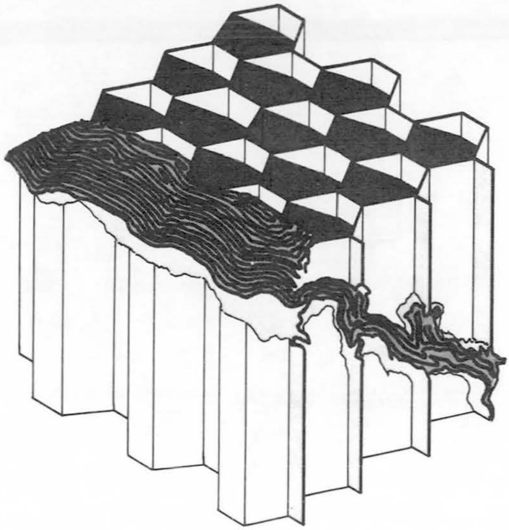

Figure 38.–Partially crushed hexagonal cell.

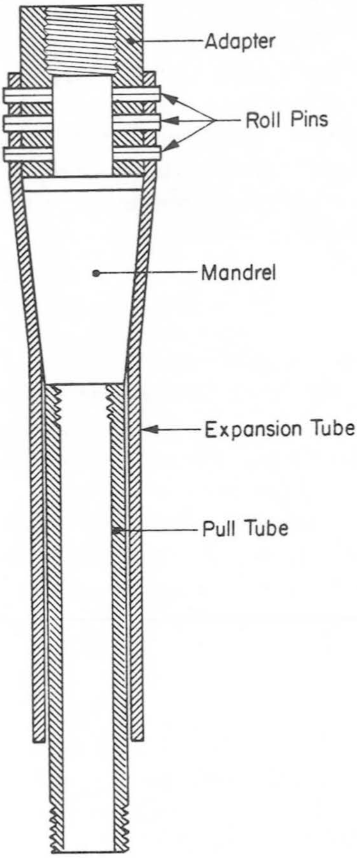

Figure 39.–Expansion-tube shock absorber.

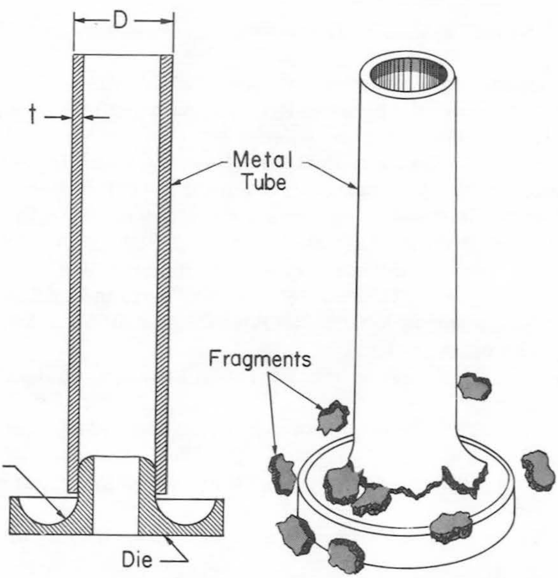

Figure 40.–Sketch illustrating fragmenting process.

an energy absorber of this type with a resisting load of 20 000 lb throughout the stroke.

Frangible Metal Tubing

A highly efficient energy-absorption process, employing frangible metal tubing as the working element, was investigated by McGehee (ref. 45). The fragmenting process (fig. 40) absorbs energy through the force developed when a frangible tube is pressed over a die. The die is shaped so that the portion of the tube in contact with the die is split into segments; the segments are broken into small fragments. A fluctuating force is developed by the fragmenting. The breaking and dispersing of the tube segments permit the entire length of the working element to be employed as the working stroke. The load-deflection characteristics for frangible metal tubing were developed experimentally.

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Vibration Technology

Excessive mechanical and structural vibrations constitute one of industry's most costly problems. Human annoyance and poor mechanical performance provide more-than-adequate justification for developing new technology applicable to vibration control. Electrical systems especially are sensitive to vibration phenomena. Because space equipment must be highly reliable, NASA has emphasized the development of practical design procedures to control or eliminate vibration.

Vibration technology has grown immensely in the last 10 years as a result of the space program. NASA's contributions to vibration in electrical systems are surveyed on the following pages. Vibration fundamentals are discussed in the first section. Vibration environments for transportation, manufacturing, and consumer equipment are then reviewed. Environmental vibration data gathered on NASA projects are reviewed and referenced. Although environmental vibrations occur randomly in nature, periodic trends often can be modeled.

Analytical techniques and related mathematical analyses, as they apply to industrial design, are reviewed in the next section. Damping data, essential to vibration analyses, have been gathered in many NASA studies and are discussed with respect to their functional mechanisms and mathematical representations. Results available on calculations of natural frequencies for beams, rotors, structures, and sloshing liquids are discussed with reference to fundamental calculating methods and for experimental comparisons. Methods are reviewed for calculating the vibration response of complicated structures (e.g., rotors, bearings, beams, structures, vibration isolators, and absorbers) to environmental loading. Some of the finest technical work on vibration is available in these referenced documents.

NASA has developed experimental techniques employing scale-models in place of full-scale testing. The techniques, which are presented in "Experimental Techniques," involve the testing of subscale structures for response, and can be used to set up experiments in allied fields. Qualification tests for components and damping-determination tests also are discussed.

The last major section in this chapter is concerned with vibration absorbers and isolators that have evolved from NASA projects. Mass absorbers that rely on a mass vibrating out of phase with a system's motion have been a major NASA concern. Elasto-plasto-viscous dampers, foams, low-frequency active-suspension systems, and pneumatic devices represent other conceptual and developmental advances by NASA in vibration isolators.

The literature surveyed in this chapter reflects vibration technology developed by NASA. There is every reason to believe that industrial firms can apply this technology to their own use.

FUNDAMENTALS

All systems and components fail to some degree, regardless of whether the failure is concerned with performance or overall life. The measure of strength (as with shock phenomena) is referred to as the fragility level. The term fragility is used as a quantitative index of the strength of equipment subjected to shock and vibration. The fragility index, expressed in terms of a system's acceleration response $\ddot{x}$ is referred to in terms of g, i.e., gravitational acceleration. The mode of failure often is called fatigue. Fatigue failures occur as a result of the vibration of a system or its components through a finite number of stress cycles. If the frequency and magnitude of the stress cycles exceed the fragility limit, fatigue-failure results.

Mathematical analysis, experimental analysis, or both are used to determine whether a system meets design specifications. Familiarity with the fundamentals of vibration analysis is necessary to understand the material presented in this chapter; therefore, a single-degree-of-freedom isolated system (fig. 41a) is described. The symbol m denotes the mass of the isolated system; the isolator's stiffness and damping are described by k and c respectively. The response x to the environmental vibration input q is a measure of the system's fragility. The relative displacement of the system is expressed by $q - x$. The environment q is described mathematically by some magnitude q_0 and frequency Ω. Mathematically, the system is described by the equation of motion

$$m \frac{d^2x}{dt^2} + c \frac{dx}{dt} + kx = kq_0 \sin \Omega t + cq_0 \Omega \cos \Omega t.$$

A dimensionless response plot (fig. 41b) represents its solution. The response ratio y/q is the ratio of input motion to system response. The natural frequency, which is a function of the system mass and elasticity, has special significance in the design of systems:

$$\omega_n = \sqrt{\frac{k}{m}}.$$

The frequency ratio is the ratio of the system's natural frequency to its input function frequency, Ω.

The response plot shows that the magnitude of the response of a system whose natural frequency equals its forcing frequency increases as the system's damping c decreases. This is called resonance and must be avoided unless sufficient damping is present in the system to control its response. The damp-

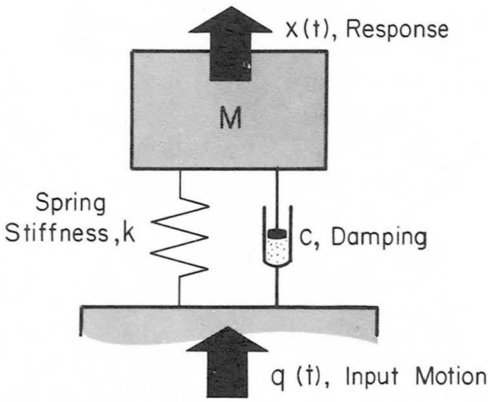

Figure 41a.–Mathematical analysis of a single-degree-of-freedom isolated system–mathematical model of shock-mounted resistor.

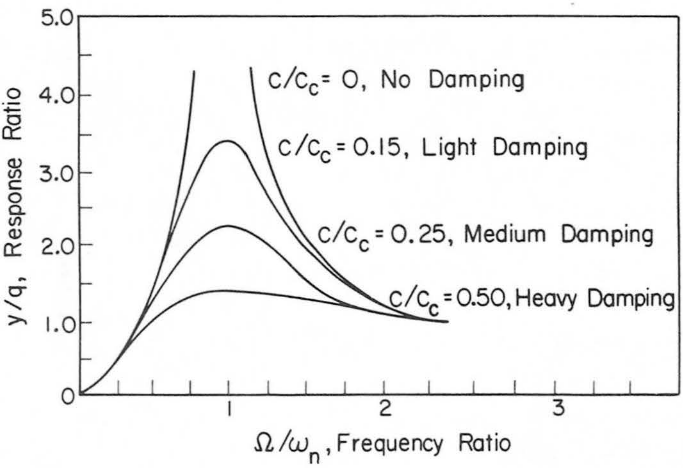

Figure 41b.–Mathematical analysis of a single-degree-of-freedom isolated system–response plot.

ing ratio c/c_0 is a measure of a system's damping; c represents the actual damping constant and c_c the critical damping constant. The critical damping constant is a function of the system's mass and elasticity:

$$c_c = 2\,\omega_n\,m\,.$$

In addition to the mathematical procedure summarized above, experiments can be used to determine whether system response exceeds the fragility level of a component. Dimensionless parameters such as frequency ratio, damping

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ratio, and response ratio may be used to design a scaled experiment if a full-scale model cannot be tested. Properly designed experiments can be used to obtain general design data on response and failure resulting from simulated vibration environments; the 1/40-scale dynamic model of the Apollo-Saturn V umbilical tower configuration (fig. 42) is an example. NASA has refined vibration testing techniques through the use of similar dynamic models.

ENVIRONMENTS

The environment of a piece of equipment or a component usually is a major consideration in its design and development. Specified vibration environments can be used in conjunction with analysis and experiment in the design process. A vibration environment is a disturbance–displacement, velocity, acceleration, or force–whose time duration is infinite and whose magnitude is either periodic or random.[6] Periodic phenomena, which are more often labeled steady-state vibrations, can be subdivided into harmonic[7] and nonharmonic classes for mathematical description. Examples of periodic environmental forces $F(t)$ are shown in figure 43.

Nonharmonic periodic phenomena can be represented in a series of harmonic functions by using a Fourier series expansion (a linear combination of sine and cosine functions). The components of the series have frequencies that are multiples of the fundamental period of the harmonic function. In the case of a convergent series, the amplitude of each component decreases as its frequency increases. When A_n, B_n, and τ are sized properly, $F(\tau)$ represents the disturbance (fig. 44) resulting from a faulty gear on a rotating drive shaft:

$$F(\tau) = \sum_{n=1}^{\infty} A_n \cos n \pi \tau + \sum_{n=1}^{\infty} B_n \sin n \pi \tau .$$

System response to random disturbances has become important as the demand for better system performance has increased. Vibration environments are handled mathematically with infinite combinations of periodic functions considering frequency spectra. Fortunately certain properties of random functions (fig. 45) can be handled statistically. It is possible to predict the probability of finding the instantaneous magnitude of vibration environment within a certain force range F to $(F + \Delta F)$. Mean and mean-square values are obtained by computing averages. Fourier series analysis is used to determine the frequency content of a vibration environment. Procedures for predicting random vibration in modern flight vehicles have been summarized by Barnoski et al. (ref. 1).

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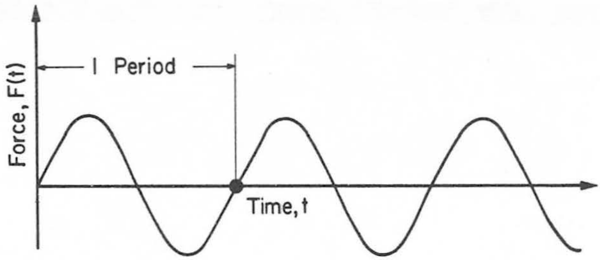
[6] By random is meant any relationship between the magnitude of quantity and time, measured during a certain time interval, that will never exactly recur.

[7] In mathematical terms, a harmonic is some form of a sine function.

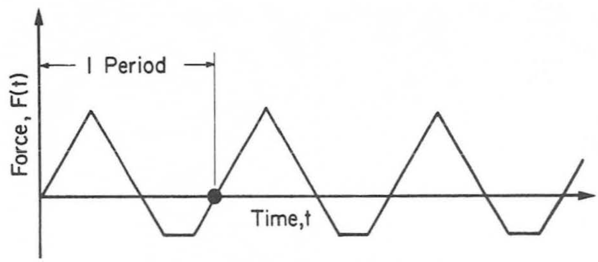
Figure 42.–One fortieth scale model of the Apollo-Saturn V space vehicle.

Electronic equipment is subjected to many types of vibration environments. These have been classified for use in design situations similar to those where shock loadings are obtained (i.e., transportation, manufacturing, and consumer equipment).

Schock and Paulson (ref. 2) have published a survey of shock and vibration environments in four major modes of transportation–aircraft, railroad, ship, and truck. Information and test data pertinent to this transportation survey are the results of a NASA study of transportation and handling; this survey includes an extensive bibliography of the three hundred reports reviewed. All information and test data were collated, analyzed, and combined to provide a unified reference source of transportation environments. Environmental data were characterized in harmonic form–in acceleration as a function of frequency envelopes (fig. 46)–to permit presentation of varied data on a single



(a) Harmonic



(b) Nonharmonic

Figure 43.—Periodic phenomena.

graph or chart. Although data lose detail in such a presentation, they retain their utility for analysis, experiment, and design. To maintain a reference for acceleration levels, response levels were accepted only if given at the cargo floor. Vibrations in the cargo area were found to have dominant frequencies for particular vehicles, loads, locations, and speeds. Even though conditions and circumstances varied, results showed that vibration levels were relatively constant over wide frequency ranges.

Extensive shock and vibration measurements on a variety of aircraft are reported in reference 2; these measurements were carried out by the Wright Air Development Division, Wright-Patterson Air Force Base, during normal service conditions—taxi, ground run-up, take-off, straight and level flight, turns, descents, landing, and landing roll. The maximum acceleration envelopes, which vary from 0.1 to 8.0 g between 10 and 100 Hz are given for several aircraft in this report.

Data descriptive of railroad vibration environments are classified in a category called over-the-road operation. Transient vibrations (not including coupling) that occur during starts, stops, slack run-outs, and run-ins have been separated. A single plot for standard draft-gear and soft-ride equipment (a combination air- and coil-spring system in the vertical direction and a pendulum system with snubbers in the horizontal direction) includes data from

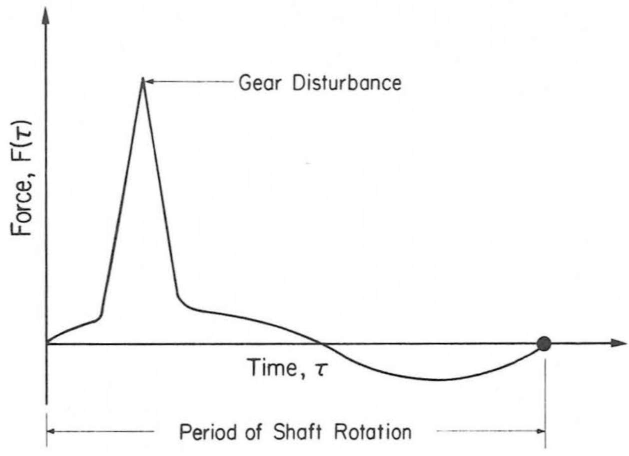


Figure 44.–Faulty gear environment.

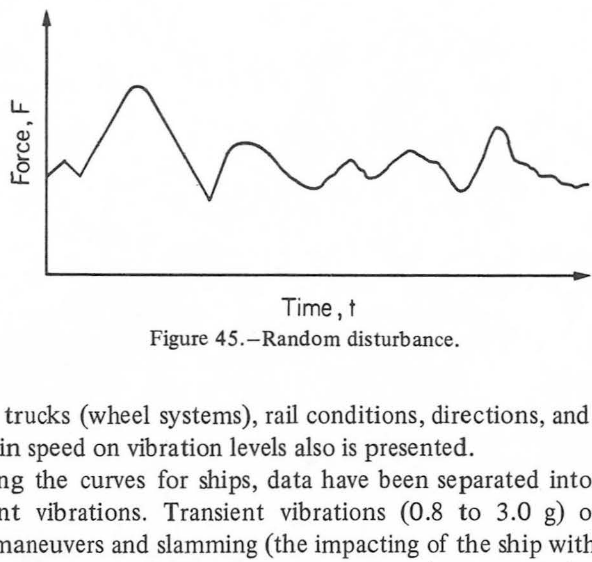


Figure 45.–Random disturbance.

all types of trucks (wheel systems), rail conditions, directions, and speeds. The effect of train speed on vibration levels also is presented.

In plotting the curves for ships, data have been separated into continuous and transient vibrations. Transient vibrations (0.8 to 3.0 g) occur during emergency maneuvers and slamming (the impacting of the ship with water after the bow has left the water). Continuous vibrations (0.05 to 1.0 g) occur during normal operations, including rough sea environments.

Environmental vibration data for trucks include peak acceleration values resulting from travel over rough roads, ditches, potholes, railroad crossings, and bridges. The lower curve (fig. 47) describes the vibration environment from paved roads. The effect of cargo load on vibration environment is given also in this report. Vibration data for trucks were obtained from tests conducted with

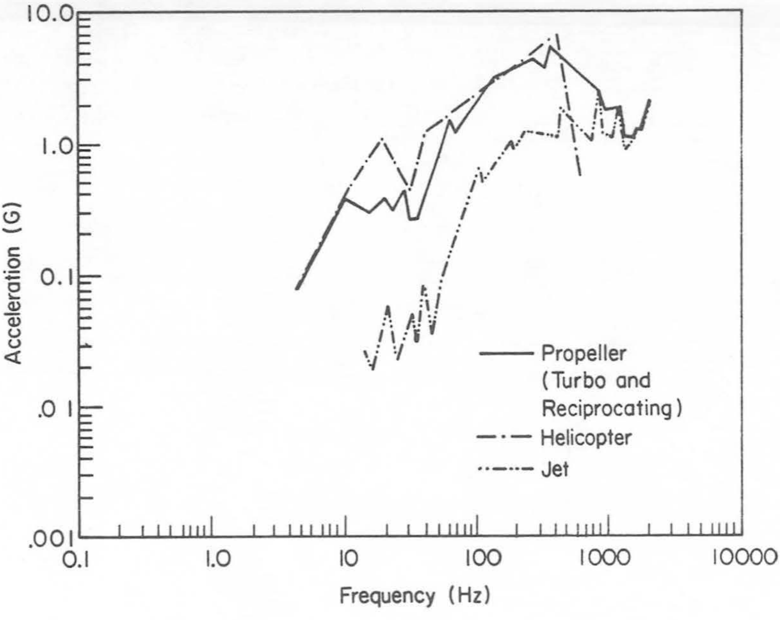


Figure 46.–Acceleration as a function of frequency (envelope for aircraft).

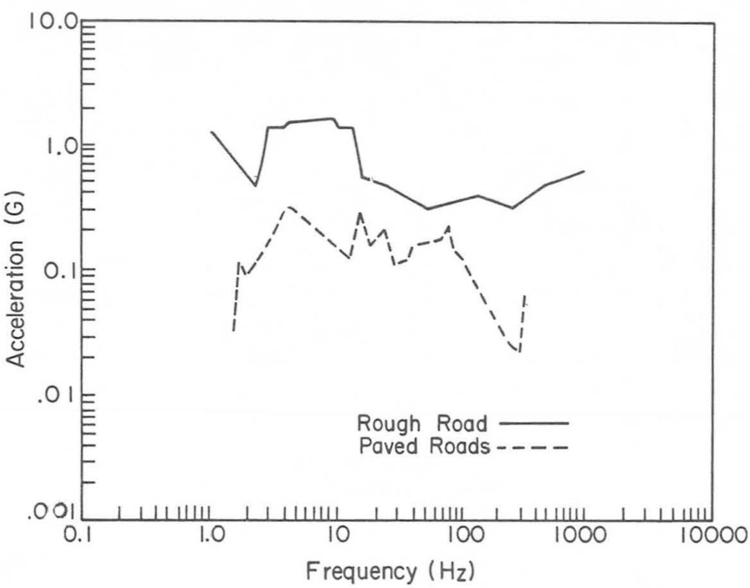


Figure 47.–Truck-acceleration envelopes as functions of road condition.

three standard commercial semitrailers traveling over first-class asphalt roads; each truck had a different suspension and was either fully loaded or empty. Curves of the data showed that vibration levels of low-frequency components were unaffected by load, but that vibration levels of higher-frequency components were reduced on loaded trucks.

Equipment may cause any of a variety of problems harmful to electrical systems; therefore, every equipment environment must be considered individually to determine the specific nature of its electrical system. Vibration environments are the result of vibrations transmitted through either material or acoustical links between vibration source and equipment. If the attenuation in an acoustical link is not sufficient, or if the link acts as an amplifier, then either the source of vibration must be eliminated or an adequate vibration isolator must be provided.

ANALYTICAL TECHNIQUES

Analytical techniques are used in the mechanical design of electrical systems to determine the response of system components to vibration environments. If a response excites force or motion[8] levels that affect equipment performance either temporarily or permanently, appropriate steps must be taken to reduce the response levels. The fundamental analytical techniques used in elastic-damped systems subjected to vibration and, therefore, fatigue failures, are well documented (refs. 3 and 4). The use of analysis has been advanced by development of the digital computer; for example, a system can be modeled in a fine network of finite elements for detailed system response studies.

Through a simulation model of a basic system concept, the digital computer permits parameter variation with the change of a computer data card. Thus, system parameters can be varied (parameter-variation study) until the best design is obtained.

Modeling, described in chapter 2, allows determination of values of analytical response studies. The basic properties of a system (i.e., mass, elasticity, and damping) and its environment are used in formulating a mathematical model for a system. NASA-generated work in determinations of basic damping constants, surveyed in the following pages along with methods of system natural frequency and response calculations, can be translated into industrial use.

Damping

Damping phenomena are difficult to determine or to use effectively in analytical response studies of vibrating systems. In the past, the dynamics of a design were handled with natural-frequency calculations. Although they

[8] Displacement, velocity, or acceleration.

identify resonant frequencies (i.e., where the exciting frequency and natural frequency of a system are identical), such calculations are not sufficient to determine their seriousness. The system's response at resonance really determines whether a natural frequency identified resonance is likely to cause vibration problems. Therefore, predicted resonant conditions often are unimportant because system damping values are large. Now the designer has techniques for determining system response and can use all available data in deciding on the validity of a design.

The analytical representation of damping for a single-degree-of freedom system as formed by Reed (ref. 5) has been described in chapter 2. This model applies equally well to shock and vibration studies, as does work on material, elastomeric, and structural-joint damping. Two additional forms of damping—air and liquid sloshing—which control the vibration response of a system, are surveyed in this chapter.

Stephens and Scavullo (ref. 6) conducted an investigation to determine the effect of air damping on the response of plates, cylinders, and spheres. These components were attached to cantilever springs (fig. 48), and the free decay of an induced vibration was measured at pressure levels from atmospheric to 4×10^{-2} torr. The magnitude of air damping reported (fig. 49) exceeded structural damping (fig. 50) of the system. Air damping associated with plates was directly proportional to pressure and amplitude, independent of shape, and a nonlinear function of the surface area. The empirical relationship for air damping of plates developed in this study,

$$\delta = 22 \frac{\rho X A^{4/3}}{m},$$

is dependent on air density ρ, amplitude X, area A, and mass m. The sphere and cylinder had viscous damping characteristics that were in general agreement with the theories of Stokes and Lamb.

Powell and Stephens (ref. 7) investigated the vibration characteristics of aluminum honeycomb and polyurethane-cored sandwich panels over a wide range of environmental pressures. Natural frequencies and damping constants were determined for several vibration modes over a pressure range of 760 torr (1 atmosphere) to 10^{-6} torr. The results showed that no significant changes in either natural frequency or damping occurred between 1 torr and 10^{-6} torr. A decrease in pressure from 1 atmosphere to 1 torr produced a frequency increase of from 2 to 10 percent. Damping decreased linearly due to reduced acoustical radiation. Changes in natural frequency and damping associated with pressure depended on the mode shape (i.e., system configuration at a particular natural frequency) of the panel.

