CHAPTER 41
EQUIPMENT DESIGN

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INTRODUCTION

Equipment is defined here as any assembly of parts that form a single functional unit for the purposes of manufacturing, maintenance, and/or recordkeeping, e.g., an electronic package or a gearbox. Designing equipment for shock and vibration environments is a process that requires attention to many details. Frequently, competing requirements must be balanced to arrive at an acceptable design. This chapter guides the equipment designer through the various phases of a design process, starting with a clear definition of the requirements and proceeding through final testing, as illustrated in Fig. 41.1.

ENVIRONMENTS AND REQUIREMENTS

The critical first step in the design of any equipment is to understand and clearly define where the equipment will be used and what it is expected to do. The principal environments of interest in this handbook are shock and vibration (dynamic excitations), but the equipment typically will be exposed to many other environments (see Table 20.1). These other environments may occur in sequence or simultaneously with the dynamic environments. In either case, they can adversely affect the dynamic performance of the materials used in a design. For example, a thermal environment can directly affect the strength, stiffness, and damping properties of materials. Other environments can also indirectly affect the dynamic performance of an equipment design. For example, thermal environments can produce differential expansions and contractions that may sufficiently prestress critical structural elements to make the equipment more susceptible to failure under dynamic loading.

The preceding example illustrates the need to understand all of the design requirements, not just the dynamic requirements. A comprehensive set of require-

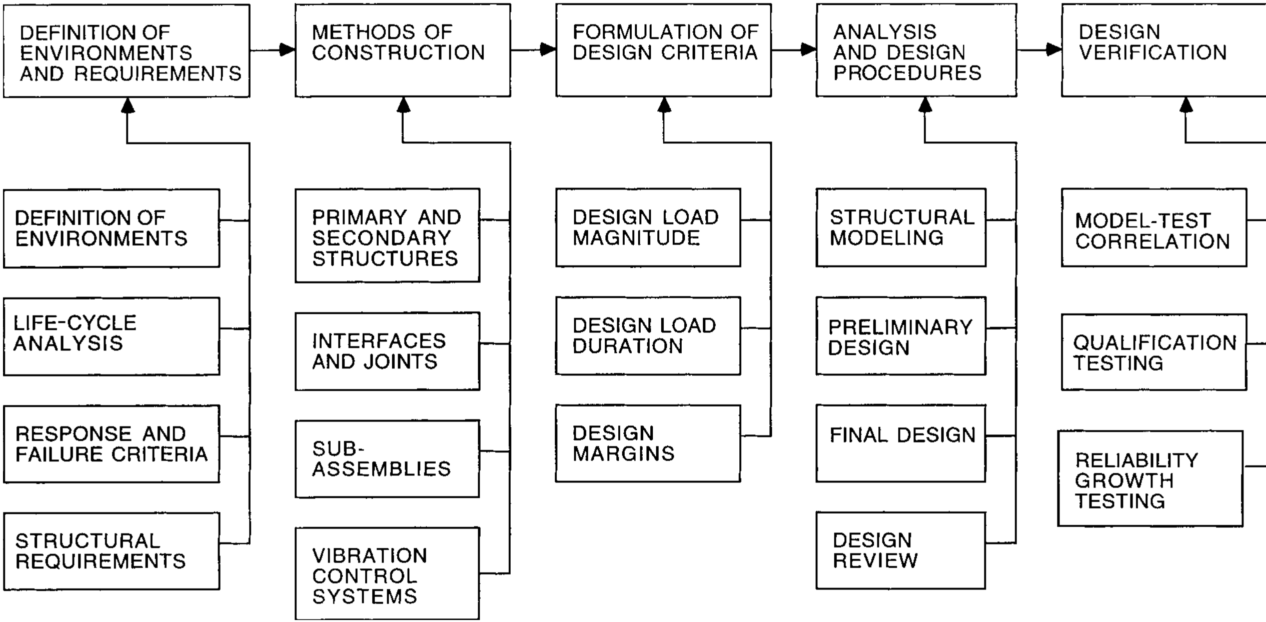


FIGURE 41.1 Steps in equipment design procedure for shock and vibration environments.

ments (or equipment specifications) must be developed so that no aspect of the design's performance is left uncontrolled. Unfortunately, different types of requirements often lead to difficult design tradeoffs that must be resolved. Priorities must be established in these situations. For example, a low-cost weak material may be preferred over a more expensive stronger material if the operational stresses can be kept low. This example reflects the fact that many requirements are not purely technical. Cost, schedule, and safety issues are additional requirements that are always on the mind of project management. Still other requirements can be more emotional (e.g., aesthetic appeal).

The approach to equipment design presented in this chapter is the systems engineering concept of minimizing the life cycle cost, where the *life cycle* is defined as all activities associated with the equipment from its initial design through its final disposal after service use. Stated simply, the design process should consider and minimize the costs incurred over the complete life of the equipment. Extra effort put forth early in the design phase can often have a large payoff later in the life of the equipment. For example, the cost of correcting a problem in manufacturing can be many times greater than the cost of making the correction during the design phase. Additional costs, such as disposal and recycling of the equipment after it has passed its useful life, can be minimized with proper attention early in the design phase.

DYNAMIC ENVIRONMENTS

Shock and/or vibration (dynamic) environments cover a wide range of frequencies from quasi-static to ultrasonic. Examples of different dynamic environments and the frequency ranges over which they typically occur are detailed in the various chapters and references listed in Table 28.1. The classification of vibration sources and details on how measured and predicted data should be quantified are presented in Chap. 22. From a design viewpoint, dynamic excitations can be grouped as follows.

Quasi-Static Acceleration. Quasi-static acceleration includes pure static acceleration (e.g., the acceleration due to gravity) as well as low-frequency excitations. The range of frequencies that can be considered quasi-static is a function of the first normal mode of vibration of the equipment (see Chap. 21). Any dynamic excitation at a frequency less than about 20 percent of the lowest normal mode (natural) frequency of the equipment can be considered quasi-static. For example, an earthquake excitation that could cause severe dynamic damage to a building could be considered quasi-static to an automobile radio.

Shock and Transient Excitations. Shock (or transient) excitations are characterized as having a relatively high magnitude over a short duration. Many shock excitations have enough high-frequency content to excite at least the first normal mode of the equipment structure, and thus produce substantial dynamic response (see Chap. 8). The transient nature of a shock excitation limits the number of response cycles experienced by the structure, but these few cycles can result in large displacements that could cause snubbing, yielding, or tensile failures if the magnitude of the excitation is sufficiently large. Frequent transients can also result in low-cycle fatigue failures (see Chap. 34).

Periodic Excitations. Periodic excitations are of greatest concern when they drive a structure to respond at a normal mode frequency where the motions can be dramatically amplified (see Chap. 2). Of particular concern is the repetitive nature of the response that can accumulate enough cycles to cause fatigue failures at excitation levels less than those required to cause immediate yielding or fracture. The most basic form of a periodic excitation is the sinusoidal excitation caused by rotating equipment. However, other periodic excitations may include strong harmonics that might be damaging, e.g., the vibrations produced by reciprocating engines and gearboxes (see Chap. 38). All harmonics of the periodic excitation must be considered.

Random Excitations. Random excitations occur typically in environments that are related to turbulence phenomena (e.g., wave and wind actions, and aerodynamic and jet noise). Random excitations are of concern because they typically cover a wide frequency range. All natural frequencies of the structure within the frequency bandwidth of a random excitation will respond simultaneously. Assuming the structure is linear, the response will be approximately Gaussian, as defined in Chap. 11, meaning that large instantaneous displacements, as well as damaging fatigue stresses, may occur.

Mixed Periodic and Random Excitations. Mixed excitations typically occur when rotating equipment induces periodic excitations that are combined with excitations from some flow-induced source. An example would be a propeller airplane, where the periodic excitation due to the propeller is superimposed on the random excitation due to the airflow over the fuselage (see Chap. 29, Part III). It is important to compute the stresses in the equipment due to both excitations applied simultaneously. The same is true of shock excitations that may occur during the vibration exposure.

OTHER ENVIRONMENTS

Other environments may have an effect on material properties and/or help define what materials and finishes can be used during the design and construction of the equipment. The more important environments that should be considered are as follows.

Temperature. Material properties can change dramatically with temperature. Of particular concern for dynamic design are the material stiffness changes, especially in nonmetallic materials such as plastics and elastomers (see Chap. 33). Most plastics show a dramatic reduction in stiffness at higher temperatures. Material strength and failure modes will also change with temperature. Some metals will exhibit high-strength ductile behavior at room temperature, and then shift to low-strength brittle behavior at low temperatures (see Chap. 34). Thermal strains can also induce stresses and deformations in structures that need to be considered as part of the design process. A thorough understanding of the expected operating and nonoperating temperatures, plus the amount of exposure time in each temperature range, is required when designing equipment structures for dynamic environments.

