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Design and Analysis of Vibration Test Fixtures for Payloads

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DESIGN AND ANALYSIS OF VIBRATION TEST FIXTURES FOR PAYLOADS

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Abstract

The project addresses the design principles and performs finite element analysis (FEA) of vibration fixtures used for testing of space payloads at the MIT Lincoln Laboratory. A review of standards that govern vibration testing of space and general payloads is presented. A set of principles and a tutorial for the design of vibration test fixtures is presented. The validation of the design principles is achieved through FEA of a vibration test fixture that involves a structure with panels, rails and support blocks. The FEA code PATRAN/NASTRAN is used and analysis of the three-dimensional computational results leads to design modifications that involve an added third panel. The redesigned fixture has increased stiffness, as well as reduced stress and displacement of its rails.

Acknowledgments

We are extremely grateful for all the support and guidance we've received from all the people who became involved in our project. First, we would like to thank our MIT Lincoln Laboratory supervisor, Dr. David Freeman, for introducing new concepts for our fixture and providing useful suggestions towards our project's success. David also assisted in making sure the project was meeting our deadlines and helped point us out to the engineers around the lab who could also assist in modifying our design. We would also like to thank Anne Vogel and Mike Mastovich for providing references regarding vibration test standards and test fixture properties.

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Chapter 1 - Introduction

MIT Lincoln Laboratory, created in 1951, is a federally funded research and development center (FFRDC). The Laboratory, which employs approximately 1,500 technical staff members with a variety of degrees as illustrated in Figure 1, focuses on developing and implementing advanced technology to support national security. The goal of this MQP is to generate and validate a set of design principles for vibration test fixtures that can be used throughout the Laboratory, by engineers of various disciplines.

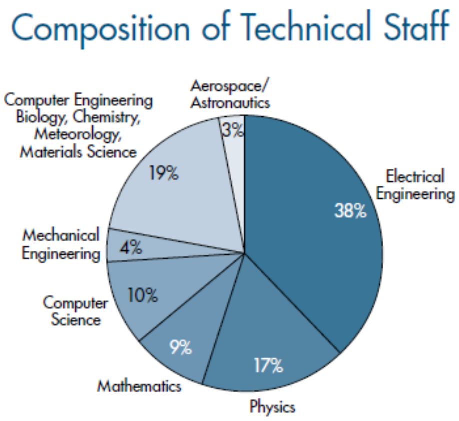

Figure 1 - Staff Composition

Between 1990 and 2010, the MIT Lincoln Laboratory has fielded seven space payloads. Between 2010 and 2013, the Laboratory is expected to deliver six new space payloads. To meet the demand created by this increase, engineers are taking on more roles and becoming more involved in all aspects of design and testing at the system and subsystem level.

Additionally, more and more experienced engineers are retiring. As “Workforce Demographics Among Engineering Professionals, A Crisis Ahead?” predicted in 2001, many engineers have reached retiring age, leaving the less experienced engineers to approach various aspects of engineering with minimum advice from the practiced and knowledgeable engineers[1]. One of the ways Lincoln Laboratory is addressing this is by exploring ways to share lessons learned. This project is a part of this effort designed to give inexperienced engineers insight into one of the most potentially destructive environmental tests, vibration testing.

[1] http://www.caee.utexas.edu/org/ccis/a_ccis_report_21.pdf

Environmental testing assesses the payload under conditions similar to the expected environments the system will encounter, and demonstrates whether the system will be negatively affected by these environments. This type of testing is conducted to validate the design of the payload and to identify any workmanship defects. Environmental testing includes several types of tests, such as shock and vibration tests, thermal cycle tests, electromagnetic compatibility tests, and thermal vacuum tests.

1.1 Project Objectives

The main objective of this MQP is to generate a set of principles for the design of vibration test fixtures, specifically fixtures used in the testing of space payload. The greatest vibratory forces for space payloads occur during launch. The vibration tests are designed to demonstrate the payload can withstand the expected vibration environment without being adversely affected. However, the vibration test itself can be destructive to the test article. In a vibration test, a test fixture connects the part to be tested to the table that will vibrate, called a shaker table or vibration table (pictured in Figure 2), and transmits the force from the table to the part. A forcing function is inputted, and the response of this part due to the function is measured. For engineers unfamiliar with the possible complications of vibration testing and test fixtures, the test could provide misleading results, or even cause damage to the payload being tested.

Figure 2 - Environmental Test Lab’s Vibration Table

Another objective of this MQP is to generate a tutorial for designing a vibration test fixture. The design principles discussed in this MQP will be validated through a design study of a vibration test fixture to support an MIT Lincoln Laboratory space payload and finite element analysis of the model.

1.2 Project Approach

The objectives are accomplished by the following approaches and methodologies. In order to assess the current practices for vibration testing, the General Environmental Verification Standard (GEVS) and the Military Standard 1540 Revision E (1540E) governing space payloads are reviewed. In addition, the Military Standard 810F (810F), which covers general types of vibration environments, is also reviewed.

A comprehensive review of vibration test fixtures leads to a set of design principles and a tutorial that governs the design of a vibration test fixture.

The validation of the established design principles is achieved through Finite Element Analysis (FEA) of the test fixture shown in Figure 3. The FEA is performed using 3D program PATRAN/NASTRAN. Based on the computational results, design modifications are

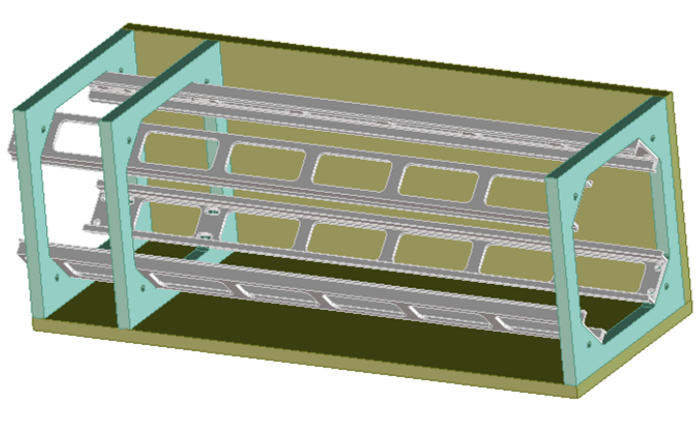

Figure 3 - Screenshot of Original CAD Model

made in accordance with the design principles, in order to increase the stiffness of the fixture and meet the stress and displacement requirements. The redesigned fixture is analyzed using PATRAN/NASTRAN, and the effectiveness of the design modifications is evaluated.

Chapter 2 - Vibration Test Standards

Vibration testing is used throughout the design process. Different types of tests are utilized as the process progresses, all of which focus on maintaining the integrity of the subsystem or system. These tests follow a set of standards which define the test as well as state the requirements that must be met for a part to be considered acceptable. In order to get a better understanding of vibration testing, various standards were reviewed to provide a basis for the process of testing. The following chapter is devoted to summarizing those standards.

2.1 GEVS[2]

The General Environmental Verification Standard, more commonly known as GEVS, is one standard that deals specifically with vibration testing. The standard was created by NASA, as a way to create an organization-wide basis to vibration test parts and is summarized in this section.

GEVS specifies sine sweep vibration testing, which slowly sweeps through a specified range of frequencies to demonstrate the modal responses of a part at a given frequency at various levels of assembly. The purpose is to test prototype/protoflight parts for the low-frequency sine environment they will encounter during flight as well as to test the workmanship of the part.

Though not specifically mentioned in the standard, it should be noted that sine sweep vibration testing can be potentially destructive. If a part's natural frequency is within the testing frequency range, the slow pace of the frequency sweep will cause the part to linger at its natural frequency for a long period of time, which could lead to destructive vibration. The

[2] All information provided in the following section, pages 9-15, is taken from the General Environmental Verification Standard.

Tables 1 and 2 describe the test factors and durations of the specified tests and the test requirements.

Table 1 - Test Factors/Duration

Sine Vibration Test	**Prototype Qualification**	**Protoflight Qualification**	**Acceptance**
Level	Limit Level x 1.25	Limit Level x 1.25	Limit Level
Sweep Rate[1]	2 octaves/minute	4 octaves/minute	4 octaves/minute

1 – The sweep direction should be evaluated and chosen to minimize the risk of damage to the hardware. If a sine sweep is used to satisfy the loads or other requirements, rather than to simulate an oscillatory mission environment, a faster sweep rate may be considered, e.g., 6-8 octaves/minute to reduce the potential for over stress.

Excerpt from Table 2.2-2 from NASA-STD-7000, pg 2.2-5

Table 2 - Test Requirements

	Payload/Spacecraft	**Subsystem/Instrument**	**Unit/Component**
Sine Vibration Tests Required	Test performed to simulate any sustained periodic mission environment, or to satisfy other requirement (loads, low frequency transient vibration).	Test performed to simulate any sustained periodic mission environment, or to satisfy other requirement (loads, low frequency transient vibration).	Test performed to simulate any sustained periodic mission environment, or to satisfy other requirement (loads, low frequency transient vibration).
	Test must be performed for ELV payloads, if practicable, to simulate transient and any sustained periodic vibration mission environment.	Test must be performed for ELV payload instruments and for ELV payload subsystems if not performed at payload level of assembly due to test facility limitations; to simulate sine transient and any sustained periodic vibration mission environment.	Test must be performed for ELV payload, instruments, and components to simulate sine transient and any sustained periodic vibration mission environment.

Excerpt from Table 2.4-1 from NASA-STD-7000, pg 2.4-2

2.1.1 Space Transportation System (STS) Payload Vibration Testing

For STS payloads, sine vibration is required to simulate the inputs from applicable sources, such as retro/apogee motor resonant burning, ignition/burnout transients or control-jet firings, which are expected to occur during flight. Each payload should be analyzed to determine what expected vibration environments it will encounter, and then should be tested at the calculated vibration environments.

Swept sine vibration tests are required at the payload, instrument, and component levels to ensure each level of the assembly can handle to expected vibration it will encounter during flight. Specifically, for the payload level test, the payload should be in a similar arrangement to the configuration it is expected to be in when the stress will occur during flight.

The sine vibration environment must have a 99.87% probability of not exceeding the maximum expected level. This is to be estimated with 50% confidence, which is equal to the mean plus the three-sigma level for normal distributions. Furthermore, if parametric statistical methods are used to determine the limit level, the data should be tested to validate the assumed distribution. The qualification is defined as the limit level multiplied by 1.25, and the test input frequency range should be limited to a range of 5 to 50 Hz.

If fatigue life is to be tested as well, a prototype verification program must be created, and the length of the tests must be modified accordingly. The prototype should be exposed to at least four times the intended life of the part, and should take into account both vibroacoustic and sine vibration environments.

2.1.2 Expendable Launch Vehicle (ELV) Vibration Testing

ELV Payload Sine Sweep Vibration Testing

For ELV payloads, prototype/protoflight parts should undergo a design qualification test to verify the part can survive the low-frequency launch environment, and to ensure quality workmanship. The sine vibration environment must have a 97.72% probability of not exceeding the maximum expected level estimated with 50% confidence. This is equal to the mean plus two-sigma level for normal distributions. The payload should be tested at qualification level, or flight level limit times 1.25.

While all ELV payloads should ideally be tested, it is not always practical for payloads with large masses or extreme lengths to be tested. If the sine sweep vibration test is impractical at the payload level, it must be performed at the lowest possible level of assembly.

For the actual test, the payload must be in the same configuration at it would be during launch. It should be attached to a vibration fixture, by use of a flight-type launch-vehicle attach fitting, referred to as an adapter, and attachment hardware, or separation system. With one axis parallel to the thrust axis, sine sweep vibration should then be applied to the base of the adapter. To ensure accurate simulation of the flight sine transient vibration, the test sweep rate should be 4 octaves per minute, with lower sweep rates utilized when necessary to match the length and rate of change of frequency of any flight sustained vibration. The test should be executed from the frequency range 5 to 50 Hz, in each test axis.

For sine sweep testing on protoflight parts, loads induced while sweeping through resonance must not exceed 1.25 times flight limit loads; if necessary, test levels can be lessened at critical frequencies. Furthermore, if the model used for the loads analysis has adequate detail and the specific responses are recovered from the results, acceleration responses of certain critical items may also be restricted to 1.25 times flight limit levels, if necessary. To aid in shaker control, the minimum input test level is ± 0.1 g.

Prior to protoflight sine sweep testing, a low-level sine sweep test should be performed in each axis, to obtain data at locations determined by notching analysis. The results should be analyzed to ensure the response is similar to the predictions.

GEVS does not have one standard test level, because sine vibration tests must simulate the flight environment, and test levels required to simulate the environment are dependent on variables such as the overall weight and length, as well as mass and stiffness distributions. Instead, mission specific test levels must be created. Preliminary levels can be estimated by using the part's user manual sine vibration level for spacecraft base drive analysis. The frequency level should not exceed 50 Hz, and the notching levels should be based on the equivalent loads to the user manual c.g. load factor loads. Additionally, other resources, such as coupled loads analysis for similar spacecraft or spacecraft interface dynamic response data from flight measurements, can be used for the base drive input, together with an appropriate uncertainty factor.

Finally, before and after each vibration test, the payload should be scrutinized and tested, to ensure it is still functional within the specified requirements.

ELV Payload Subsystem and Component Sine Sweep Vibration Testing

The ELV payload subsystem, including instruments, and components, should also be subject to a sine sweep vibration test along each of the three orthogonal axes, to validate the design and check for any workmanship defects. Overall, the requirements for ELV payload subsystems are very similar to the requirements for ELV payloads. Again, for the sine vibration environment, there must be a 97.72% probability of not exceeding the maximum expected level, estimated with 50% confidence, which is equal to the mean plus two-sigma level for normal distributions. The qualification test level remains the same, at the limit level times 1.25. Furthermore, if fatigue life is to be considered, the same procedure described for SVS payloads should be utilized for ELV payload subsystems and components. In fact, almost all of the requirements for ELV payloads remain for ELV payload subsystems. The only notable difference occurs when determining mission specific test levels.

As with ELV payloads, it is impossible to create one specific test level that would be effective for every payload subsystem. However, the process for determining mission specific test levels is where vibration testing for ELV payloads and ELV payload subsystems differ. The primary factor influencing the sine sweep test levels is the test data from structural model spacecraft sine sweep tests. If the test data is not available, there are two other courses of action.

If acceleration-response time histories are accessible, coupled loads analysis dynamic responses should be employed for all pertinent flight loading conditions. Shock response spectra, or SRS, techniques for temporary flight events should be utilized to determine equivalent sine sweep vibration test input levels, but is not required for pogo-like flight events. This is done by dividing the SRS by Q, where $Q = C_c / 2C$. Lacking test data, Q is typically assumed to be between 10 and 20, with the lower values being more conservative.

Additionally, the subsystem/component responses from a base drive analysis should be included in the test level, with the spacecraft sine sweep test levels as input. If the spacecraft sine sweep rates are not incorporated in the base drive analysis input, the responses must be corrected to account for them. Both the subsystem/component rates and the spacecraft rates should be corresponding. If the shaker can only apply translational, not rotational, accelerations, and substantial rotational responses are expected, the test levels should be increased to ensure appropriate response levels. Furthermore, for situations where the part is expected to undergo considerable rotations and translations, it is acceptable to implement suitable notching or limiting, and make the c.g. rotational and translational acceleration response levels the maximum level that cannot be exceeded.