Woolam (ref. 8) investigated the damping of flat plates that were vibrating normal to their planes in air. A dimensional analysis was made to choose the experimental parameters, which included amplitude of vibration, frequency,

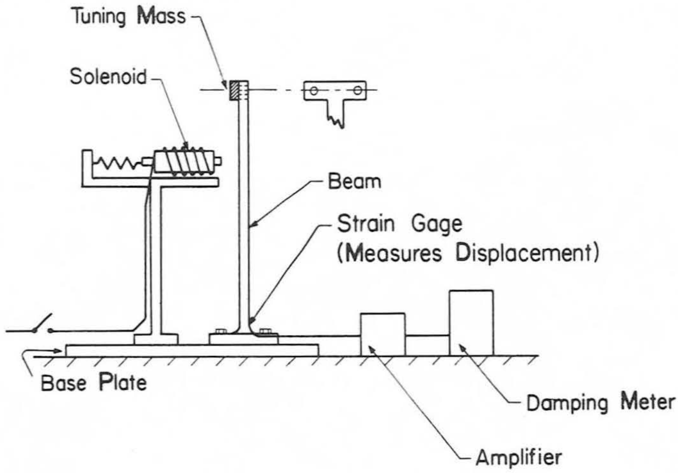


Figure 48.–Test apparatus and instrumentation for air-damping experiments.

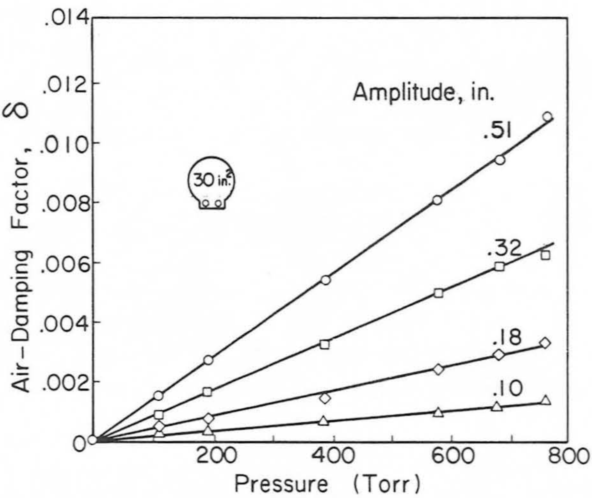


Figure 49.–Variation of air damping with pressure.

plate thickness, and plate area. Oscillatory drag coefficients (damping) were larger than drag coefficients for plates under steady fluid flow. System response errors are introduced, therefore, when damping constants for flat plates vibrating in air are calculated by conventional steady-state fluid theories rather than by dimensional analysis.

Liquid sloshing has been important in NASA space flight vehicles; damping

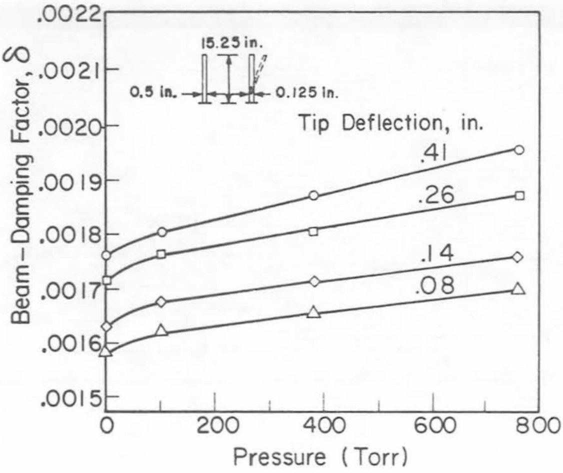


Figure 50.—Variation of structural damping with pressure.

resulting from the sloshing in tanks is also of interest to industry. Bauer (ref. 9) investigated the behavior of propellants in the tanks of large space vehicles; he also studied the effects of propellant oscillations on response characteristics of the vehicle and the effects of baffles on damping.

Sumner and Stofan (ref. 10) experimentally determined the viscous slosh-damping characteristics of several liquids (mercury, acetylene tetrabromide, and several mixtures of water and glycerine) with a wide range of kinematic viscosities. The tests (fig. 51) were conducted with three rigid spherical tanks, 9.5-, 20.5-, and 32-in. in diameter. The average first-mode damping ratio was independent of the excitation amplitude over the range investigated. The relationship between the first-mode damping ratio δ, the viscosity ν, and the tank diameter D was established experimentally (fig. 52):

$$\delta = 0.131\left(\frac{\nu \times 10^4}{\sqrt{gD^3}}\right)^{0.359}$$

Clark and Stephens (ref. 11) studied the simulation parameters important in experimental investigations of low-gravity slosh phenomena and the scaling laws for extrapolating data to full-scale spacecraft systems. Small tanks were used to amplify the importance of surface tension and viscous forces with respect to gravity or inertial forces in low-gravity slosh with earth gravity tests. Slosh frequency and damping data were obtained from measurements of the free decay of a fundamental antisymmetric slosh mode in rigid, unbaffled, cylindrical tanks. The fundamental parameter that defines the low-g slosh

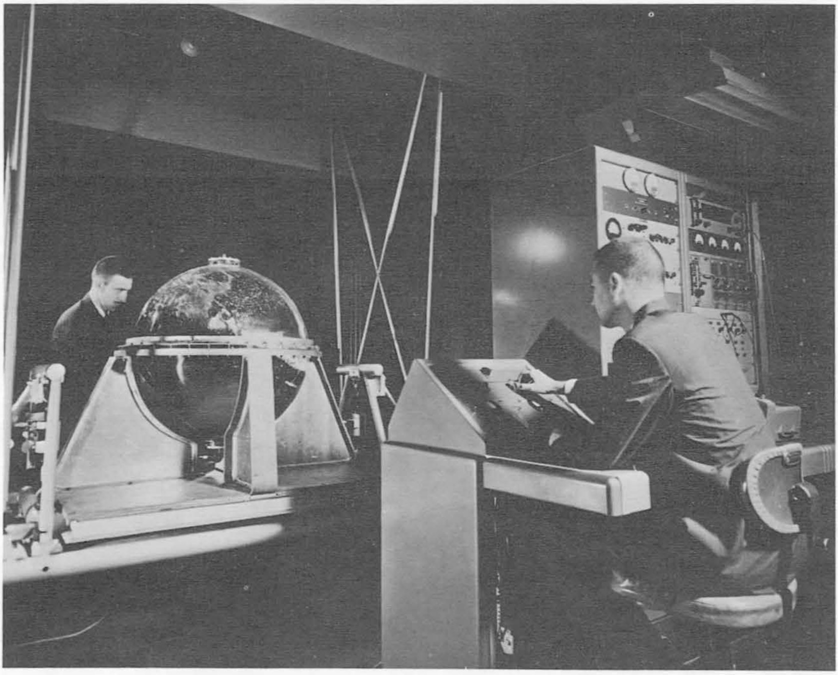


Figure 51.–Large experimental slosh-test facility.

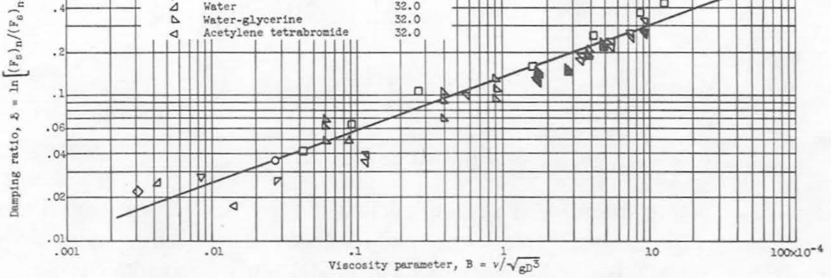


Figure 52.–First mode-damping ratio as a function of viscosity parameter for liquid sloshing in a spherical tank.

region is the *Bond* number, which provides an estimate of the ratio of gravity forces to capillary forces:

$$N_{BO} = (\rho/\sigma)\, g L^2 ,$$

where ρ is the liquid density, σ is the surface tension force, and L is the characteristic tank length. Damping was dependent on both surface tension and viscosity under simulated low-gravity conditions. For low-gravity conditions, slosh damping is relatively high, natural frequency relatively low—with the net result a marked increase in the time required to damp any slosh motion.

Natural Frequencies and Mode Shapes

Even though its response to environmental vibration can be calculated, a system's first natural frequency and mode shape often provide sufficient information for an adequate design. Unless damping is defined, a response prediction is useless. The natural frequencies of a linear system reflect the unique arrangement of that system's mass and elasticity parameters. A mode shape is the configuration that a system assumes when it vibrates at a particular natural frequency.

After formulation of a mathematical model and selection of a lumped- or continuous-parameter representation have been completed (chapter 2), two basic methods of solution are available. In Stodola's method (ref. 12), a matrix iteration involves substitution of a trial-mode shape into the system's equations of motion; mode shapes, obtained from successive iterations, approach the first one and eventually no further changes occur. The following matrix equation represents the system shown in figure 53 mathematically:

$$\underset{\substack{\text{Modal}\\\text{Vector}}}{\begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix}} = \omega^2 \underset{\text{Flexibility Matrix}, A}{\begin{bmatrix} \overset{*}{a}_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{bmatrix}} \underset{\text{Mass Matrix}, M}{\begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{bmatrix}} \underset{\substack{\text{Trial Modal}\\\text{Vector}}}{\begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix}} \text{Trial}$$

First Mode Matrix

The natural frequency (ω^2) can be calculated from this information. Before calculating higher natural frequencies and mode shapes, preceding mode shapes must be "swept" from the mathematical model with a sweeping (generated from previous mode shapes) matrix.

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*a_{ij} are influence coefficients which are derived from the system's stiffnesses k_{ij}.

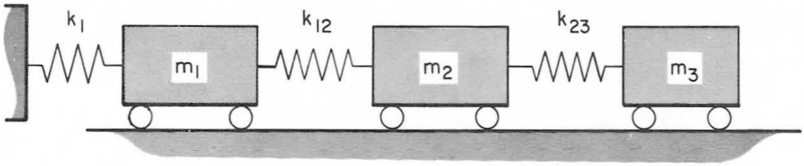

Figure 53.–Spring-mass system.

$$\begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix} = \omega^2 \quad [A] \quad [M] \quad [S] \quad \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix}^{\text{Trial}}$$

Sweeping Matrix, S
Higher Mode Matrix

Another matrix iteration results in convergence in the next natural frequency and mode shape. This method is subject to error at high frequencies because the sweeping matrix does not remove all the components of the previous mode shapes.

The Holzer method (ref. 13) is a trial-and-error procedure in which a trial frequency is substituted in a mathematical model. If the system boundary conditions have been satisfied, the trial frequency is the natural frequency of the system. Values of the residual matrix obtained in the solution are plotted (fig. 54) for several trial frequencies to determine which frequency makes the matrix zero. This frequency is the natural frequency of the system. However, the mode shape must be calculated to determine which natural frequency has been obtained. One advantage of this method is accurate determination of high natural frequencies; a disadvantage is the large amount of computer time required by trial-and-error methods. These two basic techniques allow the use of new mathematical methods and the digital computer to calculate natural frequencies of systems. Several computer codes are available for natural frequency calculation in NASA's Computer Software Management and Information Center (COSMIC) at the University of Georgia.

NASA has participated in developing methods for calculating natural frequencies and mode shapes of large complicated systems; these methods involve the establishment of rational models and computational methods. Shaker (ref. 14) developed a method, modified from the Stodola method, for calculating natural frequencies and normal mode shapes of a branched Timoshenko beam.[9] Although it can be applied to any type boundary condition, Shaker's

[9] A Timoshenko beam mathematical model includes the additional beam flexibility due to transverse shear deformation and the additional inertia due to rotation of beam elements with respect to each other. Both effects lower the beam natural frequency.

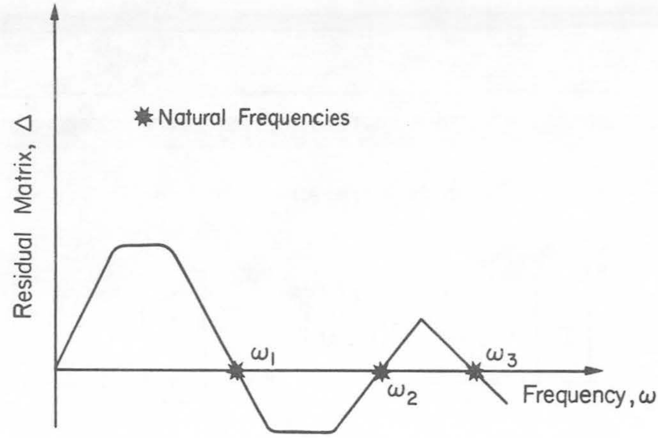

Figure 54.–Holzer residual plot.

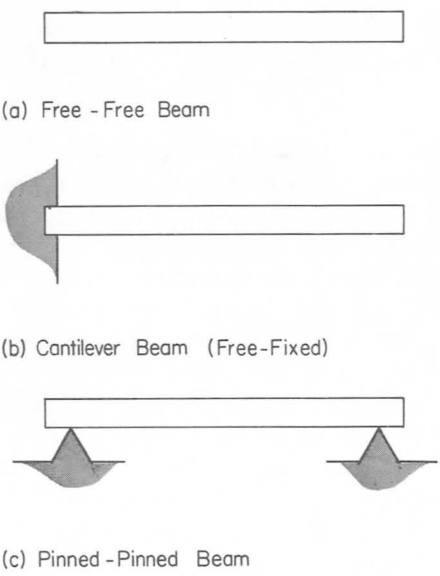

Figure 55.–Beam-boundary conditions.

method is used particularly for the free-free beam (fig. 55) and the cantilever-free beam. Alley et al. (ref. 15) have developed a recurrence solution for calculations involved in similar beam problems; first-order differential equations are used to describe the model, and a modified Holzer solution

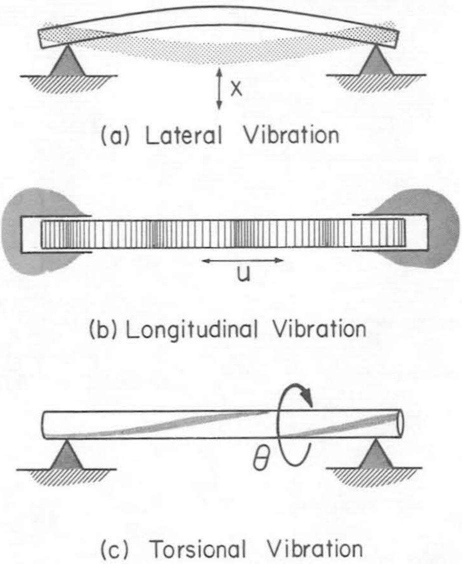

Figure 56.–Beam modes of vibration.

procedure is applied. Wingate (ref. 16) developed a method for calculating natural frequencies and mode shapes of longitudinal and torsional modes (fig. 56) of vibrating beams and beam-like structures. Using a variable-section [10] unconstrained multibranch beam, the system is modeled with lumped-parameter influence coefficients, and the matrix-iteration (Stodola) method is used. Bullock (ref. 17) used Holzer's method to calculate lateral and torsional vibration modes of the Saturn space vehicle. His results were compared to test results on the full-scale Saturn and the 1/5-scale test vehicle. The vehicle's first lateral natural frequency was found to be higher than the calculated one; test data and analytical results were in close agreement at higher frequencies.

Techniques similar to those used for beams have been applied to natural frequencies and mode shapes of flexible rotors. A detailed model of a rotor, given by Eshleman and Eubanks (ref. 18), includes effects of axial torque and gyroscopic moments[11] on lateral vibration of a rotor. Ekong, Eshleman, and Bonthron (ref. 19) solved rotor problems (including gyroscopic effects in the model) by transfer matrices using a modified Holzer approach. Gunter's (ref. 20) monograph on dynamic stability of rotor-bearing systems provides a detailed account of all rotor-bearing problems including the rotor and its bearing supports. The monograph contains original work on internal-friction damping. Cavicchi (ref. 21) prepared a theoretical analysis of rigid rotors in

[10] This refers to the cross-sectional area of the beam.

[11] Gyroscopic moments arise from rotor-spin-rotor-vibration interaction.

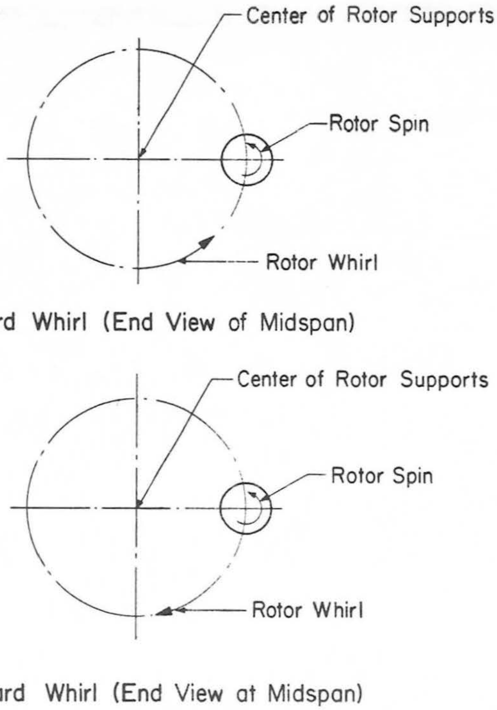

Figure 57.–Rotor whirl.

undamped flexible bearings. He determined frequencies of backward and forward whirl (fig. 57); and formulated the mathematical model in terms of nondimensional parameters for general results. Cavicchi's report also contains numerous design charts.

The calculation of natural frequencies and mode shapes of all structural components is important in analyzing an entire structure. Thompson and Clary (ref. 22) determined the resonant frequencies and modal patterns of shallow sandwich disks (fig. 58) with radially symmetric thicknesses tapered linearly from a maximum at the center. In one case the disks were pin-supported in the center; in another, the disks were hung on soft supports. Differences in the response of aluminum honeycomb and styrofoam core models are indicated and discussed. Lower-mode natural frequencies calculated in this investigation were in fair agreement with experimental results for the honeycomb core but not for the styrofoam core.

McCarty and Stephens (ref. 23) investigated the natural frequencies of fluids in spherical and cylindrical tanks (fig. 59) over a range of fluid depths and tank sizes. Data for horizontal circular cylinders are given in terms of the nondimensional frequency parameter

$$\gamma_n = \omega_n \sqrt{\frac{\ell}{g} \frac{1}{\tan h \dfrac{n\pi h}{\ell}}}$$

and the liquid depth h to cylinder radius ratio $h/2R$. Similar data are given for upright cylinders and spheres.

The preceding natural frequency calculations have been related to generalized structures; specific investigations on launch-vehicle structures have been conducted by Alley and Gerringer (ref. 24), Leadbetter et al. (ref. 25), and Loewy and Joglekar (ref. 26). Matrix notation and beam theory were used in all cases to calculate the natural frequencies of launch vehicles. Leadbetter's work contains design data on joint stiffness. Loewy used the Holzer method, formulated in matrix notation, to obtain the natural frequencies of clustered launch vehicles. He formulated the problem so that it could be adapted to the analysis of general structures, thus increasing its value for potential industrial use. In addition, Loewy's work contains considerable information on mathematical modeling of structural elements, branches, and liquid sloshing. A typical clustered vehicle analyzed by these methods is shown in figure 60. Peele, Thompson, and Pusey (ref. 27) used Loewy's techniques to investigate the three-dimensional vibration of an asymmetrical launch vehicle. Adelman and Steeves (ref. 28) determined the lateral vibration of a 1/40-scale dynamic model of the Apollo Saturn V vehicle. Good agreement was found between analytical and experimental results. This work is a valuable example of natural frequency and modal shape calculation on a complicated structure.

For general vibration analyses of large structures, the SAMIS system (ref. 29), a NASA-developed large-capacity computer program useful for natural frequency calculations, is recommended. As is the case with any computer code, the problem must be modeled to fit the format of the program. A second

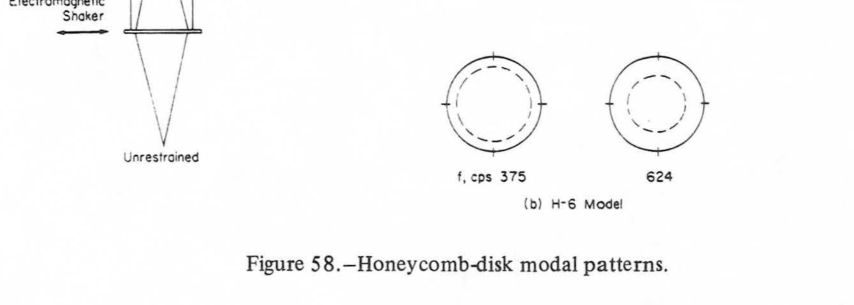

Figure 58.–Honeycomb-disk modal patterns.

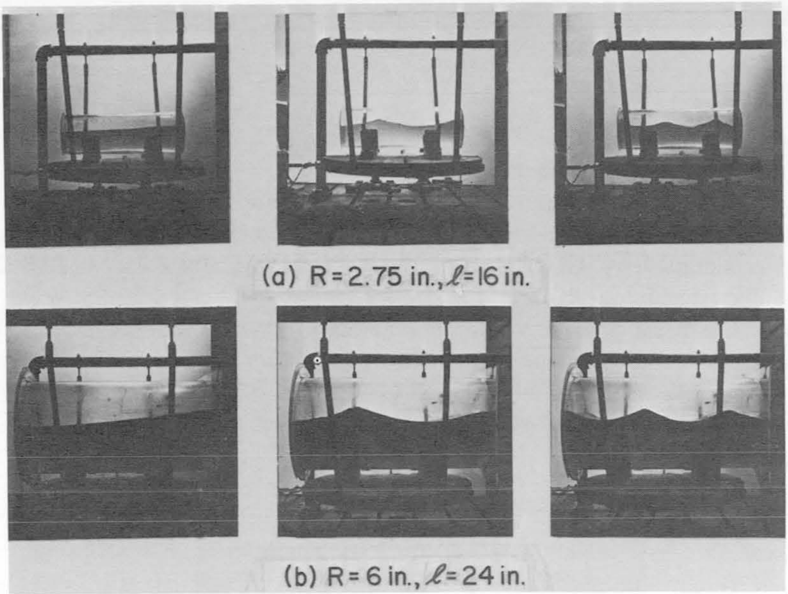

Figure 59.–Longitudinal modes of vibration of two horizontal circular cylinders.

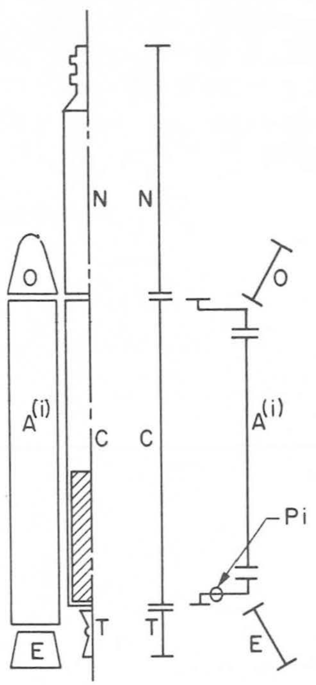

Figure 60.–Model of clustered vehicle.

large computer program developed by NASA is NASTRAN (NASA Structural Analysis) (ref. 30). NASTRAN, a finite element program, is capable of performing static and dynamic analyses of large complicated systems. This code reflects the latest techniques in modeling and numerical calculation. Both programs can be obtained through NASA's COSMIC center.

Response

NASA has been instrumental in improving methods for calculating the response of elastic systems to vibration environments. Recent work on rotors, beams, bearings, absorbers, and structures constitutes a major portion of the current technical literature in this area. The mathematical model used for response calculations is identical to that used for natural frequency calculations except for additional terms for damping and forcing phenomena. The object is to find the system vibration response, described by the vector $\underline{x}$, to the environment forcing vector, $\underline{F}(t)$:

$$[M]\,\ddot{\underline{x}} + [c]\,\dot{\underline{x}} + [k]\,\underline{x} = F(t)\,.$$

The system's parameters, described by mass matrix M, damping matrix $[c]$, and stiffness matrix $[k]$, provide a set of differential equations that must be integrated by one of two basic techniques.