Humidity. Humidity can have an effect on material properties, especially plastics, adhesives, and elastomers (see Chap. 33). Some nonmetallic materials can swell in humid environments, resulting in changes in stiffness, strength, and mass. Humid environments can also lead to corrosion in some materials that ultimately produce lower strengths.

Salt/Corrosion. Ocean and coastal environments are of particular concern because the corrosion they commonly produce can lower the strength of a material. Corrosion and oxidation can also cause clogging or binding in flexible joints. Protective finishes, seals, and naturally corrosive resistant materials are needed when equipment is designed to withstand long durations in ocean and coastal environments. Corrosive environments can also occur in power plants and chemical processing industries.

Other. Other environments might affect the dynamic performance of equipment. Two such examples are vacuum and electromagnetic fields. Vacuum environments (e.g., space vehicles or aircraft at high altitudes) can cause pressure differentials in sealed structures, which produce static stresses that are superimposed on the stresses due to dynamic responses. Vacuum environments also lack the damping provided by the interaction of the structure with the air. Electromagnetic fields can interfere with the functional performance of electronic subassemblies, and sometimes induce vibration of steel panels.

LIFE-CYCLE ANALYSIS

Dynamic design typically concentrates on the service environment, but there are other conditions during the life of a product that may require special consideration. The definition of all of the different conditions (environment magnitudes and duration) that the equipment will be exposed to during its total life, from manufacture to disposal, is commonly referred to as a *life-cycle analysis*.

Manufacturing Conditions. The life of equipment typically begins when it is manufactured. Manufacturing-induced residual stresses and strains due to plastic deformations, excessive cutting speeds, elevated adhesive cure temperatures, or welding can adversely affect the initial strength of materials. Understanding the material properties after manufacturing-induced excitations (and possible rework) is a critical first step in a life-cycle analysis.

Test Conditions. Equipment often undergoes factory acceptance or environmental stress screening tests (see Chap. 20) before it is put into service. These test environments can induce initial stresses and strains that reduce the resultant strength. An example is a pull test of a wire bond. The test should produce failure in a poor bond, but may also cause permanent plastic deformation in the ductile wire. When predicting the overall fatigue life of an item of equipment, any initial tests must be considered as excitations that will accumulate damage.

As discussed in Chap. 20, at least one sample item of any new equipment must pass a *qualification test* to verify that it can survive and function correctly during its anticipated shock and/or vibration environments. This qualification test generally represents the most severe dynamic environment the equipment will experience, and hence the equipment must be designed for this test environment. However, since the sample item used for the qualification test is not delivered for service use, the qualification test does not have to be included in the life-cycle analysis.

Shipping and Transportation. Once an equipment item is manufactured, it probably will be transported to its operating destination. This transportation environment can often induce excitations that will not be seen in service use. Examples include shock excitations from handling between shipping phases (e.g., dropped packages when unloading a truck), and low-frequency vibration excitations induced by repeated roadway imperfections as seen by a ground transportation vehicle. Special features may need to be added to the equipment, such as additional support parts, to help it survive shipping excitations. One example is a temporary part that is installed between two assemblies that would normally be vibration-isolated in use. The temporary part eliminates excessive displacements due to large low-frequency shipping excitations. Once the system arrives at its destination, the temporary part is removed so the two assemblies can then move freely.

In some cases, the transportation environments may be so much more severe than the service environment that special shipping containers need to be designed to attenuate the transportation excitations. Vibration-isolated shipping containers are often used when transporting sensitive equipment (see Chaps. 30 through 32).

Service Conditions. The most obvious condition to understand is the service environment of the equipment. A significant portion of the design process should be devoted to accurately determining the dynamic environments under which the equipment must operate. A thorough understanding of the service dynamic environments will help to ensure that the equipment will function both properly and economically. Standard dynamic environments that have been developed for various commercial and military applications may be used to help determine the service excitations (see Chap. 19). These standards, however, should be used with care because they often provide conservative shock and/or vibration estimates that may result in equipment that is overdesigned and more costly than necessary.

When the equipment is to be used in multiple locations, a larger set of dynamic environments must be considered. For each environment, the type, magnitude, duration, and other conditions (e.g., temperature range) should be itemized. For items of equipment that will be produced in large quantities, a statistical approach that groups the dynamic environments into histograms should be considered (see Chap. 20). While the specification of service environment magnitudes and durations is often the responsibility of another organization, the designer must review the desired requirement thoroughly and often request additional information.

DYNAMIC RESPONSE CONSTRAINTS AND FAILURE CRITERIA

Important requirements that need to be defined before equipment is designed are the allowable dynamic responses and failure criteria. Often there will be multiple constraints that need to be satisfied.

Displacement. Displacements due to dynamic excitations must always be considered when the equipment is made up of several subassemblies. The overall motion (or sway space) of an equipment item must also be considered when it will be mounted near other structures. This is often a concern with vibration-isolated equipment. Displacements can also be a concern for position-sensitive equipment such as printing, placement, optical, and measurement devices.

Velocity. Velocity response is of concern for all structures, because the modal (relative) velocity of the structural response at a normal mode is directly proportional to modal stress.[1] This fact can be used to estimate the stress due to the response of a structure at any given normal mode frequency, as will be detailed later.

Acceleration. Some products are most susceptible to acceleration responses. For example, an electrical relay or switch may unlatch when the acceleration acting on the mass of the contact is large enough to cause it to change state. Furthermore, quasi-static acceleration excitations are proportional to stress in the equipment structure.

Permanent Deformation and Factors of Safety. A critical part of the requirements definition process for dynamic environments is to clearly state the allowable amount of permanent deformation that the equipment will tolerate. Some equipment can still function acceptably after being subjected to brief, high-excitation conditions that cause some plastic deformation. Other equipment may not tolerate any yielding that could cause misalignment or interference. Some customers may specify factors of safety that must be met as part of a development specification. These are typically calculated based on stresses relative to the allowable material yield and/or tensile strengths.

Fracture, Fatigue, and Reliability. Equipment intended for use over a relatively long-exposure duration should carry with it some clearly defined fatigue and/or reliability requirement. The equipment design team should establish a reliability goal in terms of fatigue life. This is of particular concern when a premature failure of the equipment can result in severe economic damage or personal injury.

STRUCTURAL REQUIREMENTS

Structural and physical requirements must be defined before the start of a design. For equipment that will be used as part of a larger system, the physical requirements may be negotiable, especially in terms of mounting points and final geometry. These requirements are typically specified as part of an interface agreement, often called an *interface control document* (ICD), between the product development teams.

Volume. The overall volume requirement for an equipment item is an obvious requirement, but it may necessitate some design study. One example would be a combination of a minimum natural frequency and a radiating thermal environment requirement. A smaller design typically has a higher natural frequency due to the stiffness vs. length cubed effect in bending (see Chap. 1). However, this is contrary to the need for a large surface area to facilitate radiation heat transfer. As with most design problems, these effects need to be balanced within the allowable volume. The volume should also include allowances for any displacements that may occur over the life of the equipment.

Mass. Mass or weight requirements can conflict with other equipment requirements. For example, equipment that has a maximum mass requirement may also have a shock and/or vibration-isolation requirement (see Chaps. 30 through 32). The resulting equipment will need to be designed with a low-stiffness isolation system such that the required level of isolation can be reached while still meeting the maximum mass requirement. Other conflicting requirements are minimizing mass while maximizing stiffness and conduction heat transfer. When a mass needs to be controlled accurately, care should be given to the control of both the density and geometry of the parts, especially when the materials used are alloys of high-density metals or composites.

Materials. High-strength, low-weight metals are typically the materials of choice for equipment that is exposed to dynamic environments. While this is usually a wise choice, other factors should be considered. In many cost- and time-conscious industries, procurement organizations limit the number of materials from which a product can be made. This is a practice that can save money and limit inventories of expensive specialty materials. The designer needs to understand this situation and learn to work with the available choices of materials. A second concern is that these materials must often be selected in certain stock thicknesses and shapes. One benefit of these measures is that the physical properties of standard materials are often well documented. If not, the designer should strive to work toward a common material property database that can be linked to the available material choices.

Damping properties can be measured for polymers, elastomers, and adhesives using the procedures detailed in Chap. 37. The damping properties of adhesives are an important factor to consider when choosing between options. Adhesives that join lightly damped members can significantly reduce the overall response of the equipment assembly. Fatigue (or fracture) properties for most common materials can be found in Chaps. 34 and 35, as well as Refs. 2 to 4.

Finally, the designer should review the other required environmental conditions that may cause further constraints on the available choices of materials. When feasible, the designer should use common materials that have well-defined properties. Materials that are more exotic should be considered only when they are essential and their properties are well-documented and controlled.