Finally, as with ELV payloads, ELV payload subsystems must be tested before and after each vibration test, to make sure it is still functional within the given requirements.

2.2 1540E[3]

Another standard commonly used is the Military Standard 1540 Revision E, often shortened to just the 1540E. The original Military Standard 1540 was created by the Aerospace Corporation in 1974, to establish a standard for all Air Force space systems. Since then, it has been updated several times, slowly evolving into the current standard 1540E, introduced in 2006[4].

While GEVS requires sine sweep vibration, the 1540E calls for random vibration to ensure the part will be able to withstand the maximum predicted vibration environment, or MPE, in flight. For the set-up of the test, the part should be mounted to a fixture that has been tested and determined suitable before the vibration tests. Connecting items, such as wiring harnesses and hydraulic and pneumatic lines should be included, up to the first attachment point, and should be connected using flight-like connectors.

[3] The information in the following section, pages 15-17, is provided by the Military Standard 1540 Revision E.
[4] http://www.aero.org/publications/crosslink/fall2005/backpage.html

There are three different types of tests required: qualification, protoqualification, and acceptance. The purpose of a qualification test is to make sure the part can handle multiple rework and test cycles. The qualification test proves that later units will not fail after several acceptance tests. The qualification test validates the design and manufacturing process, as well as the acceptance program. Each type of item that is acceptance tested must be qualification tested as well. For basic qualification testing, liftoff and ascent vibration with an effective length of 15 seconds will be applied to the part, which must be tested at 6 dB above acceptance for 3 minutes/axis. Qualification tests are performed at the P99/90 level, meaning the probability is 0.99 with the confidence at 0.90.

Protoqualification tests are utilized when the part that is tested will be flown. The test reduces the amplitude and duration applied to the hardware, and can heighten the risk of the part failing during flight unless other testing and analysis and implemented. For basic protoqualification testing, the part must be tested at 3 dB above acceptance for 2 minutes/axis. Protoqualification tests are performed at 3 dB over the acceptance level of 95/50.

Acceptance tests reveal any workmanship errors, and confirm the part meets the specifications required. Basic acceptance testing must include the MPE and minimum level described in the Figure 3[5] for 1 minute/axis. Acceptance tests are performed at the P95/50 level.

$$T_Q = 4 \left(T_{MPE} + T_A / 10^{Mb/20} \right) \qquad [1]$$

In the above, T_{MPE} is the effective duration of MPE environment in flight,
T_A is the highest limit on acceptance testing,
M is the test margin over acceptance,
B is the exponent on stress for the fatigue life, and

[5] Figures 3 – 19 and Tables 5 – 9 can be found in Appendix A.

T_Q is the duration requirement

The durations for the qualification and protoqualification tests are determined from the above equation, e.q. (1). For baseline qualification testing, the standard values are normally 15 seconds for T_{MPE}, 8 minutes for T_A, 6 dB for M, and 4 for b. When these values are put into the equation, T_Q is calculated to be 3 minutes, which is the standard basic duration. If any values differ from the standard values, the required duration can be calculated. Therefore, if the part is expected to be exposed to flight vibration lasting longer than 15 seconds (T_{MPE}), the value should be changed, and the duration of the test should be recalculated.

For the test, all electrical and electronic units, including primary and redundant circuits, and even circuits that do not operate during launch, must be electrically energized and functionally sequenced to the maximum practical limit. Parameters including voltage, current, and relay contact should be examined during the test for failure or intermittent performance. Furthermore, any part that must be functional under pressure should be pressurized to the expected pressure, and watched for pressure decay.

If necessary, the vibration spectrum can be limited or notched, though the notching is limited by the minimum spectrum provided in Figure 3. Situations where the spectrum can be notched include when analysis has demonstrated the possibility for response levels above the given design limits, or when impracticable input forces will occur and must be altered.

While vibration tested is not usually required for structural loads, it can be necessary to prove structural integrity for units with critical fatigue modes of failure with a low fatigue margin. In this case, the duration should be the equivalent duration for fatigue at flight at the maximum expected level multiplied by four. Additionally, if a structural unit does not undergo a static strength qualification test, vibration testing is necessary; the unit should be tested at 3 dB above the MPE until it reaches a steady-state response. However, it must be tested for at least 10 seconds, to guarantee the strength requirements will be met.

Finally, low level vibration testing should be conducted on each axis before and after the vibration tests, to ensure the part is still functional.

2.3 Comparing GEVS to 1540E

The preceding standards, GEVS and 1540E, govern the same type of vibration environments. Both standards deal with space payloads and the resulting vibration environment encountered. The following chart compares the two standards, summarizing the pertinent information.

Table 3 - Comparison Chart

	GEVS	**1540e**
Sine Test Required	Sine Sweep Testing	Random Vibration Testing
	Prototype Qualification Test / Qualification Test	
Level	Limit Level x 1.25	Acceptance Level + 6 dB
Duration	2 octaves/min from 5 – 50 Hz	3 min/axis
Probability/Confidence	99.87/50 (STS) 95/50 (ELV)	99/90
	Protoflight Qualification Test / Protoqualification Test	
Level	Limit Level x 1.25	Acceptance Level + 3 dB
Duration	4 octaves/min from 5 – 50 Hz	2 min/axis
Probability/Confidence	99.87/50 (STS) 95/50 (ELV)	95/50
	Acceptance Test	
Level	Limit Level	Acceptance Level
Duration	4 octaves/min from 5 – 50 Hz	1 min/axis
Probability/Confidence	99.87/50 (STS) 95/50 (ELV)	95/50

2. 4 810F[6]

The military standard 810F is more diverse than the previous two standards, covering all types of vibration, not just space payloads. The standard differentiates between types of vibration environments, referred to as categories, and requires different types of tests for each category. Table 4 shows which tests are required for each category and the following sections go into further detail as to what each category and test means.

Table 4 - Vibration Environment Categories

Life Phase	**Platform**	**Category**	**Material Description**	**Test** 1/
Manufacture / Maintenance	Plant Facility / Maintenance Facility	1. Manufacture / Maintenance processes	Material / assembly / part	2/
		2. Shipping, handling	Material / assembly / part	2/
		3. ESS	Material / assembly / part	3/
Transportation	Truck / Trailer / Tracked	4. Restrained Cargo	Material as restrained cargo 4/	I
		5. Loose Cargo	Material as loose cargo 4/	II
		6. Large Assembly Cargo	Large assemblies, shelters, van and trailer units 4/	III
	Aircraft	7. Jet	Material as cargo	I
		8. Propeller	Material as cargo	I
		9. Helicopter	Material as cargo	I
	Ship	10. Surface Ship	Material as cargo	I
	Railroad	11. Train	Material as cargo	I
Operational	Aircraft	12. Jet	Installed Material	I
		13. Propeller	Installed Material	I
		14. Helicopter	Installed Material	I
	Aircraft Stores	15. Jet	Assembled stores	IV
		16. Jet	Installed in stores	I
		17. Propeller	Assembled / Installed in stores	IV/I
		18. Helicopter	Assembled / installed in stores	IV/I
	Missiles	19. Tactical Missiles	Assembled / installed in missiles (free flight)	IV/I
	Ground	20. Ground Vehicles	Installed in wheeled / tracked / trailer	I/III
	Watercraft	21. Marine Vehicles	Installed Material	I
	Engines	22. Turbine Engines	Material Installed on	I
	Personnel	23. Personnel	Material carried by/on personnel	2/
Supplemental	All	24. Minimum Integrity	Installed on Isolators / Life cycle not defined	I
	All Vehicles	25. External Cantilevered	Antennae, airfoils, masts, etc.	2/
1/ Test procedure – see corresponding section 2/ See Categories section 3/ Use applicable ESS procedure. 4/ See test configuration under Test Procedure section				

Vibration Environment Categories, taken from MIL-STD-810F, Table 514.5-I

[6] The information in the following section, pages 19-32, is provided by the Military Standard 810F.

Categories

Category 1 deals with the manufacturing and maintenance processes. All parts will undergo some type of vibration during these processes. If the environmental effect is significant enough it should be included in design calculations. In the case where multiple copies of the same part endure considerably different vibration exposure, namely different vibration levels or durations, use the maximum values to determine vibration test specifications.

Category 2 revolves around shipping and handling, where transportation will expose the parts to vibration. As with category 1, only significant vibration should be included in design calculations, and if the same parts are exposed to a range of vibration, the maximum value should be considered for the vibration test.

To ensure the part can withstand transportation, the part should be configured the way it would be during transit. The packaged part should not have low resonant frequencies, to make sure there will not be damage during shipment. Lightly suspended internal elements should be blocked to avoid low frequency relative motion between suspended elements and adjacent structures. The lowest suspension frequency should be at least two times the frequency of any low frequency spike. in addition, if the material is fixed relative to the transportation method, the vibration tests should be related to the transportation method’s orientation. If orientation can vary, the test should account for the envelopes of possible orientations.

Category 3 includes environmental stress screen vibration, shortened to ESS, in which parts are exposed to during manufacturing and maintenance. Parts can undergo several cycles of ESS before production acceptance, and repeated exposure can lead to vibratory fatigue. Maximum allowable exposure should be included in design calculations as environmental test preconditioning. Exposure strength levels are the same for each part, so the specified level should be used. Duration, however, varies part to part, and the maximum allowable production and maintenance should be used.

Category 4 consists of truck/trailer/tracked restrained cargos, which refers to broadband vibration due to the interaction of the vehicle with the road and discontinuities in the surface. The first type of environment is described as truck transportation over U.S. highways, characterized by transportation from the plant to any continental United States storage facility or user installation. The part is usually transported anywhere from 3200 to 6400 kilometers, or 2000 to 4000 miles, over paved highways by a large truck and/or tractor-trailer. For truck transportation over U.S. highways, exposure levels should be determined by Table 5. The exposure duration is usually one hour per 1609 kilometers, or 1000 miles, per axis.

The second type of environment, mission/field transportation, involves the transportation of cargo between 500 to 800 kilometers, or 300 to 500 miles, over unimproved roads under combat conditions. This environment can be further broken down by its methods of transportation, namely two-wheeled trailer and wheeled vehicles or tracked vehicles. For two-wheeled trailer and wheeled vehicles, exposure levels should be determined by Table 5, and the duration is 32 minutes per 51.5 kilometers, or 32 miles, per axis. For material too large for two-wheeled trailer, the composite wheeled levels in Table 5 should be used, and the duration for those should be 40 minutes per 804.6 kilometers, or 500 miles, per axis. For tracked vehicles, a narrow band random-on-random vibration exciter control strategy is required to reach the required exposure levels detailed in Figure 4, while the duration is simply the environmental life cycle.

Category 5 discusses truck/trailer/tracked loose cargo, which includes cargo that is free to bounce and collide with either other cargo or the side of the vehicle during transportation over irregular surfaces. To test for this type of random shock environment, a set-up similar to that of Figure 5 should be constructed. The movement of the bed is a 2.54 centimeter diameter orbital path at 5 Hz, and duration of 20 minutes is sufficient for 240 kilometers, or 150 miles.

Category 6 references truck/trailer/tracked large assembly cargo. For this category, it must be understood that the part and the vehicle vibrate as a flexible system. The actual transport vehicle should be used as the vibration exciter, and the part should be configured and secured the way it would be during actual transportation. The vehicle should be driven over a prepared test course at a specified speed that will adequately represent the road conditions and speed for the expected course. The distance travelled over each segment of the course should be determined in accordance with the Life Cycle Environment Profile.

Category 7 deals with aircraft jets, where the vibration is broadband random with the maximum occurring during takeoff. For low frequency, specifically below 15 Hz, it is assumed that cargo will not respond dynamically, and the cargo experiences it as steady inertial loads. For large cargo items, especially those with natural frequencies below 20 Hz, the items may interact with aircraft structural dynamics, which could have serious consequences. Such cargo should be evaluated by aircraft structural engineers, who must contact the System Program Office. For vibration qualification tests, the System Program Office is again to be contacted to determine the exposure level for the aircraft type. The duration is to be taken from the Life Cycle Environment Profile. Alternatively, Figure 6 details the cargo compartment zone functional qualification levels for certain aircraft, based on the worst case zone requirements, thereby providing a mildly conservative value. If Figure 6 is used to determine the exposure level, a duration of one minute per takeoff should be implemented, with the number of takeoffs obtained from the Life Cycle Environment Profile.

Category 8 deals with propeller aircraft, whose vibration is characterized by relatively high amplitudes. The assumptions valid for category 7 hold true with category 8, and the System Program Office must be contacted with regards to parts with natural frequencies below 20 Hz. As with category 7, the exposure level and duration should be taken from the System Program Office and the Life Cycle Environment Profile, respectively. If no exposure level criteria is available from the System Program Office, measurements of cargo deck vibration in the aircraft will need to be taken.

Category 9 discusses helicopter aircraft. The cargo transported via helicopter will encounter continuous wideband, low-level background vibration with strong narrowband peaks superimposed. However, determining the vibration the cargo will be subject to inside the helicopter is a complex function, dependent on the interaction of the cargo with the helicopter structure and the cargo's location within the helicopter. As such, measurements of the vibration of the cargo in the particular helicopter are required; category 14 will go into more detail. Cargo transported below a helicopter as a sling load will experience low level random vibration, along with low frequency motions because of the sling suspension modes. The sling stiffness and suspended mass should ensure that suspension frequencies do not correspond with the helicopter main rotor forcing frequencies because destructive vibration can occur. In addition, suspension frequencies must not be within a factor of two of forcing frequencies. Currently, there is no data to characterize slung cargo vibration levels. Instead, the material should meet the levels provided by category 24. For either type of helicopter vibration, the duration should be taken from the Life Cycle Environment Profile.

Category 10 is about the vibration environment of cargo on the surface of the ship, which is fundamentally identical to the vibration environment for material installed on ships. As such, the exposure level and duration are discussed in category 21.

Category 11 concerns the vibration environment of trains, which is characterized by low and moderately wideband levels, with more intense lateral vertical axis vibration as opposed to longitudinal. Figure 7 offers a broad description of vibration levels, and should be used for the duration provided from the Life Cycle Environment Profile. If the levels provided by Figure 7 are significant to the material, measurements should be taken to determine the actual environment.

Category 12 consists of material installed in jet aircraft. Airframe structural response, due to events such as landing impact, is important for the outer regions of flexible structures. The vibration environment varies with airframes, so the vibration must be evaluated through measurement. One such type of vibration that must be measured occurs when there is cavity noise induced vibration, caused by airflow across a cavity created by an opening in the aircraft skin. For small cavities, the vibrations will most likely only be important in the immediately surrounding areas, while for large cavities, the vibrations will most likely greatly affect the overall vibration environment. Additionally, installed material can potentially produce intense vibration, depending on the mounting and the material's characteristics, and therefore must be evaluated individually. For vibration testing, the exposure level can usually be found in the responsible program office for the particular aircraft. If appropriate criteria are not available, actual measurements of the vibration environment are recommended. As a last resort, Figure 8 and Table 6 can be utilized to determine the level, which should be an envelope of both the jet and aerodynamic noise induced. For material weighing more than roughly 72 kilograms, or 160 pounds, the installation should be evaluated to account for dynamic interaction. Again, the duration of the test should be taken from the Life Cycle Environment Profile.