The most common technique used for vibration response analysis is called modal analysis. The response, determined as some unique combination of previously defined mode shapes, requires a natural frequency analysis before the response analysis; more important, the damping must be expressed as some multiple of the mode shape. This latter requirement often is not consistent with physical reality; and, in addition, the modal analysis requires a linear mathematical model. The second technique, called a marching method or initial value analysis, utilizes a step-by-step integration of the system's equations of motion and depends on the initial state of the system and any environmental disturbances as time proceeds.

A finite-element model must be used to solve practical problems because few structures can be represented as a continuum; therefore, the structure is replaced by an assemblage of structural elements (e.g., bars, beams, and plates) known as finite elements. These elements coincide with the members of the actual structure. The finite-element model is used with the force (flexibility) method or the displacement (stiffness) method of response calculation. For example, this modeling technique is used in NASTRAN.

Craggs (ref. 31) presents a transition-matrix method for solving the set of equations of motion that represents a dynamic system. The *Taylor series* is used to find a recurrence relationship; for free vibration, this relationship provides a value for the response at one step in terms of the response at a previous step by means of a transition matrix. For forced-vibration-response calculation, additional terms must be used in the recurrence relationship; the actual number of terms depends on the nature of the forcing function. This method has two advantages over standard techniques: (1) a substantial decrease in computation time, and (2) an increase in the accuracy of the solution (if a high-order Taylor series is used to evaluate the transition matrix). With an arbitrary forcing pulse and a prescribed boundary motion, the response value can be determined concisely and in terms of matrix operations by this general procedure. The theory has been applied in several cases. Using the finite-element-displacement technique to idealize the structure, good agreement has been obtained between analytical results and available experimental work.

Nowak (ref. 32) analyzed the lateral vibration response of aerospace vehicles in terms of distributed parameter[12] or wave-transmission concepts, rather than in terms of the normal mode approach. Transmission models and matrices, developed for uniform beam segments, included the following effects: (1) shear compliance and rotary inertia; (2) axial loading; (3) longitudinal displacement because of axial loading; and (4) distributed, uniform, lateral loading. Local-state variables were related to characteristic variables of the beam by using a transformation technique. This allowed factorization of the solution into propagation and end-effect matrices. The technique was applied to nonuniform beam that was approximated by a step-beam structure in combination with a damped spring-mass sloshing model.

Using a linear analytical model, Archer and Rubin (ref. 33) calculated the axisymmetric launch-vehicle steady-state response to applied axisymmetric sinusoidal loads. A finite-element technique was used to construct the total launch-vehicle stiffness matrix and mass matrix by subdividing the prototype structure into a set of axisymmetric shell components, fluid components, and spring-mass components. Natural frequencies and mode shapes of the coupled system were obtained by using mass and stiffness matrices. The steady-state response due to simple harmonic loads was determined. A standard modal-response procedure was used to express total displacement, velocity, acceleration, and force responses as the linear superposition of individual modal responses based on assumed modal damping. The computer program description for this analysis is given in volume II of Rubin and Wang (ref. 34). Thompson (ref. 35) used this computer program to determine the longitudinal excitation response of a scaled model of a liquid-propellant multistage launch vehicle. A two-dimensional mathematical model consisting of axisymmetric shell, fluid, and mass components was used, and good agreement was found between computed and experimental results (fig. 61) for longitudinal vibration.

Pinson and Leonard (ref. 36) assessed the applicability of lumped-parameter analysis to a complex Saturn V launch vehicle. Analytic results were in general agreement with test results relating to structural modes; however, discrepancies were found when system response occurred either to propellant modes or in

[12] See Modeling, chapter 2.

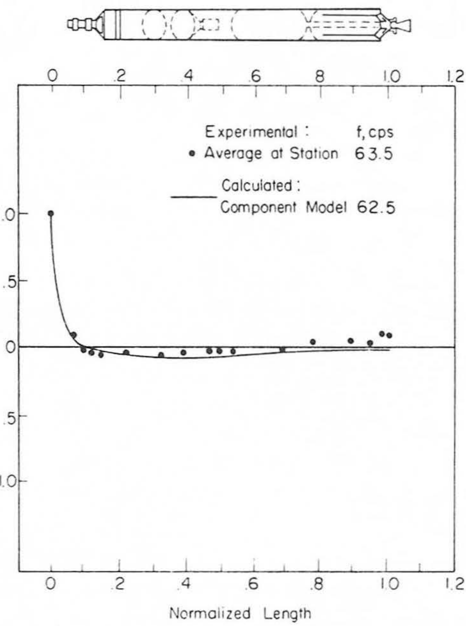

Figure 61.–Comparison of measured and calculated longitudinal response of a multistage launch vehicle at $t = 0$.

coupled-structural propellant modes. Archer (ref. 37) discusses the state of the art, basic criteria, and practices recommended for obtaining and analyzing natural vibration modal data.

Chang and Bieber (ref. 38) proposed a hysteresis model to simulate structural damping to obtain better structural-response analyses. Their results showed that nonlinear force-deflection relationships must be preserved in system equations of motion if analytical consistency between structural vibrations and hysteresis damping is to be achieved. In a related report, Hultgren (ref. 39) analyzed the lateral plane vibration of a simple structure possessing hysteresis damping. Damping was assumed to exist at some stage between a nonuniform, flexible beam and a nonuniform, rigid beam. Equations of motion were developed, and numerical methods used in solving the equations were presented. A users' manual and a sample problem for the computer program are included in the report.

Minimization of resonant-vibration amplitudes in dynamically loaded structures is an important aspect of design. McMunn and Jorgensen (ref. 40) reviewed the literature on optimization techniques and min-max structural response problems. In later work, McMunn (ref. 41) studied multiparameter optimum damping by applying mathematical programming techniques to the minimization of steady-state resonant response of dynamic systems. The optimum set of damping parameters was found for a system with specified geometry.

Rotor-response studies, motivated by the need for high-speed rotating machine designs, have potential in the design of rotating electrical systems where the capability for rotor speeds up to 100 000 rpm is advantageous. Since rotor-support bearings often have more flexibility and damping than other components in the rotor-bearing system, support-bearing effects have been emphasized in NASA studies. In a comprehensive report on steady-state and dynamic properties of journal bearings in laminar and superlaminar flow regions, Orcutt (ref. 42) presented data for steady-state load capacity, friction torque, and dynamic-stiffness and damping coefficients. The theoretical analysis of the tilting-pad bearing for laminar and turbulent flow is summarized, and design charts generated by this analytical and experimental program are included. Pan (ref. 43) analyzed gaseous squeeze-film bearings and reported design data on their stiffness. Chiang, Malanoski, and Pan (ref. 44) published design data resulting from a theoretical study on spherical squeeze-film hybrid bearings.

Graham (ref. 45) made an extensive report on a dynamic axial-pump-rotor. Analog and digital computer simulations of the rotor, its bearings, and fluid flow effects were used to study the axial-pump response, and a set of equations of motion was derived. A parametric study was performed in which rotor motion was determined for rotor unbalance, linear-bearing spring rate, nonlinear-bearing spring constant, viscous-bearing damping, and velocity-squared damping. The greatest reduction in deflection amplitude was obtained by applying viscous damping at the bearing mounts. The digital computer program can be applied to a wide variety of turbomachines.

Gunter (ref. 46) examined conditions that can lead to nonsynchronous whirling in a rotor-bearing system (fig. 62) by studying the effects of internal friction damping in the rotor and fluid-film bearings on rotor-bearing systems. The system equations of motion were programmed on the analog computer; stability studies produced rotor orbits. It was found that rotor stability could

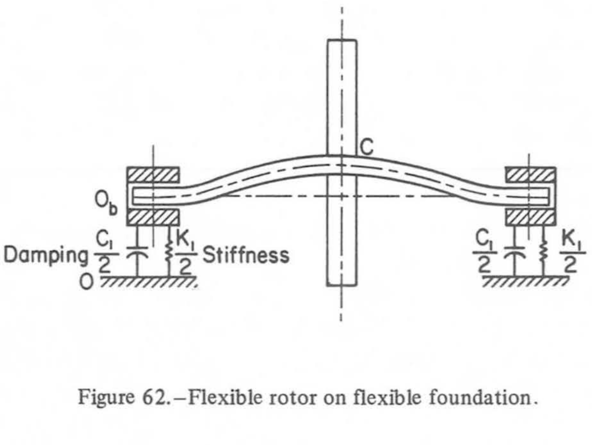

Figure 62.–Flexible rotor on flexible foundation.

be increased with asymmetric stiffness and that the addition of damping in the supports was not necessary.

NASA has conducted response studies on dynamic vibration absorbers. As a result of these studies it was found that in a vibration system (fig. 63), an attached vibration absorber can be "tuned" to apply a force that cancels undesirable motion. Vibration absorbers, however, function only at one frequency (the natural frequency of the absorber equals the forcing frequency of the vibration system). Jones (ref. 47) conducted a study to define, analytically, the effects of a dynamic absorber on the modal damping of a system to which it was attached. The mathematical model consisted of a damped dynamic absorber attached to the mass of a simple spring-mass system. The investigation showed that the absorber-damping value which optimized modal damping was 163 percent of the value that optimized transmissibility.

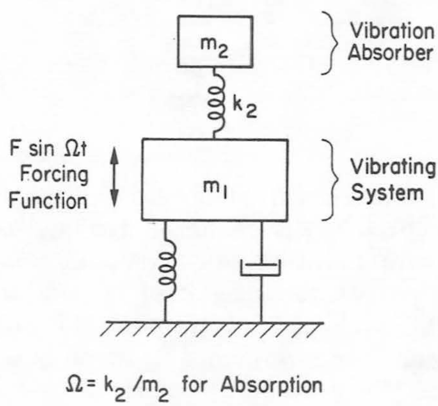

Figure 63.–Dynamic vibration absorber.

Srinivasan (ref. 48) analyzed a set of parallel-damped vibration absorbers (fig. 64) by adding a subsidiary, undamped absorber mass to a system containing a conventional-damped vibration absorber. The analysis showed that an undamped antiresonance can be obtained in a dynamic-absorber system that exhibits a well-damped resonance. Results indicated that the parallel-damped dynamic vibration absorber has definite advantages over the conventional-damped vibration absorber.

Flannelly and Wilson (ref. 49) demonstrated analytically the feasibility of using a synchronous, dynamic vibration absorber based on gyroscopic resonance. Linearized equations of motion were solved for the response of the body (to which the absorber was attached) as a function of excitation frequency, gyroscopic speed, excitation force, and gyroscopic configuration. A method for solving nonlinearized equations was included. For a parallel arrangement of two gyroscopic absorbers (one damped and one undamped),

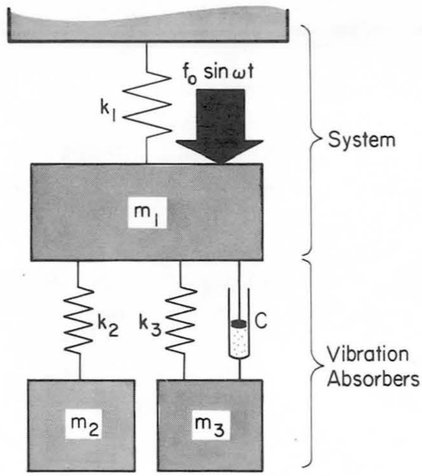

Figure 64.–Parallel-damped vibration absorber.

both an undamped-antiresonant frequency and a damped-resonant frequency were obtained. The effects of elastic restraint, damping, and flexibility in the support structure were examined analytically. An analysis indicated the feasibility of simultaneous synchronization of an absorber to two excitation frequencies and isotropic absorption in the plane of rotation. Srinivasan (ref. 50) conducted analytical and experimental studies on gyroscopic absorbers. In his studies he found that gyroscopic absorbers provide for two degrees of translational freedom and three degrees of rotational freedom. He also developed computer programs to calculate: (1) null and natural frequencies, (2) responses in the direction of excitation, (3) responses orthogonal to the direction of excitation, and (4) responses of the gyroscope.

EXPERIMENTAL TECHNIQUES

Experimentation plays a key role in developing and qualifying new equipment, in quality control, and in trouble shooting old equipment. Nothing substitutes for a well planned test; it is of no value to test equipment without adequate planning or data analysis. Testing applicable to shock isolation was reviewed in chapter 2. NASA has had an important role in the development of experimental techniques, especially in scale-model testing and data reduction.

Experimental Models

The large size of space vehicles has provided the motivation for developing subscale models and tests. Subscale, experimental model design usually is based on dimensional analysis or the system equations of motion. In scaling a vibration test model, the important dynamic characteristics of the prototype must be preserved if meaningful results are to be obtained. Nondimensional parameters, selected from dimensionless analysis or the system equations of motion, are used to scale the system mass distribution, damping, and elasticity; results then are related to full-scale models through these parameters. Scaling techniques may involve complete reduction of the system's geometry. The dynamic properties of the system, including mass, stiffness, and natural frequency, are scaled geometrically. Mixson et al. (ref. 51) designed a 1/5-scale model of the Saturn SA-1 launch vehicle (fig. 65). The length-ratios on the Saturn were reduced by 1/5 or

$$\frac{L_M}{L_F} = \frac{1}{5} ,$$

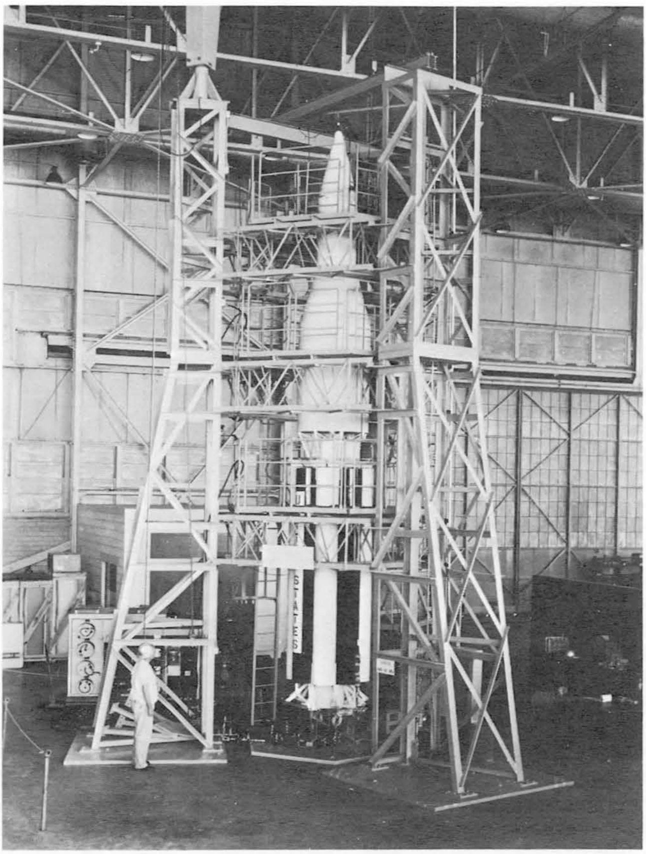

Figure 65.–One-fifth-scale Saturn model.

where L_M represents a typical component length in the scale model and L_F a typical component length in the full-scale vehicle. Lateral natural frequencies do not scale directly,

$$\left[\frac{f_M}{f_F}\right]_{\text{lateral}} = \left[\frac{EI_M m_F}{EI_F m_M}\left(\frac{L_F}{L_M}\right)^3\right]^{1/2} = 5 \,,$$

and test results must be divided by five to obtain the full-scale model natural frequency. A full-scale model test conducted at Marshall Space Flight Center (fig. 66) was compared to the 1/5-scale model tests by Mixson and Catherines (ref. 52). Comparison of test results for lateral bending modes showed good agreement. In addition, damping values were in close agreement for most tests.

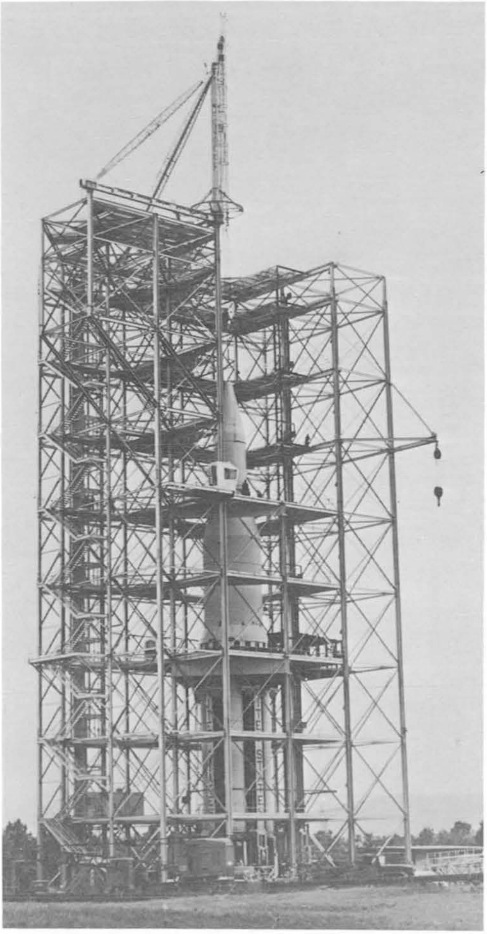

Figure 66.–Full-scale Saturn vibration test vehicle.

Catherines (ref. 53) also reported that torsional vibration of the 1/5-scale model of the Saturn SA-1 agreed closely with those of the full-scale model.

Test Equipment and Instrumentation

Standard-vibration test equipment and instrumentation have been developed to obtain the high performance required for valid experimental results. The equipment used for excitation on a test model usually is called a shaker. The many types of shakers, including mechanical, electromagnetic, hydraulic, electrodynamic, and piezoelectric, are described in reference 54. Instrumentation (e.g., transducers, amplifiers, and recorders) is also documented and described in many experimental texts. The intent here is to survey only new developments in vibration test apparatus, such as mounting techniques to simulate working conditions and instrumentation setups to obtain specific data.

The development of the experimental apparatus for full-scale testing of the Mercury capsule has been reported by Hieken (ref. 55). A variety of test fixtures was designed and constructed for this program. The basic test article was the device used to suspend the capsule in a free-free configuration; a difficult simulation of a physical situation, this configuration implies that no external restraint is applied to the test model. This is a frequent requirement in NASA and is not uncommon in related industries. The spring-suspension system that yields a minimum restraint in the longitudinal direction is shown in figure 67.

In the torsional mode of vibration, Catherines (ref. 56) used a Saturn test model in an eight-cable suspension system. This system prevented torsional restraint while two electromagnetic shakers, oriented 180-deg apart, were used to generate the exciting torque. The two shakers were controlled by one oscillator. In lateral vibrations, a two-cable suspension system (fig. 68) was used by Mixson et al. (ref. 51) to test the 1/5-scale model of Saturn SA-1. The model weight was supported through outriggers by a support yoke attached to two vertical cables; the cables were attached to a pair of movable rollers at the top of the test tower. Electromagnetic shakers were oriented to apply a force normal to the plane of the vertical suspension cables.

The full-scale free-free tests on the Saturn Dynamic Test Vehicle (SA-D) have been described by Watson (ref. 57). The suspension system consisted of cables running upward from outrigger points located in the tail section of the vehicle and through a group of 12 springs to a hydraulic cylinder fastened at the 72-ft level of the tower. The springs present in each support cable provided a relatively soft suspension system (compared to that of the vehicle) with a total maximum lateral spring rate of 560 lb/in.

Another cable-supported model was used by Thompson (ref. 35). Each of the two support cables passed through an overhead pulley and terminated at a leaf spring located near the model base.

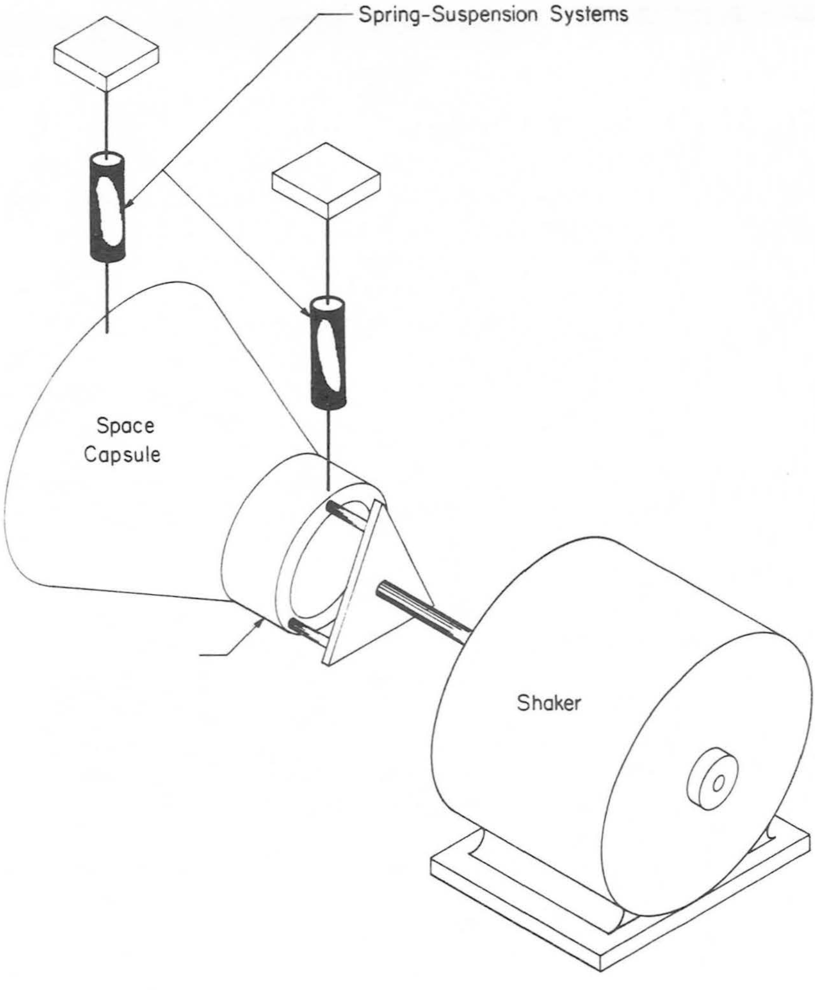

Figure 67.–Schematic of Mercury capsule test-suspension system.

Figure 69 shows a shaker suspended by two cables in order to apply a lateral force to the 1/40-scale dynamic model of the Launch Umbilical Tower. This vibration test was conducted with a cantilever-supported model.

Schoenster and Pearson (ref. 58) describe a pneumatic suspension system (fig. 70) for longitudinal vibration testing of large launch vehicles. This low-frequency support system, which did not affect the structural dynamic characteristics of the Thor-Agena vehicle while maintaining it at a predetermined elevation, is used where the test specimen is too heavy to be supported directly on the shaker and where a wide range of weights make it necessary to vary its stiffness. An automatically controlled, air-bellows system was

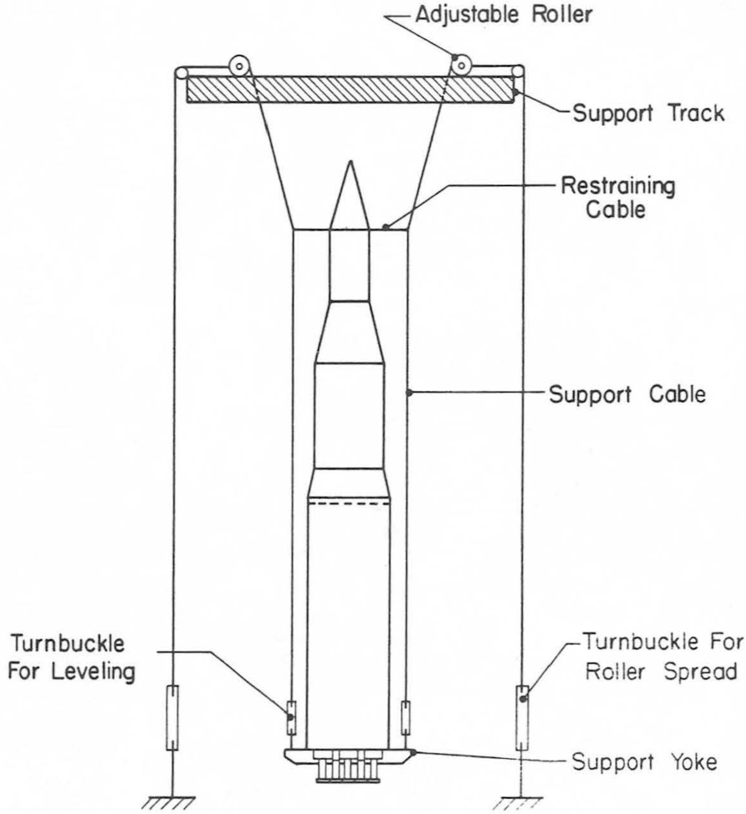

Figure 68.–Two-cable suspension system for one-fifth scale Saturn model.

developed; the controller adjusted the pressure in the air bellows to compensate for weight changes during simulated propellant flow from the vehicle.

