CHAPTER - 2

REVIEW OF LITERATURE

A brief review of the existing literature on launch vehicles dynamic environment analysis, structural vibration testing, design and analysis of vibration test fixtures, vibration test setup and different types of qualification tests are presented in this chapter. Various areas covered include the vibration test specifications generation, basic design concepts for test fixtures, material characteristics, dynamic characteristics, vibration shakers and multi shakers operations. A brief review is presented on the latest techniques to modify the dynamic behavior of the test fixtures, advances in vibration testing and measurement are also reviewed in brief.

2.1 LAUNCH VEHICLE DYNAMIC ENVIRONMENT ANALYSIS

Launch vehicle subsystems during transportation [48, 52], liftoff [72] and its atmospheric ascend experiences fluctuating loads [2, 49] by jet noise as well as aerodynamic noise [109]. The acoustic loads [30, 40, 43] generated by the exhaust jet of the
launch vehicle is more at the liftoff because of the jet reflections on
ground [33, 36, 108]. During atmospheric ascent the aerodynamic noise is generated by boundary layer turbulence [31], flow separation, shock wave oscillation and its interaction with boundary layer. The vibration due to aerodynamic noise increases with dynamic pressure (q) and reaches the maximum level at q maximum [38, 44]. Author told that nearly half of the failures observed during the flights of satellite launch vehicles are due to mechanical vibration and shock [151]. For success of launch, it is very essential to ensure that the systems could survive the expected vibration environment of the vehicle.

For simulation [29, 41] of vibration environment, test specifications are necessarily realistic and adequate. If the equipment operates smoothly during a test, there should be a very strong probability that it performs its intended task correctly in the real environment. Hence, the specification must be at least as severe as the real environment and also representative of the real environment [51, 214]. Generally at the design stage and the initial development stage, existing standards like the Military Specifications or Joint Service Specifications are used as guidelines for formulating the specifications [149]. MIL-STD-810F describes the procedures for test specifications formulations. The
specifications cause either an overtest or an undertest with regards to the overall spectrum. To overcome this, it is essential to develop a correct specification using the telemetry data acquired for a few flights [205] and from acoustic testing [69, 70].

The space launch vehicle is instrumented with vibration sensors to study the flight environment [35]. Onboard vibration levels are measured through telemetry during the previous flights [200]. The recorded data is analyzed to deduce the information about the flight vibration pattern, maximum vibration levels experienced, the vibration spectra and generation of a vibration envelop for reviewing the vibration test specifications. Based on the flight data of an aerospace vehicle, the linearly averaged spectra for previous flights are estimated for each of the three axes. These spectra are shaped according to the guidelines and the tailored spectra are obtained. Using these spectra, the break points are decided and a safety factor is applied on every break point to tailor the vibration specifications as shown in figures 2.1& 2.2 [151].
Fig. 2.1 Linearly averaged flight data

Fig. 2.2 Tailored vibration spectra
2.2 STRUCTURAL DYNAMIC TESTS

Two kinds of dynamic tests i.e. dynamic characterization test [113] and vibration test available for dynamic environment. The dynamic characterization test is a modal survey test [126], to determine structural characteristics represented by mass, damping and stiffness matrices \( m, D, \) and \( K \). The vibration test is used to simulate critical flight loads in terms of excitation force and to measure structural response. Vibration test demonstrates strength and adequate dynamic behavior of the structures.

During vibration test, certain limitations exist to reach any desired force and response level depending on the size and characteristics \( (m, D, \) and \( K) \) of the structure to be tested [93]. During dynamic characteristic test, such limitations generally do not exist because the required results \( (m, D, K \) or equivalent quantities) do not principally depend on the excitation conditions. The Dynamic tests are used to subject the test articles to the specified vibration levels and thus to qualify them against the expected flight loads. Dynamic characteristics are extracted from the measured responses during testing to verify relevant model predictions under realistic loading conditions.
Various types of dynamics tests have different purposes [1]; different frequency ranges of applicability, and also different advantages and disadvantages, so it is very important to tailor the test specifications to fit the needs, reliability, schedule, and cost. There are three reasons for conducting structural dynamic tests: qualification, workmanship, and verification [204].

The primary reason for conducting structural dynamic tests of launch vehicles is to design qualification [141, 156]. Dynamic tests of launch vehicle structures are used to simulate the flight dynamic environments. Typical vibrations are so severe that they would cause failure of many electronic components, mechanisms, optics, and structures [96]. The tests typically represent a simulation of the dynamic environments defined from a statistical analysis of previous launch vehicles [91]. Author described the environmental testing of NASA’s Evolutionary Xenon Thruster prototype model 1 reworked engine [194]. Random vibration testing is conducted at a level of 10 g\(_{\text{rms}}\) in three axes at 2 min per axis. Pre and post sine resonance surveys showed some significant differences indicating some changes to the test hardware.