Category 13 focuses on material installed in propeller aircraft. The vibration environment is mainly caused by the propellers. The frequency spectrum is made up of a broadband background with superimposed narrow band spikes, due to various random sources along with lower level periodic components from the rotating elements. For propeller aircraft which maintains constant speed, the fixed frequency produced is illustrated by Figure 9. For propeller aircraft which vary speed, a set of spectra that has bandwidths encompassing the varying propeller speed must be used to define vibration levels. If possible, measurements should be taken during flight to determine exposure levels. If not, a combination of Table 7 and Figure 9, which are based on C-130 and P-3 aircraft measurements, can be used to determine levels. The durations should again be taken from the Life Cycle Environment Profile.

Category 14 references helicopter vibration, which consists of dominant peaks superimposed on a broadband background, made up of a mixture of random vibrations and lower amplitude sinusoids. The peaks, occurring at the frequency of each component and harmonics of these frequencies, are sinusoids due to the rotating components. The exposure levels should be obtained from actual measurements because vibration levels vary greatly between helicopter types. If this is possible, the duration should be taken from the Life Cycle Environment Profile. If this is not possible, levels can be obtained from Figures 10 and 11, and Table 8, which envelope worst-case environments. For the duration, it should be tested for 4 hours in each of the three axes, assuming a 2600 hour operational life. If the duration needs to be modified, e.q. 2 should be utilized to determine the new duration.

$$t_f = 4.0\ (A_D / A_T)^M \qquad [2]$$

In the above, t_f is the actual test time per axis,

A_D is the default test amplitude,

A_T is the actual test amplitude, and

M is 6 (Material exponent for sinusoidal vibration)

Category 15 encompasses assembled jet aircraft stores, which can experience three different vibration environments: external captive carriage, internal captive carriage, and free flight. External captive carriage vibration is usually caused from any combination of the following: engine noise due to turbulence, aerodynamic turbulence disseminated over the surface, vibration of the aircraft transmitted through the attaching structure, or vibration produced by internal material and local aerodynamic effects. Regarding internal captive carriage vibration, there are two types of environments: when the aircraft bay is open to the environment, and when the bay is closed. For free flight vibration, the vibration is due to engine exhaust noise, vibration and noise made by internal material, and boundary layer turbulence. For the exposure levels of the three vibration environments, the levels should be obtained from flight measurements. If it is not possible, Figures 12 and 13, and Table 9 should be used to determine test levels and spectra. The duration should be taken from the Life Cycle Environment Profile.

Category 16 includes material installed in a jet aircraft store, which will undergo the same store vibration discussed in category 15. If possible, in-flight measurements should be used to determine exposure levels. If not, Table 9 and Figures 13 and 14 should be used. The test duration should be taken from the Life Cycle Environment Profile.

Category 17 revolves around propeller aircraft stores. While there is no current data for the measurement of the vibration environment, it is most likely that the store will encounter vibration similar to the vibration of the carrying aircraft, discussed in category 13. As such, measurement of the actual vibration environment is vital. For initial estimates of the vibration environment, Table 7 and Figure 9 should be utilized, while Figure 13 should be used for maneuver buffet vibration. The duration level should be taken from the Life Cycle Environment Profile.

Category 18 discusses assembled stores carried externally on helicopters. The stores undergo complex periodic waveforms due to vibratory energy travelling through the attachment points and excitation due to periodic pressure changes caused by the rotor. The best way to determine exposure levels are through in-flight measurements, in which case the duration should be taken from the Life Cycle Environment Profile. If that is not possible, preliminary estimates can be obtained from Figures 10 and 11, and Table 8. In this case, the duration should be 4 hours per axis, testing in all three axes, for a 2500 hour operational life. If this needs to be modified, equation 2 should be used to determine the new duration.

Category 19 focuses on tactical missile carriage launch vibration environments. Currently, there is no source covering the vibration environment, so measurement of the actual environment is crucial. Preliminary estimates of the free flight vibration environment can be found in Table 9 and Figures 12 and 14. The duration should be taken from the Life Cycle Environment Profile.

Category 20 deals with ground vehicles. The vibration environment encountered is affected by variables such as the terrain discontinuities, speed, and suspension system, leading to a broadband random vibration with considerable peaks and notches. For wheeled vehicles, although there is no generalized model, Table 5 can be used in most circumstances to determine the spectra. For a track-laying vehicle environment, depicted in Figure 4, a narrowband random vibration should be superimposed at certain frequencies over a broadband random base. If possible, the exposure levels should be determined from actual measurements. If not, the guidelines provided in categories 4 and 5 should be followed. The duration of the test should be taken from the Life Cycle Environment Profile.

Category 21 covers marine vibration. The spectra for marine vibration have a random element, caused by the varying of speeds and sea states. There is also a periodic element, as a result of both resonance in the hull and the rotation of the propeller shaft. Comprised of the natural environment forcing function caused by the waves and wind, the created forcing function due to the operation of the propellers and various parts, the structure of the ship/watercraft, the way the material is mounted to the structure, and the material response, ship and watercraft vibrations are very complicated. As such, measurements of the specific environment must be used to determine exposure conditions, and the duration should be taken from the Life Cycle Environment Profile. For a rough idea of onboard life exposure on a ship, Figure 15 should be used, along with the sinusoidal requirements from MIL-STD-167, Type I, given that the levels envelope the maximum value for each frequency. The duration should be two hours per axis, testing in all three axes. For watercraft other than a ship, there are no current guidelines available.

Category 22 is comprised of parts mounted directly on a turbine engine. The vibration spectra are made up of a broadband background with superimposed fine band spikes, which are caused by the rotation of the rotor. The background is the total of the random flow turbulence and low-level partly-sinusoidal peaks created by rotating machines. Measured values should be taken from the actual environment to determine exposure levels, and the duration should be taken from the Life Cycle Environment Profile. However, measured data is not always available, and there is rough data that can be utilized. For turbines maintaining a constant speed, Figure 16 illustrates the fixed frequency spikes. For turbines with varying speeds, Figure 16 should still be used, but the spikes should be adjusted to incorporate the engine rpm range. If the engine has multiple rotors running at different speeds, Figure 16 should be modified to contain the spikes for each rotor. The duration should still be taken from the Life Cycle Environment Profile.

Category 23 touches on material carried by or on personnel. Any vibration intense enough to affect material would be unendurable the human body. As such, there are no vibration tests required if the material is carried on the human body.

Category 24 references minimum integrity tests for all material. Material that already has been completely environmentally tested, or has already undergone testing equivalent in level and duration, does not need to be tested again. Additionally, if the material is considered to be delicate, and may not be able to withstand the levels, the complete environment life cycle should be assessed to make certain the material is shielded from vibration and shock during all phases. The exposure levels and durations should be obtained from Figure 17 for general use or from Figure 18 for helicopter material. For material under 36 kilograms, or 80 pounds, the vibration should be applied directly to the hard-mounted material. For other material, the vibration should be applied to the subassembly to prevent unnecessarily high loads.

Finally, category 25 deals with material that includes cantilever elements attached externally, and the special problems these material encounter that are uncommon but will most likely result in quick failure. When cantilever elements are submerged in a fluid flow, the elements can vibrate because of self excited vibration and forced response to pressure changes, usually caused by one of three mechanisms. The first such mechanism is referred to as flutter, which is when the vibration of a cantilever beam with a lean cross section in a flow produces lift forces and moments that strengthen and magnify the vibration. Flutter is not caused by an environmental forcing function, but is intrinsic to a design, and should be dealt with by flutter engineers. While any wing can flutter, most fixed wing aircraft and external stores do not because the first bending mode frequency is separated from the first torsion mode frequency; any wing with close bending and torsion mode frequencies should be examined by flutter engineers. Most helicopter wings also do not experience flutter because of the comparatively low air speeds. With regards to ground vehicles, only the waving of a cloth cover of an open truck constitutes flutter, and that can be solved by tying it down tightly. However, flutter can cause a serious problem for underwater portions of a watercraft.

Flutter will lead to either immediate failure or intense fatigue and wear failure. Ideally, the design should be made to prevent flutter. However, if flutter does occur, it must eliminated by changing the modal mass, stiffness, damping, or aerodynamic shape. At times, multiple changes will have to be made. Although the problems should be corrected by eradicating flutter, if it becomes necessary to determine exposure durations, the Life Cycle Environment Profile should be used.

Another mechanism occurs when air flows over a blunt cross section, a cross section whose depth and height are roughly equivalent, and vortices are shed from one side then the other, creating a fluctuating force. The vortices run along the length of the cantilever and spread downstream as singular elements, dispersing quickly. If the excitation frequency created by the oscillating force is roughly the cantilever resonant frequency, it will lead to vibration. This mechanism is also self-excited as opposed to environmental, but the vibration engineer should rectify the problem. E.q.s 3 and 4 demonstrate how to calculate shed frequency as well as force generated.

$$f = 0.22\ V / D \quad [3]$$

$$F = (0.5\rho V^2 DL)\sin(2\pi f t) \quad [4]$$

In the above equations, f is the frequency,
V is the velocity,
D is the cantilever cross section diameter,
F is the force,
ρ is the density,
t is the time, and
L is the exposed length (perpendicular to cross section)

This mechanism is not usually a problem in fixed wing aircraft, simply because most aircrafts do not have external cantilevers with a blunt cross section. Helicopters and ground vehicles are more likely to use blunt cross section cantilevers because of the low speeds. If such cantilevers are used, the shed frequencies and cantilever frequencies must not be closely spaced. Watercraft wings often utilize blunt cantilevers that can be subject to vortex shedding, which can lead to serious problems.

Vortex shedding often leads to rapid fatigue and/or wear failure. The ideal solution is to separate the shed frequency and the resonant frequency by at least a factor of 2. If this is not possible, a solid design, that includes materials with strong fatigue properties, no high stress points, and damping, could survive long enough to be useful. Measurements of the cantilever movement in its operating environment should be taken to define modal responses and exposure levels. If tests are needed, response control is imperative, and the duration should be taken from the Life Cycle Environment Profile.

The third primary mechanism occurs when fluctuations in a fluid lead to forced vibration of the cantilever, and this forced vibration is the same response to aerodynamic turbulence. There are two pertinent types of excitation that leads to the forced vibration. The first is broadband random turbulence behind any reasonably blunt material blocking flow. The second is vortices, created when pressure is different on two sides of a wing and resulting in a rotating flow from the high pressure side to the low pressure side.

This mechanism can be very damaging to fixed wing aircraft, subject to turbulence forced vibration. Harsh broadband flow turbulence can continue downstream behind the blunt material blocking flow for three to five times the highest cross sectional dimension of the material. Helicopters may also experience turbulence, though at lower speeds it is less likely to cause problems. It is still recommended not to place any cantilevered material near known turbulence. For ground vehicles, forced vibration problems are exceedingly unlikely, but turbulence can affect large trucks, which will affect the handling of smaller vehicles around the truck. With regards to watercraft, the hulls and underwater material are usually designed to create even flow along the sides, so turbulence should not be a problem. However, there are often squared off sterns, creating high levels of turbulence downstream.

Turbulence is not predictable and should be avoided as much as possible by keeping material far from known turbulence areas. If a problem should occur, the part causing turbulence should be discovered and moved. If this is not feasible, the procedure governing vortex shedding problems should be implemented.

Test Procedures[7]

Test Configuration: Configure the test item for each test, as it will be in the corresponding life cycle phase. In cases representing transportation, include all packing, shoring, padding, or other configuration modifications of the particular shipment mode. The transportation configuration may be different for different modes of transportation.

[7] The information in the following section, pages 32-41, is taken verbatim from the Military Standard 810F, with the exception of the changing of pointers, to refer to sections within this paper as opposed to the Standard.

a. Loose cargo. The method contained herein is a general representation based on experience as well as measurement, and is not tailorable. The most realistic alternative for truck, trailer, or other ground transportation is to utilize Procedure III. Note that Procedure III requires the transportation vehicle and a full cargo load.

b. Restrained cargo. Procedure I assumes no relative motion between the vehicle cargo deck or cargo compartment and the cargo. This applies directly to material that is tied down or otherwise restrained such that no relative motion is allowed considering vibration, shock, and acceleration loads. When restraints are not used or are such as to allow limited relative motions, provide allowance in the test set up and in the vibration excitation system to account for this motion. Procedure III is an alternative for ground transportation.

c. Stacked cargo. Stacking or bundling of sets or groups of material items may effect the vibration transmitted to individual items. Ensure the test item configuration includes appropriate numbers and groupings of material items.

Before starting test, review pretest information in the test plan to determine test details (procedure(s), test item configuration(s), levels, durations, vibration exciter control strategy, failure criteria, item functional requirements, instrumentation requirements, facility capability, fixture(s), etc.).

a. Select appropriate vibration exciters and fixtures.

b. Select appropriate data acquisition system (e.g., instrumentation, cables, signal conditioning, recording, and analysis equipment).

c. Operate vibration equipment without the test item installed to confirm proper operation.

d. Ensure that the data acquisition system functions as required.

All items require a pretest standard ambient checkout to provide baseline data. Conduct the pretest checkout as follows:

Step 1. Examine the test item for physical defects, etc. and document the results.

Step 2. Prepare the test item for test, in its operating configuration if required, as specified in the test plan.

Step 3. Examine the test item/fixture/exciter combination for compliance with test item and test plan requirements.

Step 4. If applicable, conduct an operational checkout in accordance with the test plan and document the results for comparison with data taken during or after the test.

Procedure I[8] governs general vibration tests, and should be implemented as follows:

Step 1. Perform a visual inspection of the test item and an operational check. If a failure is noted, proceed as in the following failure section.

Step 2. Conduct fixture modal survey and verify that fixture meets requirements, if required.

Step 3. Mount the test item to the test fixture in a manner dynamically representative of the life cycle event simulated.

Step 4. Install sufficient transducers on or near the test item/fixture/vibration exciter combination to measure vibration at the test item/fixture interface, to control the vibration exciter as required by the control strategy, and measure any other required parameters. Mount control transducer(s) as close as possible to the test item/fixture interface. Ensure that the total accuracy of the instrumentation

[8] Use Procedure I for those cases where a test item is secured to a vibration exciter and vibration is applied to the test item at the fixture/test item interface. Steady state or transient vibration may be applied as appropriate.

system is sufficient to verify that vibration levels are within specified tolerance and to meet additionally specified accuracy requirements.

Step 5. Conduct test item modal survey, if required.

Step 6. Perform a visual inspection of the test item and, if applicable, an operational check. If failure is noted, proceed as in the following failure section.

Step 7. Apply low level vibration to the test item/fixture interface. If required, include other environmental stresses.

Step 8. Verify that the vibration exciter, fixture, and instrumentation system functions as required.