A typical instrumentation system for vibration-response tests is shown in figure 71. Crystal accelerometers with cathode followers and voltage amplifiers were used as fixed and movable probes in determining mode shapes at resonant frequencies. The excitation force was measured by a crystal-type force gage inserted at the driving element of the shaker. The force delivered to the model was monitored quantitatively on an rms voltmeter and qualitatively on an oscilloscope.

Test Procedures

NASA has established a number of test procedures to determine system damping, response and natural frequencies; in addition, it has developed qualification tests concerned with life and performance reliability. Test

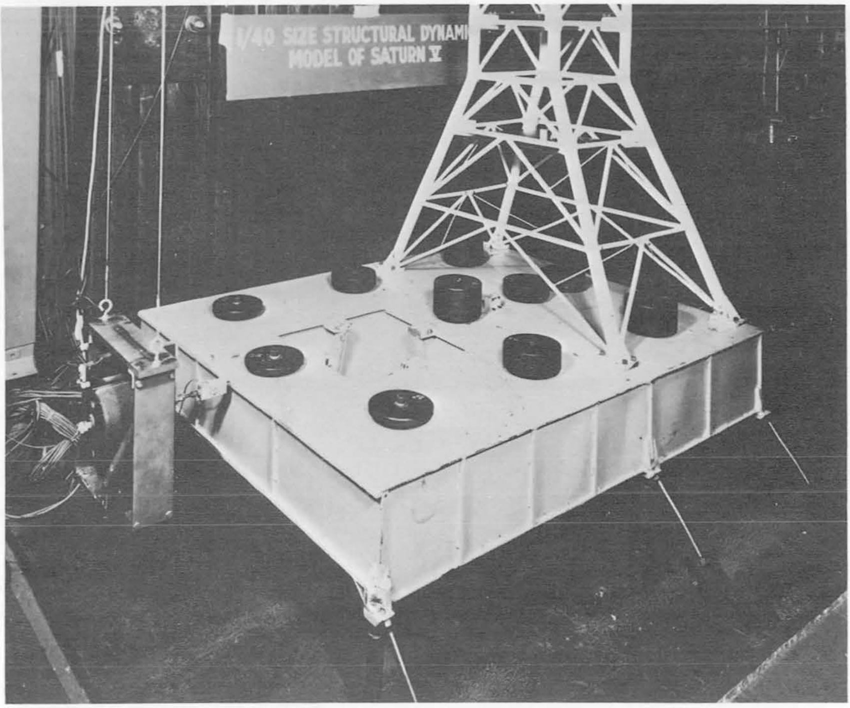

Figure 69.–Shaker orientation on cantilevered Launch Umbilical Tower model.

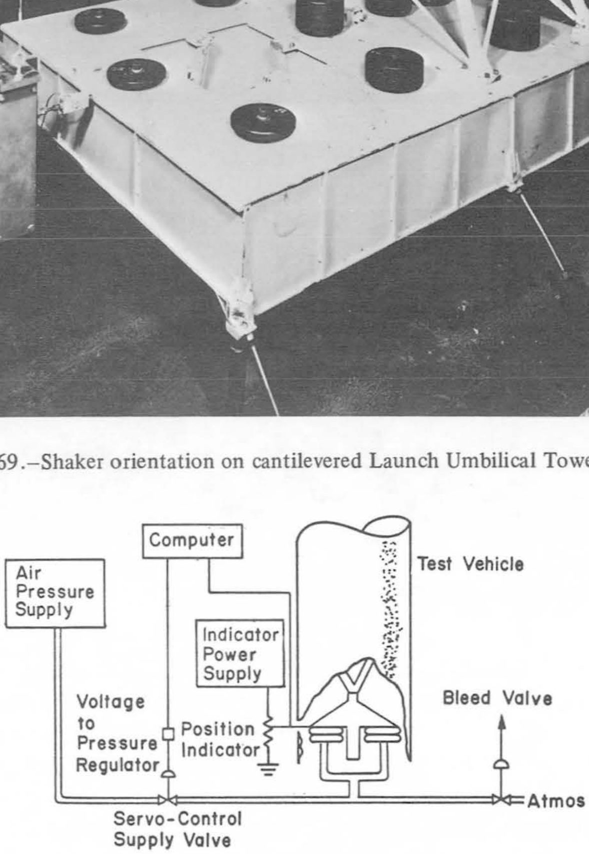

Figure 70.–Pneumatic spring-support system.

procedures involving basic determination of system damping were described in chapter 2; the determination of system natural frequencies, mode shapes, and forced response is presented below.

Two approaches can be used to determine the vibration characteristics of a system. Mixson et al. (ref. 51) used a frequency sweep with constant shaker force over the given frequency range. The output of fixed accelerometers was

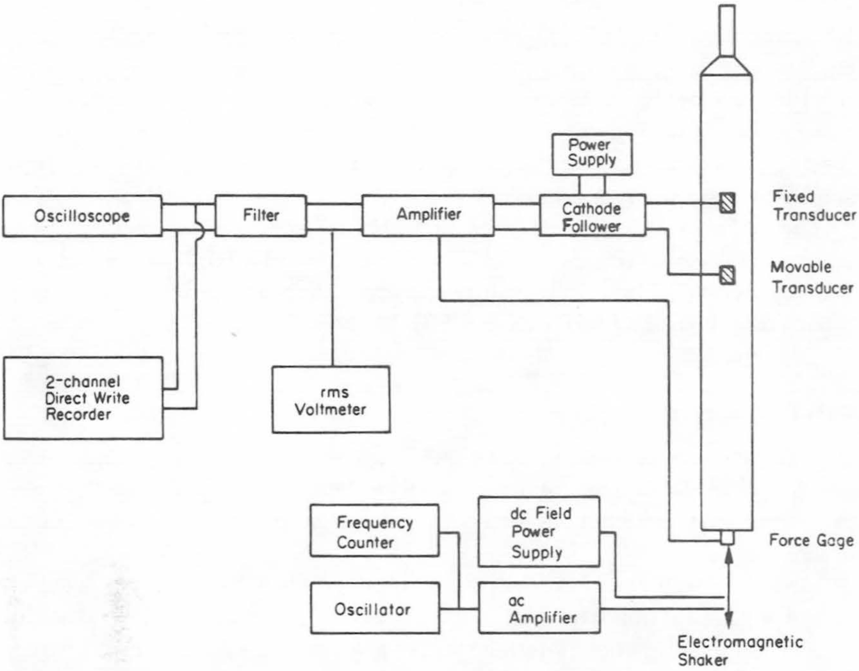

Figure 71.–Typical Instrumentation System for vibration-response tests.

recorded and used to determine the approximate frequencies of resonant-response peaks. A survey of the vibration-mode shape at each response peak was then made by tuning the frequency of the shaker to produce maximum model response. The survey included recordings of the portable accelerometer response as it was attached to intermediate locations. Damping values were obtained by instantaneous cutting of the input to the shaker at the frequency of the natural mode of interest. Damping values were obtained from fixed-accelerometer output recorded on an oscillograph. The decaying amplitude was read from the oscillograph and plotted on semilogarithmic paper. The damping measure δ, log decrement, was obtained through the relationship

$$\sigma = \frac{1}{n} \log_e \frac{x_0}{x_n},$$

where x_0 is the initial vibration amplitude and x_n is the amplitude after n cycles.

The second method for vibration-response testing, used by Catherines (ref. 56), involved a frequency sweep at a rate in octaves per minute with constant force input. The principal lateral-response frequencies were obtained by monitoring the acceleration level measured at the vehicle tips. Each resonance then was tuned to its maximum response, and resonant frequencies, mode shapes, and damping of the structures were determined. The data were recorded on analog tape and digitalized by means of a 24-point-per-cycle conversion. Digital data were reduced by using a harmonic analysis to determine normalized mode shapes. Damping values were obtained in a manner similar to that used by Mixson (ref. 51).

The sinusoidal-sweep test normally used to simulate transient-vibration conditions often overloads a model, resulting in overdesign. To overcome this problem, a new short-duration, high-intensity, shaped random-vibration test, reported by Fitzgerald and Kula (ref. 59), has been developed.

Data Reduction and Analysis

Test data may be received in various forms, depending on the recorder used and the ultimate purpose of the data. Raw data are seldom of practical use; therefore, data reduction or analysis is necessary before conclusions can be made.

Jones (ref. 60) described, in general terms, some techniques employed to reduce flight-vibration data from a Saturn vehicle and discussed the reduction of such data as random vibration, shock, and bending mode. The techniques included the one-third-octave analysis system (which allows a "quick-look" at data), the random-wave analysis system, the low-frequency analysis system, the digital analysis system, and the sonic analysis system. Clevenson (ref. 61) described a generalized regression analysis of test data from which response predictions can be made. Detailed flight-measured vibration data, obtained during Lunar Orbiter flights, were compared with vibration levels specified as flight acceptance requirements. These data also were compared with predictions based on results of a regression analysis of vibration data compiled for a number of major launch vehicles. The regression analysis does not provide as good a basis for comparison as flight-vibration measurements because the former is conservative; regression analysis can be useful, however, in estimating vibration levels for new vehicles.

VIBRATION ABSORBERS AND ISOLATORS

The serious problem of vibration control continues to motivate the development of vibration absorbers and isolators. A vibration isolator (fig. 1) is a resilient support that tends to attenuate steady-state disturbances applied at the moving base. The mass constitutes the isolated item (in a more complex system the mass would not be a rigid body), and the isolator is composed of stiffness, k and damping, c. A vibration absorber (fig. 2) is a device that dissipates energy to modify the response of the mass that houses a vibration source. Classical isolation devices (e.g., springs and dashpots) are not surveyed here; attention is given instead to new concepts developed by NASA in vibration isolation and absorption.

Plastic Foam

Plastic foams are useful in shock and vibration isolation both as direct isolators and as material for encapsulation of instruments and other fragile devices. Methods of rapidly producing plastic foam and activating polyurethane by heat have been developed by NASA (ref. 62).

Mechanical Isolators

Many useful mechanical devices for shock and vibration isolation have also been developed by NASA. Flexible ring baffles (fig. 72) can be used to solve liquid slosh problems in large tanks. The design of these baffles is presented by Stephens and Scholl in reference 63 where they also discuss the effectiveness of flexible and rigid ring baffles for damping liquid oscillations in large-scale cylindrical tanks.

Figure 73 shows a viscous-pendulum damper that can be used to suppress structural vibrations. Lead slugs moving in a pendulous motion in a viscous liquid are used. If the damping is used as a tuned absorber, the pendulum natural frequency of the lead slugs is designed to match the structural-excitation frequency. In an allied concept for tall flexible structures, suspended plastic-covered chains (fig. 74) are used to damp wind-induced structural

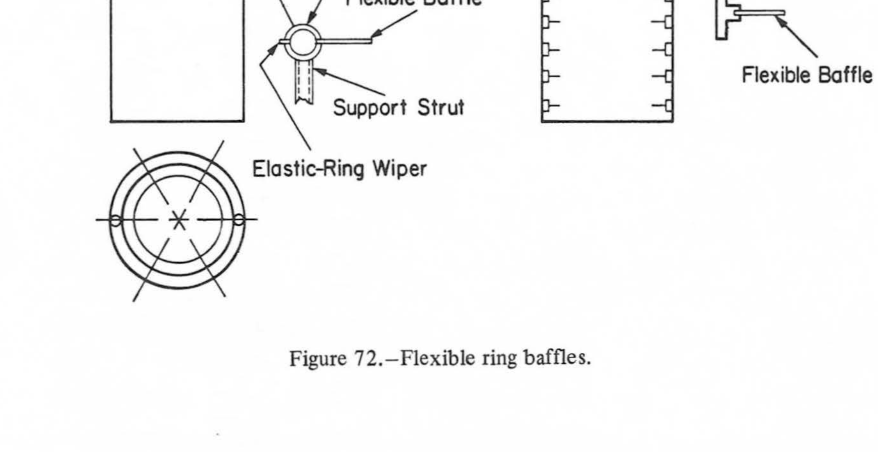

Figure 72.–Flexible ring baffles.

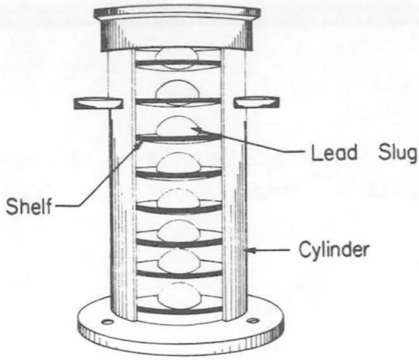

Figure 73.–Viscous-pendulum damper.

vibrations. Further descriptions and design information about the concepts shown in figures 73 and 74 can be obtained from NASA (ref. 64).

For the vibration control of structures with small amounts of damping, Bieber et al. (ref. 65) have tested the effectiveness of 11 commercially available damping treatments. The treatments were applied to cantilever beam specimens of various thicknesses and over various percentages of surface area. Coatings were effective in controlling local resonances in packages and structures.

Peck (ref. 66) reported on the analysis and design of elasto-plasto-viscous point vibration dampers. This report contained numerous feasible design concepts. One such concept, similar to that used on Mariner IV, is shown in figure 75. Peck's report also includes theoretical analysis and design techniques needed for application of this concept to industrial use. A basic energy-absorption mechanism, elastoplastic hysteresis, is supplemented by rubber-damping elements in a report by Lohr (ref. 67). The variable-stiffness damper (fig. 76) utilizes an increase in the energy absorption of polymeric-like materials with increasing temperature. This isolator or absorber concept can be used in the attenuation of shock and vibration in many different configurations.

Wire-mesh housing (fig. 77) provides an excellent support for the shock and vibration isolation of electronic components and is described by Kerley in reference 68.

Synchronous Vibration Absorbers

Synchronous vibration absorbers, described previously in the vibration response section, have been improved significantly in recent years as a result of NASA's interest. The design and analysis of gyroscopic configurations (fig. 78) of the synchronous gyroscopic absorber are described by Srinivasan (ref. 50).

The gyroscopic absorber can be tuned to a band of structural resonances (fig. 79) because of its variable speed feature, whereas a normal synchronous vibration absorber can be tuned to only one exciting frequency. Srinivasan has

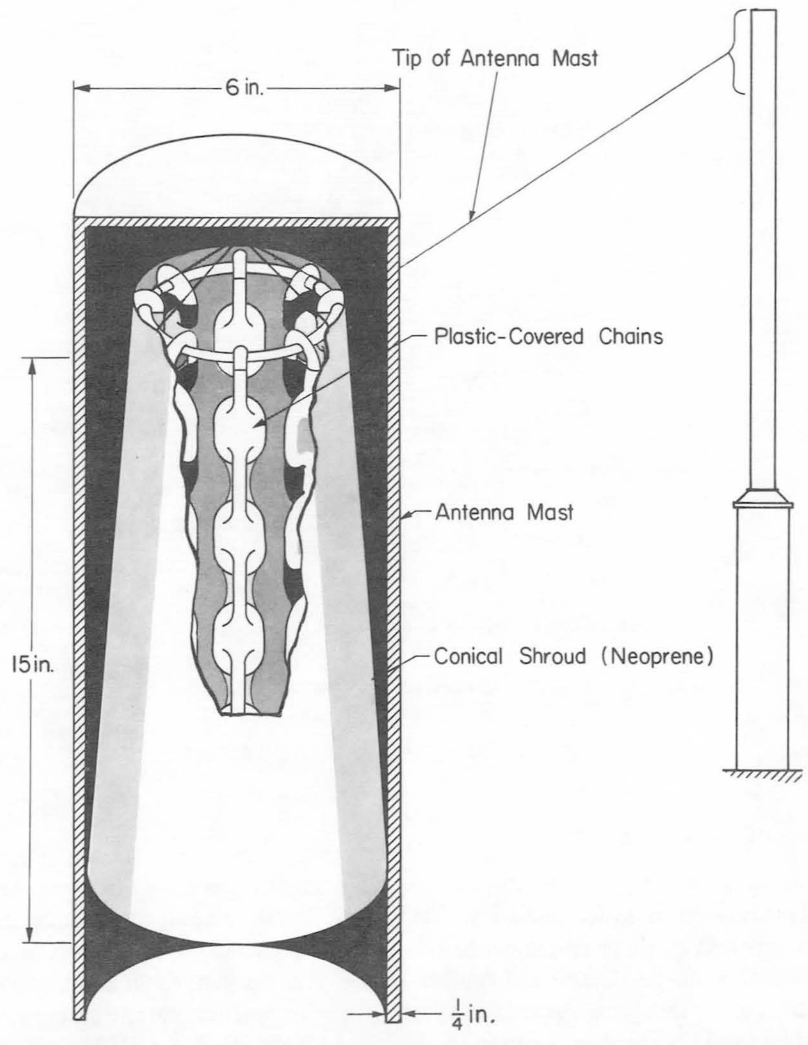

Figure 74.–Suspended-chain damper.

described the use of two gyroscopic configurations as synchronous vibration absorbers, which allow two excitations to be controlled.

Parallel damped linear vibration absorbers were investigated by Srinivasan (ref. 48); a subsidiary, undamped absorber mass was added to a conventional damped-absorber system to form parallel absorbers. Analysis showed that undamped antiresonance can be obtained in an absorber system that exhibits well damped resonance. While the bandwidths of frequencies between the damped peaks were not significantly increased, the amplitude of the main mass was considerably smaller within the operational range of the absorber.

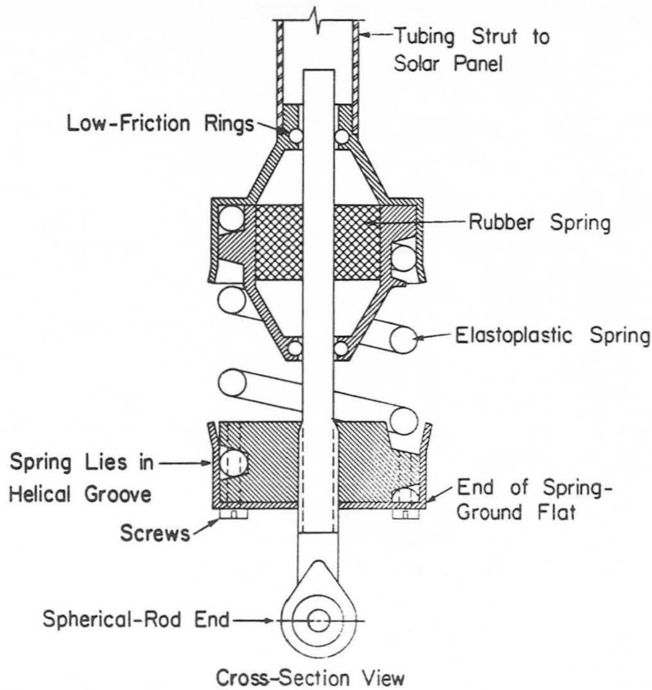

Figure 75.–Elasto-plasto-viscous point damper.

Suspension Systems

Because of their complexity, active suspension systems have not been used extensively in vibration isolation. NASA continues to sponsor work in the development of these systems, which have many possibilities for application to practical problems. Dixon and Pearson (ref. 69) have described an automatically controlled air-spring suspension system for vibration testing; the suspension system could be used as a vibration absorber. Figure 80 shows a schematic of the suspension system with its instrumentation and load. The low-frequency air suspension system consists of an air spring with variable stiffness (air pressure) and a closed-loop control of the air-spring height.

The active isolation system shown schematically in figure 81 was studied both analytically and experimentally by Leatherwood, Stephens and Dixon (ref. 70). This active vibration isolator, which can isolate transient and steady-state disturbances and maintain a predetermined equilibrium position of an isolated body, has three parts. Sensing elements measure the dynamic response of the flexible load and position the actuator; electrical control networks compare signals of the sensing elements with preset standards to provide an output to the actuator; and an actuator applies a force to the

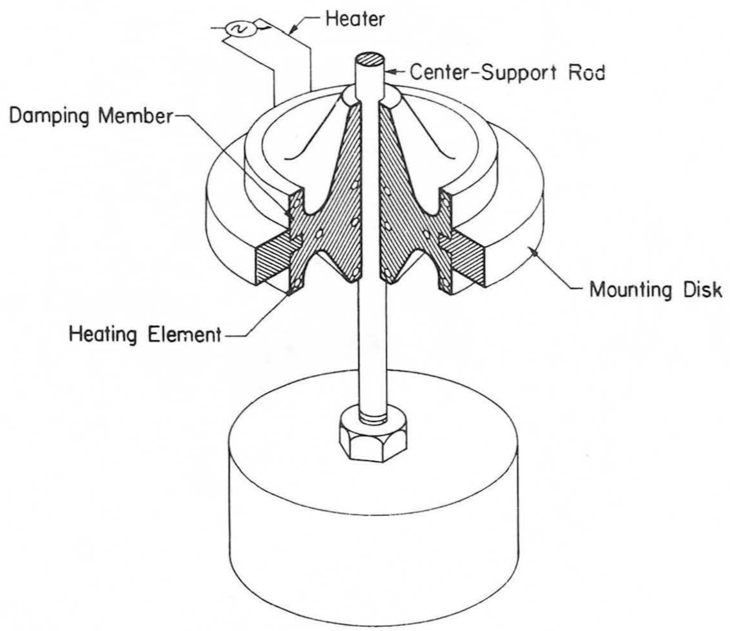

Figure 76.–Variable-stiffness polymeric damper.

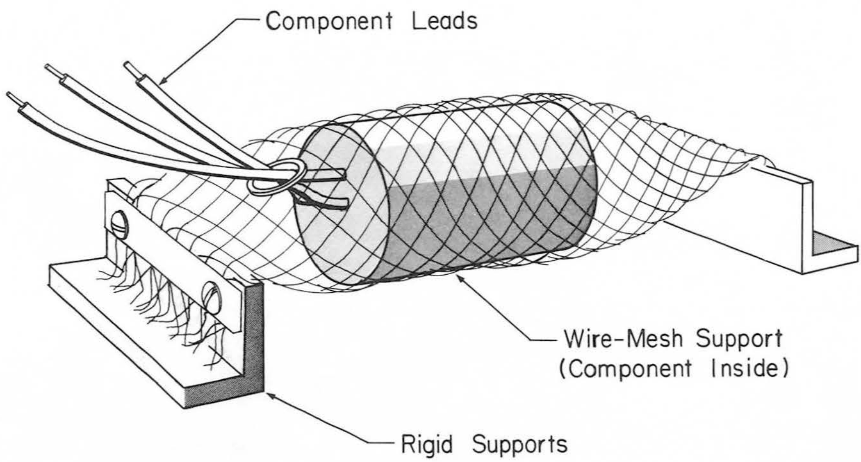

Figure 77.–Wire-mesh isolator.

isolated mass or structure to null its response and maintain a fixed equilibrium position.

Reed (ref. 71) describes a low-frequency vibration isolation mount (fig. 82) that can be designed for any allowable stroke length. Low-frequency isolation that utilizes linear springs usually requires prohibitively large deflections. This

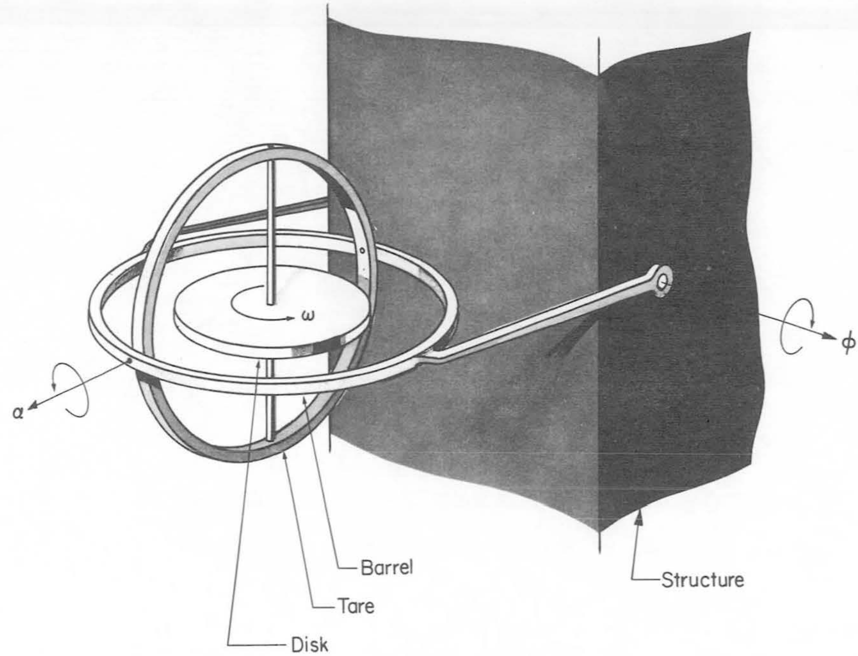

Figure 78.—Schematic of gyroscopic vibration absorber.