A secondary reason for conducting structural dynamic tests of launch vehicle structures is to identify workmanship defects,
which if undetected would cause problems or failures in flight [107]. Most workmanship defects are detected at lower levels of assembly, but there are some interface and interconnection problems that can only be detected in the system level tests. It is, however, important that the test levels in workmanship dynamics tests be low enough so that they do not cause problems that would not occur in flight. Surface tension propellant tank plays a major role in aerospace industry. These tanks are manufactured with very thin material of the order of 2 to 3mm. Surface tension propellant tanks are subjected to different structural tests at component levels. Mechanical vibration tests are the most complicated, risky, and expensive component level tests applied to the propellant tank [73, 81]. The main purpose of vibration testing of propellant tank is to demonstrate that the tank can withstand, with margin, the dynamic accelerations expected during transportation, handling, launch and placement on orbit. These tests have provided valuable data to correlate and correct analytical models used to predict structural damages.

The third reason for conducting structural dynamics tests of spacecraft is to verify dynamic models [168]. This is the justification for modal testing, and tests to verify jitter and in-flight vibration models. In these cases it is also important that the test
levels and durations be such that the tests are nondestructive. Author described a numerical analysis of the dynamic response of the mechanical structure and the fairing inner acoustic cavity of the Brazilian Vehicle Satellite Launcher [155]. Finite element and boundary element methods are used for low frequency analysis. The high frequency vibro acoustic behavior is analyzed using a statistical energy analysis model.

2.3 FORCE LIMIT VIBRATION TESTING

Flight equipment is exposed to random vibration excitations during launch. In the dynamic testing, the random vibration design levels are applied at the equipment mounting interface. For lightweight aerospace structures, the mechanical impedance of equipment and of the mounting structure is typically comparable. So the vibration of the structure and load involves modest interface forces and responses. Most of the high amplification resonances and mechanical failures in conventional vibration tests are test artifacts associated with the essentially infinite mechanical impedance and unlimited force capability of the shaker.

The classical test approach is using motion controlled envelope to input acceleration levels. The motion control induces
excessive vibration responses at equipment resonant frequencies. This problem has been recognized and emerged proposals for alternative vibration control techniques. Force limit approach [77, 84, 138] is the most recent technology developed as an alternate vibration control technology. This method is also called as dual control method. It limits force levels as well as acceleration levels at the input to the test article. This technique was first proposed by Murfin in 1968 and recently developed by Smallwood [83] and Scarton [97, 103].

The force limit vibration testing technique has [78, 79, 124] applied to eliminate over testing caused by the mechanical impedance of the shaker in conventional vibration tests. With this newly developed technique, the acceleration input is automatically notched at the resonance frequencies of the test item by specifying a force limit [140]. It is becoming standard practice in NASA aerospace programs to measure and limit the input forces in vibration tests of structure-like hardware, [123, 137]. Force limiting makes vibration tests more realistic by simulating the impedance characteristics of the flight mounting structure [129]. The most straightforward method of defining force limits [92] is based on the interpretation of the quasi-static limit loads. Then, by Newtons Second Law, the force limit is equal to the limit load times the
weight of the test item. For random vibration test, it is common to multiply the measured rms force by a peak factor of 3 for comparison with the limit loads [135].

The process of deriving an acceleration specification for a vibration test is illustrated in Fig2.3. The ragged curve in Fig2.3 represents a hypothetical measurement of the acceleration at the interface of the mounting structure and the item to be tested. The vibration test specification [76] is typically derived by averaging, enveloping, and adding a margin to the data. Unfortunately, the acceleration notches at the load anti resonance frequencies, where the interface force is a maximum and the acceleration is a minimum, are lost in this smoothing process. The load is very responsive at the anti resonance frequencies [71] and acts as a dynamic absorber to reduce the input.

![Coupled System Resonances](image)

**Fig. 2.3 Dynamic Absorber Effect of Load on Interface Acceleration**
Eliminating the notches in the acceleration input is resulted in overtesting in conventional vibration tests by typically 10 dB to 20 dB [142]. In force-limited vibration tests, both the interface acceleration and force are controlled [99]. The reaction forces at the load anti resonance frequencies [102] are limited to values predicted for the flight mounting, and the notches in the input acceleration are automatically restored. Equation 1, which may be derived from Norton’s and Thevinen’s equivalent circuit theorems, provides a theoretical basis for dual control of force and acceleration in vibration tests:

\[ 1 = \frac{A}{A_0} + \frac{F}{F_0} \]  

Eq [2.1]

Where:
- \( A \) is the interface acceleration,
- \( A_0 \) is the source free acceleration,
- \( F \) is the interface force, and
- \( F_0 \) is the source blocked force.