Step 9. Apply the required vibration levels to the test item/fixture interface, as well as any other required environmental stresses.

Step 10. Verify that vibration levels at test item/fixture interface are as specified. If the exposure duration is 1/2 hour or less accomplish this step immediately after full levels are first applied, and immediately before scheduled shut down. Otherwise, accomplish this step immediately after full levels are first applied, every half-hour thereafter, and immediately before scheduled shut down.

Step 11. Monitor vibration levels and, if applicable, test item performance continuously through the exposure. If levels shift or a failure occurs, shut down the test in accordance with the test shut down procedure. Determine the reason for the shift and proceed in accordance with the test interruption recovery procedure.

Step 12. When the required duration has been achieved, stop the vibration. Depending on the test objectives, the test plan may call for additional exposures at varied levels prior to shut down. If so, repeat steps 6 through 12 as required by the test plan before proceeding.

Step 13. Inspect the test item, fixture, vibration exciter, and instrumentation. If failure, wear, looseness, or other anomalies are found, proceed in accordance with the test interruption recovery procedure.

Step 14. Verify that the instrumentation functions as required and perform an operational check of the test item. If a failure is noted, proceed as in the following failure section.

Step 15. Repeat steps 1 through 14 for each required excitation axis.

Step 16. Repeat steps 1 through 15 for each required vibration exposure.

Step 17. Remove the test item from the fixture and Inspect the test item, mounting hardware, packaging, etc. Refer to the following failure section if there are failures.

Procedure II[9] deals with loose cargo transportation, and should be conducted as follows:

Step 1. Perform a visual inspection of the test item and an operational check.

Step 2. Conduct test item modal survey, if required.

Step 3. Place the test item(s) on the package tester within the restraining fences in accordance with the test facility and setup requirements.

Step 4. Install instrumentation sufficient to measure any required parameters. Ensure that the total accuracy of the instrumentation system is sufficient to meet specified accuracy requirements.

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[9] Use this procedure for material to be carried in/on trucks, trailers, or tracked vehicles and not secured to (tied down in) the carrying vehicle. The test severity is not tailorable and represents loose cargo transport in military vehicles traversing rough terrain

Step 5. Operate the package tester for one-half of the prescribed duration.

Step 6. Perform a visual inspection of the test item and an operational check. If a failure is noted, proceed as in the following failure section.

Step 7. Reorient the test item(s) and/or the fencing/impact walls in accordance with the setup requirements.

Step 8. Operate the package tester for one-half of the prescribed duration.

Step 9. Perform a visual inspection of the test item and an operational check. If a failure is noted, proceed as in the following failure section.

Procedure III[10], regarding large assembly transportation, should be carried out as follows:

Step 1. Perform a visual inspection of the test item and an operational check. If a failure is noted, proceed as in the following failure section.

Step 2. Mount the test item(s) on/in the test vehicle as required in the test plan.

Step 3. Install transducers on or near the test item sufficient to measure vibration at the test item/vehicle interface and to measure any other required parameters. Protect transducers to prevent contact with surfaces other than the mounting surface.

Step 4. Subject the vehicle containing the test item to the specified test conditions.

Step 5. Perform a visual inspection of the test item and an operational check. If a failure is noted, proceed as in the following failure section.

Step 6. Repeat steps 1 through 5 for additional test runs, test loads, or test vehicles as required by the test plan.

[10] This procedure is intended to replicate the vibration and shock environment incurred by large assemblies of material installed or transported by wheeled or tracked vehicles. Use the specified vehicle type to provide the mechanical excitation to the test material. Generally, measured vibration data are not used to define this test.

Procedure IV[11] regulates assembled aircraft store captive carriage and free flight, and should be executed as follows:

Step 1. With the store suspended within the test chamber and the instrumentation functional, verify that the store suspension system functions as required by measuring the suspension frequencies.

Step 2. If required, conduct a test item modal survey.

Step 3. Place the test item in an operational mode and verify that it functions properly.

Step 4. Apply low level vibration to the vibration exciter/store interface(s) to ensure that the vibration exciter and instrumentation system function properly. For acceleration feedback control, use an initial input level 9 dB down from the required forward test monitor transducer spectrum. For force feedback control, use a flat force spectrum where the response at the test monitor accelerometer is at least 9 dB below the required test monitor value at all frequencies. For bending moment feedback control, use an initial input level that is 9 dB down from the required test monitor transducer spectrum.

Step 5. Adjust the vibration exciter(s) such that the test monitor transducers in the excitation axis meet the test requirements. For acceleration control, identify the test monitor transducer spectrum peaks that exceed the input spectrum by 6 dB or more (frequencies may differ fore and aft). For force feedback control, identify major peaks from the force measurements to check monitor accelerometer transfer functions. For both cases, equalize the input spectra until

[11] Apply Procedure IV to fixed wing aircraft carriage and free flight portions of the environmental life cycles of all aircraft stores, and to the free flight phases of ground or sea launched missiles. Steady state or transient vibration may be applied as appropriate.

the identified peaks equal or exceed the required test levels. The resulting input spectra should be as smooth and continuous as possible while achieving the required peak responses. (It is not necessary to fill in valleys in the test monitor transducer spectra; however, it is not acceptable to notch out the input in these valleys.) For bending moment control raise and shape the input spectrum until it matches the required spectrum (peaks and valleys).

Step 6. When the input vibration is adjusted such that the required input response (A1) is achieved, measure the off-axis response(s) (A2, A3). Verify that off-axis response levels are within requirements using the following equations. If the result obtained from the equation is greater than the value established for the equation, reduce the input vibration level until the achieved input and off-axis response levels balance the equation. Apply these equations at each peak separately. Use the first equation for testing that requires vibration application in two separate mutually perpendicular axes, and use the second equation for testing that requires vibration application in three separate mutually perpendicular axes.

$$2 = (R_1/A_1 + R_2/A_2) \text{ or, } 3 = (R_1/A_1 + R_2/A_2 + R_3/A_3)$$

In the above equation, R_i is the test requirement level in
g2/Hz or (N-m)2/Hz or (in-lb)2/Hz for i = 1 - 3, and
A_i is the response level in
g2/Hz or (N-m)2/Hz or (in-lb)2/Hz for i = 1 – 3

For example, for testing that requires vibration application in three separate mutually perpendicular axes, and when vibration is being applied in the vertical axis, use the equation below as follows.

$$3 = (R_1/A_1 + R_2/A_2 + R_3/A_3)$$

In the above equation, R_1 is the vertical axis test requirement level,
A_1 is the vertical axis response level,
R_2 is the horizontal axis test requirement level,
A_2 is the horizontal axis response level,
R_3 is the longitudinal axis test requirement level, and
A_3 is the longitudinal axis response level

For vibration being applied in the horizontal axis, use the equation below as follows.

$$3 = (R_1/A_1 + R_2/A_2 + R_3/A_3)$$

In the above equation, R_1 is the horizontal axis test requirement level,
A_1 is the horizontal axis response level,
R_2 is the vertical axis test requirement level,
A_2 is the vertical axis response level,
R_3 is the longitudinal axis test requirement level, and
A_3 is the longitudinal axis response level

For vibration being applied in the longitudinal axis, use the equation below as follows.

$$3 = (R_1/A_1 + R_2/A_2 + R_3/A_3)$$

In the above equation, R_1 is the longitudinal axis test requirement level,
A_1 is the longitudinal axis response level,
R_2 is the vertical axis test requirement level,
A_2 is the vertical axis response level,
R_3 is the horizontal axis test requirement level, and
A_3 is the horizontal axis response level

Step 7. Verify that vibration levels are as specified. If the exposure duration is 1/2 hour or less, accomplish this step immediately after full levels are first applied, and immediately before scheduled shut down. Otherwise, accomplish this step immediately after full levels are first applied, every half-hour thereafter, and immediately before scheduled shut down.

Step 8. Monitor the vibration levels and test item performance continuously through the exposure. If levels shift, performance deviates beyond allowable limits, or failure occurs, shut down the test in accordance with the test shut down procedure. Determine the reason for the anomaly and proceed in accordance with the test interruption recovery procedure.

Step 9. When the required duration has been achieved, stop the vibration. Depending on the test objectives, the test plan may call for additional exposures at varied levels prior to shut down. If so, repeat steps 6 through 9 as required by the test plan before proceeding.

Step 10. Inspect the test item, fixture, vibration exciter, and instrumentation. If failure, wear, looseness or other anomalies are found, proceed in accordance with the test interruption recovery procedure.

Step 11. Verify that the instrumentation functions as required and perform an operational check of the test item for comparison with data collected in the pretest standard ambient checkout. If a failure is noted, proceed as in the following failure section.

Step 12. Repeat steps 1 through 11 for each required excitation axis.

Step 13. Repeat steps 1 through 12 for each required vibration exposure.

Step 14. Remove the test item from the fixture and inspect the test item and mounting hardware. Refer to the following section if there are failures.

If qualification tests are required, the following definitions should be used:

Failure: "Material is deemed to have failed if it suffers permanent deformation or fracture; if any fixed part or assembly loosens; if any moving or movable part of an assembly becomes free or sluggish in operation; if any movable part or control shifts in setting, position or adjustment, and if test item performance does not meet specification requirements while exposed to functional levels and following endurance tests." Ensure this statement is accompanied by references to appropriate specifications, drawings, and inspection methods.

Test completion: "A vibration qualification test is complete when all elements of the test item have successfully passed a complete test. When a failure occurs, stop the test, analyze the failure, and repair the test item. Continue the test until all fixes have been exposed to a complete test. Each individual element is considered qualified when it has successfully passed a complete test. Qualified elements that fail during extended tests are not considered failures and can be repaired to allow test completion."

Failure[12]

If there are failures:

For general testing, and in tolerance interruptions:

Interruption periods during which the prescribed test conditions remain in tolerance (e.g., power interruptions that do not affect chamber temperature) do not constitute a test interruption. Therefore, do not modify the test duration if exposure to proper test levels was maintained during the ancillary interruption.

For general testing and out of tolerance interruptions:

A logic diagram for these methods is on Figure 19. If undertested: if test condition tolerances fall below the minimum tolerance value (i.e., environmental stress less severe than specified) resulting in an undertest condition, the test may be resumed (after reestablishing prescribed conditions, except as noted in the individual methods) from

[12] The information in the following section, pages 42-43, is taken verbatim from the Military Standard 810F, with the exception of the changing of pointers, to refer to sections within this paper as opposed to the Standard.

the point at which the test condition fell below the lower tolerance level. Extend the test to achieve the prescribed test cycle duration.

If overtested: if an overtest condition occurs, the preferable course of action is to stop the test and start over with a new test item. But, as shown on figure 5-1, if there is no damage to the test item, continue the test, realizing that if the item fails the test from this point on or fails subsequent tests, you have a "NO TEST" unless it can be shown that the overtest condition had no effect on the test item. Overtest conditions can damage the test item and cause subsequent failures that may not have occurred otherwise, thus failing a test item because of an invalid test. However, if damage resulting directly from an overtest occurs to a test item component that has absolutely no impact on the data being collected, and it is known that such damage is the only damage caused by the overtest (e.g., rubber feet on bottom of a test item melted by high temperature where those feet have no impact on the performance of the test item), the test item can be repaired and the test resumed and extended as in the undertest condition. Coordinate with the customer before repairing and continuing to test an item after it has been overtested. This coordination is aimed at preventing customer objections if the test item fails during the remainder of the test program (claims that the test was invalid past the point of the overtest because the overtest caused undiscovered damage to a critical component).

Method specific:

(1) When interruptions are due to failure of the test item, analyze the failure to determine root cause. With this information, make a decision to restart, to replace, to

repair failed components and resume, or to declare the test complete. Tailor this decision to the test and the test objectives.

(2) If a qualification test is interrupted because of a failed component and the component is replaced, continuation of the test from the point of interruption will not verify the adequacy of the replaced component. Each replaced component must experience the full vibration requirement prior to its acceptance.

Chapter 3 - Design Properties of a Vibration Test Fixture

The vibration test fixture is a very important element of the vibration test setup. A vibration test fixture is required to allow the mounting of a specific test specimen onto a vibration table and is part of the vibration load path. A fixture should uniformly transmit all the forces produced by the vibration table to the test specimen. It is normally impossible to fix a test specimen directly to the shaker table itself, so the fixture acts as a transmission. The test fixture should simulate both the specimen and vibration equipment interfaces. In this chapter we will discuss important properties of a test fixture and several design principles to follow when still in the design phase.

3.1 Physical Design Properties

When designing a vibration test fixture from scratch, it is important to consider its various physical properties, such as material, size, weight, stiffness, and shape of the fixture. These physical properties should be carefully chosen as to not affect the results of the vibration test, since one small mistake could lead to the accidental destruction of a test specimen.

3.1.1 Material

As a general rule of thumb, the material used to fabricate a vibration test fixture should be as low in mass as possible without affecting the stiffness of the fixture because the material mass has a great influence on the shaker's thrust ability. For a mass M and linear acceleration a, the equation

$$F = M \times a$$

shows that for a constant force, the greater the mass of the fixture, the less acceleration it will provide for the vibrations. To prevent a fixture from inhibiting a shaker's full potential, the fixture is commonly sculptured to remove unwanted material and reduce the mass. The dynamics of the fixture remain unchanged, but the stiffness and strength are compromised and considered. The fixture sees high stress levels from the applied vibration or the specimen response and must be strong enough to transmit the forces and survive the tests.

Of the materials generally considered for fixture construction, steel, aluminum and magnesium are the most common. The ratio of Young's Modulus to density

$$E/\rho$$

is a controlling factor for the natural frequency of a fixture, and since this ratio is roughly the same for these three materials the choice of material will not greatly affect the range in which the natural frequency of the fixture falls between. It will, however, in the case of magnesium, affect the cost, and in the case of steel, the weight. When the shakers are operating at their full performance level, for example, the weight of the fixture dominates the selection of the material.

Table 5 compares the Young's modulus, density, and ratio of the three metals.

	Steel	Aluminum	Magnesium
Young's Modulus (E) N/m^2	$20{,}7 \times 10^{10}$	$6{,}9 \times 10^{10}$	$4{,}14 \times 10^{10}$
Density (ρ) kg/m^3	7840	2770	1800
E/ρ N m/kg	$2{,}65 \times 10^7$	$2{,}49 \times 10^7$	$2{,}3 \times 10^7$

Table 5 - Physical Properties of Metal

Composite materials are ideal for fixtures which test large and heavy specimens, but the fabrication process of these materials is highly difficult. Although magnesium is a lighter metal, the availability of indigenous fabrication techniques and other various fabrication issues have compelled engineers to choose an aluminum alloy as the fixture material for most of the applications. It should also be noted here that wooden fixtures have been used successfully at low frequencies and that a reasonable fixture can be made of laminated wood, which has a high damping coefficient.

3.1.2 Mass and Stiffness

The acceleration level possible with the vibration shaker table is inversely proportional to the total mass it has to drive. In order to get the most energy out of the vibration shaker table, a test fixture should carry the force from the mounting holes in the table to the mounting points on the test specimen with a minimum of loss and distortion. In other words, the fixture should be as rigid as possible. Therefore, an ideal fixture would have zero mass and infinite stiffness. However, this statement in itself is a paradox, which is the basis of fixture design principles.