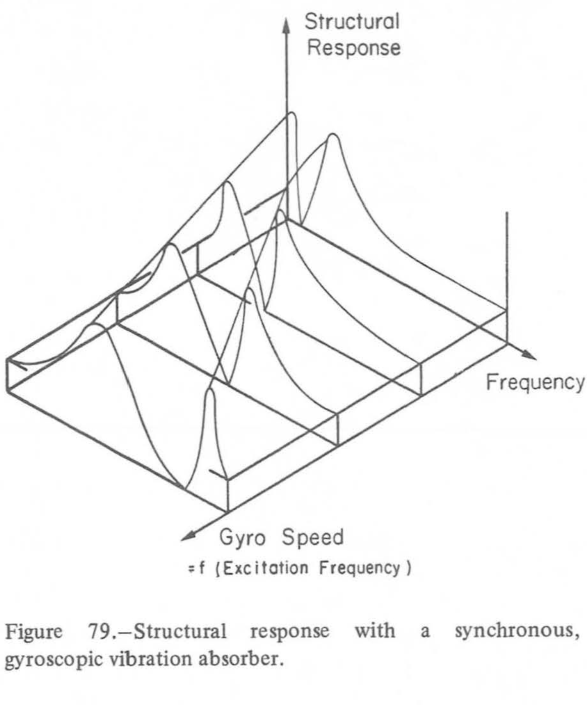

Figure 79.—Structural response with a synchronous, gyroscopic vibration absorber.

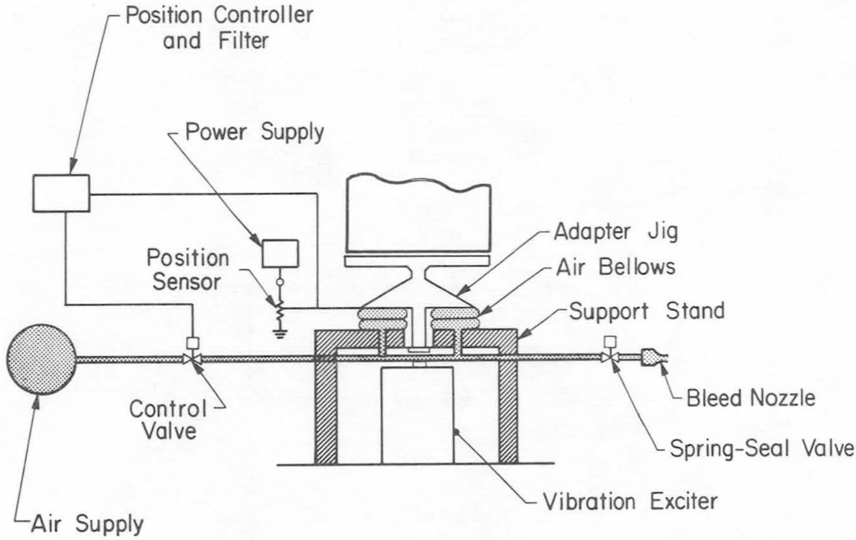

Figure 80.–Automatically controlled air-spring suspension system.

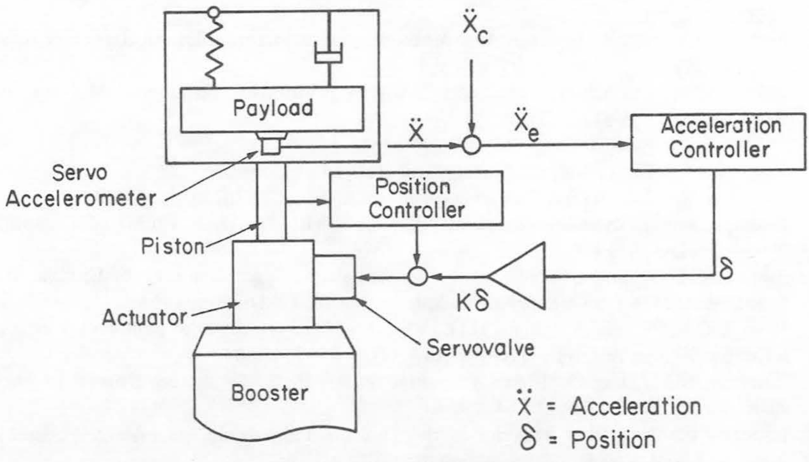

Figure 81.–Active vibration isolator.

mount, a piston-cylinder device with one side of the piston subject to vacuum and the other side open to the atmosphere, provides a near-zero spring constant for virtually any length stroke and eliminates the need for electronic-servo systems. Around the vacuum chamber, a rubber-like rolling diaphragm serves as a nonsliding seal and has very little resistance to motion. This mount, however, can become unstable in a rotational moment.

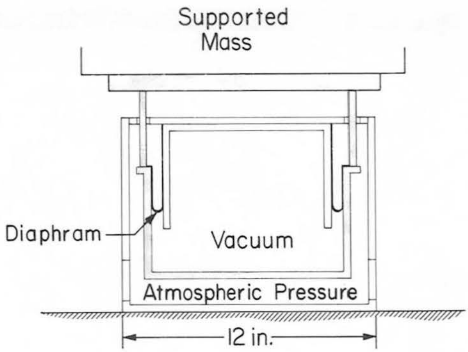

Figure 82.–Low-frequency vibration isolation mount.

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Application to Electronic Components and Subassemblies

The components (e.g., tubes and transistors) of electronic hardware mounted near a source of vibration exhibit some form of motion. Even if the strength of the vibration source is small—for example, a fan or air conditioner—a component mounting tuned to the source frequency may cause damaging fatigue over a finite time. Many problems in mounting electronic components and subassemblies have been solved during the space program. Severe vibration environments that result from a launch vehicle's high-frequency acoustical loading and low-frequency vibration loading have motivated these technological developments. Application of this work to industrial problems could result in a reduction in cost and inconvenience.

In recent years, electronic subassemblies have become convenient functional entities in an electronic package. In this chapter NASA's shock and vibration technology is interpreted as it relates to mounting electronic components and subassemblies in their chassis. Failures include component failure, mounting system failure, relative motion, and performance degradation.

Technology in shock and vibration mounting involving electronic hardware relates to individual components because different component classes are susceptible to different failures. Classes include heavy components (e.g., transformers), fragile components (e.g., cathode-ray tubes), components subject to performance degradation (e.g., relays and wires), and connectors that undergo deflection.

In solving mounting problems, the component mount is viewed as a spring and damper; the component's mass forms the remainder of a simple mass-spring system (fig. 83) that has characteristics typical of any shock and vibration system—i.e., a natural frequency and a response to external disturbances. Motion of the mounting base is dependent on the motion of the system's chassis or cabinet. For practical purposes the mount of an average electronic component, because of its negligible weight, is uncoupled from the chassis. Notable exceptions include transformers, solenoids, and other components with sufficient weight to negate this practice.

FAILURE MECHANISMS

In determining failure mechanisms of electronic components, the fragility of both a component and its mount must be considered; failure of a mount often

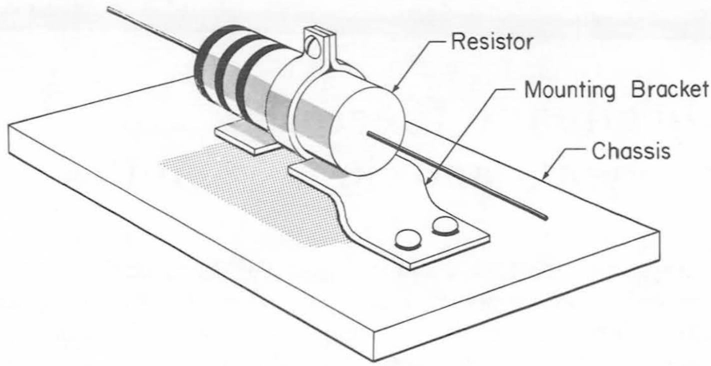

Figure 83.–Shock mounted resistors.

constitutes failure of a component. For example, failure of a lead used in wiring a component diminishes component performance.

A given design dictates the causes of component failure. In a liberal sense, performance is defined as anything short of permanent failure. Temporary lack of performance caused by shock or vibration is a more conservative definition of failure; this definition is applicable if proper equipment function can be maintained after exposure to a shock and vibratory environment. Performance degradation may or may not constitute failure, depending on the ability of the equipment to compensate for it.

The two basic failure mechanisms in any structure are strength and deflection. If either mechanism lessens performance of an electronic component, failure results. Failure due to deflection (or relative motion) of parts (e.g., relays, clamps, indicators, and latches) that make up electronic equipment results from shock loading or low-frequency vibration. Resonance and moderate- to high-frequency vibration response often cause failure in meters, tubes, and numerous variable components. The level of shock and vibration response that causes failure normally is given in the form of an acceleration-shock spectrum. Acceleration is a measure of force, which in turn, is a measure of stress in a body.

A fragility-level shock spectrum for an electronic component is shown in figure 84. At higher-frequency response levels, the allowable acceleration must decrease to compensate for a larger number of cycles until finite failure occurs. Finite failure also may be caused by high-frequency resonant conditions. In either case, the failure results from fatigue of a structural member in the component.

At this point, the frequency content of shock and vibration loading on a component should be discussed. Loading, component response, or both can consist of either a discrete frequency at varied acceleration levels or a random frequency-amplitude loading. It is difficult to assign damage levels to random vibration response. Waterman (ref. 1) attempted to correlate sinusoidal vibration levels with random vibration-damage levels, so that he could assess failure

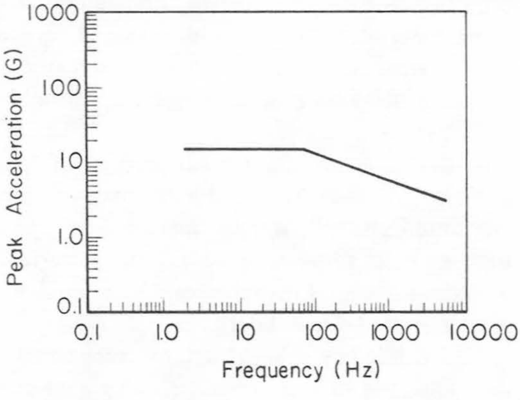

Figure 84.–Fragility level for an electronic component.

on a conventional reversed stress basis. He began with fundamentals and developed theory and methodology based on internal work done by damping forces. The introduction of a damage function allowed dimensional agreement of theoretical with experimental data. Waterman conducted structural-fatigue tests on both steel and aluminum beams, and developed examples of design charts.

Rich and Roberts (ref. 2) developed a technique for determining the safe life of components by using available fatigue data. In addition, they showed methods for assessing damage after varied loads and frequencies have been applied to a component; a reliability index for a specific component can be established in this way. This approach was used, for example, on the S-IC (the first stage of the Saturn V vehicle) where vibration environments were a major cause of electronic-component failure.

Clevenson and Steiner (ref. 3) calculated the fatigue life of components at several random loading spectra. Results of this fundamental investigation, where 100 approximately identical specimens were examined under random loading, showed that classical linear-cumulative damage theory was nonconservative.

The preceding theories and methods have been developed for, or by, NASA to assess the fatigue of components subjected to random vibration environments. Since industrial vibration environments are normally random, these theories and methods are directly applicable to the design of industrial electronic-component mounts.

SUBASSEMBLY DESIGN

Control of shock and vibration response often can be obtained through proper mechanical design of an electronic subassembly. This design is warranted if a subassembly's function can be obtained at no additional cost. If the cost is greater, then the subassembly must be sold as special equipment for use in shock and vibration environments. In any event, it is better to incorporate a shock and vibration capability in a basic design than to add a mounting to overcome these disturbances.

A printed circuit that functions in a small amount of space is subject to malfunction from vibration. The physical characteristics of the circuit board—the components mounted on it, its weight, and stiffness, and its mounting in the chassis—contribute to its response to shock and vibration. Printed-wiring fracture and other circuit malperformances can be related to the shock and vibration response of a printed-circuit board.

The board is modeled like any other structural component, most likely as a plate. Stiffness, damping, and mass distribution, along with external excitation and support conditions make up the system's mathematical model. The technology of fundamental shock and vibration response, examined in Chapters 2 and 3, can be applied to printed-circuit boards. Techniques of natural frequency prediction have been applied to printed-circuit boards by Arnold and Strother (ref. 4). They tested printed-circuit boards of various sizes, thicknesses, weights, and mounting configurations to obtain design data. The results of all tests were correlated into an equation that enabled them to predict natural frequency as a function of known board parameters (e.g., length, width, thickness, weight, modulus of elasticity, and mounting configuration). Figure 85 shows some examples of printed-circuit board vibration. Response technology is particularly applicable to design and mounting involving large printed-circuit boards that contain many components and intricate printed wiring. In these cases, the strategic location of supporting posts or other stiffening techniques must be considered during layout of the printed wiring.

Predicting printed-circuit board response and designing printed-circuit boards for limited shock and vibration response has been facilitated by the work of Derby and Ruzicka (ref. 5), who used structural composites with viscoelastic material layers to induce system damping. The work performed on

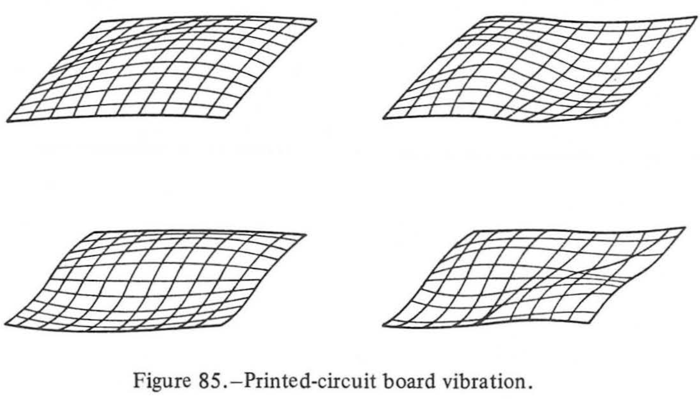

Figure 85.—Printed-circuit board vibration.

structural composites can be applied in designing printed-circuit boards that must function in severe vibration environments. These men have published a variety of design data, including resonant frequencies and structure-damping factors. Bieber et al. (ref. 6) reported test results of 11 commercially available damping specimens. In these tests, treatments were applied in various thicknesses and over various percentages of specimen surface area.

Mounting printed-circuit boards may increase the sensitivity of the circuit to shock and vibration if the board must be removable. In these cases the vibration model usually does not have rigid mounting provisions on the two sides. Further, a tradeoff exists between ease of board removal and the possibility of its being either loosened or removed during shock and vibration exposure. The possibility of shock-induced board removal is usually reduced by either increasing the number of pins on the connector or using a mechanical hold-down device. In either case, a tool–either a specially designed card extracter or a screwdriver–is often required to remove the board.

Potted assemblies (e.g., welded modules) require seals and filler material to ensure good thermal conductivity and to restrict relative motion of compo-

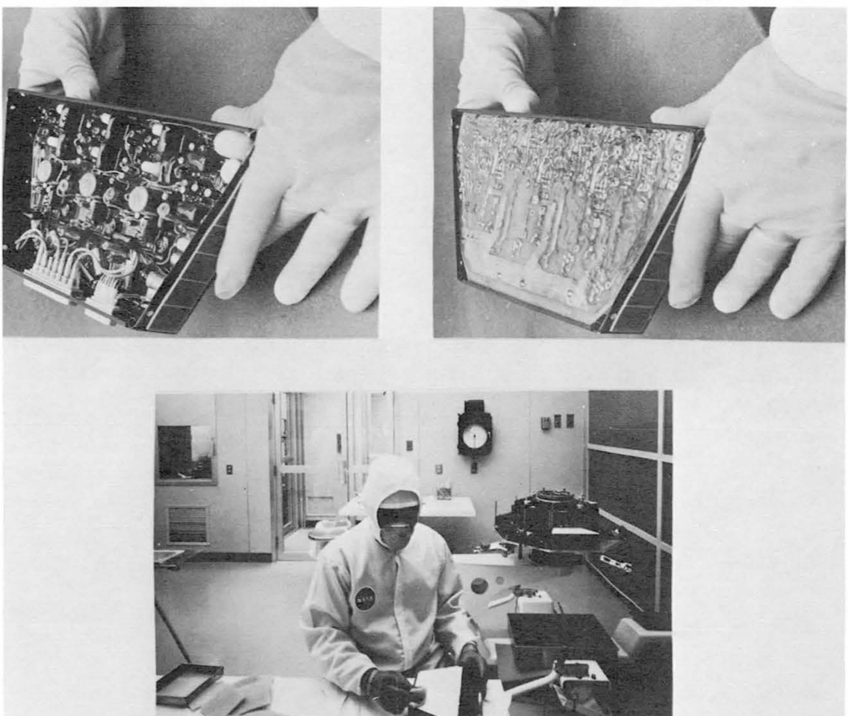

Figure 86.–Encapsulation of an electric subassembly.

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nents within the assembly. An electronic-circuit module in the process of being encapsulated is shown in figure 86. Provided the requirements have been met, a potted assembly is generally more reliable than a printed-circuit board because of improved structural integrity and reduced likelihood of component resonances causing lead breakage. A potted assembly, however, often has a greater mass and is more susceptible to unwanted loosening and removal with shock excitation than a printed-circuit board. Unwanted loosening of a potted assembly can be avoided by increasing the number of pins in the header, increasing the retention force of the receptacles, or employing mechanical hold-down devices. Some encapsulation techniques in which polyurethane foam is used have been reported by Le Doux (ref. 7).

COMPONENT MOUNTING

Only a few electronic components are individually sensitive to shock or vibration environments; the method by which a component is mounted, however, may substantially reduce its useful life. Many failures are related to elements that must accompany a mounted electronic component, including lead breakage and insulation breakdown; both can be attributed to the mounting configuration. The actual mechanism that holds a component to a chassis depends on the component. This survey relates only NASA-generated technology to the mounting of electronic components; however, Forkois and Woodward (ref. 8) have prepared a general discussion on shock and vibration design of electronic components and chassis.

The NASA-generated concepts for shock and vibration isolators discussed in chapters 2 and 3 can be used to design useful electronic-component mounts. With the exception of heavy components (e.g., transformers) that change the mass distribution of chassis or components (e.g., power resistors) that add structural integrity to a chassis, mounts can be designed using a single-degree-of-freedom vibration model. The shock-mount ideas covered in chapter 2 included the mechanisms of:

- Materials deformation
- Fluid compression
- Coulomb friction
- Materials yielding.

In chapter 3 the following mechanisms for vibration isolation were developed:

- Elasto-plasto-viscous damping
- Foam deformation
- Pneumatic compression
- Mass absorption
- Structural damping
- Viscoelastic damping.

The mountings used for components usually supported by input/output leads are important, particularly in high-frequency vibrational environments. Generally it is necessary to prevent relative motion of the component with respect to its mounting terminals in order to avoid lead breakage from fatigue. This can be accomplished by securing larger components, such as electrolytic capacitors with hold-down devices of the ordinary cable-clamp variety. For smaller components, with size and mass similar to resistors or capacitors, control of lead lengths during the mechanical layout of terminals or a protective polymer spray may be sufficient.

Foam encapsulation (mentioned above with respect to electronic subassemblies) is a way to mount a group of electronic components for protection from shock and vibration disturbances. Brackets with viscoelastic coatings or inserts (fig. 87) can provide excellent vibration isolation to fragile components. Ruzicka (ref. 9) has documented the shape factors and elasticity and damping constants required to design mountings. This fundamental work was sponsored by NASA in order to provide a basis for the design of vibration-controlled structures and machines.

The breakdown of insulators associated with certain components, (e.g., sleeves on electrolytic capacitors and mounting power transistors) is also caused by relative motion induced by high-frequency vibration. The techniques of support or encapsulation are again used to reduce the likelihood of possible abrasion and eventual failure of the insulation.

Heavy components such as transformers and solenoids must be mounted

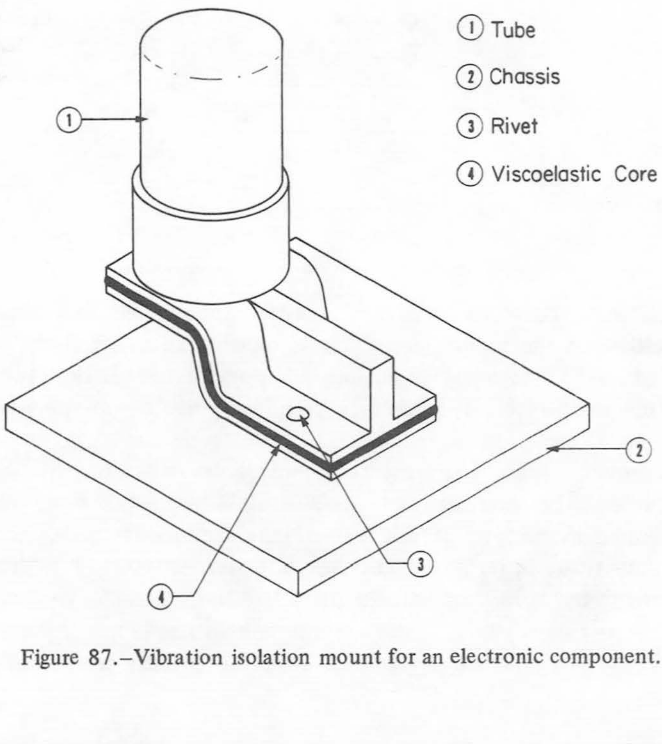

Figure 87.–Vibration isolation mount for an electronic component.

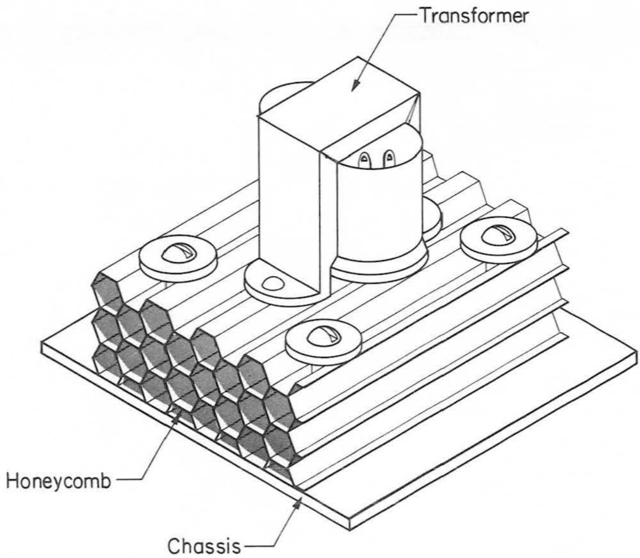

Figure 88.–Honeycomb vibration mounting for a transformer.

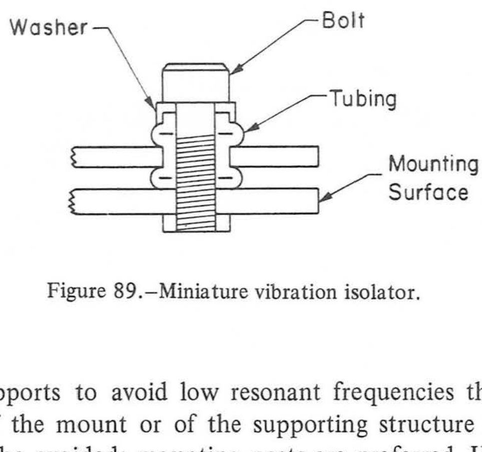

Figure 89.–Miniature vibration isolator.

on inflexible supports to avoid low resonant frequencies that cause fatigue failures either of the mount or of the supporting structure itself. Cantilever supports should be avoided; mounting posts are preferred. Honeycomb plate (fig. 88) makes an excellent lightweight, rigid support for components in which weight is a problem.