Equation [1] is exact but difficult to apply because each term is a complex function of frequency, and phase is ignored in current vibration test controllers. An approximate form of equation [1] useful for control of vibration tests is the extremal control equation:
|A| ≤ |A_s|, and |F| ≤ |F_s|  

Eq [2.2]

Where the free acceleration and blocked force are replaced by specifications that envelope the interface values in the coupled system. Vibration controllers have the capability for extremal control i.e., to select the larger of two or more feedback signals in each narrow frequency band. Old controllers allow only one specification. In this case, to implement dual control, a filter is used to scale the shaker force control signal to an equivalent acceleration. New controllers allow separate specifications for each control channel so that Equation [2] may be implemented without any complications.

2.4 DESIGN OF VIBRATION TEST FIXTURE

Vibration tests are performed to qualify launch vehicle structures for real time environments. Electrodynamic shaker systems are used to generate an input spectrum that envelops the actual dynamic environment. Vibration test specification mandates a particular input to the shaker that is needed to satisfy the requirement. The interface between the shaker and the test article is the test fixture [45]. The test fixture includes the shaker armature, expander head (or slip plate) as well as the attachment
test fixture. Fig 2.3 shows a typical schematic of the elements of a fixture. Ultimate goal of a fixture designer is to make the fixture infinitely stiff and massless [132]. This essentially implies that the fixture would be resonance free over the test frequency range of interest. Practically the test fixtures experience resonant behavior due to the mass/stiffness characteristics of the armature, expander head (or slip plate) and attachment fixture.

Fig. 2.4 Typical vibration test fixture components
For a developing space structure, because of the complexity of structure and uncertainty of environmental condition in a general way, there is no mature specification for the environmental vibration test. So, it is very important to study the characteristic and optimal design of the fixture attaching the tested structure to vibration table [50]. Author claimed that the fixture design was a challenge task, especially to a new complicated tested structure [162]. In the environmental vibration test, the tested Structure-Fixture-vibration Table (SFT) must be regarded as a whole system in order to analyze the effects caused by fixture on the vibration test. This is an active system with feedback control. The effects caused by the fixture are concerned with the number, the location of measured points and the feedback control manners.

Author described the key issues in the design of vibration test fixture [203]. According to his opinion, test fixture allows for ease of mounting of the test specimen to the vibration table. Test fixture allows mounting the test specimen in each of the orthogonal directions with minimum cross talk. Test fixture ensures the absence of fixture resonances in the testing frequency band. Weight of the moving mass and force limitations of the shaker plays a predominant role in the fixture design. Test fixture ensures uniform distribution of vibration energy to the test specimen.
The dynamic coupling effects of a vibration test fixture [197] are explained by considering a vibration fixture and test specimen using a simple model. He considered two different test fixtures with a test specimen. A simple two DoF model is used to represent the test article and a simple single DoF model is used to represent the first resonant frequency of each of the test fixtures. The test article is tested to maximum frequency shown in Fig 2.5. Both fixtures considered for the test have resonant frequencies beyond the test frequency range but fixture 1 has its first resonant frequency very close to the test range upper frequency.

The systems are coupled [42] together, and the resulting frequency response is shown in Fig 2.6 for the two fixtures. From Fig 2.7 it is understand that the resonant frequencies of the test specimen are only slightly different comparing the two different fixtures. However, the amplitude of the response is significantly different. This means that the response levels are different. So while the frequencies are not changed significantly, the mode shapes are significantly different. This means that the test specimen will be exposed to different levels of acceleration than desired due to the use of the flexible fixture. This clearly shows that the fixture should be as stiff as possible and should be well beyond the test frequency range of interest.
Fig. 2.5 Mode superposition of each of the SDoF systems
Fig. 2.6 Uncoupled fixture and test specimen

Fig. 2.7  
(a) Uncoupled fixture test specimens
(b) Coupled fixture test specimens.
2.5 STRUCTURAL DYNAMIC MODIFICATIONS

Structural dynamic modification using transfer function is used to the design of vibration test fixture [217]. The response at any point on the fixture is predicted utilizing the experimental data. Structural dynamic modification of the fixture is performed so that the spectra at the mounting points meet the specified reference spectrum. The values of the added masses on the fixture are used as design variables for removing the under tests in environmental vibration test.