OTHER REQUIREMENTS

It is important to consider other requirements that can adversely affect the finished equipment if not considered early in the design process.

Safety. For those items of equipment where a failure or malfunction during service use might result in severe economic damage or personal injury, safety must be a primary concern. Safety issues should also receive top priority during all other life cycle phases, including manufacturing, handling, and transportation. A qualified safety engineer should be involved in all phases of the design process.

Cost and Schedule. Cost is an important concern that must be considered by every designer developing new equipment. Of particular importance is the life cycle cost of the equipment. It is often less expensive overall to spend time early in the design phase to define and understand the equipment requirements. This can often reduce costly changes to the design further along in its development. However, as previously discussed, safety requirements must always receive careful consideration in making cost and schedule decisions.

Disposal/Recycle. Disposal and recycling requirements should always be considered in the design. Some markets now require that the final disposal of an equipment item include recycling of its materials. Products may also be remanufactured, that is, some types of equipment that have completed their service life might be refurbished, with worn parts repaired or replaced, and then returned to service.

Other. The designer should be aware that equipment needs to function well in ways other than its prime task. Additional features that will help other groups work with the equipment should be considered early in the design phase. Included here are such features as handles, additional holes for lifting equipment, modular design, and adjustable interfaces. When conflicting requirements make a straightforward design difficult, it is sometimes desirable to convene a design team comprised of engineers in such disciplines as systems operation analysis and testing, electromagnetic compatibility, high-reliability parts, cost control, manufacturing, and thermal analysis, as well as shock and vibration.

METHODS OF CONSTRUCTION

Equipment designed to withstand shock and/or vibration excitations must typically be stronger than equipment that only has to withstand gravity or static acceleration loads. This dictates that the equipment have a well-defined primary structure that can withstand the dynamic excitations, as well as carry the additional excitations that might be internally generated. Basic construction methods should be considered early in the design process to facilitate the modeling and analysis procedures discussed later.

PRIMARY STRUCTURE

Primary structures are those that carry the greatest loads and support the secondary structures and subassemblies. The design and analysis of any product should start with particular attention to primary structure. The primary structural elements often have to be designed early in the product development cycle to allow for long lead-time material and tooling acquisition. Simple lumped parameter (see Chap. 2) or beam/plate finite element models (see Chap. 28, Part II) can be used to perform initial stiffness and natural frequency calculations for primary structures. There are many ways to build primary structures.

Machined Parts. Machined parts are often used for primary structures. The machining operations can be customized to place holes and attachment points for secondary structures where needed. For economic reasons, machine operations can be used to remove unnecessary material or allow thicker sections where needed. Machined parts are typically used for low-volume production. Unfortunately, machining operations can also reduce the strength of the parent material by introducing microcracks that might lead to fatigue or fracture. Machined parts may need to be heat-treated after machining to develop the necessary strength and ductility for the intended use.

Castings/Forging. Casting or forged parts are typically used for high-production-volume structural elements because they usually can be formed in near-final shapes that reduce the need for machining operations. Cast materials typically have lower strength and ductility than wrought or forged materials (see Chap. 34). Cast materials also can suffer from various manufacturing defects, such as porosity and shrinkage, which can increase part variability. This variability should be factored into the stress and strength analysis of the part.

Forged parts typically have higher strengths than cast and wrought materials. The forging process can shape material grain and orient the strength along specific part directions. Forged parts are used when the very highest strengths are needed to resist high excitations, e.g., in aircraft landing gear and construction equipment linkages. The forging process does tend to be expensive because of the hard tooling that is needed to form parts under high temperatures and pressures.

Plates/Sheet Metal. Sheet and plate parts are often used for primary structures, especially when they are formed into more rigid three-dimensional shapes. Sheet and plate material can often be bent, cut, and then joined to other parts to give strength and stiffness where needed. Automobile bodies are excellent examples of how sheet metal can be used to form rigid and reliable structures. Modern computer-controlled laser and water-jet cutting techniques can be used to form complicated sheet or plate metal geometries economically for even low-volume production. The important thing to remember with sheet or plate metal construction is that parts need to be stiffened in the out-of-plane (normal to the surface) direction. Care should also be given to minimizing large unsupported areas that can vibrate, especially with acoustic excitation. An example of stiffened construction for the base of an equipment item with a shock-isolated subassembly is shown in Fig. 41.2.

Beam Frames. Beam and tube construction is a very efficient way to make primary structures that span large distances, especially when built into trusses or frames. Beams and tubes also have the advantage of high material strength because of the manufacturing processes, such as extrusions, that form them into their continuous cross sections. The most difficult part of designing a beam or tube structure is determining the best way to join the pieces. Welding can often reduce the strength of

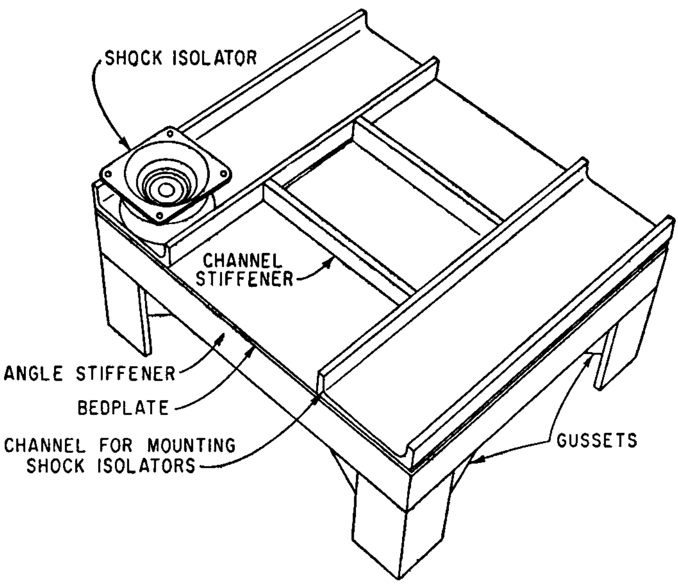


FIGURE 41.2 Illustration of stiffened primary structure for equipment with a shock-mounted subassembly. (*H. M. Forkois and K. E. Woodward.*[5])

the material at the joints, requiring additional fittings or gussets to maintain the necessary overall strength. Care should also be given to locating any holes or secondary attachment points at low-stress locations on the beams.

Composite Structures. Composite structures have proven to be efficient primary structures, especially when high strength and low weight are prime concerns. Composite materials can be laid up into plate, beam, and large thin-wall structures. Boat hulls and filament-wound pressure vessels are good examples of large composite thin-wall structures. Composite materials can be mixed, taking advantage of different strength, stiffness, thermal conductivity, and thermal expansion properties for each layer. However, care is required when designing joints for composite structures. See Chap. 35 for details on the properties of composite materials.

SECONDARY STRUCTURES

Secondary structures are those structures used to attach subassemblies to primary structures. Secondary structures typically do not have the more stringent strength and stiffness requirements of the primary structures, so they can be designed later in the development cycle, often allowing changes in geometry to accommodate changes in subassemblies. Secondary structures can also evolve as more cost-efficient materials or manufacturing processes are developed.

Plates/Sheet Metal. Plate and sheet metal parts are often used for nonstructural members such as covers. In this case, the products need only to support their own weight or some minor additional weight due to cables, sensors, or other secondary assemblies. As with all plate structures, care should be given to minimizing large unsupported areas.

Composite Structures. Composite structures can also be used for secondary structures. Their high strength-to-weight ratios make them attractive options for covers and other molded thin-wall sections that need to support some subassemblies.

Plastic Parts. Plastic parts can be used for both primary and secondary structures. Plastics can form adequate primary structures, especially for smaller, low-weight consumer products that are not subjected to extreme conditions. When combined with other materials, such as metal stiffeners in selected areas, plastics can be used effectively for even larger products. The wide range of colors, finishes, and shapes make plastic materials a common choice for secondary structures that are visible to the consumer. They also make excellent low-cost parts when they do not need to be exposed to intense shock and/or vibration excitations.

INTERFACES AND JOINTS

Interfaces are the junctions between the various structural elements that form the equipment. The manner in which the structural elements are jointed together at interfaces is very important in the construction of equipment because the interface friction at joints is the primary source of the damping (energy dissipation) in the equipment that restricts its dynamic response to vibration and, to a lesser degree, shock excitations. There are five basic devices used to make joints in the construc-

TABLE 41.1 Typical Damping Ratios for Equipment with Various Types of Joints

Method of construction	Typical damping ratio for equipment
Welded and spot-welded	0.01
Riveted	0.025
Bolted	0.05
Bonded	0.01 to 0.05*

* Heavily dependent on the type of adhesive and its thickness.

tion of equipment, namely, (*a*) continuous welds, (*b*) spot welds, (*c*) rivets, (*d*) bolts, and (*e*) adhesives. Typical values of the damping ratio in fabricated equipment using these various types of interface joints are summarized in Table 41.1.