In components such as electron tubes, relays, crystals, and switches, relative motion causes either degradation or failure. Relative motion sometimes causes a displacement mode type of failure in which a component, although not overstressed, fails to be in its proper location with respect to some reference. This can constitute temporary or permanent failure. For example, if a switch accidentally closes due to shock, then the component has failed. Relative motion can also cause performance degradation locally in switch, relay, and potentio-

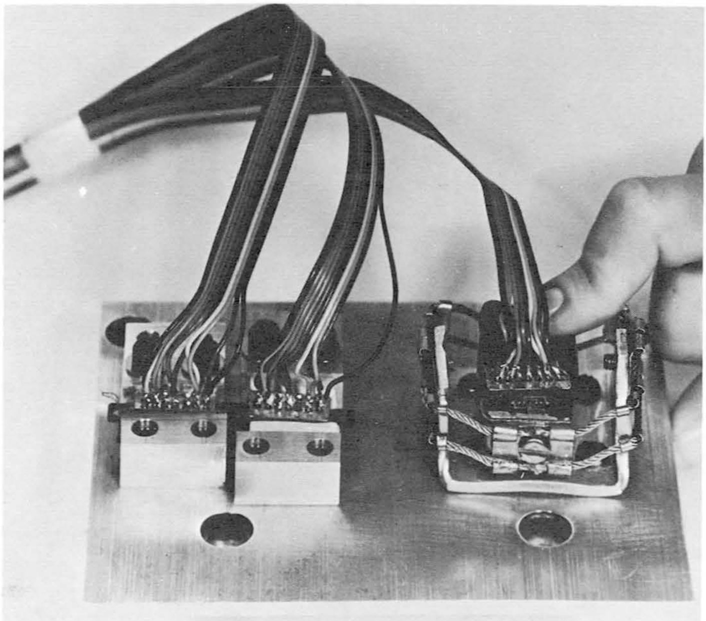

Figure 90.–Cable shock mount.

meter contacts. Local chatter due to some vibration disturbance eventually leads to permanent failure.

A number of shock- or vibration-sensitive components commonly are located on the front panels of electronic equipment; often therefore, it is helpful to isolate this panel from the rest of the unit. Adjustable components (e.g., potentiometers, chokes, and variable capacitors), as well as meters, switches, and indicator lamps are sensitive to shock and vibration. Figure 89 shows a miniature vibration isolator, which is suitable for either component mounting or isolation of an entire panel. Rubber tubing prevents contact with the vibrating chassis. Even though panel isolation is used, however, the designer should ensure that adjustable components will have locking mechanisms to prevent rotation. An effective cable shock and vibration mount (fig. 90) has been designed by James J. Kerley, Jr. of NASA. The cables provide a flexible mounting that is coulomb damped by the relative motion of its strands, an effective technique in shock and vibration mounting of small, fragile components. An associated innovation, described in Chapter 3 (fig. 77), is the wire-mesh isolator. It provides good protection for a sensitive component subjected to shock or vibration. This idea, which originated at Goddard Space Flight Center, could be extended to enclose large equipment or subassemblies (such as the relays shown in figure 91) or the mounting of switches, tubes, chokes, potentiometers, and variable capacitors.

Large fragile electronic components such as cathode-ray tubes require spe-

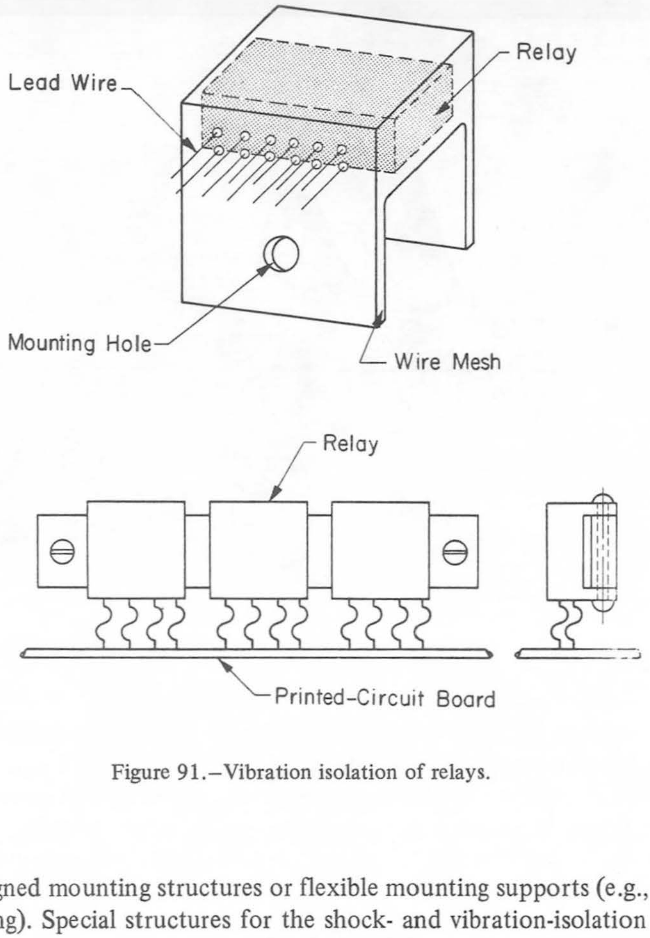

Figure 91.–Vibration isolation of relays.

cially designed mounting structures or flexible mounting supports (e.g., a pneumatic spring). Special structures for the shock- and vibration-isolation mounting of electronic components can be designed by applying NASA-developed technology concerned with damping mechanisms and structural-response calculations.

In many cases, the design of component mounts for shock and vibration isolation involves single-degree-of-freedom systems; and the shock-spectra method of design reviewed in chapter 2 is especially useful. If the weight, fragility, and input-disturbance waveform of a component are known, a designer can devise a shock or vibration mount with the aid of a simple shock-spectra chart. This technique is particularly attractive where a large number of fragile components are subject to the same environment; in this case only one shock spectrum is needed for a given input disturbance waveform.

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Application to Electronic Equipment

The annoying rattles of equipment housing, automobile bodies, or heating ducts demonstrate the effect of shock and vibration disturbances on lightweight flexible structures. The shock and vibration responses of a structure cause its failure as well as the failure and performance degradation of mounted components.

Natural frequency and response analyses of plates, beams, and structures have been developed by NASA for the design of space vehicles, electronic equipment, and components; these analytical results already have been useful to structural engineers in the design of buildings, containers, and packages. This technology also is directly applicable to the design of chassis, mounting shelves, closure panels, and electronic equipment.

New techniques (e.g., stiffening, damping, and absorption) in panel design have resulted in more reliable functioning of panel-mounted components. These design techniques for chassis and cabinets are also applicable to electronic packaging. NASA has designed effective suspension systems that isolate an entire piece of equipment from its shock and vibration environment. Shock isolation concepts (discussed in chapter 2) apply in both industrial and military electronics. Low- and high-frequency vibration suspension concepts will be related to fragile electronic equipment in this chapter. Innovations in testing electronic equipment and components also are considered.

DESIGN CRITERIA

Design criteria for electronic equipment packaging are expressed in terms of the response of chassis, panels, and other mounting fixtures. Displacement and acceleration-response data allow determinations of acceptable shock and vibration levels at the point of component mounting. The measure of cabinet- or chassis-response acceptability refers to either the equipment itself or the environment the equipment provides to the component performance. A hostile component environment is no more acceptable than a structural failure. The design criterion can be either deflection or stress, depending on the sensitivity of the mounted component. Resonant responses, therefore, that cause both deflection and stress failure must be avoided.

PANEL DESIGN

The basic structural members used in electronic equipment housing are plates, beams, and gussets. Since a component usually is mounted on a plate-like structure, the technology available for the design and analysis of panels and panel-like structures will be discussed in this section. If a panel is flexible compared to its supporting members, the panel can be analyzed. Where members of a cabinet are of about equal flexibility, however, an analysis must be conducted on the entire structure.

Coatings, sandwich construction, honeycomb construction, viscoelastic cores, constrained polyurethane foam, and fiberglass cores have been studied by NASA in research concerned with reducing the shock and vibration response of panel-like structures. Powell and Stephens (ref. 1) used sandwich panels and determined their vibration characteristics over a wide range of environmental pressures or densities. Natural frequencies and damping were determined for several vibration modes in panel with aluminum honeycomb and polyurethane-foam cores. The results showed no significant changes in natural frequency or internal damping at low pressures.

Structural composites with viscoelastic shear-damped cores provide a tremendous vibration-attenuation capability. NASA has sponsored investigations that contain design data on structural damping, including shape factors (refs. 2 and 3). Figures 92 and 93 show the viscoelastic shear-damped solid plates and viscoelastic shear-damped honeycomb plates investigated by Ruzicka.

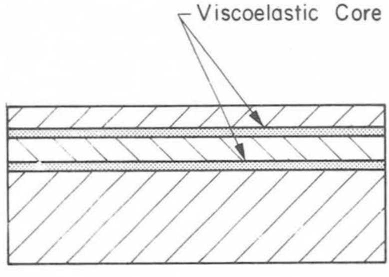

Figure 92.–Viscoelastic shear-damped solid plates.

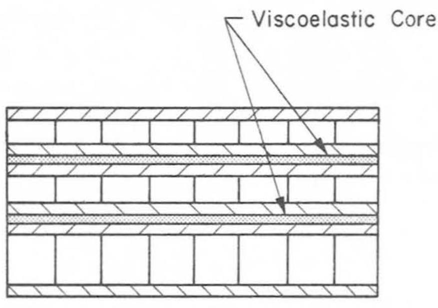

Figure 93.–Viscoelastic shear-damped honeycomb plates.

Plate of honeycomb construction provides high stiffness with low weight. Design data on hexagonal cell structures were generated by McFarland (ref. 4), while Lewallen and Ripperger (ref. 5) investigated the crushing-strength and energy-dissipating characteristics of Trussgrid aluminum honeycomb. These available data are useful in designing cabinets, panels, and chassis that provide vibration-free housing and mountings for electronic equipment.

Eleven commercially available damping treatments were applied by Bieber et al. (ref. 6) to cantilever beams commonly used for internal support structures in missiles and spacecraft. These treatments can be used for panels as well as other structures where local damping is desired. Elastomeric foams were characterized by Liber and Epstein (ref. 7), who showed methods for determining the dynamic response of foam-isolated structures. In their analytical technique foam can be used for external isolation of equipment or the isolation of components within a chassis.

Analysis of the response of panels to shock and vibration environments was reviewed in Chapters 2 and 3. Modeling techniques by Regier (ref. 8) provide guidelines for the preparation of models usable with NASA-developed analytical methods. A typical panel, modeled for vibration analysis, is shown in figure 94. Analytical methods for natural frequency and response calculations, examined in Chapter 3, apply to panels. In addition, the SAMIS and NASTRAN programs, both developed by NASA, can be used for the design of a panel containing many mounted components.

CHASSIS AND CABINET DESIGN

Since the design problems of cabinets and chassis are mutual, they are considered together. Although the design of electronic chassis and cabinets involves techniques and concepts similar to those used for panels, additional considerations include the design of structural joints, the use of supporting structural members, and the need for more elaborate mathematical models. Two extremely desirable features in a structure are rigidity and low weight. Low weight usually implies low cost, and rigidity is associated with lack of vibration response. In combined shock and vibration environments it is best to have rigid structure and to mount the entire structure on a shock-isolation device. If vibration is not a problem, shock isolation can be obtained by flexible structural design.

In cabinet design, Forkois and Woodward (ref. 9) have provided techniques for stiffening joints, panels, and channels with gussets, angles, and welded joints. A typical stiffened cabinet section is shown in figure 95. While many of the techniques reviewed in the report still are valid, recent NASA work has yielded new technology that can increase the capability of the cabinet designer.

The most notable improvement in cabinet design has been the development of rigid, lightweight, damped panels and structural shapes. In addition, new joint-design techniques, materials, and absorption devices are available for elec-

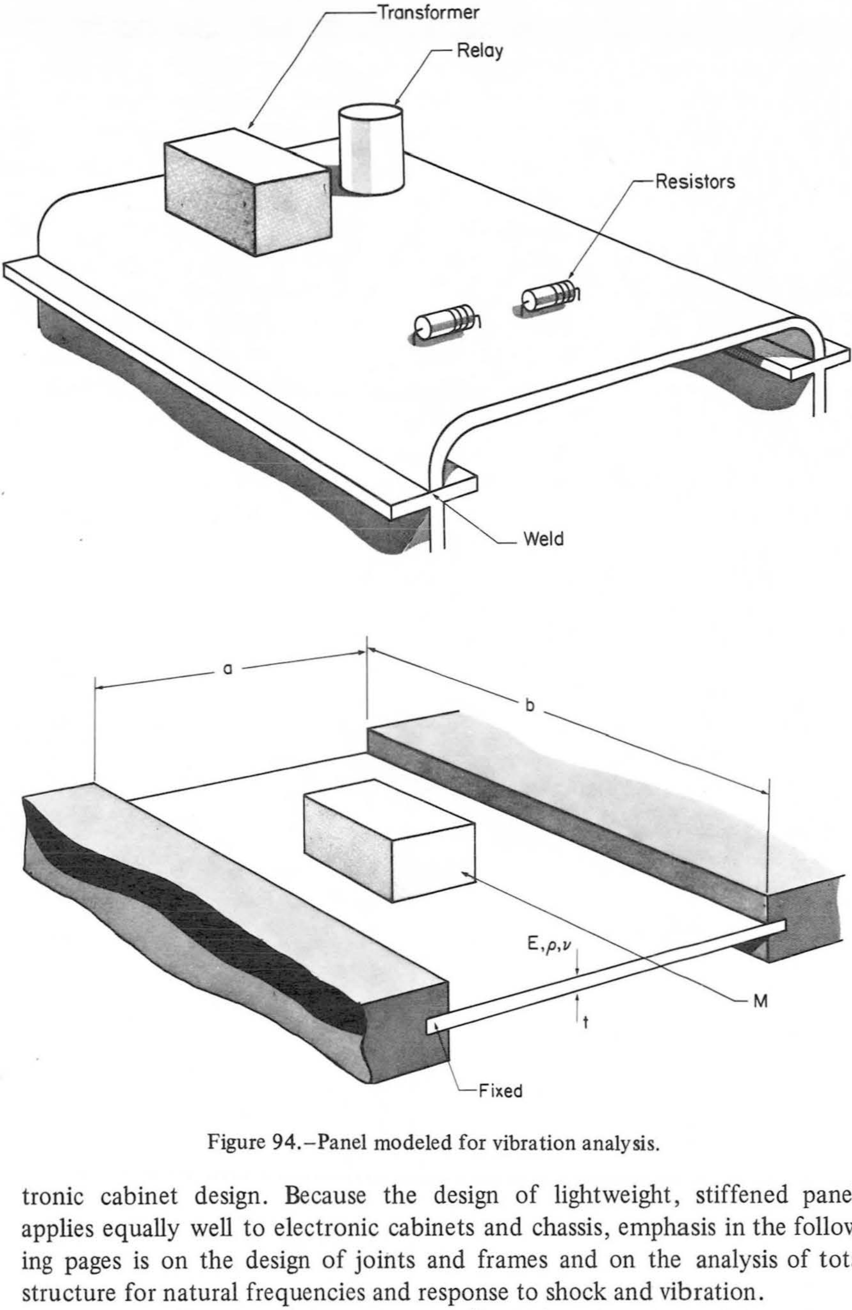

Figure 94.–Panel modeled for vibration analysis.

tronic cabinet design. Because the design of lightweight, stiffened panels applies equally well to electronic cabinets and chassis, emphasis in the following pages is on the design of joints and frames and on the analysis of total structure for natural frequencies and response to shock and vibration.

In one structural fastening technique (fig. 96), enlarged rivet holes are used to obtain extra coulomb damping in a joint. NASA's Jessie M. Madey attenuated the structural response of unmanned satellites in this way. Joints constructed of aluminum on anodized aluminum also have been found to be excel-

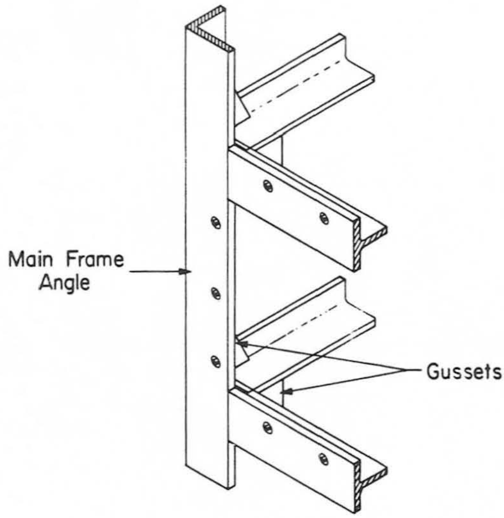

Figure 95.–Cabinet stiffened with welded channel.

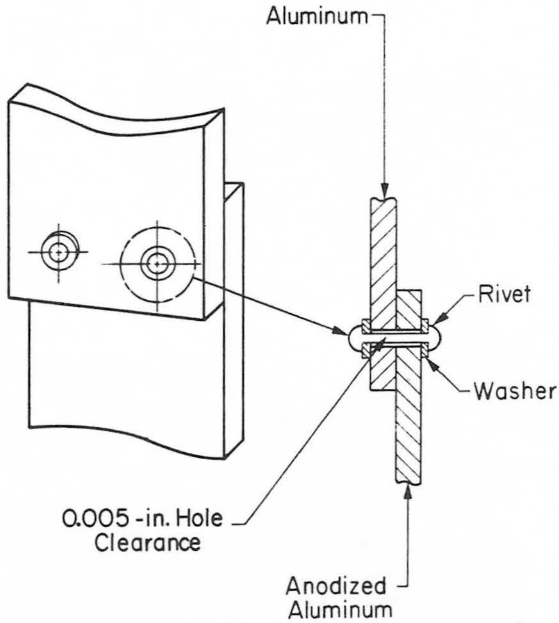

Figure 96.–Coulomb-damped joint.

lent vibration attenuators. These techniques are useful in designing the damped, lightweight structures that house electronic equipment.

Structural composites with viscoelastic inserts can be used as frames and supports. The constrained viscoelastic material acts as a barrier against noise and transmission of vibration. Since structural-loss factors and stiffness values for these composites have been documented in NASA reports (refs. 2 and 3), design data on damped, structural shapes are available for use in electronic chassis and cabinet design. Some typical structural elements investigated by Derby and Ruzicka (ref. 2) are shown in figures 97 and 98.

NASA has contributed to analytical and experimental response determinations of structures loaded in shock and vibration. Data documented in chapters 2 and 3, include Regiers' (ref. 8) work on the modeling of structures for shock and vibration analysis. McMunn (ref. 10) reports on multiparameter optimum damping in linear dynamic systems. Although in its initial stages, as far as design application is concerned, this work has potential application in controlling the response of structures such as cabinets and chassis.

Vibration response may become a problem in an existing cabinet or chassis because some excitation occurs nearby. Since it is not practical to redesign and rebuild the cabinet, a vibration response "fix" is used. Vibration mass absorbers (chapter 3) provide one way to solve the problem if the response has a single frequency. The parallel-damped dynamic vibration absorber is another

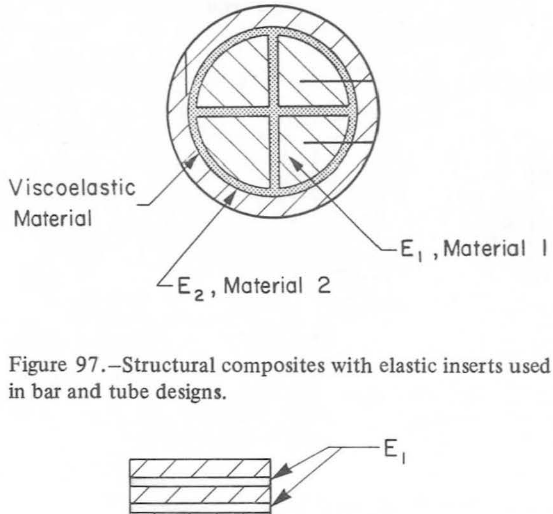

Figure 97.–Structural composites with elastic inserts used in bar and tube designs.

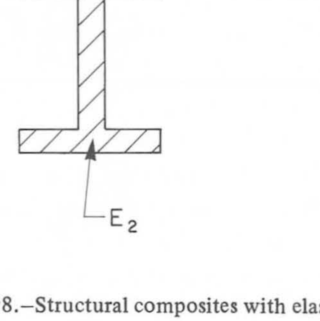

Figure 98.–Structural composites with elastic inserts.

passive, response inhibitor for a single-frequency vibration response. If the response frequency varies, then the gyroscopic absorber, which has multiple excitation control capabilities, can be used. Figure 99 is a schematic of a cabinet with a vibration absorber attached. Other passive absorption devices also are examples of NASA-generated innovations that can modify existing equipment. A viscous-pendulum damper attached to a cabinet is shown in figure 100. The slugs sloshing in fluid absorb structural vibration response. The chain damper shown in figure 101 acts as an absorber through friction. Also, built-in slosh tanks (fig. 102) will alleviate structural response problems locally but must be incorporated in the design of the original structure.

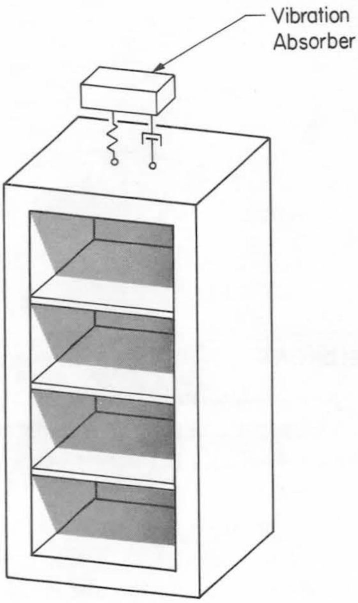

Figure 99.–Cabinet with an attached vibration absorber.

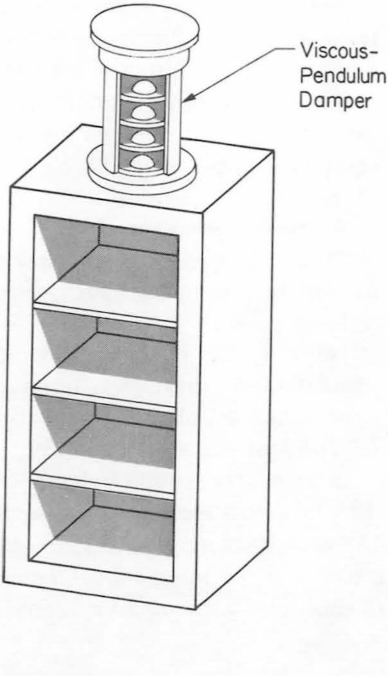

Figure 100.–Cabinet with an attached viscous-pendulum damper.

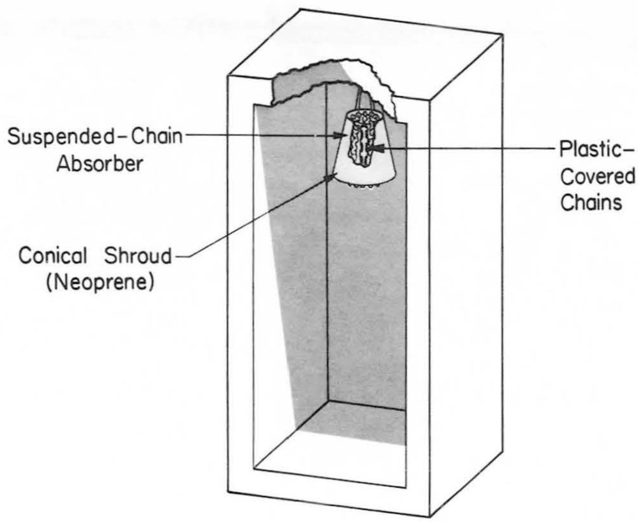

Figure 101.–Suspended-chain absorber.

VIBRATION SUSPENSIONS

Complex electronic equipment (such as tape recorders and electron microscopes) is extremely sensitive to vibration environments and should be isolated from the surroundings with a low-frequency mount. NASA has used vibration mounts in several instances for electronic equipment.

A low-frequency vibration mount, which can be used to obtain large vibration strokes, was designed by Reed (ref. 11). The mount (fig. 103), with equipment in place, can have a 1-Hz natural frequency, yet support the weight of the isolated equipment.

A low-frequency air-spring suspension system, reported by Dixon and Pearson (ref. 12), may be useful to industry. Figure 104 shows the low-frequency suspension system used for electronic equipment tests. The fact that variable isolated mass of the electronic equipment is allowed makes it particularly attractive for testing heavy generator sets or other similar equipment. An ordinary vibration shaker will not support the weight of heavy equipment; this low-frequency isolation mount eliminates the need to design new suspensions for different pieces of equipment.