Dynamic design is the process obtaining desired dynamic characteristics by specifying the right shape, size, configuration, materials [168]. Desired vibration characteristics include reduced noise levels, avoidance of resonances, higher dynamic stability and desired mode shapes. Conventionally, the dynamic designs have been carried out via intuitive cut and try prototype testing. It involved testing of several prototypes configured as per proposed designs/modifications. This is very time consuming and expensive and usually did not lead to an optimum solution. Structural dynamic modification methods are the techniques that intend to determine the changes in the vibration characteristics as a result of certain design modification like addition of a mass, spring, and
stiffener. The techniques form the basis for performing dynamic
design at the computer level.

Environmental tests are performed on spacecraft prior to
launch in order to verify the ability of the structure to survive the
dynamic environment [47]. The tests are implemented by mounting
the whole spacecraft on the moving armature of a large
Electrodynamic structure. The objective of the test is to reproduce
environment at the base of the spacecraft and excites to actual
launch environment. The Mariner Mars 1971 spacecraft designed
and built at JPL, presented a fair amount of dissymmetry, mainly
due to the weight difference between the two large tanks of the
propulsion subsystem used at Mars encounter in placing the
spacecraft in orbit around the planet. As a result of this
dissymmetry, it was anticipated that an overturning moment of
substantial magnitude at the base of the spacecraft could possibly
occur during the environmental sinusoidal vibration test at certain
resonant frequencies. A large overturning moment would have
presented potential damaging effects on the armature of the shaker
by creating large lateral and rotational armature displacements
and possibly subsequent damage to the spacecraft. Therefore, a
detailed dynamic analysis was made to calculate the armature
displacements during a computer simulation of the vibration test.
The dynamic characteristics of the shaker and the spacecraft had to be known in order to carry out this analysis and simulation. The method outlined in the previous section was implemented. A combination of modal test and analysis was employed to obtain the models. Knowing the dynamic characteristics of the shaker armature was of primary importance, since it was in essence the support structure for the whole spacecraft in the testing configuration. It was believed that dynamic modal tests rather than static tests would be preferable to obtain those dynamic characteristics. To this end, a rigid mass of about 500 kg was mounted on the shaker armature, and eight natural frequencies, mode shapes, and damping are determined experimentally.

2.6 MULTI SHAKER VIBRATION TESTING

Single shaker system is having limitations to test large structures. It is not at all practical to provide a fixture for a large specimen if single shaker is used. Heavy specimens are handled easily on multi shaker systems. Multi shaker system generates more force to drive the specimen. Several shakers can even be arranged to eliminate the fixture, with definite cost savings. Multi shakers [167] can be used separately and individually for the testing of smaller specimens, when not needed to form one large single testing system. Simultaneous, multi axial vibration testing in six
degrees of freedom is possible with the help of multi shakers [88, 104]. Several commercially available test controllers have the sophisticated computational capability, data throughput speed and convolution algorithms to solve the problems associated with MIMO control [218]. A variation of MIMO is known as MESA where more than one shaker is used to provide force in single direction. In the case of MESA the control system is capable to feed all control points in phase. If the shakers go out of phase, very large moments are generated in the specimen or shaker. It has been recognized that the multi axes testing provides a more realistic representation of actual field conditions [143]. However, the little research is conducted in systematically studying the differences between the multi axial system and single axis system but these studies are not incorporated in the standard testing procedures.

A multi-point excitation vibration test strategy [188] aims to realize a specified vibration spectrum on a full aerospace vehicle, as an alternative to the section level testing being carried out currently, using modal excitation methodology. The strategy is based on a modal formulation and is demonstrated first on a uniform beam of circular cross-section, which denotes an idealized structural model of a generic aerospace vehicle. The study proposes two strategies, one based on single-point excitation and the other based on two point excitations, while both sine sweep...
and random excitations are used for generating the responses. Experiments are carried out on the same two configurations for generating the response spectrum and the results show that there is a good match between the theoretical predictions and experimental observations. The study also proposes a general formulation to quantify the difference between the desired spectrum and the achieved spectrum, which can be used to optimize the locations of the excitation points. The above strategy is also applied to a non-uniform beam, which represents an aerospace vehicle structure more realistically. The study brings out the fact that a multi-point excitation is better able to achieve the desired response spectrum on a full aerospace vehicle.

2.7 CHARACTERIZATION OF ELECTRODYNAMIC SHAKER

The performance envelope of an Electrodynamic shaker system is strongly influenced by three modes of vibration and the voltage/current capacities of the power amplifier that drives it [150]. Other limiting factors are the designed stroke (displacement) of the table, the moving mass and the total mass of the shaker, the thermal power limit ($i^2R$) of the coil and the stress safety factor of the armature. The structure of an Electrodynamic shaker bears
some resemblance to a common loudspeaker but is more robust. At the heart of the shaker is a coil of wire, suspended in a fixed radial magnetic field. When a current is passed through this coil, an axial force is produced in proportion to the current and this is transmitted to a table structure to which the test article is affixed.