A fixture that is designed to be as light and stiff as physically possible helps prevent unwanted cross axial motion, or “cross talk”. The stiffness prevents any deflection by the specimen load and also transfers motion with high fidelity. This quality is called transmissibility, which is a comparison of the output to the input. At a transmissibility of 1.0, the output faithfully follows input. Ideally, a dynamic test fixture couples the motion from the vibration shaker table to the specimen with zero distortion at all amplitudes and frequencies. This ideal is approached if the test frequency range is narrow or if the test specimen is small.

Although a fixturing problem may seem very complicated, many of the design parameters are actually determined before-hand. The mass of the fixture, for instance, is limited by the weight of the test object and the force available from the shaker table. Weight is a drawback as it infringes the capability of the vibrator to deliver enough acceleration to the specimen. Cost considerations influence whether several single-purpose fixtures or one general purpose fixture will be built. Weight and cost considerations almost always compromise fixture performance. After performance evaluation, the test fixture dynamic characteristics are improved by using structural dynamic modification method with transfer function application. The shape is determined in part by the fact that it has to fit the exciter table at its bottom surface and the test object at one of its other surfaces.

3.2 Resonance

When considering all the materials, size, and weight of the fixture, it is important to note that each property will affect the natural frequency of the fixture. The purpose of the fixture is to relay all the vibrations from the shaker directly to the test specimen without adding any additional forces or vibrations. So, the test fixture has to be designed to be as rigid as possible within the weight limits of the machine to not induce mechanical impedance to the specimen during testing. It is also important for the fixture to have energy conservation throughout the entire vibration test. A vibration test fixture shall be designed to transmit vibration energy from the machine table to the test item in a uniform manner over the mass.

The most important factor in fixture design is a high natural frequency above the frequency of interest, since the various test conditions can impose restrictions on the fixture. A typical test specification calls for testing to 2000 Hz with no fixture resonances below 1000 Hz and an allowable variation in vibration levels between 1 and 2 kHz of not more than 2 ; 1 between any pair of points on the table.

3.3 Mounting A Test Article To Vibration Test Equipment

When designing a vibration test fixture, it is important to consider how the fixture is connected to both the shaker table and the test specimen. An efficient test fixture will follow a set of guidelines that allows a vibration test to be run multiple times without failure.

The fixture should allow for the ease of mounting of the test item to the vibration machine, allow for vibration testing in each of the three orthogonal directions with minimum cross talk (motion in the two orthogonal directions not being tested), and ensure the absence of fixture resonances within specified test frequency range by tailoring the dynamic response of the fixture and the table.

Details of the shaker table, such as pattern of the attachment holes, bolt size and thread sizes, are known as what the fixture attaches to. The number and the size of these points must be sufficient to transmit energy of vibration equally to the fixture and the test specimen without any losses while being able to sustain the vibration shaker's force capacity. Details of the test specimen, such as the mounting interface, pitch circle diameter (PCD), and predicted dynamic characteristics of the test specimen are also considered when mounting the specimen onto the fixture.

Since a vibration test fixtures simulates the real mounting interface for both the specimen and the vibration shaker on either side, it is important to ensure that the details of the dynamic test specifications are used to simulate the dynamic environment. The test specification intensities must be known for two reasons:

1) The inertia force acting on the test item $F=W_t A$ must be with stood by the bolts or other fasteners connecting the test item to the shaker.

2) There is a limit to fixture weight that can be allowed without exceeding the force rating of the shaker, since mass of the moving table (W_a) and mass of the test specimen (W_t) are fixed.

3.3.1 Testing Within Orthogonal Directional Axes

A vibration test is performed in one of the three orthogonal directional axes, one at a time. The reason for testing these axes individually is to allow for unit control and a standardization of test specifications. This test procedure does not simulate real environmental problems, but a vibration test can also be coupled with an environmental test to duplicate a more realistic real-life scenario.

The isolation of each orthogonal directional axis allows the tester to observe the part's response in various situations. It also allows the testers to focus on the specific physical properties of the part. If possible, avoid cross talk during testing by maintaining the center of gravity of test specimen and fixture at or close to vibration machine table. The C.G. should be precisely calculated and each fixture should be rigidly coupled to the armature or slip table. If the C. G of the payload is kept low and aligns with the armature centre, then cross axial stress is minimized

3.4 Accelerometer Placement

Deciding the placement of the single-axis accelerometers is one of the most critical decisions when designing a procedure for random vibration tests. The various locations of an accelerometer can greatly affect the range of data received during a vibration test, so it is important to know which locations are the most effective for the part and which are the least.

The placement of accelerometers on the test specimen is crucial to reading the results of a vibration test. As mentioned in the topics before, the test fixture is designed to uniformly distribute all of the vibration force upon the test specimen. Therefore, the accelerometers must be efficiently placed on various locations of the test specimen and the shaker table itself to ensure that the output of the machine matches with the force on the test specimen.

When performing a vibration test, it is recommended to attach at least three accelerometers on the testing assembly. Any number of accelerometers can be placed on the part, as long as the three crucial spots are covered. Traditionally, the control accelerometers have been located three areas: on the shaker table, on the fixture, or on the test specimen. When placing accelerometers on the test specimen, it is wise to consider the CG of the test specimen and an arbitrary location near the edge of the part to ensure that there is no resonance on the outer edges of the part. These locations are crucial because it is easy to monitor any loss of force between the fixture and the part when comparing the accelerometer data.

When recording the data provided by the accelerometer, it is also important to note which accelerometers may possibly experience over-testing or under-testing based on the conditions of the test. For example, it is unwise to always select and record the largest accelerometer signal because some discrepancies can mask the true accelerometer signals. Make sure all the signals and data are analyzed.

3.5 Test Fixture Design Summary

For a successful vibration test fixture design, the following guidelines need to be considered:

- Optimum fixture would have its lower natural frequency about 50% higher than the highest required forcing frequency (avoid resonances, which can magnify acceleration forces)
- Understand test specimen and test specification
- Analyze preliminary designs and try to keep the lowest natural frequency about 50% higher than the highest forcing frequency
- Avoid sharp changes in the cross section
- Consider the stiffness-to-mass ratio for optimum design
- Keep fixtures as small as possible
- Avoid bolted fixture assemblies, except where ribs may be required for stiffness and damping
- Keep fixture designs simple
- Design symmetrical fixtures
- Design for dynamic similarity
- Consider the effective length of the bolt thread engagement when calculating the effective bolt spring rate
- Torque all bolts
- Use fine threads, instead of coarse threads, on bolts wherever possible
- Balance fixtures with test specimen

Chapter 4 - Validation of Design Principles

In order to ensure the design principles stated in Chapter 3 are accurate and all encompassing, analysis was conducted on a provided vibration test fixture. The CAD model was inherited and finite element analysis (FEA) was conducted to determine how the fixture would perform during vibration testing. Then, a design change was made, and further analysis was conducted to determine the effectiveness of the design change. This chapter presents the steps taken to prepare the CAD model for FEA, a description the results of the analysis, an explanation of the design iterations and the reasoning behind it, and finally a description of the resulting analysis of the model with the design change.

4.1 Model Description and Preparation

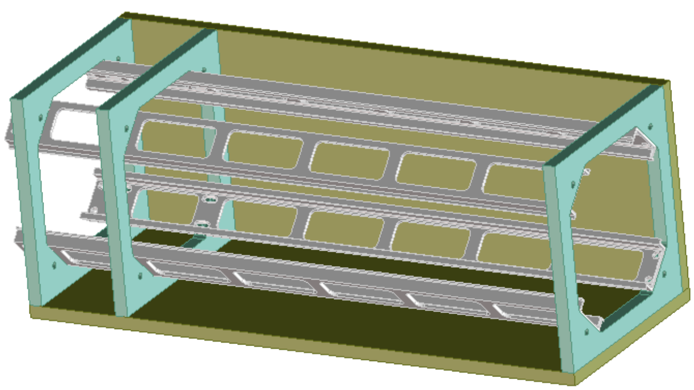
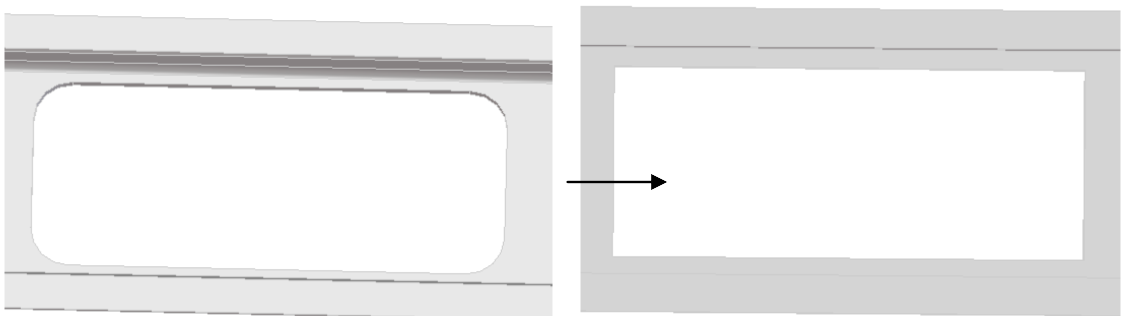


Figure 20 - Screenshot of Original CAD Model

The CAD model used in the design iteration shown in Figure 20 was inherited. The model represents a vibration test fixture, with one panel for support on the bottom and one panel for support on the side. There are three support blocks: one in the back, one close to the end of the back for extra support, and one in the front. Finally, there are four rails in each corner of the support blocks, connecting them.

The first step in the FEA was to de-feature the model, which means removing the characteristics unnecessary to analysis, such as bolt holes and chamfers. This step is important because, while the holes and chamfers are necessary to model, they are immaterial to the analysis, and could complicate the analysis. If the model is not de-featured before FEA is conducted, the analysis will take much longer to run, and the results can potentially be misleading, by creating stresses or forces around bolt holes or chamfers when, in reality, there may be none.



Screenshot of Original Cross Section of Rail Figure 21 Screenshot of De-Featured Cross Section of Rail

For this model, the bolt holes on the bottom and side support panels were removed, as well as the eight bolt holes in the two support blocks in the back. The chamfers on the rails, pictured above, were removed also. The bolt holes on the front support block were kept, because the part being tested was to be connected through those holes. Additionally, the bolt holes where the rails were to be connected to the support blocks were kept, as a reference to where the connections needed to be made in the FEA software. The finalized model, de-featured and ready to import into the FEA software, is shown in Figure . The last step in the CAD software was to save the

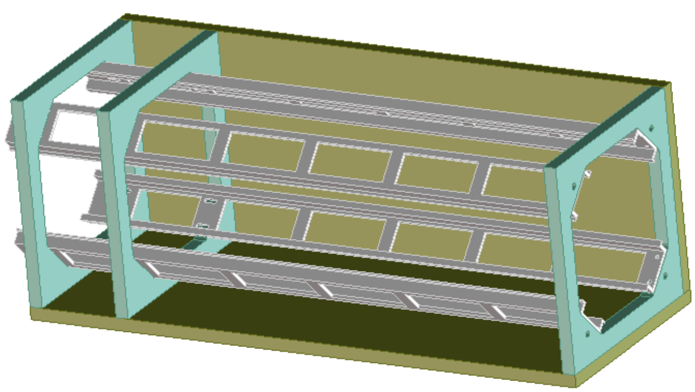


Figure 22 - Screenshot of De-Featured CAD Model

model as a .STEP file as opposed to the default .pkg, because the .STEP file can be translated into the FEA software.

The .STEP file should be brought into the FEA software, in this case Patran, modeled as solids. For simplicity later, the first step was to separate the part into groups: the two panels made up the Panel Group, the three support blocks made up the Support Group, and the four rails made up the Rail Group. Next, midsurfaces were created in all of the nine solids. The material assumed for all of the parts is Aluminum, and the material was created, with the properties inputted as 10^7 psi for the Elastic Modulus, 0.33 for Poisson's Ratio, and 0.000252 lb/in^3 for the density.

2D shells were then created for each of the midsurfaces by group, with the Aluminum properties imported. A coordinate system was created at the origin, then tilted 45 degrees from the global coordinate system. The Rail Group was then isolated, and a paver mesh with quad4 elements and a global edge length of 0.15 was added to the rails, with the node analysis coordinate frame referencing the new coordinate system as opposed to the global. Then the Panel and Support Groups were isolated, and a paver mesh with quad4 elements and a global edge length of 0.25 were applied to both groups. The node analysis coordinate frame for both the Panel and Support Groups was the global coordinate system. After the mesh was added, the nodes were equivalenced, to remove any duplicate nodes.

Once the mesh was created, Multi-Point Constraints (MPCs) were created. The first step was to create a node at the center point of each of the bolt holes. Rigid Body Element 2's (RBE2s) were created first around the four bolt holes on the front support block to keep the holes rigid, and then around the twenty-four bolt holes connecting the rails to the support blocks to keep the holes rigid and to ensure the rails stayed connected to the blocks. Six RBE2s were also added to each support block, three per side the panels were on, to ensure connection between the support blocks and the panels. A screenshot of the meshed model with all 46 RBE2s is shown in Figure 23 on the next page.

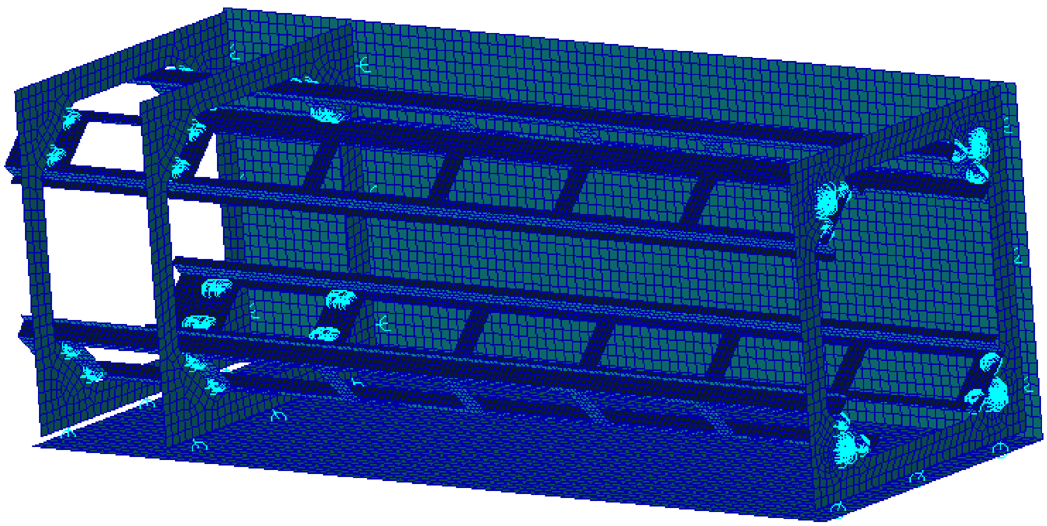


Figure 23 - Meshed Model with all 46 RBE2s

The next step was to add the boundary conditions. The vibration table the fixture was to be attached to had bolt holes every two inches. This pattern was applied to the bottom of vibration fixture, for a total of 44 nodes, as shown in Figure 24. All of the chosen nodes were fixed with [0 0 0] for both rotational and translational movement. At this point, both modal analysis and random vibration analysis was conducted on the fixture alone.