In many cases, good isolation of sensitive electronic equipment can be obtained through the ingenious use of standard commercial isolators. Conn (ref. 13) used standard isolators, for example, in an isolation system for the tape recorder in the International Satellite, UK-2. The lightweight isolation system (a small unit with good load-carrying capabilities and damping characteristics)

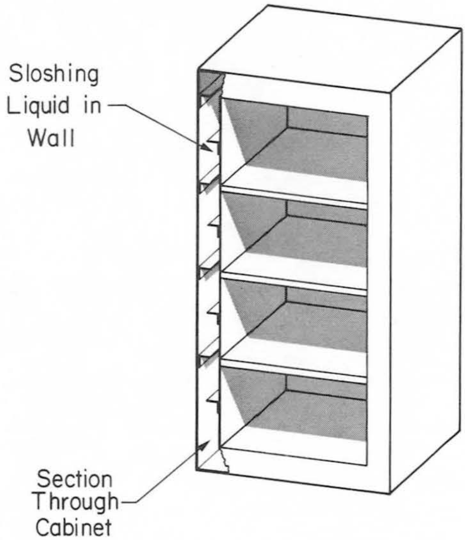

Figure 102.–Liquid slosh damped cabinet.

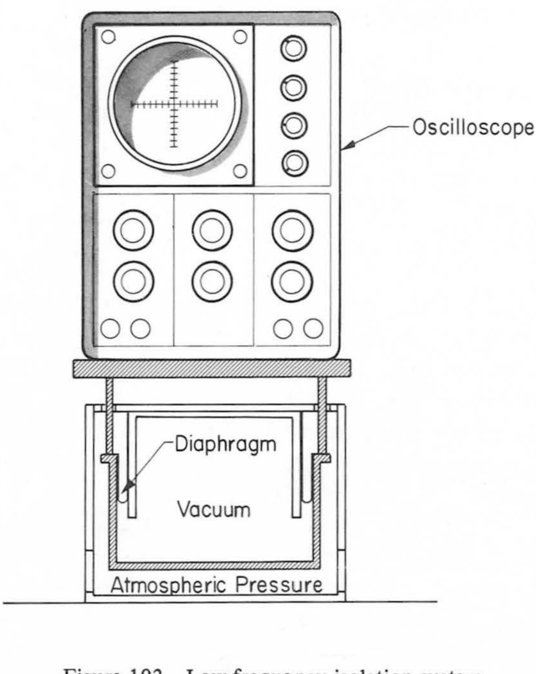

Figure 103.–Low-frequency isolation system.

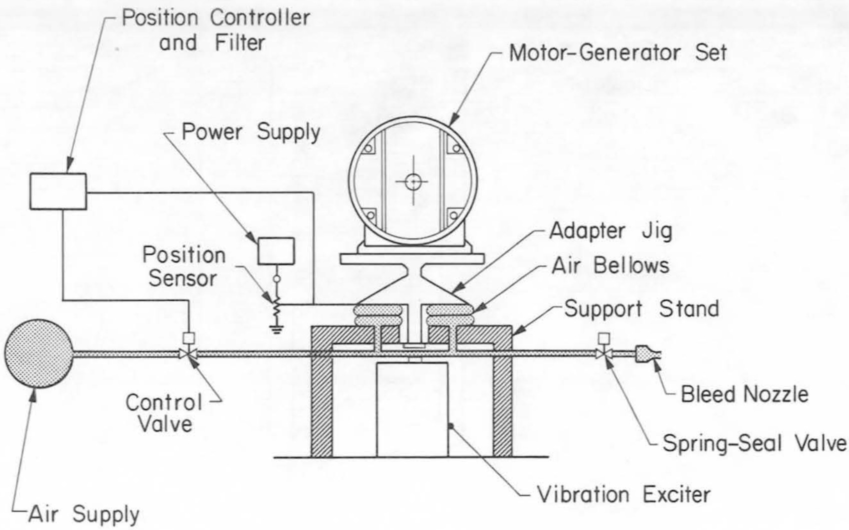

Figure 104.–Test support for electronic equipment.

has widespread application in electronic equipment subjected to vibrational excitation in all three planes.

Weinstock (ref. 14) conducted a design study on a two-axis servomechanism leveling system controlled by gyroscopes and level sensors mounted on a massive conventional pneumatic isolator (fig. 105). At frequencies below 0.012 Hz, the system is controlled by the level sensors; the gyroscopes maintain control from 0.012 to 25 Hz. Above 25 Hz, the leveling system is locked out, and the test device and the massive frame act as a rigid body mounted on springs. This results in a 1.0-Hz damped vibration system and provides a stable, seismically inactive test platform for instrument calibration and performance testing.

A unidirectional isolator, called the spacecraft isolator (ref. 15), consists of a flexible elastomeric support with coulomb-friction energy dissipation between the inner and outer isolator rings (fig. 106). This isolator, a useful addition to vibration isolation technology in the 10- to 100-Hz frequency range, was designed to isolate the payload from longitudinal vibration of the launch vehicle. Industrial application might be as a uniaxial vibration isolator of equipment from high-frequency environments.

In figure 107 steel cables are shown suspending a system for shock and vibration isolation. The length, size, and pretensioning of the cables can be varied to obtain the desired support characteristics. Coulomb friction between cable strands adds valuable damping since it does not transmit high-frequency effects. This method of suspension was developed at NASA Goddard Space Flight Center for testing components and equipment for launch vehicles.

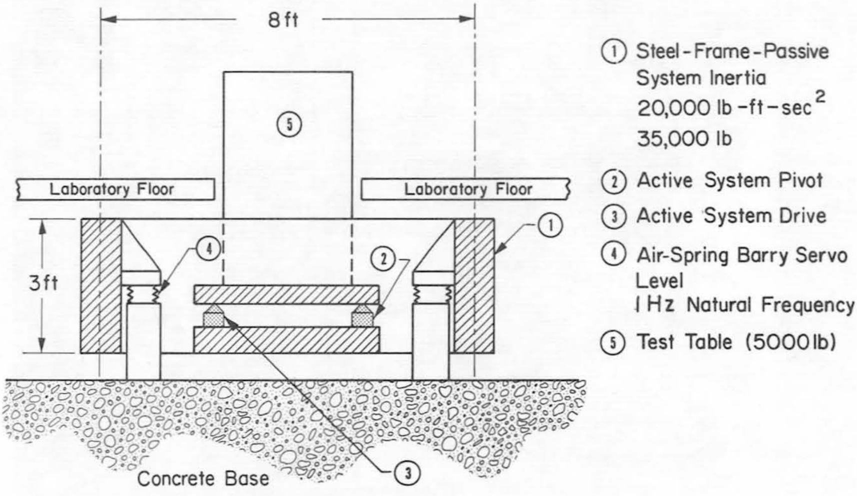

Figure 105.–Active (servo-control) base-motion isolation system.

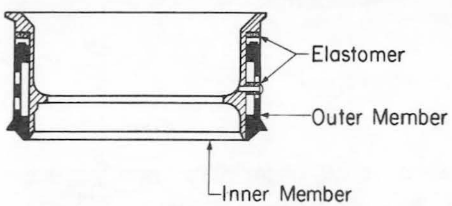

Figure 106.–Mounting assembly.

SHOCK SUSPENSIONS

The cable-mounting suspension system shown in figure 107 provides both good shock and vibration isolation. It is likely that longer cables would be valuable in obtaining a softer suspension system.

Figure 108 shows typical shock suspension systems that could use isolator components (discussed in Chapter 2); several of these components were based on material deformation mechanisms. One vital consideration in designing a shock-suspension system is the number of times it will be exposed to shock-excitation cycles. When the absorbers in figure 108 are useful for a limited number of duty cycles, they could be very effective as a crash-protective device for electronic equipment mounted in automobiles, trucks, planes, and trains. Existing special-purpose equipment may reduce design time.

Multiple shock-design concepts, which include fluid-compression devices (e.g., liquid and pneumatic springs) coupled with orifices and coulomb-damping mechanisms, can be used to isolate most types of electronic

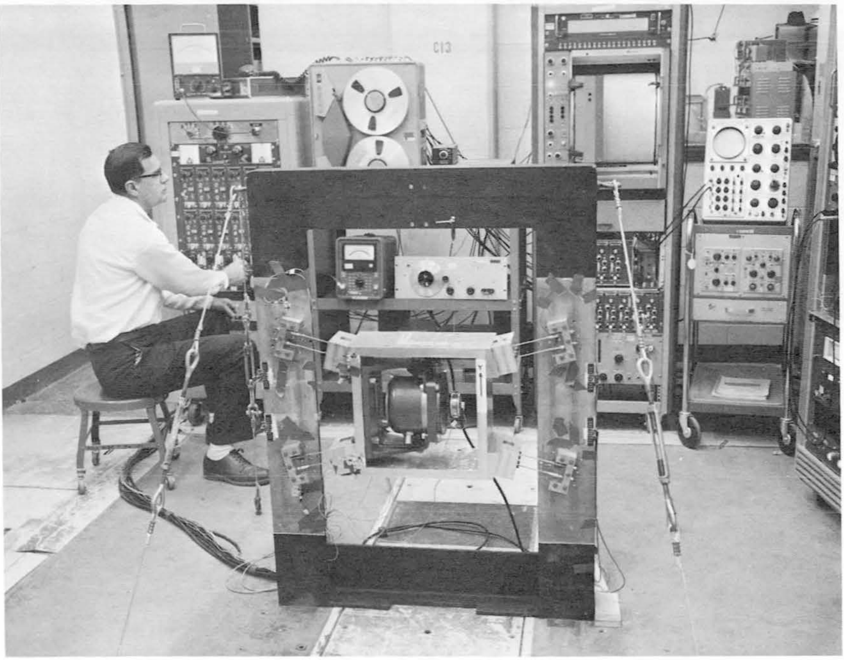

Figure 107.–Cable isolation.

equipment from shock disturbances. One typical, major design problem is the space required for satisfactory shock isolation since large shock attenuation requires considerable movement of the isolated equipment. The quality of a shock suspension system can be determined with shock-isolator synthesis and optimization techniques described in chapter 3.

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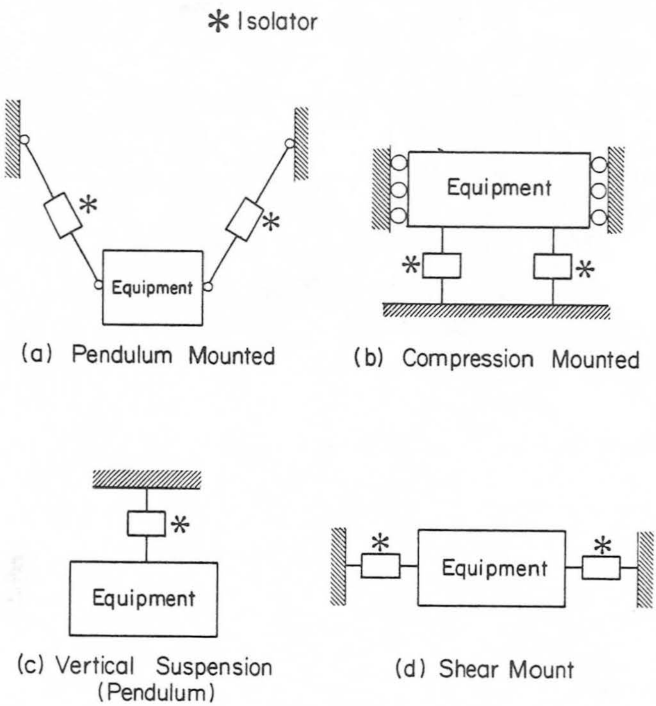

Figure 108.–Typical stock-suspension systems.

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Chapter 6

Application to Rotating Machines

Because design technology has developed so rapidly, it is now possible to design machines that operate smoothly at 50 000 rpm and above. Several factors are responsible for these technological advances. The digital computer has made possible the analytical solution of many complex problems in fluid-film bearings and rotor behavior. New experimental techniques on high-speed equipment have made possible the testing and improvement of machines under actual operating conditions. Design technology has been advanced by NASA in the areas of fluid-film bearings, high-speed rotor behavior, and design and development of turborotors that operate with a stable low-level amplitude. The industrial development of rotor-bearing techniques was motivated by the need for designs for jet engines, turbines, and centrifugal compressors. Even though sophisticated techniques are now available, design processes are being refined and new concepts in bearings and rotors are continually introduced. The extensive technological literature on rotating-machine design includes Gunter's monograph (ref. 1), which describes most of the techniques needed for designing high-speed machines. The monograph has a number of references to English literature pertinent to this field. Tondl (ref. 2) and Dimentberg (ref. 3) have written comprehensive texts, which reflect the nonEnglish technical literature in rotor dynamics.

In this chapter, NASA-generated shock and vibration technology is reviewed as applied to electric rotating machines (fig. 109) such as tape recorders, electric motors and generators and disk and drum recorders. Design criteria, such as stability, response, relative motion between rotor and stator under various environmental conditions (e.g., shock, vibration, and unbalanced mass) are related to this equipment. The design technology related to rotor and rotor support, fluid-film bearings and ball bearings is reviewed for the equipment listed above. In addition, existing standards on rotor response are discussed from the point of view that optimal machine performance should be obtained at minimum cost. Shock mounting of rotating machines is discussed with respect to performance and survival during and after application of a shock load to a machine support.

Vibration control, an important aspect of design, involves controlling the effect of machine-mass unbalance or gear inaccuracies. On the other hand, vibration isolation involves the inverse problem of environmental effects on the electric rotating machine. The majority of the new design technology available for vibration control and isolation is analytical.

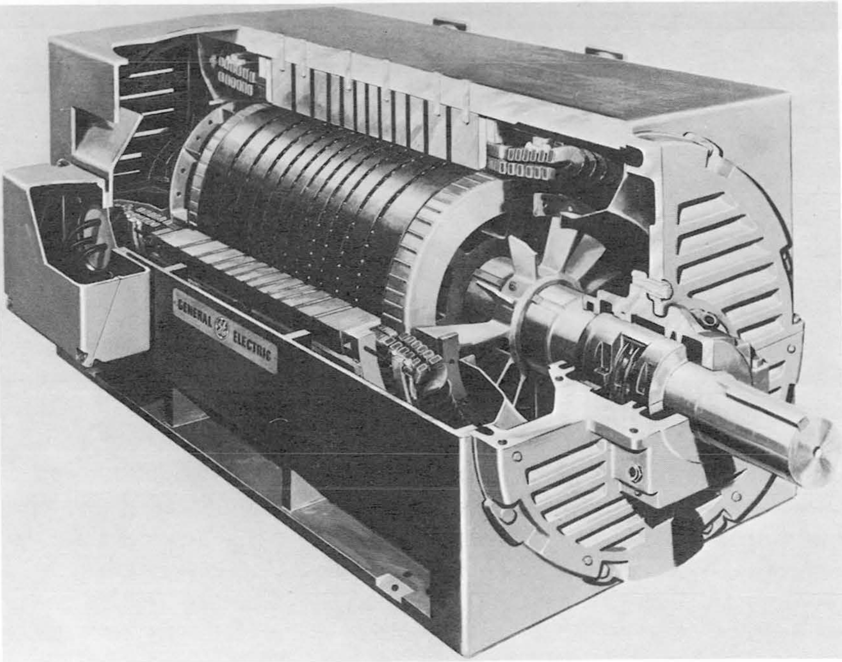

Figure 109.–Electric motor. (Courtesy of General Electric Co.)

ROTOR-BEARING DESIGN

Since high-speed rotating machinery involves many complicated phenomena, analysis and experimentation are important aspects of the design process. The rotor-bearing system includes a rigid or flexible rotor and a minimum of two support bearings. An example of a high-speed rotating machine is the generator rotor shown in figure 110. The rotor is rigid or flexible according to the relationship between its first flexural natural frequency[13] and the rotor's speed. If a rotor must run near or above its first flexural natural frequency–not uncommon in high-speed machines–the local response of the rotor contributes to its running quality. Support bearings are selected for minimum resistance to rotational motion, stability, and minimum response. Roller or ball bearings usually are used in machines running at low speed (somewhat less than 10 000 rpm); various types of sleeve bearings are used above this speed. Support bearings are an important part of a dynamic system because they have stiffness and damping qualities. In fact, support-

[13] Flexural natural frequency: a natural frequency associated with the bending of a rotor.

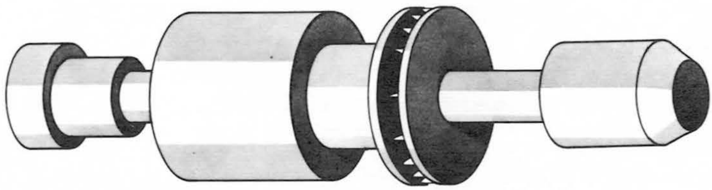

Figure 110.–High-speed generator rotor.

bearing damping is vital to response control when a rotor-bearing system passes through one of its critical speeds.[14]

CRITERIA

The criteria used to design a rotating machine depend upon its ultimate function, performance, and speed range. In high-speed machinery with fluid-film support bearings, rotor stability is of major importance. Because fluid-film bearings have nonlinear force-deflection characteristics, it is possible for the motion of a rotor to grow until the system destroys itself. The response of a rotor is always a design criterion because the rotor must operate smoothly over a predetermined speed range even though there are critical speeds in this range. Rotor-response velocity is a valid failure criterion for frequencies in the range of 10 to 1000 Hz; often, however, relative motion between rotor and stator is the design criterion selected. Relative motion, for example, is important in a drum recorder. The allowable level of response velocity for any machine depends upon its strength and the way response affects performance. Examples are tape reels that excite moving tape to cause wow, flutter, or both if the motion of the reels is anything other than constant rotary velocity.

The effect of rotor-bearing response on surrounding equipment is a final consideration when design criteria are selected. Excessive rotor response, transmitted through the machine bearings to the foundation, may cause malfunctioning of adjacent equipment and human discomfort.

ENVIRONMENTS

The disturbances that cause vibration in a rotor-bearing system may originate within the machine, as a result of its rotational motion; or outside, as a result of a shock and vibration. Within a rotating machine, the principal cause of vibration response is mass unbalance that imposes a one-per-revolution excitation on the system. The mass unbalance of a system is related to a mass distribution in the rotor. In reality, because factors such as material

[14]A critical speed is so defined because of its proximity to a system's natural frequency. In other words, the rotor rotational speed provides the system excitation. This is also called resonance.

heterogeneity, machining errors, keyways, slots, and windings cannot be eliminated in machines, complete rotor balance is never achieved. Much time and effort is expended each year on the dynamic balance of rotating machines; both stationary and transportable balancing equipment is used.

Asymmetric shaft stiffness, another source of excitation commonly found in electric motors, causes instability in a system rather than a large controllable response. This instability can be removed only by redesigning the rotor geometry. Asymmetric shaft stiffness, a self-excited phenomenon, is caused by the motion of the rotors. Similar phenomena are internal friction in the rotor and external friction caused by disks or gears shrunk on the rotor. These phenomena also cause rotor instabilities.

The rotor response caused by pulsating torque applied to a flexible rotor can be controlled by damping. Unstable motion of a rotor can result, however, from certain combinations of rotor geometry, speed, and torque pulsations. The cutting of flux at integeral poles causes natural torque pulsations in an electric motor or generator.

Exterior shock and vibration affect rotor response because they are transmitted through the supports, and can induce rotor instability.

ANTIFRICTION-BEARING SUPPORTS

Antifriction bearings such as rollers, needles, and balls are utilized to support low-speed equipment. Figure 111 shows a typical ball bearing that can be used in electric motors. The lack of damping to control rotor motion is one disadvantage of antifriction bearings which are inflexible, linear springs. The major advantages of these bearings are their stiffness and linearity; such features provide high critical speeds with no stability problems. Unfortunately, however, the speed and load of antifriction bearings are limited. Design techniques for determining the life and load capacity of these bearings are well documented in the manufacturers' literature.

FLUID-FILM BEARINGS

Technological advances developed through NASA research programs concerned with fluid-film bearings were used in designing the high-speed rotors that are part of space power systems. Fluid-film bearings, which either generate support pressure through rotation (self-acting) or are externally pressurized, have a potential for all electrical rotating machines. Analyses of load capacity and dynamic characteristics are essential features of good machine design. In this case, dynamic characteristics (stiffness and damping) of fluid-film bearings and rotor equations of motion are used to analyze rotor response to mass unbalance or any other disturbance.

Methodology and design charts are available in NASA reports, cited in

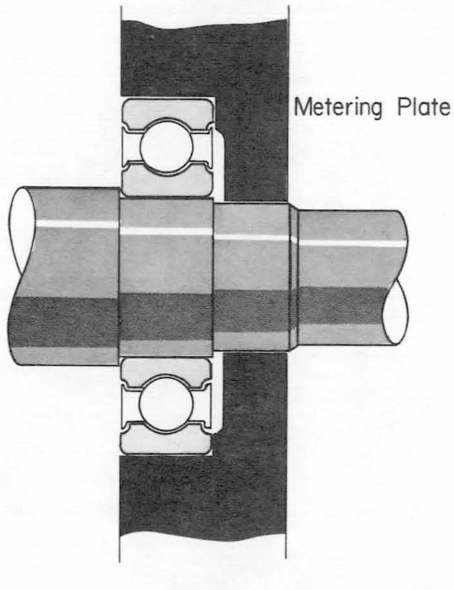

Figure 111.–Ball bearing for electric motor.

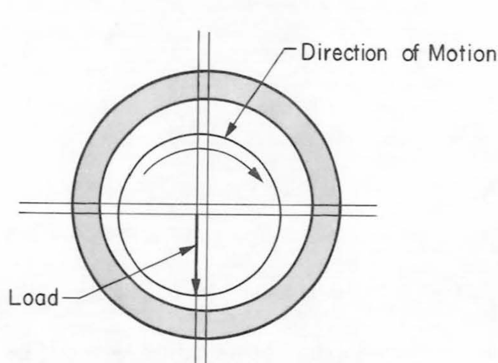

Figure 112.–Full journal bearing.

chapter 3, for common bearing configurations such as: (1) full journal (fig. 112), (2) tilting pad (fig. 113), and (3) squeeze film (fig. 114).

Hydrodynamic-bearing equations for a full journal bearing are available in Gunter's monograph. This monograph contains references to reports and papers that include design data and dynamic and steady-state characteristics of self-acting, incompressible, and compressible fluid-film bearings. These reports and papers also present methods for calculating bearing stability and frictional resistance to rotational motion. Sternlicht (ref. 4) has summarized useful

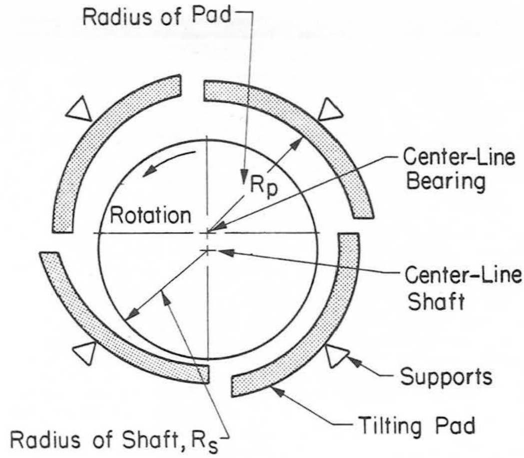

Figure 113.–Schematic of a tilting-pad bearing.

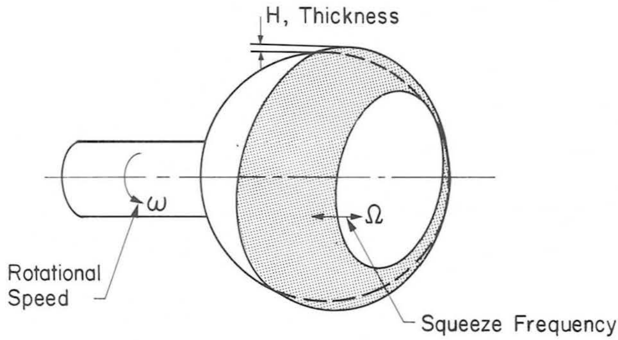

Figure 114.–Squeeze-film bearing.

design techniques for developing rotor-bearing systems; he has an excellent design procedure and refers to design charts.

One basic disadvantage of the full journal bearing is its tendency to become unstable and induce whirling at high speeds. This problem manifests itself with high-speed, lightly loaded rotors. Cunningham et al. (ref. 5) experimented on the stability of herringbone-grooved gas-lubricated journal bearings to high-fluid compressibility. This study confirmed the analytical results of Vohr and Chow (ref. 6) who showed that stability is achieved in full journal bearings when herringbone grooves are utilized. The geometry of the grooves is important in determining stable rotor speeds. The grooved bearing is particularly applicable to electrical rotating systems since they are lightly loaded and would be subject to stability problems if full journals were used. Tilting-pad bearings are stable as long as the pads track the motions of the journal center. An important advantage of the tilting-pad bearing is its stability under all operating conditions. A light-load capacity is not a significant problem with electric machines.