Fig2.8 show the magnetic circuit used to create the intense magnetic field required by the shaker. A permeable (ferrous) inner pole piece transmits flux from one end of an axially magnetized permanent magnet or electromagnet, say, the North face. A permeable “back structure” conducts flux from the opposite pole of the magnet to a permeable disk with a hole in its center surrounding the inner pole piece. This creates a radial flux field in the air gap between the round face of the North-polarized inner pole piece and the round hole in South polarized outer pole piece. The air-gap between these pole pieces is minimized to reduce the reluctance of the magnetic circuit thus maximizing the intensity of the fixed magnetic field.

The force provided by the machine is proportional to the magnetic flux passing through the coil, to the current flowing through the coil and to the length of wire within the flux field. In general, shaker coils use heavier conductors to accommodate heavier currents. The coil, coil form and table structure
combination is called the armature assembly. The test object is rigidly mounted to the armature assembly. Some shakers have interchangeable armatures, providing a small table for high-acceleration level testing of light objects and a large table for mounting heavy objects. In older designs, the coil is wound around the outer diameter of a stiff, thin-walled tube. Modern armature designs typically use epoxy bonding techniques to affix a rigid epoxy-stabilized coil to a light magnesium table structure [152].

The armature must be accurately centered in the narrow gap between the inner and outer poles. It must be allowed to move axially while being restrained from all other motions. This is accomplished by a soft elastic suspension system. In small shakers, a pierced compliant disk provides radially distributed cantilevers between the load table and the shaker body. In larger units, guide rollers support and center the armature while separate elastomeric shear elements provide the axial compliance. This compliant connection between the armature assembly and the shaker body forms an obvious spring/mass/damper vibration system with one degree-of-freedom. Here, the test object and armature assembly move together, relative to the shaker body. Adding two more degrees-of-freedom completes the shaker mechanical model. First, the armature structure is recognized as
being elastic rather than rigid. This is modeled by treating the coil
and table as separate masses connected by a spring and damper.
Second, shakers are frequently isolated from the building floor by
use of compliant mounts that allow the entire machine to translate
vertically. This is modeled by attaching the shaker body mass to
ground using a spring and damper. The shaker’s electrical model
must account for the resistance and inductance of the armature
coil. The coil resistance $R$ defines the minimum impedance
exhibited at the shaker input terminals. This resistance increases
substantially with temperature (the resistance of copper wire
increases by about 40% per 100° C) and increases slightly with
frequency (due to the skin effect). The coil inductance $L$ is large
because the coil couples strongly with the iron of the pole pieces,
causing the complex electrical impedance, to increase with
frequency. The interplay between the electrical and mechanical
domains is not a “one-way street.” When the coil moves within the
magnetic field, a voltage is generated across the coil in proportion
to the velocity. This “back EMF” is seen in the electrical domain as
an increase of the coil impedance and reflects the mechanical
activity into the electrical circuit. These interactions are reflected in
the composite mechanical-electrical model of Figure 2.9.
Three modes of vibration dominate the mechanical response. At very low frequency (often below the range of operation), the compliant isolation mounts allow the entire shaker to translate as a rigid body with almost no relative motion between the components. This deformation shape is termed the Isolation Mode. In the low end of the operating range (10 to 40 Hz, typical) the Suspension Mode dominates. In this shape, the table and coil move together relative to the shaker body. Motion in this mode is limited by the design stroke of the machine. At or beyond the high frequency limit of operation, the (undesired) Coil Mode is encountered. Here, the coil moves out-of-phase.

Fig. 2.8 Electrodynamic shaker
A dynamic mathematical model involving three vibration modes (isolation, suspension and coil) [119] with appropriate electromechanical cross coupling to a two element electrical circuit provides clear understanding of shaker system behavior. Maximum performance is achieved by driving the model with a composite current that respects stroke limit, drive coil power limitation; amplifier voltage/current limits and rated force capacity. A simple graphical means of conservatively estimating this behavior over the usable range of payload has been presented.
Current (as opposed to voltage) is clearly established as the desired reference for all structural transfer function relationships within a shaker. The ability to separate electromagnetic damping from other sources of damping has been demonstrated. Low frequency performance is dictated by the shaker’s design stroke and further limited by the need to isolate the machine from the laboratory building. The effects of isolation on achievable stroke have been investigated with particular emphasis on selecting appropriate damping factors. The interplay of maximum amplifier voltage in the neighborhood of suspension resonance has been reviewed with some interesting new findings regarding table velocity limit. High frequency performance is shown to be limited by the “coil mode” resonance. Exceeding the rated high frequency limit can result in overstressing the armature structure of the machine. Clever design placement of the coil mode frequency and the resulting anti-resonance above suspension frequency can actually improve the high frequency performance of a shaker. Power analysis discloses the Electrodynamic shaker to be a thermodynamically inefficient machine. As machine payload is increased, efficiency decreases while line power factor improves. The laboratory thermal load (for fixed intensity shakes) is almost independent of test item weight. Power analysis discloses that an isolated system can be designed to improve mechanical delivery in the low frequency region.
An Electrodynamic shaker is modeled as a mixed electrical/mechanical system with an experimentally derived two port network characterization [121]. The model characterizes the shaker with amounted load can be predicted. The characterization depends on the measurements of the shaker input voltage and current, and on the acceleration of the shaker armature with several mounted loads. The force into the load is also required and can be measured directly or inferred from the load apparent mass.