Welded Joints. Welded joints must be well designed, and effective quality control must be imposed upon the welding conditions. The most common defect is excessive stress concentration which leads to low fatigue strength and, consequently, to inferior capability of withstanding shock and vibration. Stress concentration can be minimized in design by reducing the number of welded lengths in intermittent welding. For example, individual welds in a series of intermittent welds should be at least 1½ in. long with at least 4 in. between welds. Internal crevices can be eliminated only by careful quality control to ensure full-depth welds with good fusion at the bottom of the welds. Welds of adequate quality can be made by either the electric arc or gas flame process. Subsequent heat-treatment to relieve residual stress tends to increase the fatigue strength. See Refs. 6 and 7 for further information on welded joints.

Spot-Welded Joints. Spot welding is quick, easy, and economical but should be used only with caution when the welded structure may be subjected to shock and vibration. Basic strength members supporting relatively heavy components should not rely upon spot welding. However, spot welds can be used successfully to fasten a metal skin or covering to the structural framework. Even though improvements in spot welding techniques have increased the strength and fatigue properties, spot welds tend to be inherently weak because a high stress concentration exists in the junction between the two bonded materials when a tension stress exists at the weld. Spot-welded joints are satisfactory only if frequent tests are conducted to show that proper welding conditions exist. Quality can deteriorate rapidly with a change from proved welding methods, and such deterioration is difficult to detect by observation. However, accepted quality-control methods are available and should be followed stringently for all spot welding. See Refs. 6 and 7 for further information on spot-welded joints.

Riveted Joints. Riveting is an acceptable method of joining structural members when riveted joints are properly designed and constructed. Rivets should be driven hot to avoid excessive residual stress concentration at the formed head and to ensure that the riveted members are tightly in contact. Cold-driven rivets are not suitable for use in structures subjected to shock and vibration, particularly rivets that are set by a single stroke of a press as contrasted to a peening operation. Cold-driven rivets have a relatively high probability of failure in tension because of residual stress concentration, and tend to spread between the riveted members with consequent lack of tightness in the joint. Joints in which slip develops exhibit a relatively low fatigue strength. See Refs. 6 and 7 for further information on riveted joints.

Bolted Joints. Except for the welded joints of principal structures, the bolted joint is the most common type of joint. A bolted joint is readily detachable for changes in construction, and may be effected or modified with only a drill press and wrenches as equipment. However, bolts tend to loosen and require a means to maintain tightness. Furthermore, bolts are not effective in maintaining alignment of bolted connections because slippage may occur at the joint; this can be prevented by using dowel pins in conjunction with bolts or by precision fitting the bolts; i.e., fitting the bolts tightly in the holes of the bolted members. See Refs. 6 and 7 for further information on bolted joints.

Adhesives. Adhesives are gaining increased usage as a method of attaching structural elements. When stringent manufacturing controls are used to ensure consistent material properties and area coverage, adhesives can be used in most joints between structures. Adhesives have an advantage over other types of joints when some flexibility and damping is needed in the joint. Adhesives are also good at filling uneven gaps in parts manufactured to wider tolerances. See Ref. 7 for details.

SUBASSEMBLIES

Most types of equipment, especially large items, require subassemblies to perform various functions to satisfy the overall function of the equipment. These subassemblies must be supported on the primary or secondary structures in a way that ensures they will function correctly. Subassemblies can often be treated as lumped masses, but they may need additional dynamic analysis when they are large or sensitive to dynamic effects. Subassemblies and their support structures often need to have their own requirements allocated to them. Examples are given below.

Electronic Assemblies. Many equipment items include one or more electronic assemblies. The designer must ensure that the environment seen by the electronic assembly is low enough for it to function correctly for the intended duration. Often, electronic assemblies will be purchased with specific dynamic requirements that, if exceeded, may cause malfunction or permanent damage. The design of support structures for the electronic assembly must ensure that the input dynamic environment to the assembly is within the specified dynamic requirements. Otherwise, the assembly must be mounted to the equipment through shock or vibration isolators (see Chaps. 30 to 32).

When it is necessary to design new electronic assemblies, several specific procedures need to be followed. First, the designer should establish a dynamic requirement for the assembly, as discussed earlier. Then, parts that can withstand this requirement must be selected. If some parts cannot be procured (at a reasonable cost) to withstand these levels, then isolation of a subassembly or the whole assembly must be considered. Finally, the design of the electronic circuit boards to which parts will be mounted requires specific attention.

Electronic circuit boards, also called *printed wiring boards* (PWBs) or *printed wiring assemblies* (PWAs), are often constructed of laminated fiberglass or other composite materials. These boards form a flexible plate that, if not supported adequately, can deflect easily and deform or break sensitive electrical part connection leads. Frequent attachment points, stiffening ribs, heat sinks, and plates should be considered early in the design of the electronics. It is often desirable to take advantage of the damping characteristics of adhesives used to bond stiffeners and heat sinks to boards to reduce dynamic deflection. See Ref. 8 for details on the design of electronic equipment for vibration environments.

Mechanical Assemblies. Mechanical assemblies require special attention when they deliver dynamic excitations to the structures that support them. Mechanical items, such as hydraulic cylinders or electrical motors, can induce large dynamic excitations to their support structures. Structural fittings need to withstand these excitations and often allow removal or adjustment of the mechanical assembly after its original manufacture. Dynamic excitations can also affect the performance of mechanical assemblies. For example, dynamic accelerations can act on imbalanced masses in rotating equipment to cause additional shaft displacement or speed errors. These disturbances need to be either limited or isolated.

Optical Assemblies. Optical assemblies need special consideration when used in dynamic environments. Optics must often be mounted using strain-free exact constraints. Overly constrained mounts are statically indeterminate, causing unpredictable and unwanted deformations. The dynamic parameters of the optical elements by themselves must also be well understood so that the effects of any dynamic excitations can be kept to an acceptable level. Of considerable concern is the lightly damped and brittle nature of glass optics.

SHOCK AND VIBRATION CONTROL SYSTEMS

As mentioned in several of the previous sections, many systems need to be designed to provide some sort of vibration isolation for sensitive assemblies contained within them. Shock and/or vibration isolation is typically achieved by what is essentially a low-pass mechanical filter (see Chaps. 30 through 32). These isolation systems can be very effective and should be considered early in the equipment design cycle, but are often considered later as a fix for a poor design. Passive shock and vibration control can also be achieved by careful attention to the damping characteristics of the materials used in the construction of the structure (see Chap. 36). Finally, applied damping treatments can be used to suppress unwanted dynamic responses (see Chap. 37).

DESIGN CRITERIA

Based upon a thorough evaluation of the environments and requirements summarized in the preceding section, specific design criteria must now be formulated. These criteria may cover any or all of the environments previously summarized, but it is the shock and vibration (dynamic excitations) environments that are of concern in this handbook. The dynamic environments are usually specified as motion excitations (commonly acceleration) at the mounting points of the equipment to its supporting structure. However, there may be situations where the equipment is directly exposed to fluid flow, wind, or aeroacoustic loads, which cause fluctuating pressure excitations over its exterior surfaces that can produce a significant contribution to the dynamic response of the equipment. An example would be a relatively light item of equipment with a large exterior surface area mounted in a space vehicle during launch. In this case, the dynamic excitation design criteria must also include pressure excitations over the exterior surface of the equipment, as detailed in Chap. 29, Part III. Nevertheless, attention here is restricted to dynamic inputs in the form of motion excitations at the mounting points of the equipment. It is assumed these dynamic excitations will be described by an appropriate frequency spectrum, as summarized in Table 20.2.

DESIGN EXCITATION MAGNITUDE

The procedures for deriving the magnitude of the dynamic excitations for design purposes are essentially the same as those used to derive qualification test levels in Chap. 20. The principal steps in the procedure are as follows:

Determination of Excitation Levels. When the structural system to which the equipment is to be mounted is available, the shock and vibration levels should be directly measured in terms of an appropriate frequency spectrum (see Table 20.2) at or near all locations where the equipment might be mounted. If the structural system is not available, the shock and vibration levels must be predicted in terms of an appropriate frequency spectrum at or near all locations where the equipment might be mounted using one or more of the prediction procedures detailed in other chapters of this handbook and summarized in Chap. 20. These measurements or predictions should be made separately for the shock and/or vibration environments during each of the life-cycle phases discussed in the previous section.