Squeeze-film bearings (fig. 114) depend on a viscous-flow process in which the pressure in a liquid-bearing film increases above ambient when the bearing gap is closing and decreases below ambient when the bearing gap is opening. The high-frequency oscillation of the bearing surface and the nonlinear nature of the gaseous flow in the film combine to increase the time-averaged film pressure above that of the ambient; as a consequence, load carrying capacity is generated. Pan (ref. 7) published a NASA-sponsored work on an asymptotic analysis of gaseous squeeze-film bearings. Design charts are included for conical squeeze-film bearings that will support radial and thrust loads. Chiang et al. (ref. 8) published results of a fundamental NASA study on spherical squeeze-film bearings. These bearings can be used in the design of motors and generators that must accept thrust loading. In addition, heavier load-carrying ability is generated with squeeze-film bearings. These bearings are useful for large electric generators with heavy rotors.

ROTOR RESPONSE

The response of electrical rotating machines to mass unbalance and other excitations can be determined using recent NASA-developed rotor-bearing technology. In addition, critical speed calculations, discussed in chapter 3, are applicable to electrical rotating machines. The modeling of a rotor-bearing system must reflect the mass and flexibility of the rotor and the damping and stiffness of its fluid-film supports. The generation of a rotor design by solving a system of equations of motion (including a description of the fluid-film bearings and the rotor continuum) is not yet possible. Instead, fluid-film-bearing dynamic characteristics must be used to calculate support stiffness and damping constants. A schematic of a typical rotor-bearing system (analytical model) is shown in figure 115.

A very stiff rotor, often used on fluid-film or antifriction-bearing supports, simplifies critical-speed and rotor-response calculations because the rotor is modeled as a rigid body. Cavicchi (ref. 9) has calculated critical speeds for a rigid rotor of varied stiffness and reported them as nondimensional design plots. In analyses for response and stability, the rotor often is considered rigid with damped-spring supports. This is especially true if the fluid-film bearings are much more flexible than the rotor.

Lukas (ref. 10) reported on the design of a high-speed turbocompressor supported on a spiral-grooved, pump-in thrust bearing (fig. 116); he analyzed the power loss of a gas-lubricated journal bearing, its load carrying capacity, and its response. This technology, used by NASA to build a radial flow turbocompressor in a brayton cycle space system, is an example of the way in

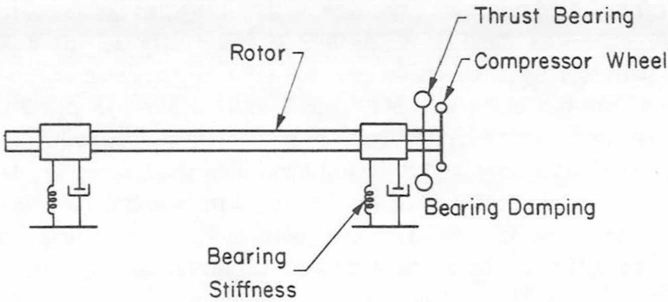

Figure 115.–Analytical model of a rotor-bearing system.

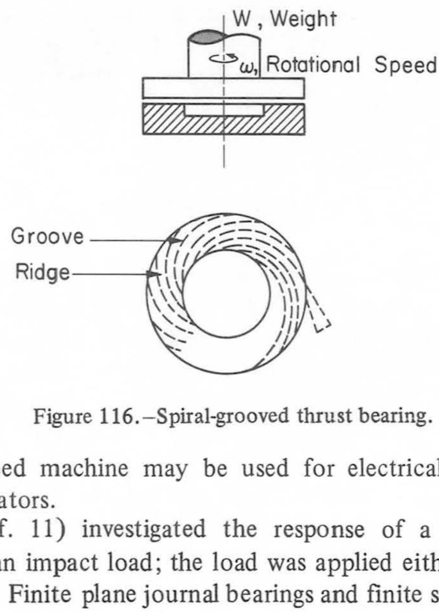

Figure 116.–Spiral-grooved thrust bearing.

which a high-speed machine may be used for electrical equipment such as motors and generators.

Malanoski (ref. 11) investigated the response of a rigid-bearing simple rotor system to an impact load; the load was applied either to a rotor or to a pedestal support. Finite plane journal bearings and finite spiral-grooved journal bearings were analyzed by applying a unit-step impulse to the rotor-bearing system and determining its response. This technology is directly applicable in the design of electric machines that undergo the type of shock that occurs in transportation or blast loading.

Gunter (ref. 12) has shown the effect of internal friction on the stability of a rotor. This friction is often caused by the relative motion between a disk shrunk onto the rotor and the rotor. Gunter calculates whirl orbits of an unbalanced rotor with internal friction damping. This type of data is important in shrink-fit designs.

Shaker (ref. 13), Nowak (ref. 14), Alley (ref. 15), and Loewy (ref. 16) reported on methods for determining critical speeds of multistation flexible supports; these speeds are important in many machine designs. The response of a rotor to external excitation can be determined with methods used by Nowak, Loewy, and Craggs (ref. 17). Analysis of the critical speeds and the forced response of a rotor-bearing system determines two things: (1) the acceptability of the selected bearing and (2) the limits of unbalance that will provide a smooth-running motor. This technology is directly applicable to electrical rotating machines.

In any electric machine in which torque fluctuations are present, torsional vibration may occur. Wingate (ref. 18) has reported an analysis of beam-like structures with multiple branches and variable sections that could be applied to these rotor systems. Hankinson (ref. 19) described a fatigue-resistant shear pin (fig. 117) that can be used to couple a rotor subject to high torsional loads, such as those usually obtained between reciprocating machines (e.g., diesel engines and electric generators).

Procedures for the design of rotor-bearing systems, well documented in NASA reports and other technical literature, can be used to design any type of machinery. Many circuit problems have benefited because of design charts for fluid-film bearings.

STANDARDS

Certain standards set up for industry by organizations such as the National Electrical Manufacturing Association and the American National Standards Institute are used in quality specifications of machinery. Standards for rotating machines provide acceptable performance levels for vibration-velocity response. Experimental data gathered by Muster (ref. 20) showed that velocity is a valid measure of failure in rotating machines. Levels of vibration-velocity response vary, depending on the function of a piece of equipment; for example, an

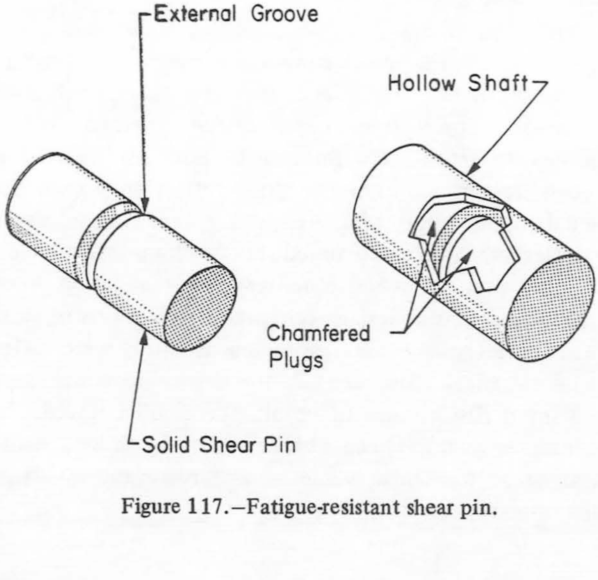

Figure 117.–Fatigue-resistant shear pin.

allowable vibration level for a drum recorder would be lower than that for an electric motor.

In specifying the level of vibration-velocity response, it is easier to control performance than to specify mass, unbalance, damping, or any of the other factors that combine to determine response. Thus, response usually reflects the quality of a rotating machine.

SHOCK MOUNTING

In a shock environment, such as that encountered in a railroad car or package, an electric machine should be mounted on flexible supports. The extent of stiffness and damping required in the supports to attenuate an input disturbance depends on a design standard. If high-frequency low-amplitude motion disturbs the performance of a system such as a disk recorder, for example, then there should be as little damping as possible in the supports. If the strength of the system is the primary consideration, then damped supports tuned to the proper frequency to act as attenuators (rather than amplifiers) are recommended. In general, motion is necessary in good shock isolation. New concepts in shock isolators and absorbers now being considered are

- Material deformation
- Fluid compression
- Mass acceleration
- Friction
- Pneumatic bags
- Honeycomb struts
- Filament-wound toroids
- Reversing aluminum tubes
- Expansion-tube absorber
- Frangible-metal tubing.

The use of any of the shock-isolation concepts considered in chapter 2 depends on its ultimate purpose and, in many cases, multishock protection capability is sought. Concepts such as reversing aluminum tubes and material-deformation devices are effective for a single-shock loading. The advantage of a pneumatic bag lies in its nonlinear load-deflection curve; friction devices provide some damping without high-frequency wave transmission.

The analytical techniques required to shock-mount an electric machine include computer simulation and other response-calculation procedures. In the discussion of shock-mount designs (chapter 2) solutions using shock spectra, optimum isolator techniques, and nonlinear isolators were reviewed. Basic to the design of electrical equipment is the degree to which an isolator must attenuate an input disturbance to obtain the desired level of fragility. Some electric motors, for example, can sustain a high-shock load without impairing their survival or performance, while a tape recorder, which must maintain uniform tape velocity, can sustain little loading.

Since mass and stiffness affect the performance of a shock isolator in an electrical system, they must be considered in its design. Figure 118(a) shows a model of an electric motor mounted on shock mounts; since the mounts represent the stiffness of ball bearings, springs k can be considered rigid compared to isolator springs k_2; this assumption allows representation of the motor as a simple spring-mass system [fig. 118(b)] and simplifies the analysis. In systems such as drum recorders and other high-speed machines, the fact that k is a flexible spring compared to k_2 must be reflected in the analytical model. Malanoski's work (ref. 11) on shock-excited rotor systems reflects this refinement in the analysis of shock-mounted rotating machines.

VIBRATION ISOLATION

Isolation of a rotor-bearing system from external vibratory disturbances involves modeling and analytical techniques similar to those utilized in shock-isolation design. Conceptually the vibration isolator is very different from the shock mount, and the two should not be confused. A good shock mount has a long working stroke that spreads the energy input over a longer period of time than the isolator; on the other hand, the vibration isolator has a stiffer spring-damped device. This device attenuates input force and motion

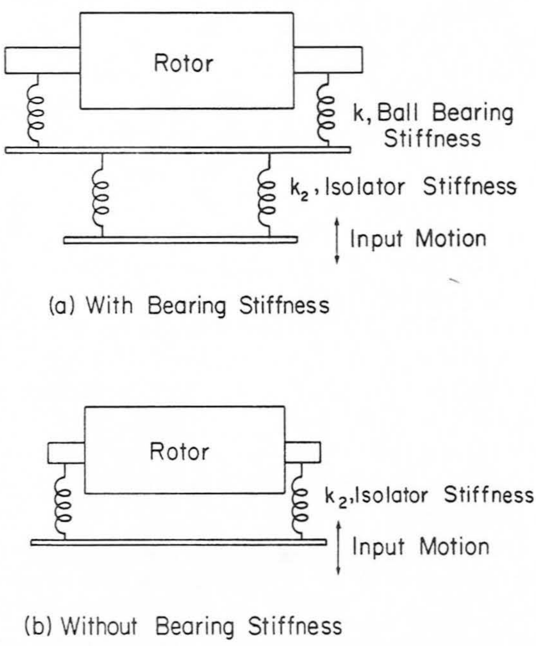

Figure 118.–Analytical model of a shock-mounted electric motor.

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through a motion-phase shift in which the input force works against an isolated-mass acceleration and/or an isolated damper, which dissipates energy in the form of heat.

Recent concepts in vibration isolation applicable to the isolation of electrical systems include elastomeric foams, elasto-plasto-viscous dampers, pneumatic devices, and elastomeric coatings.

Elastomeric foam can be shaped with ease into a given design requirement; its air-damping characteristics and properties can be synthesized. Liber and Epstein (ref. 21), for example, have described an experimental procedure for the characterization of foam for design purposes. Foam may develop into an ideal material for cushioning tape recorders and feet for electric motors and generators. The use of pneumatic devices, well documented in the literature, was discussed with respect to low-frequency suspension systems in chapter 5. Derby and Ruzicka (ref. 22) investigated the resonant-frequency, loss-factor, and thermal-conductivity characteristics of structural composites with viscoelastic-shear-damping mechanisms. Elastomeric coatings appear to eliminate surging caused by high-frequency disturbances in helical springs. The elasto-plasto-viscous point-vibration dampers developed by Peck (ref. 23) apply to design situations in which a point-vibration isolator must yield high performance. These point-vibration dampers could be used to isolate any rotating machine mentioned in this survey since basic stiffness and damping properties can be synthesized.

When a vibration isolator is used in a rotating system, the dynamic properties of the entire system must be considered. If the support stiffness (bearings) and rotor stiffness are inflexible compared to the proposed vibration isolators, the machine can be modeled as a rigid body on the vibration isolators. This is more likely to occur, however, in shock mounting than in vibration isolation; vibration isolators usually are more inflexible than shock mounts. The analytical model [fig. 118(a)] is adequate for vibration-isolation problems as well as those involving shock mounting. The analytical techniques applicable for a rotor-bearing problem also are used for the rotor-bearing vibration-isolation problem. The model in figure 119 includes additional lumped springs, masses, and dampers; figure 119 also shows the motor's rotor-bearing model, and its rotor-bearing vibration-isolation model.

VIBRATION CONTROL

Vibration control is concerned with eliminating the transmission of vibration from a motor or a recorder to the surroundings. This familiar problem can be solved in one of three ways: (1) by isolation, (2) by elimination of excitation, and (3) by absorption of excitation. The use of vibration isolators for vibration control is similar to other applications of vibration isolators; however, in vibration control, the excitation originates

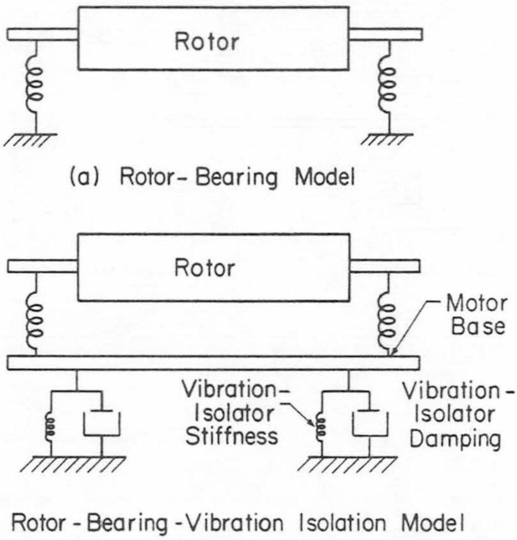

Figure 119.–Modeling of electric motor with vibration isolation.

within the machine, and the forces transmitted to the base of the machine must be attenuated.

The easiest way to control a vibration is to remove its source. Mass unbalance, gear inaccuracies, and belt fluctuations can be minimized through careful design, machining specifications, and dynamic-mass balancing. Rotor balance is mandatory in high-speed rotating machines (e.g., electric motors). Balance techniques for rigid and flexible rotors are well documented in the literature. The process of balancing flexible rotors is an involved one because the unbalanced forces are a function of rotor deflection, which changes with speed. Elimination of gear excitations, usually too costly a procedure to be practical, is useful more for internal vibration control (where excitation affects an optical, electronic, or similarly sensitive device) than for external vibration control.

Gunter (ref. 12) showed how decreased support stiffness and increased support damping raised the rotor whirl threshold. He also showed that rotor stability improved, without induced damping, when anisotropic support stiffness was applied to a rotor-bearing system.

The final vibration control measure is the mass-vibration absorber, which can have either rotational or linear motion. A survey of NASA's development of linear and gyroscopic vibration absorbers is found in Chapter 3. The vibration absorber functions by introducing to a system an opposing excitation that cancels the vibration response of the original rotating system. Although it

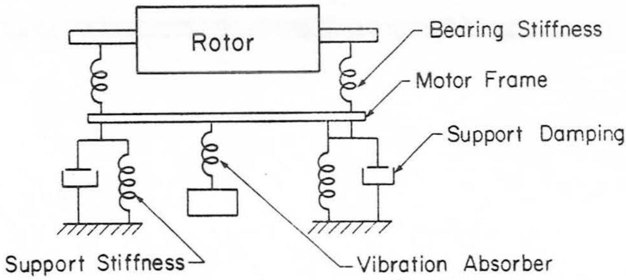

Figure 120.–Vibration-absorber application to electric motor.

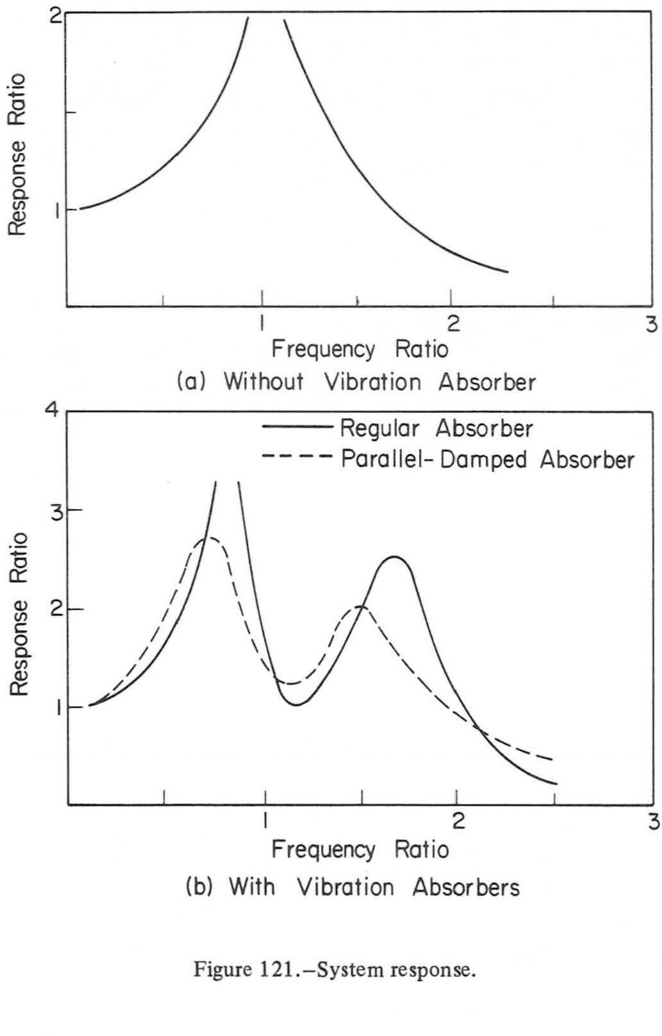

Figure 121.–System response.

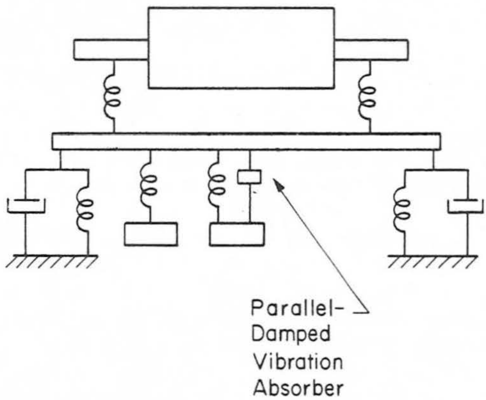

Figure 122.–Parallel-damped vibration absorber applied to electric motor.

does not eliminate internal machine stress, this mass vibration absorber can be designed to eliminate unwanted internal motion. A schematic of a vibration absorber on an electric motor or a drum recorder is shown in figure 120. The natural frequency of the absorber must coincide with the motor speed for maximum effectiveness. The main disadvantage of this system is that it is effective at only one frequency, and two resonance peaks (fig. 121) are obtained. The parallel-damped vibration absorber (fig. 122) proposed by Srinivasan (ref. 24) would be useful because the response at the two resonance points could be reduced to obtain an acceptable response at all speeds. The synchronous gyroscopic vibration absorber was developed by Flannelly and Wilson (ref. 25) for multiple-speed absorption.

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Glossary

Analytical technique. Solution of shock and vibration problems using mathematical analysis.

Band-pass filter. An electronic device used to filter all signals in a predetermined frequency range.

Critical damping. The minimum viscous damping that allows a displaced system to return to its initial disturbing position without displacement.

Critical speed. A speed of a rotating system corresponding to a resonant frequency of the system.

Degree of freedom. The minimum number of independent coordinates required to define completely the positions of all parts of a system at any instant of time.

Differential equations. Mathematical equations that describe the physical motion of a system.

Digital simulation. Simulation of the behavior of a physical system to environmental disturbances using the digital computer and numerical analysis.

Dimensional analysis. Analysis of a physical system using algebraic expressions that are a function of the dimensions of the system.

Fatigue failure. Failure of a mechanical component or system as a result of repeated stress cycles.

Fourier series. A mathematical description of a nonharmonic periodic function using a linear combination of sine and cosine functions.

Fragility level. A quantitative index of the strength of a piece of equipment subjected to shock and vibration loads.

Frequency response. The response (i.e., displacement, velocity, or acceleration) of a system described in terms of frequency.

Gyroscopic moment. A load imposed on a shaft by its gyroscopic acceleration. Gyroscopic acceleration results from the interaction of the shaft spin velocity and its vibration velocity.

Harmonic analysis. Analysis of a periodic complex waveform using sine and cosine functions.

Hysteresis. Dissipation of vibrational energy through internal friction of repeatedly stressed metals.

Influence coefficients. Mathematical coefficients that describe the influence of system loading on system deflection.

Initial value analysis. Mathematical description of the time behavior of a system using a step-by-step integration of the equations of motion and initial conditions of the system.

Logarithmic decrement. The natural logarithm of the ratio of any n successive amplitudes of decaying vibration of like sign.

Matrix. An orderly two-dimensional arrangement of the physical properties of a system.

Modal analysis. A vibration response analysis that uses a unique combination of previously determined mode shapes for its mathematical description.

Modal damping. Viscous damping expressed as a multiple of the mode shape of a system.

Model. A mathematical or experimental simulation of a component or system.

Modulus of shear. A physical constant that describes the elastic property of a metal in shear loading.

Motion. Description of the displacement, velocity, or acceleration of a system as a function of time.

Natural frequency. The reciprocal of the natural period of a system.

Natural period. A physical characteristic of a system, dependent on the magnitude and arrangement of the system mass and elasticity; observed physically by disturbing the system and noting the time required for it to complete one cycle of vibration.

Normal mode. The normal mode is the space configuration a system assumes when it is vibrating at a natural frequency.

Random vibration. A nonrecurring relationship between the magnitudes of quantity and time, measured during a certain time interval.

Resonance. A state of a system in which the magnitude of response is large due to the proximity of the forcing frequency and the natural frequency.

Scaling. Extrapolation of the properties and/or geometry of a system in its analytical or experimental model.

Shock. A nonperiodic excitation of a mechanical system characterized by sudden loading.

Shock absorber. A device that dissipates energy in order to modify the response of a mechanical system to applied shock.

Shock isolator. A resilient support that isolates a system from a shock loading.

Shock pulse. A substantial disturbance characterized by rise and decay of acceleration in a short period.

Shock spectrum. The maximum response (acceleration, velocity, or displacement) of a series of damped or undamped single-degree-of-freedom systems resulting from a specific shock excitation; an independent mass-spring-damper system is associated with each frequency.

Specific-energy absorption. The energy absorbed by a shock absorber divided by its weight.

State variables. Variables that define the state (motion or force) of a system as a function of time.

Steady-state response. The response of a system to a periodic disturbance.

Stiffness. The description of the elastic properties of a system, given in terms of pounds force per inch of deflection.

System response. A time-history description of the displacement, velocity, or acceleration of a system excited by a shock or vibration.

Timoshenko beam. A mathematical model that includes the beam flexibility due to transverse-shear deformation and the inertia due to rotation of beam elements with respect to each other. Both effects lower the beam natural frequency.

Transient response. The time description of the temporary state of a system that exists as a result of shock loading or initial vibration loading.

Transducer. A device that converts a mechanical motion into an electrical signal.

Transmissibility. The dimensionless ratio of the response of a system in steady-state vibration to its environmental vibration loading.

Vibration. The repeated motion of a system expressed as a function of time.

Vibration absorber. A device that dissipates energy as it attenuates a vibration disturbance.

Vibration isolator. A resilient support that attenuates steady-state disturbances applied at a moving base.

Whirling. The state of vibration of a rotor that is manifested in the form of an orbital motion.

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