2.8 History of Vibration Testing and Measurement

The status of testing (simulation) had advanced significantly by the mid-1950s. Kroeger stated that vibration testing procedures had undergone a pronounced evolution since 1947 [28]. His paper presented a review of vibration testing procedures leading to the present concept of the continuous spectrum or random excitation. Means used to define the vibration existing in an environment were discussed, and there is a comparison of scanning at discrete frequencies versus testing with random excitation. Crede [3] not only presents a clear description of the vibration test trends of 50
years ago, but discusses factors that influence simulation choices that are still valid today. His closing remarks were probably written because at that time random vibration was still quite controversial: Finally, the simulation of vibration environments is currently in a very tentative state. Much more information is needed, not only on the nature of environments but on the strength of equipment when subjected to vibration, before it will be possible to state with some degree of certainty that a particular testing procedure is adequate to qualify equipment for use in any designated environment. In 1956, random vibration was still a new concept. To become better acquainted with the problems with this concept, McIntosh and Granick [4] conducted a study of the response of simple beams to this form of vibration. The work in their preliminary investigation is described in their paper. The authors intended to continue and extend this investigation to determine the value of random vibration testing. The final paragraph in this paper is quoted to give the reader an understanding of the status of random vibration at that time: The real value of random vibration test techniques actually cannot be appraised until more is known about the true service environment. Investigations must be conducted on the structural conditions in widely scattered regions of missiles. These investigations ought to be of a thorough, comprehensive nature similar to those conducted on aircraft. It is essential that the
highest priority be given to these assignments, because from these studies and from laboratory experiments, we shall be looking for the answer to the question: Is it necessary to adopt random vibration testing as a general requirement? Noonan [5] reviewed the requirements for vibration testing of shipboard equipment specified under Type I - Environmental Vibration, in MIL-STD-167(SHIPS). Information was given to provide a better understanding of the importance of environmental testing. He describes some of the reasoning behind the development of the MIL-STD-167 specification that after a few revisions is still the specification required for equipment to meet Navy vibration requirements. This paper provides excellent guidance on the application of the specification, the types of equipment to be tested and on designing to meet test requirements. A milestone development that advanced our vibration test capability probably more than any other single event is described in a paper by Hansen [6]. He developed the first horizontal oil slip table for horizontal vibration testing by using a horizontal test fixture that could technically be called a “flat hydrostatic bearing.” It consists of a flat, fixed plate upon which a movable specimen table in the form of a similar flat plate is suspended on a film of oil. The specimen table is free to slide horizontally but resists with almost infinite force any vertical motion either as a complete entity or as a
spurious, resonant, standing wave such as those to which most other tables are prone. (The forces are similar to those that hold together optical flats or Johansson blocks.) The lower plate is fixed solidly to the foundation or to a large mass such as a concrete-filled steel box. The height and level of the whole fixture is adjusted by four jack screws located in the bottom corners. The movable specimen table is driven by a drive rod in the conventional manner. There was extensive discussion following Hansen’s presentation, all of which is published with the paper in the proceedings. It is believed that within two years, most all of the vibration test laboratories had a horizontal slip table. At the 27th symposium, there were a number of papers on fixture design, with special emphasis on horizontal testing. Hansen was an innovative test engineer just doing his job. He very likely had no idea that his work would have such a great impact on vibration test technology.

To give a better understanding of the status of shock and vibration activities 50 years ago, a few examples are given from papers describing the effort at that time. For example, Armstrong [7] was concerned with evaluating the validity of shock tests. The purpose of his paper was to explore the usefulness of some of the simplest dynamic and static relationships in pointing the way toward testing that can result in correct design and evaluation
decisions in cases where the state of knowledge of field condition or specimen response is incomplete and where also the capabilities of testing equipment falls short of the mark in various respects. In other words, the essence of valid shock simulation is that the tests reveal in the test specimen the effects that would result from the service condition being simulated. Morrow [8] Discusses a number of considerations in shock and vibration, the philosophy of smooth specifications, the test of components versus parts, force versus acceleration for amplitude excitation and the single frequency equivalent. His discussion relates mostly to vibration test specifications. It was noted previously that random vibration in the 1950s was a relatively new and controversial concept. Booth [9] presented a paper describing the nature of random motion and how it is generated. He said that the properties of random motion most useful to the vibration engineer are stated in simple terms. The method used to create these motions in the laboratory and operation of the major components of the required equipment is briefly described in his paper.