Determination of Maximum Expected Environments. For each life-cycle phase, the measurements or predictions of the shock and/or vibration environments made at all locations at or near the mounting points of the equipment to its supporting structure should be grouped together. Often design criteria are derived for two or more equipment items in a similar structural region. Hence, a dozen or more measurements or predictions may be involved in each grouping of data (called a *zone* in Chap. 20). A limiting (maximum) value of the spectra for the measured or predicted shock and/or vibration data at all frequencies is then determined, usually by computing a *statistical tolerance limit* defined in Eq. (20.2). The statistical tolerance limit given by Eq. (20.2) provides the spectral value at each frequency that will exceed the values of the shock and/or vibration spectra at that frequency for a defined portion β of all points in the structural region with a defined confidence coefficient γ. This limiting spectrum is called the *maximum expected environment* (MEE) for the life-cycle phase considered.

The MEE will generally be different for each life-cycle phase. From a design viewpoint, since the equipment response is heavily dependent on the frequency of the excitation, it is the largest MEE at each frequency (that is, the envelope of the MEEs for all life-cycle phases) that is important. This envelope of the MEEs is called the *maximax environment.* This same concept of a maximax spectrum is commonly used to reduce the time-varying spectra for nonstationary vibration environments, as defined in Chap. 22, to a single stationary spectrum that represents the maximum spectral values at all times and frequencies.

Equipment Loading Effects. The shock and/or vibration measurements or predictions used to compute the maximax excitation spectral levels at the mounting points of the equipment are commonly made without the equipment present on the mounting structure. Even when the equipment is present for the measurements or modeled for the predictions, the computations required to determine MEEs and the final maximax spectrum smooth the detailed variations in the spectral level with frequency. However, if the equipment is relatively heavy compared to its mounting structure, then when the equipment is actually mounted on the structure, the shock and/or vibration levels at the equipment mounting points are modified. This is particularly true at the normal mode frequencies of the equipment where it acts like a dynamic absorber, as detailed in Chap. 6. The result is a spectrum for the input excitation from the supporting structure that may be substantially reduced in level at the normal mode frequencies of the equipment. If this effect is ignored, the maximax spectrum might cause a severe overdesign of the equipment.

The equipment excitation problem can be addressed in two ways. First, if there is a sufficient knowledge of the details of the supporting structure, the input excitation spectra at the equipment mounting points can be analytically corrected using the mechanical impedance concepts detailed in Chap. 10. Specifically, let $Z_s(f)$ and $Z_e(f)$ denote the mounting point impedance of the supporting structure and the driving point impedance of the equipment, respectively. Then for a periodic vibration

$$L_c(f) = \frac{L_r(f)}{|1 + [Z_e(f)/Z_s(f)]|} \tag{41.1a}$$

where $L_c(f)$ and $L_r(f)$ are the line spectra, as defined in Eq. (22.5), for the response of the equipment mounting structure with and without the equipment present, respectively. For a random vibration,

$$W_{cc}(f) = \frac{W_{rr}(f)}{|1 + [Z_e(f)/Z_s(f)]|^2} \tag{41.1b}$$

where $W_{cc}(f)$ and $W_{rr}(f)$ are the power spectra, as defined in Eq. (22.8), for the response of the equipment mounting structure with and without the equipment present, respectively. For those situations where the driving point impedance of the equipment is small compared to the mounting point impedance of the structure, that is, $Z_e(f) << Z_s(f)$, it is seen from Eq. (41.1) that the vibration response of the equipment mounting structure is only slightly altered when the equipment is attached. However, if $Z_e(f)$ approaches $Z_s(f)$, as it often will at the normal mode frequencies of equipment mounted on relatively flexible structures, then the vibration of the mounting structure will be significantly modified by the presence of the equipment, and a correction of the design levels for the equipment loading will be required. Again assuming there is a sufficient knowledge of the details of the supporting structure, a second way to correct for the equipment loading problem is to include at least a portion of the supporting structure in the equipment model that will be used for the equipment response analysis to be discussed later.

DESIGN LIFE

For equipment that is designed for a long service life, the potential for a time-dependent failure (e.g., fatigue damage) is generally of primary concern. Hence, the total duration of the dynamic excitation exposure during all of the life-cycle phases must be determined. For shock environments, the problem reduces to simply estimating the total number of shocks that will occur during each of the life-cycle phases. For vibration environments, however, an equivalent duration for the vibration excitations during each life-cycle phase must be computed. If the vibration environment during a life-cycle phase were stationary, the task would be simple. However, vibration environments during life-cycle phases are often nonstationary (see Chap. 22). A common approach in this case is to assume any time-dependent failure of the equipment follows the inverse power law given by Eq. (20.6), where a value of $b = 8$ is often assumed for metal structures with no stress concentrations, and $b = 4$ is commonly assumed for electrical and electronic equipment, as well as metal structures with substantial stress concentrations. Using Eq. (22.6), vibration environments of different magnitudes and durations can be collapsed to a single stationary vibration environment with an equivalent damaging potential using Eqs. (20.7) and (20.8), as illustrated for automotive equipment in Table 20.4. In addition, this procedure is often used to collapse the vibration environments during each of the life-cycle phases into a single spectrum with an equivalent total duration for design purposes.

DESIGN MARGINS

Given the maximax spectra for the shock and/or vibration excitations at the mounting points of the equipment, perhaps with a correction for the loading effects of the equipment on its supporting structure, it is common to further increase the levels to allow for uncertainties in the derived maximax levels. This increase in the levels is called the *design margin,* and is commonly selected to be between +3 and +6 dB. For a periodic vibration described by a line spectrum, as defined in Eq. (22.5), or a shock described by a shock response spectrum, as defined in Eq. (23.33), +3 dB and +6 dB correspond to a multiplication of the spectral values by $\sqrt{2}$ and 2, respectively. For a random vibration described by a power spectrum, as defined in Eq. (22.8), +3 dB and +6 dB correspond to a multiplication of the spectral values by 2 and 4, respectively. Of course, other design margins, either larger or smaller, might be selected depending on the designer's confidence in the derived maximax spectrum. In any case, the maximax spectrum plus the design margin gives the final shock and/or vibration design magnitudes.

METHODS OF ANALYSIS

The analysis of structures for design purposes must involve an analytical model. This section outlines the different types of analysis methods and gives advice on how to use them for the design of equipment.

MODELING

Modeling is an essential part of the design process. Models allow designers to understand the dynamic behavior of the equipment and conduct trade-off studies and experiments without committing to hardware. Options range from the single degree-of-freedom model (see Chap. 2) to finite element method (FEM) models with thousands of degrees-of-freedom (see Chap. 28, Part II). Modern computers allow very large numerical analysis models to be created. In the limit, every detail of a structure can be analyzed. However, the economic wisdom of such a pursuit is questionable.

The decision of how much detail to incorporate into a model should be driven by a clearly defined objective related to a specific design requirement or constraint. The designer must determine the required output of the modeling effort and ensure appropriate design features are adequately represented in the model. For example, the model format and size will depend upon the need for stress results. In general, much less detail is required for displacement models than for stress models. Stress concentration resolution generally requires extensive modeling detail.

Sometimes multiple models are appropriate. For example, lumped parameter models may be sufficient in preliminary design and for conducting system sway space budget exercises. A beam model may be appropriate for a shock or vibration isolation system and excitation path design. In most cases, a finite element model is necessary to resolve stresses in detailed features. Engineering judgment must be applied to assess the need for modeling nonlinear properties and detailed features. Planning and data management are also important elements of the modeling process. The designer should consider all of the potential uses of the model prior to model construction.

Lumped Parameter Models. The simplest type of dynamic model is the single-degree-of-freedom system, for which tabulated and charted solutions are widely available (see Chaps. 1 and 2). Lumped parameter models can be used to accurately represent many mechanical structures. These include structures in which one structural element is much more flexible than the remaining structure. In such a case, the rigid portion of the structure may be adequately represented as a lumped mass connected to the equivalent spring stiffness of the flexible element. The behavior of complex structures often can be represented by very simple dynamic models. Designers should seek to recognize and exploit such simplifications wherever possible, as is illustrated later.

Distributed Parameter Models. Sometimes the mass of a structure is evenly distributed over a large span of the structure. In these cases, a lumped parameter model may require a very large number of degrees-of-freedom, and a distributed modeling approach is preferred. Distributed parameter models are on the next level of complexity in the hierarchy of modeling tools. Classical beam, plate, and shell theory provide the basis for such modeling. Poles, wings, frames, and the leads of electronic devices may be considered as beams, while printed circuit boards, panels, covers, and doors may be viewed as plates. Modeling techniques for distributed systems are provided in Chaps. 1 and 7.

Finite Element Method Models. Systems with multiple distributed parameter components become difficult to solve as geometry becomes even modestly complex. Fortunately, user-friendly software tools exist which enable designers to obtain computer solutions to distributed parameter models using the finite element method (FEM) of analysis. See Chap. 28, Part II, for details on FEM models and Chap. 27 for a discussion of the computer implementation of FEM models.