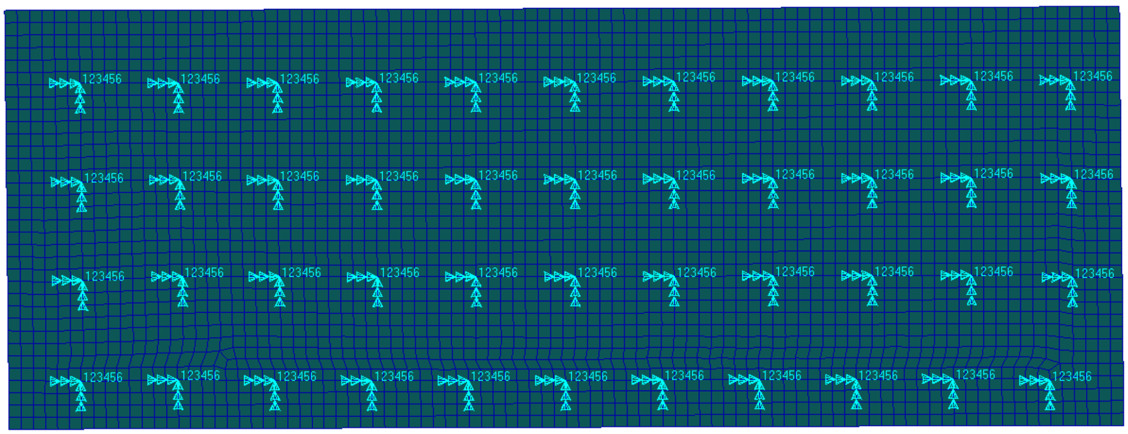


Figure 24 - Bottom Panel – Illustrating Pattern
of Boundary Conditions

Once analysis was conducted on the fixture alone, a lumped mass was added to represent the part inside the fixture. The mass was located 13.9 inches in from the center of the front block, and a node was created. A coordinate system was created, with that node as the origin, and a point element was created from the node. A 0D lumped mass element was created at the point element location, with the input properties referencing the new coordinate system and given values for the mass and moments of inertia properties. A RBE3 was then created, with the dependent node being the node where the mass was located, and the independent nodes being the center nodes of the bolt holes on the front support block. A screenshot of the transition from the initial model meshed model with all the RBE2s and RBE3s and the boundary condition is depicted below.

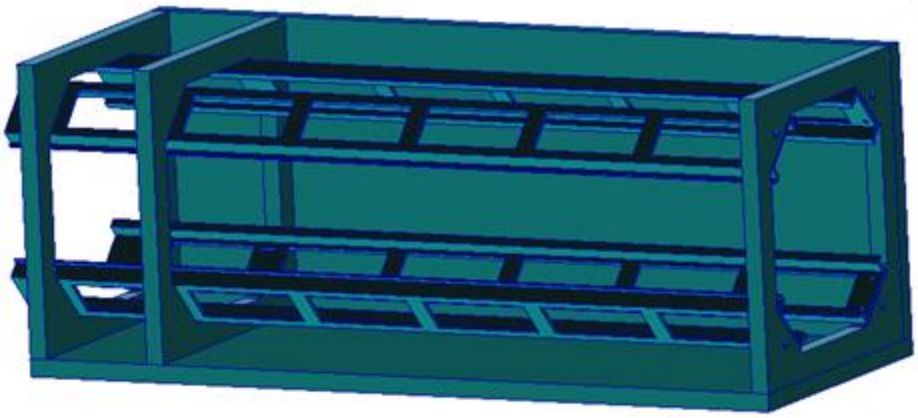
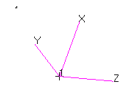


Figure 25 - Initial Model as Imported from CAD Software

Global Coordinate Frame

Coordinate Frame for Rail Analysis



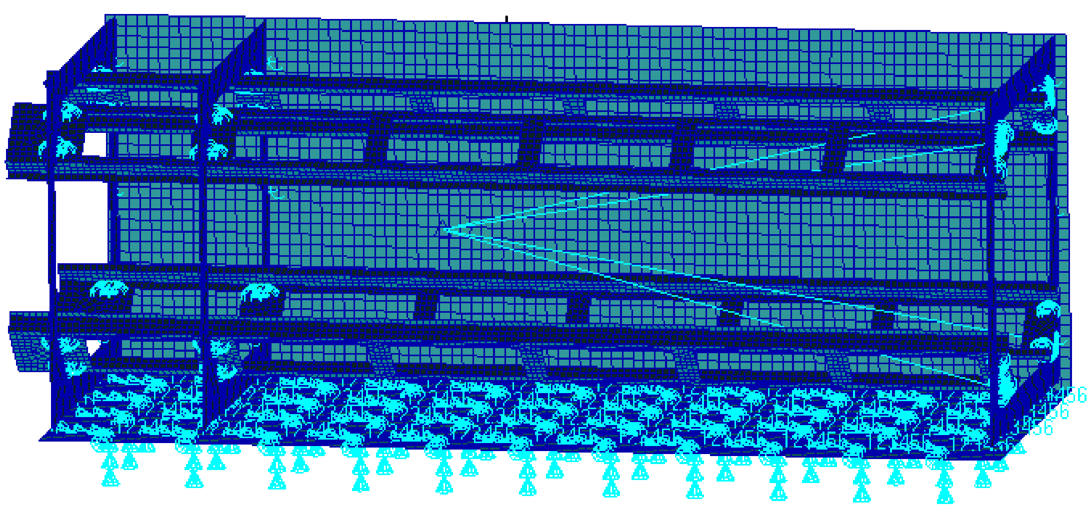


Figure 26 - Meshed Model with Mass, RBE2s, RBE3s, and Boundary Condition

At this point, the model was ready for both modal analysis and random vibration analysis. For the random vibration parameters, the damping was assumed to be constant at 0.02. The PSD curve was given, and is provided in Appendix B. The outputs desired were stress in each direction, and displacement in each direction.

4.2 Analysis

As previously stated, both modal analysis to determine the natural frequencies and random vibration analysis were conducted on the model. The first analysis was on the fixture by itself, while the second run included the fixture and the lumped mass representing the part. Screenshots of all the analysis results are illustrated in Appendix B.

Initially, all of the natural frequencies of the fixture fell between 200 and 300 Hertz, which was within the testing range. With the added lumped mass, however, the natural frequencies decreased, though the majority of modes still remained within the testing range. The seven natural frequencies above 200 Hertz affected only the rails, vibrating either a single rail or a combination of rails. The displacements and the stresses obviously increased with the added mass, which increased the forces.

For this specific fixture, the main concern was the vibratory displacement of the rails, and whether or not the rails would make contact with the part to be tested. There was 0.035 inches of clearance space between the rails and the part to be tested, and so special attention was paid to the rails to determine if they would collide with the part.

The rails vibrating caused the majority of the maximum displacements. For displacements in the Z direction, the maximums occurred in the top corner of the support blocks where there were no panels directly connected, while the maximum displacements in the X and Y direction occurred in the rails. Tables 10 and 11 show the normal modes as well as displacement and stress due to vibration for the fixture alone and the fixture plus the lumped mass.

Table 6 - Analysis Results of Fixture

Modes (Hz)		Random Vibration Displacement (in)				3-Sigma Values			
			Displacement Direction				Displacement Direction		
			x	*y*	*z*		*x*	*y*	*z*
1	211.7	*x*	0.0252	0.0225	0.00516	*x*	0.0756	0.0675	0.01548
2	214.75	*Location*	Rail	Rail	Rail/Corners				
3	216.51	*y*	0.0106	0.0121	0.00169	*y*	0.0318	0.0363	0.00507
4	217.91	*Location*	Rail	Rail	Rail/Corners				
5	239.41	*z*	0.00646	0.00578	0.0107	*z*	0.01938	0.01734	0.0321
6	257.02	*Location*	Rail	Rail	Rail/Corners				
7	262.15	**Von Mises Stress (psi)**				**3-Sigma Values**			
8	265.1		*x*	11100			*x*	33300	
9	272.47		*y*	4040			*y*	12120	
10	287.45		*z*	5150			*z*	15450	

Table 7 - Analysis Results of Fixture + Mass

Modes (Hz)		Random Vibration Displacement (in)				3-Sigma Values			
			Displacement Direction				Displacement Direction		
			x	*y*	*z*		*x*	*y*	*z*
1	7.498	*x*	0.0654	0.056	0.016	*x*	0.1962	0.168	0.048
2	11.2	*Location*	Rail	Rail	Rail/Corners				
3	37.246	*y*	0.0166	0.0176	0.00323	*y*	0.0498	0.0528	0.00969
4	216.17	*Location*	Rail	Rail	Rail/Corners				
5	217.51	*z*	0.0379	0.028	0.0263	*z*	0.1137	0.084	0.0789
6	218.18	*Location*	Rail	Rail	Rail/Corners				
7	218.64	**Von Mises Stress (psi)**				**3-Sigma Values**			
8	246.31		*x*	13700			*x*	41100	
9	262.08		*y*	12200			*y*	36600	
10	264.17		*z*	13700			*z*	41100	

4.3 Design Iteration

The analysis results proved the fixture did have natural frequencies in the testing range, and was not stiff enough. A design change was made, in which a third panel was added to the side. In order to do this, the third panel was created in the CAD software, and imported into Patran. The third panel had all the properties of the other two panels. The panel was then meshed the same way, and the nodes were equivalenced to delete any duplicate nodes where the third panel met the bottom panel. A total of nine RBE2s were added, three per support block, to connect the blocks to the new panel. The exact same tests were run: modal analysis and random vibration analysis with the same PSD curve applied on both the fixture by itself and the fixture with the added lumped mass. The new design is depicted in Figure 27.

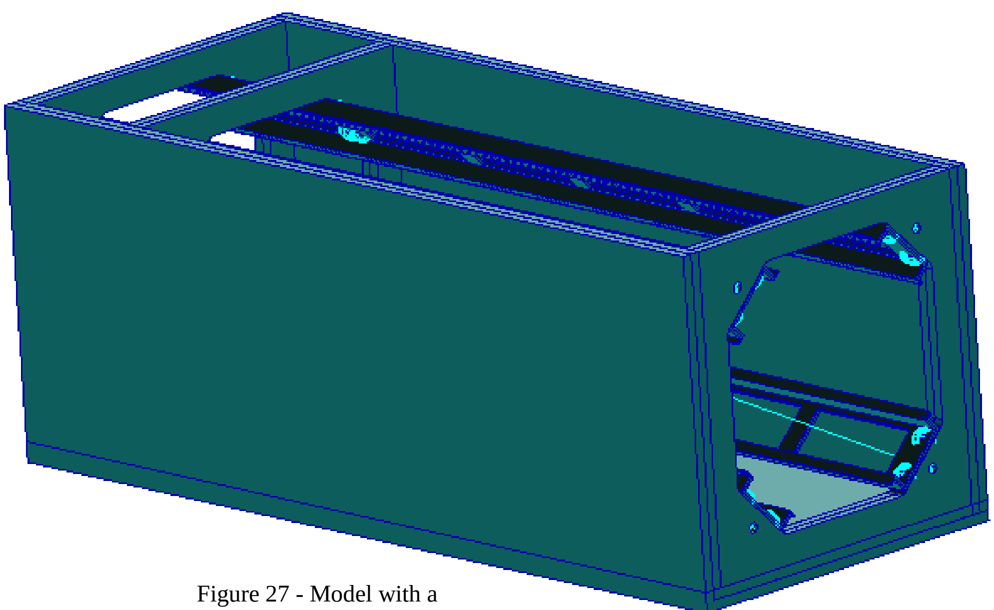
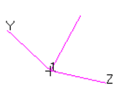
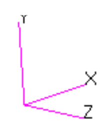


Figure 27 - Model with a Third Panel Added

Coordinate Frame for Rail Analysis



Global Coordinate Frame



4.4 Design Iteration Analysis

With the added third panel, the fixture was expected to be stiffer, and the analysis results supported this expectation. The fixture with the third panel had higher natural frequencies, though the same amount of modes was still in the testing range. The rails again caused the natural frequencies occurring during the testing range.

The stress in each direction, as well as the displacement in each direction due to the random vibration, all decreased, with the exception of the X direction. The increase in the stress and displacement in the X direction was understandable. The added third panel increased the mass, which would increase the displacement and stress, and the way the third panel was oriented affected the X direction the most. On the other hand, the added support decreased the displacement and stress in the Y and Z direction, in some cases drastically.

Overall, the advantages of adding the third panel outweigh the negatives. Though the rails will most likely make contact with the part to be tested, the original fixture had the same problem. With the design iteration, the displacements and stresses in both the Y and Z directions decreased substantially, proving the design change effective.

The planned solution is to the significant displacement of the rails is to add cushions to the part being tested, to absorb the contact and protect the part. Tables 12 and 13 on the following page show the normal modes as well as displacement and stress due to vibration of the fixture with the third panel, and the fixture with the third panel and the lumped mass.

Table 8 - Analysis Results of Fixture + Third Wall

		Random Vibration Displacement (inches)				**3-Sigma Values**			
			Displacement Direction				Displacement Direction		
Modes (Hz)			*x*	*y*	*z*		*x*	*y*	*z*
1	214.89	*x*	0.027	0.026	0.00153	*x*	0.081	0.078	0.00459
2	215.69	*Location*	Rails	Rails	Rails/Tops Support Blocks				
3	217.96	*y*	0.0102	0.0114	0.000488	*y*	0.0306	0.0342	0.00146 4
4	218.08	*Location*	Rails	Rails	Rail/Panel Sides/Tops Supports				
5	253.83	*z*	0.00135	0.00139	0.00492	*z*	0.00405	0.00417	0.01476
6	261.07	*Location*	Rail/Sides of Panels	Rails/Tops Support Blocks	Rails/Tops Support Blocks				
7	265.49	**Von Mises Stress (psi)**				**3-Sigma Values**			
8	265.54		*x*	15200			*x*	45600	
9	281.49		*y*	4080			*y*	12240	
10	357.4		*z*	3110			*z*	9330	

Table 9 - Analysis Results of Fixture + Mass + Third Wall

		Random Vibration Displacement (inches)				**3-Sigma Values**			
			Displacement Direction				Displacement Direction		
Modes (Hz)			*x*	*y*	*z*		*x*	*y*	*z*
1	12.297	*x*	0.0663	0.07	0.00156	*x*	0.1989	0.21	0.00468
2	14.245	*Location*	Rails	Rails	Rails/Tops Support Blocks				
3	55.931	*y*	0.0126	0.0134	0.00274	*y*	0.0378	0.0402	0.00822
4	217.46	*Location*	Rails	Rails	Rail/Panel Sides/Tops Supports				
5	218.13	*z*	0.00768	0.00594	0.00855	*z*	0.02304	0.01782	0.02565
6	218.73	*Location*	Rail/Sides of Panels	Rails	Rails/Tops Support Blocks				
7	218.77	**Von Mises Stress (psi)**				**3-Sigma Values**			
8	261.85		*x*	15700			*x*	47100	
9	263.16		*y*	6900			*y*	20700	
10	266.61		*z*	10600			*z*	31800	

Chapter 5 - Summary and Recommendations

5.1 Review of Test Standards

The review involved a comprehensive summary of key features of the two standards used for space payloads. After studying the test standards provided by both sources, it was concluded that our fixture meets the standards of a sounding rocket being launched into space. The vibration environments that the test specimen will encounter are simulated in computer software.