Wimpey [10] presented a method of applying the continuous spectrum concept to vibration analysis of electronic components. The acceleration spectral density spectrum concept is defined by the author, and the mechanizing of this concept and its
subsequent evaluation are discussed. Modifications of conventional vibration equipment are described and instrumentation methods for measuring the statistical variables with this technique are defined. In the 1950s, vacuum tubes (electron tubes) were still widely used. Robbins [11] described a white-noise vibration test developed for the vibration evaluation of vacuum tubes over a wide range of frequencies. White-noise, random vibration with all frequencies present is explained theoretically and compared with sinusoidal vibration. The author describes a practical test method and presents details on the white-noise generator, vibration test equipment and methods of reading the tube noise output. Five decades ago, we were already in the age of jets and guided missiles; this resulted in an increase in the severity and variety of mechanical environments for equipment in military applications.

Among the more important resulting changes was the extension of the frequency range of mechanical excitation to which electronic components were subjected. Researchers realized that much of this excitation is transient and aperiodic in nature. This is why Wohl and Schnee [12] discussed the relative effectiveness of impulse versus steady-state excitation in the field of resonance and compared vibration testing of vacuum tubes using a precision
impulse exciter. Optimum excitation, representative of the broadest range and field environment, is considered.

Fuses were important in the warheads of guided missiles. For this reason, two independent approaches were followed by Warren [13] to obtain information for designing reliable fuses. One approach determines the vibration environment of the fuse in flight and attempts to simulate it in the laboratory. The other approach is to construct the fuse to function reliably under the full output of a vibration exciter while being swept through the available frequency range. At the time the paper was written, the first approach had not been successful. As a secondary objective, however, production of an equivalent damaging effect was satisfactory. Relating to vibration testing, Yorgiadis [14] collected and published experimental acceleration-time records of various types of mechanical vibration tables. He showed that in addition to the fundamental sine wave of vibration, there are superimposed high-frequency random harmonics that are of substantial magnitude. These undesirable harmonics were found to originate in gears, ball and roller bearings, or other regions of repeated localized impact. A vibration testing machine for very heavy loads was described by Brown and McClintock [15] Large airborne cryogenic equipment weighing up to 10 tons was subjected to
sinusoidal vibration testing on the NBS-60,000 mechanical shaker. The experience gained was used to design a second machine capable of testing units weighing up to 20 tons.

Barnes and Mock [16] described two vibrators for testing small electronic components. One is a magnetostrictive unit with a flat response (20%) between 1 and 10 Hz. The second is a liquid jet vibrator producing a nearly uniform acceleration spectral density between 1 and 10 Hz. Edelman, et al [17] described a special vibrator for testing items up to 10 lb to accelerations of 10 g from 1,500 to 15,000 Hz. Greater acceleration can be reached at the many axial resonances of each vibrator.

A lot of attention was and still is paid to shock testing devices and techniques. Blake [18] presented a good argument that machines for simulating shock (as well as sinusoidal and random vibration) cannot afford to neglect the mechanical impedance of the item under test. Unfortunately, machines of that era were usually designed to duplicate the envelope of typical field motions without considering the impedance. Blake acknowledges the difficulties of dealing with this problem, but points out that finding a solution would enhance the validity of the tests. There were several other diverse papers on shock testing in the same proceedings. At the
Naval Ordnance Laboratory, there was a need for a shock testing machine to validate the design of submarine weapons and equipment. Mead [19] describes a program to develop a shock test machine called the UWX (underwater explosion). This machine may well be the predecessor of a machine developed at Naval Ordnance Laboratory at White Oak and designated as the WOX (White Oak experiment). The WOX machine still exists, and tests are being run at its present location in NSWC Dahlgren, Virginia.