FEM models can vary widely in complexity depending on the desired results. Because FEM models can place substantial demands on computer and human resources, it is important not to make the model any more complex than needed for the application. Relatively simple models that involve only a few hundred degrees-of-freedom are often adequate to compute estimates for the first few normal modes of a structure. On the other hand, models involving 10,000 or more degrees-of-freedom are often required to obtain accurate stress predictions, particularly if the structure has nonlinear characteristics. This variation in the complexity of an FEM

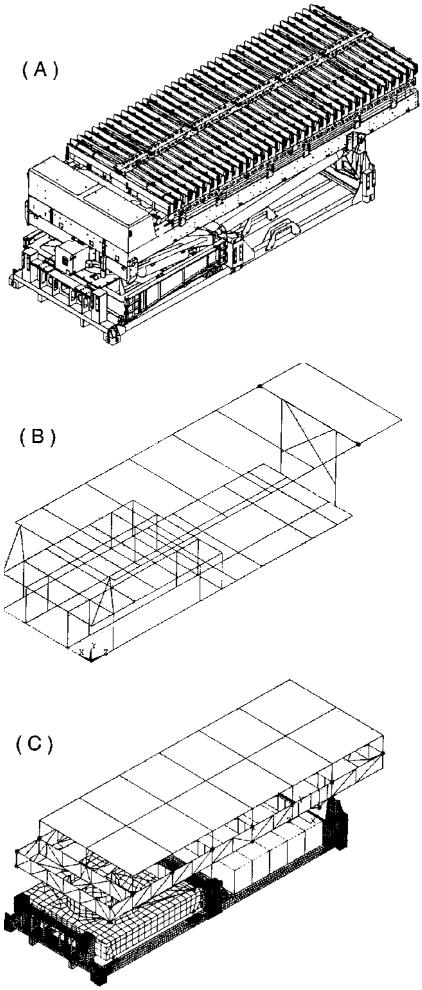


FIGURE 41.3 Illustration of FEM models for ground-based radar unit: (*A*) diagram of unit, (*B*) simple FEM model, (*C*) complex FEM model. (*Courtesy of Lockheed Martin Corporation.*)

model for different applications is illustrated in Fig. 41.3. A drawing of a ground-based radar unit in a stowed position for transportation is shown in Fig. 41.3*A*. A simple (400 degrees-of-freedom) beam approximation for the structure, which is adequate to estimate the first few normal mode frequencies of the equipment, is illustrated in Fig. 41.3*B*. In contrast, a complex (10,000+ degrees-of-freedom) model used for stress analysis is depicted in Fig. 41.3*C*. Construction and execution times of the two models are vastly different. The simple model in Fig. 41.1*B* was constructed in a day or so, and can be executed on the computer in a few minutes. Hence, it can be very useful in preliminary design where numerous analyses can be made with various different structural configurations to select a basic structural design that will have certain desired normal mode characteristics. On the other hand, the complex model, which includes nonlinear features, may take weeks of effort to construct and require hours of computer time to execute, making it practical only for final design. Model architectures must be carefully planned for specific objectives.

Statistical Energy Analysis Models. Even the most detailed FEM model becomes increasingly inaccurate at frequencies above about the 50th normal mode frequency of the structure. For equipment that is exposed to relatively high-frequency dynamic excitations, such as aeroacoustic excitations (see Chap. 29, Part III) or pyroshock excitations (see Chap. 26, Part II), FEM analysis procedures usually become costly and ineffective. In such cases, statistical energy analysis (SEA) procedures become attractive (see Chap. 11). However, SEA procedures have three important limitations, as follows:

1. They provide a vibration response averaged over a structural region, rather than at specific locations on the structure.
2. They provide a vibration response averaged over frequency bandwidths that each cover several normal modes of the structure (commonly ⅓-octave bandwidths), rather than at specific frequencies.

3. They provide accurate results only when there are at least five normal modes of the structure in the frequency bandwidth used for the analysis.

The above limitations make it difficult to translate SEA results into stresses at specific locations on the equipment structure. Nevertheless, SEA can yield valuable descriptions of the average shock and/or vibration response of structural elements in the equipment as a coarse function of frequency. Furthermore, since SEA models do not require structural details, they can be used effectively during the preliminary design phase.

PRELIMINARY DESIGN PROCEDURES

Based upon all the considerations and requirements discussed earlier, an initial design for the equipment should be made, perhaps with the assistance of a standard design handbook (e.g., Ref. 7), relevant reference books (e.g., Refs. 8 and 9), and/or specialized reference documents (e.g., Ref. 5). This initial design should be modeled by any of the procedures discussed earlier, although FEM and SEA models are preferred. A simple FEM model can be used to estimate the first few normal modes of the equipment, as well as the maximum displacements, velocities, and accelerations induced by the design shock and/or vibration excitations at frequencies up through the first few normal mode frequencies. An SEA model can be used to estimate the average accelerations of various elements of the equipment induced by the design shock and/or vibration excitations at the higher frequencies where there are at least several normal modes of the equipment in the SEA analysis bandwidths (usually ⅓-octave bandwidths). In either case, all of these responses can be evaluated by executing the model(s) for various different structural configurations.

Of particular concern early in the design process is the identification of the potential for excessive stresses in the equipment structure due to the design shock and/or vibration excitations. Since the maximum stresses in equipment structures exposed to shock and/or vibration excitations are generally due to the responses of the normal modes of the equipment, preliminary estimates of stress can be made using the relationship between maximum modal bending stress and maximum modal (relative) velocity given by[1,10]

$$\sigma_m \approx CEv_m/c \approx Cv_m\sqrt{E\rho} \tag{41.2}$$

where σ_m = maximum modal bending stress in the structure
v_m = maximum modal velocity of the structural response
c = speed of sound (longitudinal wavespeed) in the structural material
E = Young's modulus of the structural material
ρ = mass density of the structural material
C = constant of proportionality

The coefficient C in Eq. (41.2) is $C \approx 2$ for all normal modes of homogeneous plates and beams,[10] but can vary widely for complex equipment structures depending on the geometric details and the specific normal mode of the response.[11] Nevertheless, a value of C in the range $4 < C < 8$ is often assumed for the first normal mode response of typical equipment designs.[12] The first normal mode frequency of the equipment can be estimated early in the design using a simple FEM model, as illustrated in Fig. 41.3*B*. Equation (41.2) can then be applied to estimate the maximum stress in the response of any arbitrary equipment structure by assuming the following:

1. The maximum stress in the basic structure of the equipment occurs due to the response of the equipment at its first normal mode frequency.
2. The response of the equipment at its first normal mode frequency can be modeled by a base-excited single-degree-of-freedom system (oscillator), as illustrated in Fig. 23.5.

It is emphasized that this approach provides only crude estimates for maximum stress that are intended to provide guidance on desirable natural frequencies and damping ratios for the equipment design, and the possible need for a shock or vibration isolation system in the final design. Furthermore, it does not provide any information concerning the possibility of functional failures in electrical, electronic, or optical subassemblies in the equipment.

Shock Excitations. Consider a shock environment where the design excitation is described by a relative displacement shock response spectrum, as given by the maximum value of Eq. (23.33), which is denoted here as $\delta_m(f_n,\zeta)$ where f_n is the natural frequency and ζ is the damping ratio of the single-degree-of-freedom system. Since the shock response spectrum is defined as the maximum response of a single-degree-of-freedom system as a function of its natural frequency and damping ratio, it can be used directly with Eq. (41.2) to predict the maximum stress in the structure of equipment due to a response at its first normal mode frequency, specifically,

$$\sigma_m = CE(2\pi f_n)\delta_m(f_n,\zeta)/c \tag{41.3}$$

where all terms are as defined in Eq. (41.2) and the $(2\pi f_n)$ term is needed to convert the relative displacement shock response spectrum to an approximate relative velocity shock response spectrum, commonly referred to as a *pseudovelocity* shock response spectrum because it is an exact relative velocity shock response spectrum only for $\zeta = 0$. From Chaps. 8 and 23, for simple pulse-type transients, the SRS values vary only slightly with damping ratio for $\zeta \leq 0.05$. Hence, for such transients, the value of the damping ratio used to compute the SRS is not of major importance. However, for more complex transients like pyroshocks (see Chap. 26, Part II), the assumed damping ratio has a greater influence on the SRS value and, hence, must be more accurately defined.