5.2 Validation of Design Principles

The objective of the FEA analysis was to ensure that the fixture's properties were appropriate and acceptable due to the design properties we have researched. We were given the task of analyzing a test fixture for a space payload, and we had to ensure that the fixture would be capable of carrying the test specimen during the vibration test. This was accomplished through the following methods.

5.2.1 FEA Analysis of Existing Fixture

1. An inherited CAD model of the test fixture was de-featured (removing the characteristics unnecessary to analysis) to provide a base for the fixture's initial analysis.

2. Meshed the fixture using Patran and ran the simulation. Boundary conditions, loads, and nodes were applied to duplicate the real environment's testing conditions.

3. Added lumped mass to FEA model to simulate real applied loads.

4. Performed analysis of the computational results to ensure that the deflection of the rails of the fixture is acceptable, while undergoing testing with the canister loaded inside it.

The results obtained from this test show that the fixture did not meet some of the stress and displacement requirements in order to safely launch the space payload. Some design changes were necessary in order to move forward with this project.

5.2.2 Design Iterations

The objective of the design iteration was to obtain a redesigned test fixture that is to meet the required criteria. The design iteration also helped to provide a better structure to withhold some of the forces generated from vibration loads. The redesigned fixture includes an added wall and was analyzed as follows:

5.2.3 FEA Analysis of Redesigned Fixture

1. Edited the CAD model in CoCreate to incorporate the physical changes to the fixture. The model was de-featured (removing the characteristics unnecessary to analysis) to provide a base for the fixture's initial analysis.
2. Meshed the fixture using Patran and ran the simulation. Boundary conditions, loads, and nodes were applied to the FEA model to duplicate the real environment's testing conditions.
3. Added lumped mass to FEA model to simulate the real applied loads.
4. Performed analysis of the computational results to ensure that the deflection of the rails of the fixture is acceptable, while undergoing testing with the canister loaded inside it.

The results obtained show that the redesigned fixture displayed improved characteristics, as the bulkheads and rails would be sturdier and more effective if the additional reinforcement was added.

5.3 Recommendations

It is recommended that fabrication utilizes the improved design. However, although our results proved that the design iteration was successful in reinforcing our fixture, there were still a few concerns. The displacements and stress values generally decreased in all directions, but the values did not meet the requirements for the fixture to be fabricated. After closer review of the analysis, it was noted that the long rails within the fixture were the source of the problem.

5.3.1 Rails from the Sponsor

The rail supports that run along the bulkheads are the parts which deflect the most during vibration testing. The reinforcement that can be added to the support structure does not directly reinforces the rails. Also, the physical properties of the rails could not be altered because these rails are provided by an outside source. It is recommended that the rail fabricator considers using a different material or assumes modifications that can stiffen the long rails.

5.3.2 Fixture Fabrication

Due to time constraints and lack of open lab space, this MQP did not fabricate the redesigned fixture. Fabrication of the redesigned test fixture will allow the engineers in the group to continue with the project and ultimately launch the sounding rocket into space.

5.3.3 Vibration Test on Modified Fixture

Vibration tests of the redesigned fixture should be conducted in order to validate the computational analysis presented in this MQP. Computational analysis provides the opportunity to identify potential shortcomings of a vibration test before the performance of actual tests with the shaker table. Results from the vibration test should be close to the results from the FEA analysis.

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Appendix A - Tutorial for Vibration Test Fixtures

Purpose of Tutorial

The purpose of this tutorial is to provide guidance to engineers who have little or no knowledge about vibration testing. This tutorial will cover all the important steps to performing a vibration test safely, which include designing the test fixture, ensuring the test specifications are accurate, and analyzing the test results. In most laboratories, there is only a small percentage of engineers are capable of running the software or machines to perform a vibration test. There are an even smaller percentage of those who understand how vibration testing operates, and the procedures that go into testing a test specimen. This tutorial will give an overview of the procedure of the vibration testing and how to use the equipment.

This tutorial was written to be used as a guide at MIT Lincoln Laboratory. Refer to the "Design Procedure" section if you've had some experience with vibration testing and are looking to design a fixture right away.

What is Vibration Testing?

A vibration test is an acceptance test to ensure that a new design will survive its intended environment. A specific type of vibration testing, random vibration testing, is typically used to simulate real world environments. Random testing can be used to duplicate transportation vibration, seismic vibration and operating vibration.

For space payloads in particular, the greatest vibratory forces occur during launch, and the vibration tests are designed to demonstrate that the payload can withstand the expected vibration environment without being adversely affected. However, the vibration test itself can be potentially destructive.

In a vibration test, a test fixture connects the part to be tested to the table that will vibrate, called a shaker table or vibration table, and transmits the force from the table to the part. A forcing function is inputted, and the response of this part due to the function is measured. For engineers unfamiliar with the possible complications of vibration testing and test fixtures, the test could provide misleading results, or even cause damage to the payload being tested. This tutorial will cover the possible complications from vibration testing and steps to take to prevent these complications.

Vibration Testing Procedure

The procedure of vibration testing is as follows:

1. Perform Finite Element Analysis (FEA) on the model using computer software

The purpose of this analysis is to duplicate the vibration environment on a CAD model. The model is meshed and then sent through computer software to simulate a series of vibration tests. The analysis is useful because it will easily point out any design errors of the model, since the results show the maximum stresses and displacements of the model.

When performing the FEA, it is important to run tests on both the fixture and the test specimen individually. Running the analysis in this fashion ensures that both parts are checked and qualified for the vibration test. Once the fixture and test specimen are accepted, combine both the fixture and the part and run the FEA once more. The results of the FEA are used as a base to compare the results of the actual vibration test.

2. Shake fixture on shaker table

Once the FEA is complete and approved, the fixture can be fabricated according to the model. The fixture should then be bolted to the shaker table so vibration tests can be run. The results from this test should be similar to the results from the FEA analysis of the fixture itself.

If the data does not match up, check to see if there's anything physically wrong with the fixture.

3. Shake fixture and mass mock-up of test part

Once the fixture has passed its vibration test, attach a mass mock-up of the test specimen onto the test fixture. The purpose of creating a mass mock-up is to prevent the real test specimen from receiving damage from the vibrations, since a vibration test can be potentially destructive. Shake the fixture with the test specimen to obtain data, which can be compared to the FEA analysis of the fixture and specimen. If the data does not match up, check to see if there's anything physically wrong with the fixture or the test specimen.

4. Low level shake of fixture and test part

Once the mass mock-up has passed its vibration test, attach the test specimen onto the test fixture. Perform a low-level shake of the part. This data is collected in order to ensure that there isn't any damage caused from a higher level vibration test.

5. Perform specified vibration test

The fixture and test specimen undergo the specified vibration test. This data should also accurately match up with the FEA analysis and the test with a mass mock-up.

6. Repeat low level shake of fixture and test part

The low level shake is repeated to ensure that nothing moved within the test specimen during a higher level shake. The results from this test should match up with the results from the first low level shake.

It is also important to note that vibration testing is performed in each of the three orthogonal axes, one at a time, to provide control and standardization throughout the test. This is also done to minimize the movement in the other two axes, which can alter the data with unrealistic results.

Vibration Test Fixture

The vibration test fixture is a very important element of the vibration test setup. A vibration test fixture is required to allow the mounting of a specific test specimen onto a vibration table and is part of the vibration load path. A fixture should uniformly transmit all the forces produced by the vibration table to the test specimen. It is normally impossible to fix a test specimen directly to the shaker table itself, so the fixture acts as a transmission. The test fixture should simulate both the specimen and vibration equipment interfaces.

Each vibration test fixture is designed individually and tailored to fit the test specimen. The reason for this is because every object that undergoes vibration testing will have its own unique features, with varying surfaces and materials. Therefore, the composition, shape, and stiffness of the fixture will depend heavily on the details of the test specimen.

The vibration test fixture is mounted to the shaker table by using bolts. The size and placement of these bolts on the fixture will depend on the bolt pattern on the shaker table, as it is important to make sure the holes on the fixture do not interfere with the physical properties of the fixture.

Design Properties

When designing a vibration test fixture, it is important to take all of its physical properties into consideration. Although the test specimen greatly influences the design of the fixture, other properties such as material and stiffness can be altered to engineer the most efficient fixture.

There are various materials which can be used for fixture fabrication. However, the most important property of these materials is the ratio between its Young's Modulus to density. This ratio will determine the natural frequency of the material. Steel, aluminum, and magnesium all have varying Young's Moduluses and densities, but their ratios are very similar, which make them the most common materials to fabricate a fixture with. At Lincoln Laboratory, aluminum is the material of choice for fixture fabrication. It should also be noted that in reality, a fixture can be fabricated from any material, depending on special cases.

A vibration test fixture should also provide as little interference as possible – the fixture should be designed to minimize any of its mechanical impedance on the test specimen. This is directly related to the material of the fixture, since the natural frequency of the fixture should be outside and above the testing range. A fixture with less mass will also be easier for the shaker table to perform. Therefore, the fixture should be designed to be as rigid as possible.

The placement of accelerometers is also a crucial part of the design process, since the fixture should be designed to accommodate accelerometers if needed. A control accelerometer is often placed in one of three areas – the shaker table, the test fixture, or the test specimen. The location of this accelerometer is often defined by the properties of the test specimen – such as the center of gravity, the natural frequency, or some points of interest in the part (i.e. sensitive electronics).

In conclusion, there are three main design principles to follow when designing a vibration test fixture:

1. Fixture should have the least mass possible.

2. Fixture should be designed to be as stiff as possible.

3. Fixture's natural frequency should be above the vibration testing range.

All of these principles are directly related to the properties of the fixture, so it is important to keep all of that information in mind.

Design Procedure

When designing a custom test fixture, there are numerous steps that need to be taken before the fixture is fabricated.

1. Size of the test specimen

The size and shape of the test specimen will greatly influence the design of the fixture. As mentioned above, a material needs to be chosen to accommodate the test specimen. Pick a material that is strong enough to support the specimen's weight, as well as stiff enough to prevent any additional vibration. The shape of the fixture will also rely heavily on which axes to mount the test specimen on. Make sure the fixture design accommodates mounting onto the vibration table in three separate axes.

2. Figure out bolt holes pattern of the table and mounting pattern of specimen

Once the material is chosen for the fixture, research the bolt patterns of the shaker table to see where the bolts should be placed on the fixture. The holes should be placed in the fixture as to not influence its physical properties, such as its natural frequency or stiffness. FEA analysis can be run on the design model to ensure that no significant changes are made. Also make sure to note if there are any parts on the test specimen that may need additional attention so that an accelerometer can be placed to monitor its movement.

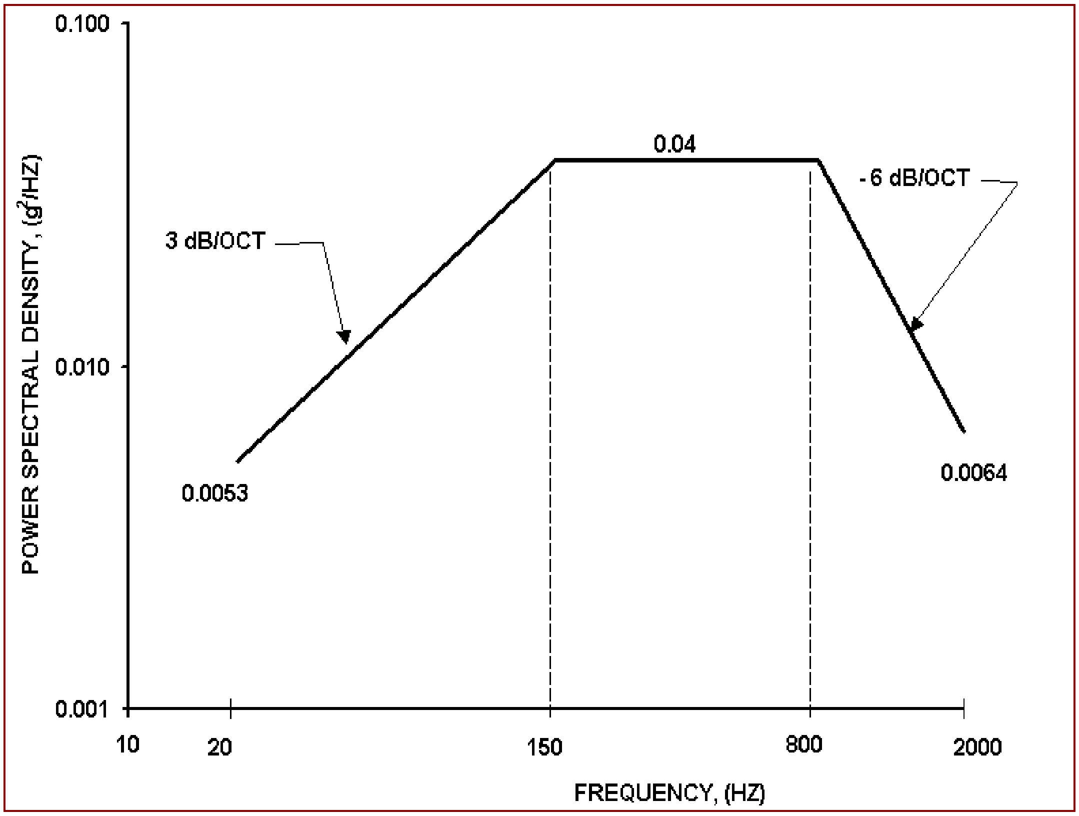
3. Create computer models and run analysis on models

The designs should be modeled as accurately as possible in CAD and sent through computer software for finite element analysis (FEA). Refer to the "Vibration Testing Process" section for guidance. Once the fixture passes the FEA, you may proceed with the fabrication of the fixture.

Taking all of these steps ensures that a vibration test fixture can be designed and fabricated efficiently. Make sure to follow each step closely and not to skip over any small detail, because a tiny design error can cause thousands of dollars of damage in the test specimen!

Appendix B

Figure 3



Curve Values	
Frequency (Hz)	Minimum PSD (g2/Hz)
20	0.0053
20 to 150	+3 dB per octave slope
150 to 800	0.04
800 to 2000	-6 dB per octave slope
2000	0.00644
The overall acceleration level is 6.9 grms. Note: This spectrum applies only to units whose mass does not exceed 23 kilograms (50 pounds).	

From Figure 6.3.5-1 Minimum Random Vibration Spectrum, Unit Acceptance Tests from TR – 2004(8583), pg 46

Table 5

TABLE 514.5C-VII. Break points for curves of figures 514.5C-1 through 514.5C-3.