Before the current design of the Navy’s Floating Shock Platform (FSP), now the Navy’s official heavyweight shock test facility, there was a program to develop a mechanical heavyweight machine described by Gareau [20] in a 1956 paper. At the end of the paper, a schematic drawing of this complex machine was included. Of course, the machine was never constructed. Westgate [21] described an unusual drop-test facility to test materials and instruments at high impact levels under controlled conditions. The facility was a 300-ft, universal, drop-test tower that was installed at Sandia Corp. (now Sandia National Laboratories) and used a then-new type of gas-energized, hydraulically controlled accelerating device known as the HYGE actuator in machine operation, stored energy in the form of compressed air is released instantaneously, and the waveform is then controlled by means of
hydraulic flow through an orifice controlled by a metering pin. Sanders [23] describe the design, operation and performance of the pneumatic impactors used to generate the boost phase of the flight of the missile. The simulation covers the high acceleration at the start of the boost phase and moderate deceleration between the booster burn-out and the missile’s combustor ignition. Finally, an inexpensive shock machine developed for testing lightweight items was described by Schatz [24] the shock pulse obtained is approximately a square wave, and a somewhat novel arresting media of lead and plastic was employed. Four papers in the area of measurement in the 1950s provide interesting information. Jones, et al [25] published a paper describing a number of small pick-ups designed to meet special requirements in a number of different applications. Essentially small pick-ups are required to avoid loading the equipment under test significantly. This way the weight of the pick-up does not affect the equipment response during the test and ensures that the measurements are accurate.

Meyer [26] describes the development and use of a statistical amplitude probability meter. It was a device that would provide a measurement of the statistical amplitude probability distribution of a forcing function. By this means, randomness is proven or, if the function is not random, will give insight to the amount of
periodicity present. Upham and Dranetz [27] describe development of a miniature recording accelerometer that can be mounted directly on a test structure. At that time, it had already been used to monitor simulated water-entry impact of torpedoes. A playback device reduces pulse-coded information to analog form for further recording, analysis or control. Transducers must be calibrated. Christensen reviewed sensors as well as calibration techniques employed in the laboratory. He discussed the virtues and vices of commonly used calibration techniques. He then described a system of optical calibration that had been developed.

It is obvious that the references discussed in this section are mostly from the 23rd Symposium. This symposium had the greatest number of attendees ever. It is viewed as a milestone symposium at a time when enthusiasm for advancing the technology was at a peak. It provides a reference point to compare today’s capabilities in the same technical areas. Of course, there have been many landmark papers published in the many symposia since then, but it is not possible to reference and discuss each of these in this brief article. Instead, I refer you to a DVD containing all unclassified symposium papers through the 78th Symposium. The DVD is available for purchase from SAVIAC. There is no question that our current capabilities in shock and vibration
testing, measurement and analysis are indeed remarkable. In the remainder of this article, some of the developments that accelerated this advancement are discussed.

2.9 Advances in Shock and Vibration Technology

Probably the most significant development that contributed to the technological advancement was the development and evolution of the computer. Most “old timers” will remember that we were still doing computations using the slide rule. Development of the electronic calculator was a huge step forward at the time. But by far the greatest increase in our computational capability is the evolution of the computer. The early computers were slow and had little memory and, in fact, were used mostly for word processing. Over the years, with development of hard drives and the rapid increase in the amount of memory available, our computational capability rapidly increased. Today the average personal computer has many times the memory capability of the early mainframe computer and is faster and easier to use. Software development kept pace with the hardware, and today the response of the equipment and structures to shock and vibration loads can be rapidly calculated. This is possible due to the development of methods like finite-element modeling. Using FEM on complex
structures produces models having 100 degrees of freedom or more. Computer programs like ANSYS can calculate the response of such structures without the burden of extensive computer time. Equations of motion in matrix form have become the usual approach to conduct dynamic analyses in both the time and frequency domain. In the area of testing, measurement and data analysis computers are also the tool responsible for our increased capability. Both shock and vibration tests are more often than not programmed and controlled by computers resulting in more precise and accurate simulation of the field environment. We are in a digital age. Using a personal computer with several gigabytes of memory, an engineer can analyze field or test data and present the results very quickly in the format of his choice. For example, from a transient acceleration time history record he can calculate velocity or displacement, plot a shock spectrum and much more.

In the area of shock data analysis, the digital age essentially began in 1965 with the development of An Algorithm for the Machine Calculation of Complex Fourier Series developed by Cooley and Tukey. This is better known as fast-Fourier transform (FFT). In the late 1960s, several papers covered the digital calculation of response spectra from earthquake records, studies of selected shock analysis methods and digital shock spectrum
analysis by recursive filtering. In addition, a monograph was published by SAVIAC in 1969 on the principles and techniques of shock data analysis. In this digital age, publications like these have resulted in an ever-growing list of vendors offering analyzers and other equipment for special purposes like machine monitoring for fault detection, response of structures to earthquakes and other applications. There has also been a rapid development of new transducers to measure almost any parameter that might be related in some way to shock and vibration. Most recently, there has been a surge in the development of MEMS devices, some of which are miniature transducers such as accelerometers. Despite all these advancements, many research efforts are underway to develop new approaches and equipment to help solve some problems that still exist. One wonders what the shock and vibration world will be like 50 years from now.