For example, assume an item of equipment must be designed to survive the U.S. Navy high-intensity shock test for lightweight equipment, i.e., a weight of less than 350 lb (159 kg), which constitutes one of the most severe shock environments any equipment would experience in a service environment. The test machine is diagrammed in Fig. 26.6, and the SRS for the shock computed with a damping ratio of about $\zeta = 0.01$ is shown in Fig. 26.7. Further assume the equipment is to be constructed from a high-quality aluminum alloy, such as 2024-T3, that has a yield and ultimate strength of 50,000 psi (345 MPa) and 70,000 psi (483 MPa), respectively.[2] For aluminum, $E \approx 10 \times 10^6$ psi (6.9×10^4 MPa) and $c \approx 2 \times 10^5$ in./sec (5100 m/sec). From Fig. 26.7, if the first normal mode of the equipment were at 100 Hz, the velocity SRS value $[2\pi(100)\delta_m)$ would be about 400 in./sec (10 m/sec). Hence, from Eq. (41.3), even assuming an optimistic value of $C = 4$ and adding no design margin, the maximum stress in the equipment structure would be about σ_m = 80,000 psi (552 MPa). Although this stress is in the nonlinear region of the material, it probably would cause a structural failure. It follows that the designer should proceed assuming a shock isolation system (see Chap. 31) will be needed in the final design. On the other hand, if the first normal mode frequency of the equipment were above 400 Hz where the velocity SRS value from Fig. 26.7 is 180 in./sec (4.6 m/sec), then the maximum stress would be about 36,000 psi (248 MPa) and the equipment might survive without a shock isolation system. However, it would be difficult to design equipment with a first normal mode frequency above 400 Hz unless the equipment is relatively small.

Periodic Vibration Excitation. Consider a periodic vibration environment where the design excitation is described by a line spectrum, $L_a(f)$, with the units of g (acceleration in gravity units) versus frequency in Hz, as defined in Eq. (22.5). In the unlikely case where the fundamental frequency f_1 of the excitation is fixed, then the stress in the equipment response can be suppressed simply by pursuing a design with no normal modes of the equipment at frequencies near f_1, or any significant harmonics thereof. In many cases, however, the fundamental frequency of periodic vibration environments varies with time, e.g., rotating machinery and reciprocating engines that produce periodic vibration environments often operate at various different rpms. Hence, the designer must usually assume that at least one of the harmonic frequencies of the periodic excitation will correspond to a normal mode frequency of the equipment, at least on some occasions. From Eqs. (2.41) and (41.2), and assuming a damping ratio of $\zeta < 0.1$, the maximum stress in the equipment structure for a periodic excitation at the equipment natural frequency is given by

$$\sigma_m = \frac{CEgL_a(f_n)/c}{4\pi f_n \zeta} \tag{41.4}$$

where $gL_a(f_n)/(2\pi f_n)$ converts the periodic excitation in gravity units to velocity, and all other terms are as defined in Eq. (41.2).

For example, assume an item of equipment must be designed to survive a periodic excitation with an amplitude of $5g$ and a frequency, at least on some occasions, of 100 Hz. Further assume the equipment has a fundamental normal mode at $f_n = 100$ Hz with a damping ratio of $\zeta = 0.025$, and the equipment structure is a steel alloy where $E = 30 \times 10^6$ psi (2.1×10^5 MPa) and $c = 2 \times 10^5$ in./sec (5100 m/sec). Using an average value of $C = 6$, the maximum stress in the equipment structure is approximated by Eq. (41.4) as $\sigma_m = 55{,}000$ psi (380 MPa). A maximum stress of this magnitude would probably not cause an immediate fracture of a high-quality steel alloy, but it might ultimately lead to a fatigue failure. A preliminary estimate of the potential for a fatigue failure could be evaluated by estimating the number of cycles during the design life when the periodic component is at the normal mode frequency of the equipment, and then making a prediction of the fatigue life using the procedures detailed in Chap. 34.

Equation (41.4) provides important guidance to the designer of equipment that will be exposed to a periodic excitation at its fundamental normal mode frequency. Specifically, the maximum stress in the equipment structure is inversely proportional to the damping ratio of the structure. Hence, unlike pulse-type shock excitations, applied damping treatments (see Chap. 37) constitute a powerful design tool for reducing the maximum stress levels induced by periodic excitations.

Random Vibration Excitation. Consider a random vibration environment where the design excitation magnitude is described by a power spectrum, $W_{aa}(f)$, with the units of g^2/Hz versus frequency in Hz, as defined in Eq. (22.8). Assume the random excitation has a frequency bandwidth that covers at least the fundamental normal mode frequency of the equipment. From Eqs. (11.35) and (41.2), and assuming a damping ratio of $\zeta < 0.1$, the rms value of the maximum stress in the equipment structure due to its response at the first normal mode frequency is approximated by

$$\sigma_{\mathrm{rms}} = \frac{CE}{4c}\sqrt{\frac{g^2W_{aa}(f_n)}{\pi f_n \zeta}} \tag{41.5}$$

where $g^2W_{aa}(f_n)/(2\pi f_n)^2$ converts the power spectrum from g^2/Hz to v^2/Hz, where v is velocity in in./sec (m/sec) and all other terms are as defined in Eq. (41.2).

As an illustration, assume an item of equipment must be designed to survive a random vibration excitation with a magnitude (including a design margin) of $0.2g^2$/Hz at its fundamental normal mode frequency. Further assume the equipment has a fundamental normal mode at $f_n = 50$ Hz with a damping ratio of $\zeta = 0.025$, and the equipment structure is an aluminum alloy where $E = 10 \times 10^6$ psi (6.9×10^4 MPa) and $c = 2 \times 10^5$ in./sec (5100 m/sec). Using a conservative value of $C = 8$, the maximum rms stress in the equipment structure is approximated by Eq. (41.5) as σ_m = 8,700 psi (60 MPa). However, this is an rms stress. The maximum stress must be estimated in terms of a probability function. From Ref. 13, the maximum stress level that will be exceeded at least once during an exposure duration of T sec with a probability of $P(T)$ is estimated by

$$\sigma_m = \sigma_{\mathrm{rms}}\sqrt{2\ln\left[\frac{f_nT}{P(T)}\right]} \tag{41.6}$$

where ln [ ] is the natural logarithm of [ ]. For example, if the total exposure duration at the design magnitude is $T = 5$ h (18,000 sec), the stress level that might be exceeded with a probability of $P(T) = 5$ percent would be about 50,000 psi (345 MPa). This maximum stress probably would not cause an instantaneous fracture of the structure, assuming it is fabricated from a high-quality aluminum alloy such as 2024-T3 that has an ultimate strength of 70,000 psi (483 MPa),[2] but it might cause a fatigue failure over a sufficiently long exposure time.

It should be noted that Eq. (41.6) is unbounded, that is, there is no limit on the maximum stress as the duration T increases. However, experience suggests that this equation yields reasonable results for durations up to the equivalent of about 1×10^6 cycles, assuming the structural response is linear. For longer-duration environments, the potential for a structural failure should be evaluated using the fatigue prediction procedure detailed in Chap. 11 for a narrow bandwidth structural response, or a narrow bandwidth random fatigue curve.[12]

Equation (41.5) provides important guidance to the designer of equipment that will be exposed to a random vibration excitation at its fundamental normal mode frequency. Specifically, the maximum stress in the equipment structure is inversely proportional to the *square root* of the damping ratio of the structure, rather than the first power of the damping ratio, as for periodic vibrations in Eq. (41.4). Hence, applied damping treatments (see Chap. 37) do not provide as powerful a design tool for reducing the maximum stress levels induced by random excitations.

FINAL DESIGN PROCEDURES

The final design of equipment for shock and/or vibration excitations is best accomplished using a detailed finite element method (FEM) model, as illustrated in Fig. 41.3*C*. By applying the design excitations to the FEM model, the stresses at critical locations on the equipment structure, as well as the displacements and accelerations at those locations where equipment motions are critical, can be predicted for any modeled structural configuration. The designer can simply modify various elements of the structure to minimize the stress, displacement, and/or acceleration responses at all locations of concern to arrive at a final design. Specialized computer programs are available to facilitate these final design procedures (see Chaps. 27 and 28, Part II). Of course, all of the environments and requirements discussed earlier must be integrated into the design. In particular, the effects of the temperature environment on the strength and stiffness of all elements of the design that are temperature-sensitive must be carefully incorporated into the structural properties.

Fatigue Damage. For equipment being designed for a long service life, a primary step in the final design process is a fatigue life prediction. This can be accomplished in one of two ways, as follows:

1. For either periodic or random vibration excitations, the design excitation can be applied to the FEM model, and a sample time-history for the stress response at any location of concern can be computed. This sample time-history can then be used to predict the fatigue life using the procedures given for metals in Chap. 34 or composites in Chap. 35.
2. For random vibration excitations, the design excitation can be applied to the FEM model and the spectrum for the stress response at any location of concern can be computed. This spectrum can then be used to make a statistical prediction for the fatigue life using the procedures given in Chap. 11 and Ref. 14.

Higher-Order Response Modes. Some design shock and/or vibration excitations may have substantial energy in the frequency range of the higher-order normal modes of the equipment. Examples include motions of the equipment mounting structure induced by pyroshocks (see Chap. 26, Part II) and aeroacoustic excitations (see Chap. 29, Part III). In these cases, statistical energy analysis (SEA) models can provide valuable support to the design process, starting in preliminary design. Specifically, the SEA model can be used much like an FEM model to modify structural elements so as to minimize the motion response of the structure at any location of interest. As previously mentioned, it is difficult to obtain accurate stress predictions using an SEA model. However, the primary source of shock- and/or vibration-induced stresses in structural elements is usually due to the structural response in its lower-order modes. Hence, the FEM model will generally provide all the required stress data needed for a proper design.

Other Sources of Information. There are many specialized technical handbooks that cover the design of equipment for dynamic excitations that address specific equipment applications or specific types of equipment. For example, Ref. 15 is the NASA Technical Handbook that covers the design and testing of equipment for space vehicle shock and vibration environments. When available, such specialized handbooks should be consulted to support the equipment design process for shock and/or vibration environments.

DESIGN REVIEWS

Following both the preliminary and final design activities, there should be a thorough review of the design details. Following preliminary design, the review should include a study of all considerations that went into the design, including the assumed environments and requirements, the formulation of the design criteria, the planned methods of construction, the preliminary design analysis, and the planned final design analysis. Following final design, the final design analysis procedures and results should be carefully checked. These reviews should be performed by an independent group of engineers that were not directly involved in the design process. In smaller organizations, employing an independent contractor for the design review should be considered. This is particularly desirable if a failure or malfunction of the equipment during its service use could result in major economic damage or personal injury.

DESIGN VERIFICATION

Uncertainty is always present in the modeling and analysis of any dynamic system. By necessity, simplifying assumptions are introduced to make the analysis tractable. Naturally, unmodeled and unexpected phenomena will be present in a given equipment design. The significance of these effects is uncertain. Testing is often the only way to confidently confirm compliance with requirements. Furthermore, testing may also be used as a design tool for structures lacking suitable models, such as those with highly nonlinear response characteristics.

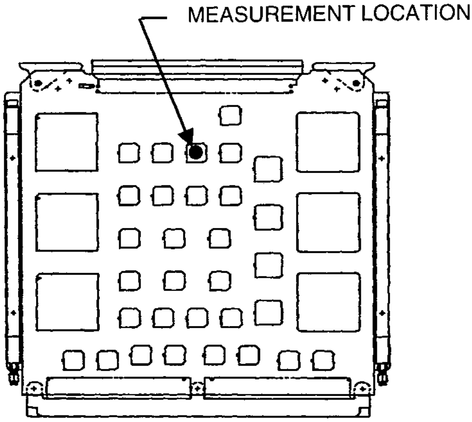
As in other phases of the equipment development, testing should be performed with a clear set of objectives. Since hardware testing can be expensive, careful planning is important to maximize benefits and efficiency. Some organizations separate testing activities from the design functions. Nevertheless, the designer should participate in determining the verification tests that will be performed. Shock and vibration test facilities are expensive to maintain and not available to many small companies and agencies. Commercial test facilities are available for such organizations. Chapter 19 describes general shock and vibration standards and Chap. 20 discusses the derivation of shock and vibration test criteria from measured or predicted excitation data.

MODEL-TEST CORRELATION

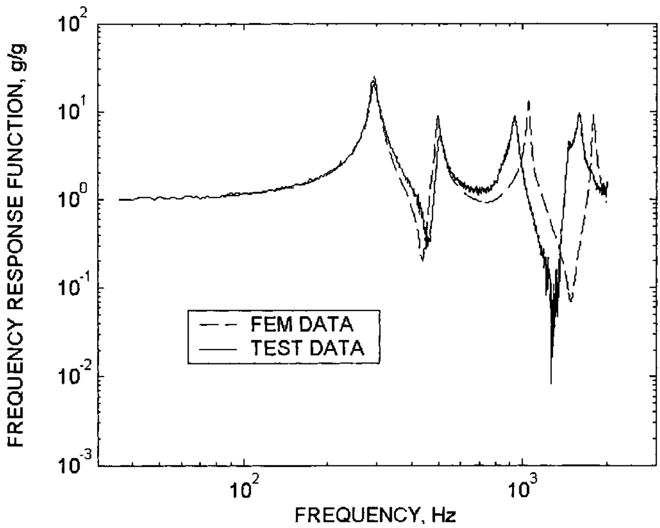
Dynamic testing often begins at low excitation levels in order to preview structural behavior and ensure proper instrumentation and test control without causing significant damage to the equipment (see *Development Testing* in Chap. 20). Data collected in the early phases of testing can be used to validate or refute models that may have been used to make design decisions. Full dynamic excitation tests also yield data useful for model correlation purposes, for example, the detection of nonlinear properties that were not modeled.

Frequency response functions, as defined in Chap. 21, are particularly well suited for model-test correlation purposes. In general terms, frequency response functions show input-output relationships. They are useful in relating inputs, such as force or motion, to outputs such as motion or strain. Frequency response functions can be experimentally generated from a variety of tests, including modal hammer impact tests and laboratory vibration tests. When properly determined, frequency response functions provide the modal parameters of the equipment, namely, natural frequencies (eigenvalues), mode shapes (eigenvectors), and damping ratios. Chapter 21 describes experimental modal analysis and modal parameter estimation techniques.

The frequency response functions for a printed wiring assembly computed using a simple FEM model (a few hundred degrees-of-freedom) and measured in a laboratory vibration test are compared in Fig. 41.4. A drawing of the printed wiring assembly is shown in Fig. 41.4*A,* and the frequency response functions com-



(*A*)



(*B*)

FIGURE 41.4 Comparison of FEM-computed and laboratory-measured frequency response functions for a printed wiring assembly: (*A*) diagram of assembly, (*B*) comparison of FEM and test data. (*Courtesy of Lockheed Martin Corporation.*)

puted using the FEM model and the laboratory vibration test are presented in Fig. 41.4*B*. The comparison shows good agreement for the lower-frequency modes, although the correlation degrades with increasing mode number. A more complex FEM model would provide better agreement for the higher-frequency modes, but often a confirmation of the first few modes is adequate for model verification purposes.

QUALIFICATION TESTING

A qualification test, as defined in Chap. 20, gives the designer and the customer confidence that the equipment will function properly in its expected service environment. It is usually a contractual requirement and commonly involves the application of all environments the equipment will experience in service, applied either in sequence or simultaneously. In particular, for equipment that will experience temperature extremes in service, a temperature test is often performed simultaneously with a vibration test using a combined temperature-vibration test facility. In any case, shock and/or vibration qualification tests occur too late in the design process to allow the cost-effective implementation of design changes. Thus, it is common practice to perform preliminary qualification-like tests before the design phase is completed to ensure the design will pass the qualification test requirement.

Qualification testing requires more than just the structural survival of the equipment within acceptable damage limits. A structure can survive the environment, but be rendered operationally useless by dynamic disturbances. Sometimes operational performance is restored when the dynamic excitation is removed, e.g., electrical circuitry can malfunction under dynamic excitation, intermittent problems can occur as gaps open and close, disruptive electrical noise can be generated, optical surfaces can be distorted, and servo-positioning systems can become unstable. The operational performance of the equipment must be closely monitored during the qualification test to identify any such malfunctions.

RELIABILITY GROWTH TESTING

A reliability growth test, as defined in Chap. 20, involves the following steps:

1. Assuming a sample item of equipment has passed the specified qualification test with no failures or malfunctions, increase the magnitude of the test level by some increment, usually 3 dB, and repeat the test.
2. If the equipment item again passes the test at this higher level, increase the magnitude of the test level again by the same increment and repeat the test.
3. Continue repeating the test at step-wise increased test levels until a failure or malfunction occurs.
4. If possible, repair the equipment to function properly and continue the testing at piece-wise increased test levels until another failure occurs.
5. Again, if possible, repair the equipment and continue the testing at piece-wise increased test levels until it is no longer feasible to make repairs that will allow the equipment to function correctly.
6. Report to the designer the details of all failures identified by the testing that could be repaired. Often simple changes in the design can be made that will suppress or eliminate the failures revealed by the tests.

The theory behind a reliability enhancement test described above is as follows. Even if the equipment is adequately designed to function properly during the qualification test, which represents a conservative simulation of the anticipated service shock and/or vibration environment, increasing the ability of the equipment to function properly during more extreme dynamic excitations will improve the reliability of the equipment in its service environment. Furthermore, by establishing the maximum shock and/or vibration excitations that the equipment can endure, it may be possible to use the equipment at a later time for another application involving more severe shock and/or vibration excitations without the need for a redesign and new qualification testing.

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