U. S. highway truck vibration exposures, figure 514.5C-1

vertical		transverse		longitudinal	
Hz	g^2/Hz	Hz	g^2/Hz	Hz	g^2/Hz
10	0.01500	10	0.00013	10	0.00650
40	0.01500	20	0.00065	20	0.00650
500	0.00015	30	0.00065	120	0.00020
1.04	g rms	78	0.00002	121	0.00300
		79	0.00019	200	0.00300
		120	0.00019	240	0.00150
		500	0.00001	340	0.00003
		0.204	g rms	500	0.00015
				0.740	g rms

Composite two-wheeled trailer vibration exposures, figure 514.5C-2

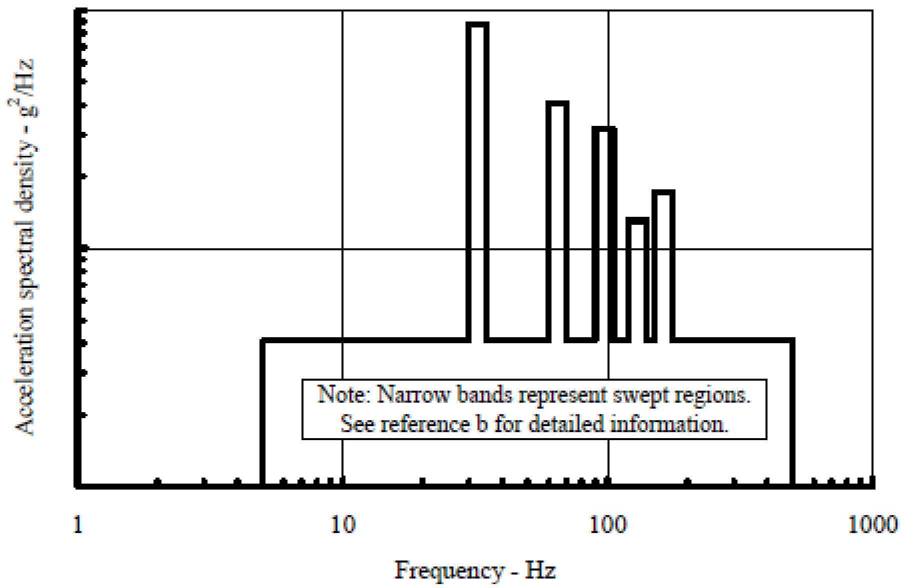
vertical		transverse		longitudinal	
Hz	g^2/Hz	Hz	g^2/Hz	Hz	g^2/Hz
5	0.2252	5	0.0474	5	0.0563
8	0.5508	6	0.0303	6	0.0563
10	0.0437	7	0.0761	8	0.1102
13	0.0253	13	0.0130	13	0.0140
15	0.0735	15	0.0335	16	0.0303
19	0.0143	16	0.0137	20	0.0130
23	0.0358	21	0.0120	23	0.0378
27	0.0123	23	0.0268	27	0.0079
30	0.0286	25	0.0090	30	0.0200
34	0.0133	28	0.0090	33	0.0068
36	0.0416	30	0.0137	95	0.0019
41	0.0103	34	0.0055	121	0.0214
45	0.0241	37	0.0081	146	0.0450
51	0.0114	46	0.0039	153	0.0236
95	0.0266	51	0.0068	158	0.0549
111	0.0166	55	0.0042	164	0.0261
136	0.0683	158	0.0029	185	0.0577
147	0.0266	235	0.0013	314	0.0015
185	0.0603	257	0.0027	353	0.0096
262	0.0634	317	0.0016	398	0.0009
330	0.0083	326	0.0057	444	0.0027
360	0.0253	343	0.0009	500	0.0014
500	0.0017	384	0.0018	2.40	g rms
3.85	g rms	410	0.0008		
		462	0.0020		
		500	0.0007		
		1.28	g rms		

Composite wheeled vehicle vibration exposures, figure 514.5C-3

vertical		transverse		longitudinal	
Hz	g^2/Hz	Hz	g^2/Hz	Hz	g^2/Hz
5	0.2308	5	0.1373	5	0.0605
8	0.7041	9	0.0900	6	0.0577
12	0.0527	12	0.0902	8	0.0455
16	0.0300	14	0.0427	12	0.0351
20	0.0235	16	0.0496	15	0.0241
22	0.0109	18	0.0229	16	0.0350
24	0.0109	119	0.0008	19	0.0092
26	0.0154	146	0.0013	25	0.0159
69	0.0018	166	0.0009	37	0.0041
79	0.0048	201	0.0009	41	0.0060
87	0.0028	273	0.0053	49	0.0017
123	0.0063	289	0.0021	105	0.0006
161	0.0043	371	0.0104	125	0.0004
209	0.0057	382	0.0019	143	0.0013
224	0.0150	402	0.0077	187	0.0013
247	0.0031	422	0.0027	219	0.0028
278	0.0139	500	0.0016	221	0.0068
293	0.0037	1.60	g rms	247	0.0325
357	0.0028			249	0.0098
375	0.0052			270	0.0026
500	0.0011			293	0.0094
2.18	g rms			336	0.0120
				353	0.0247
				379	0.0085
				431	0.0224
				433	0.0092
				500	0.0014
				1.96	g rms

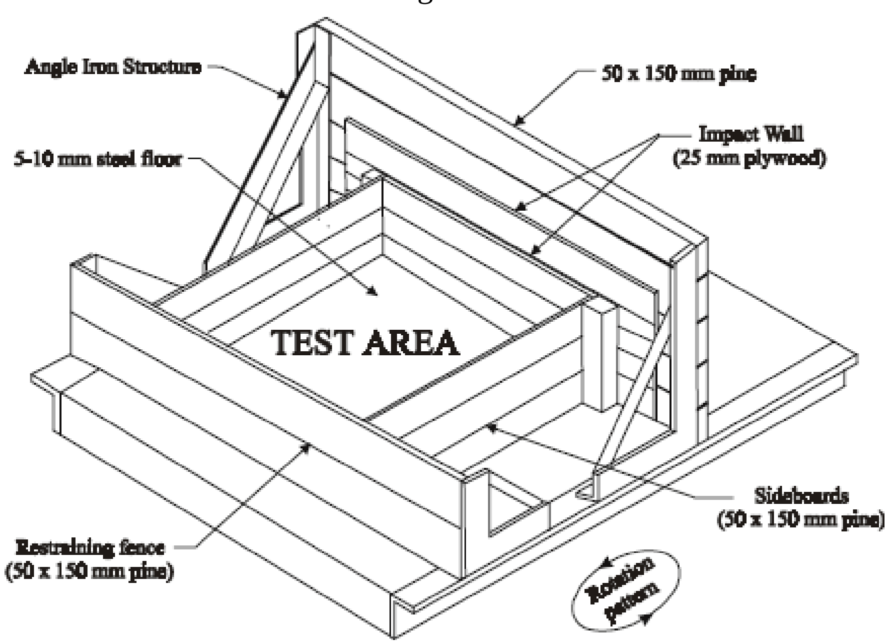
Vibration Exposure Values, taken directly from MIL-STD-810F, Table 514.5C-VII

Figure 4



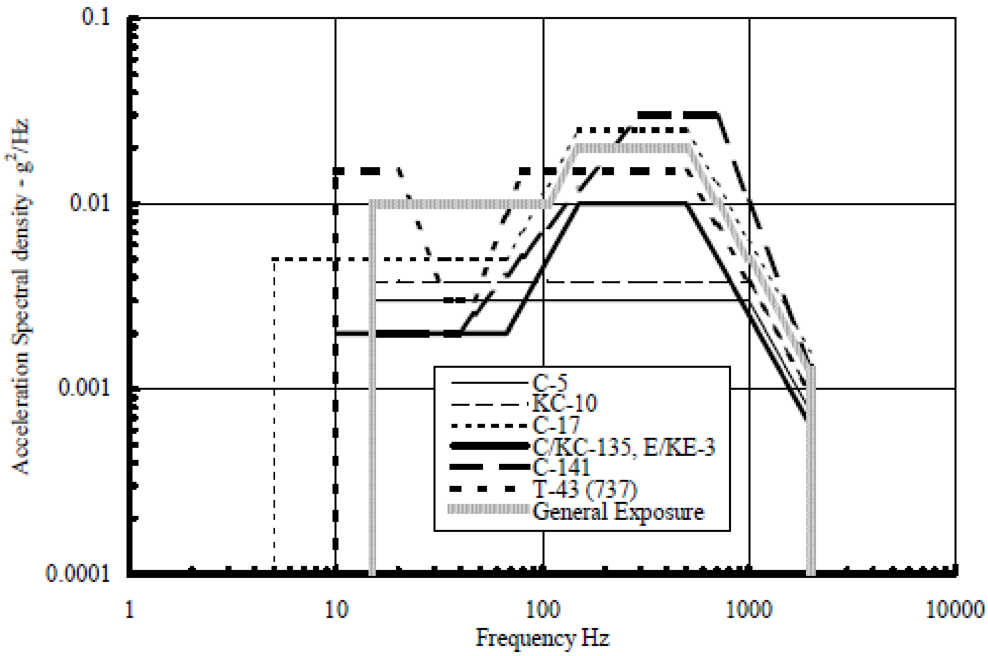
Tracked vehicle representative spectral shape, taken directly from MIL-STD-810F Figure 514.5C-4

Figure 5



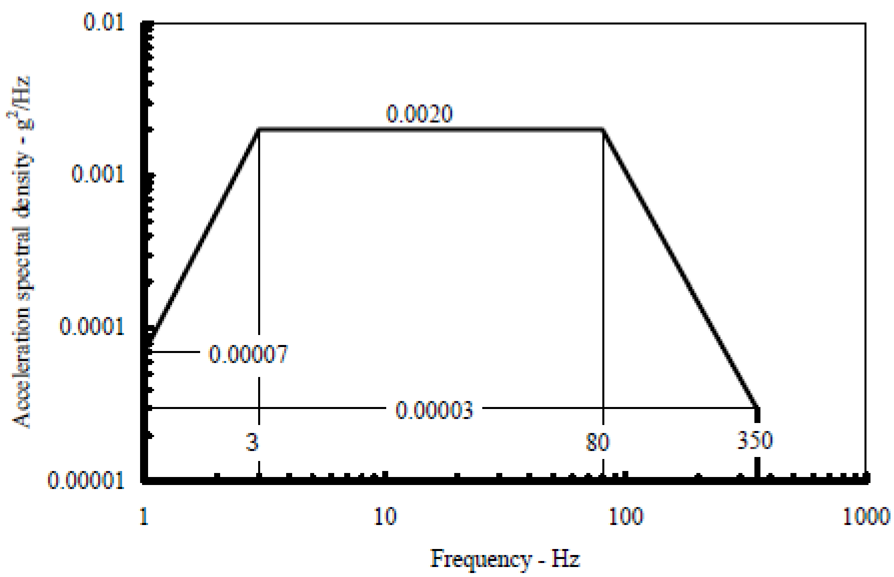
Taken directly from MIL-STD-810F Figure 514.5C-5

Figure 6



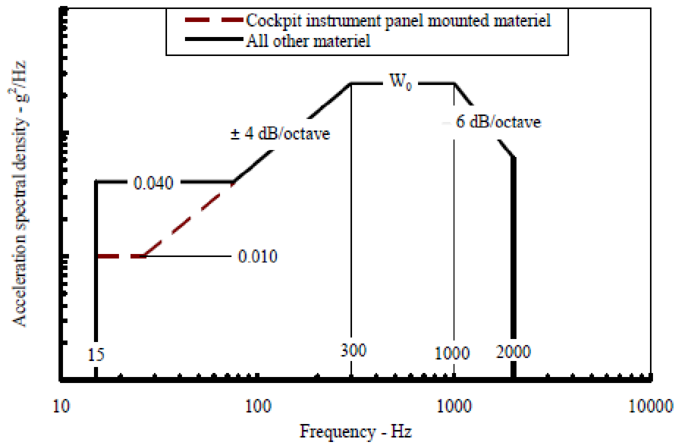
Jet aircraft cargo vibration exposure – Taken directly from MIL-STD-810F Figure 514.5C-6

Figure 7



Rail cargo vibration exposure, taken directly from MIL-STD-810F, Figure 514.5C-7

Figure 8



Jet aircraft vibration exposure, taken directly from MIL-STD-810F, Figure 514.5C-8

Table 6

<table>
<tr><td colspan="2">$W_0 = W_A + \sum_1^n (W_J)$
W_0, W_A, W_J - Exposure levels in acceleration spectral density (g^2/Hz).</td></tr>
<tr><td colspan="2">Aerodynamically induced vibration
$W_A = a \times b \times c \times (q)^2$</td></tr>
<tr><td colspan="2">Jet engine noise induced vibration
$W_J = \{[048 \times a \times d \times \cos^2(\theta)/R] \times [D_c \times (V_c / V_r)^3 + D_f \times (V_f / V_r)^3]\}$</td></tr>
<tr><td colspan="2">a - Platform / Materiel interaction factor (see Annex B, paragraph 2.4). Note that this factor applies to W_o and not to the low frequency portion (15 Hz to break) of figure 514.5C-14.
= 1.0 for materiel mounted on vibration isolators (shock mounts) and materiel weighing less than 36 kg.
= $1.0 \times 10^{(0.6 - W/60)}$ for materiel weighing between 36 and 72.12 kg.(w = weight in kg)
= 0.25 for materiel weighing 72.12 kg or more.</td></tr>
<tr><td>b - Proportionality factor between vibration level and dynamic pressure (SI units).
= 2.96×10^{-6} for materiel mounted on cockpit instrument panels.
= 1.17×10^{-5} for cockpit materiel and materiel in compartments adjacent to external surfaces that are smooth and free from discontinuities.
= 6.11×10^{-5} for materiel in compartments adjacent to or immediately aft of external surface discontinuities (cavities, chines, blade antennae, speed brakes, etc.), fuselage aft of wing trailing edge, wing, empennage, and pylons.

c - Mach number correction. Note that this factor applies to W_0 and not to the low frequency portion (15 to TBD Hz at 0.04 g^2/Hz) of figure 514.5C-8. (Annex A, paragraph. 2.3.1)
= 1.0 for $0 \leq$ Mach ≤ 0.9
= (-4.8M + 5.32) for $0.9 \leq$ Mach ≤ 1.0 (where M = Mach number)
= 0.52 for Mach number greater than 1.0

q - Flight dynamic pressure, kN / m^2 (lb/ft^2). (See Annex B, para. 2.6.1 and table 5145C-VI)</td><td>$\sum_1^n$ - Jet noise contribution is the sum of the W_J values for each engine.

d - Afterburner factor.
= 1.0 for conditions where afterburner is not used or is not present.
= 4.0 for conditions where afterburner is used.

R - Vector distance from center of engine exhaust plane to materiel center of gravity, m (ft).

θ - Angle between R vector and engine exhaust vector (aft along engine exhaust centerline), degrees For $70° < \theta \leq 180°$ use 70°.

D_c - Engine core exhaust diameter, m (ft).

D_f - Engine fan exhaust diameter, m (ft).

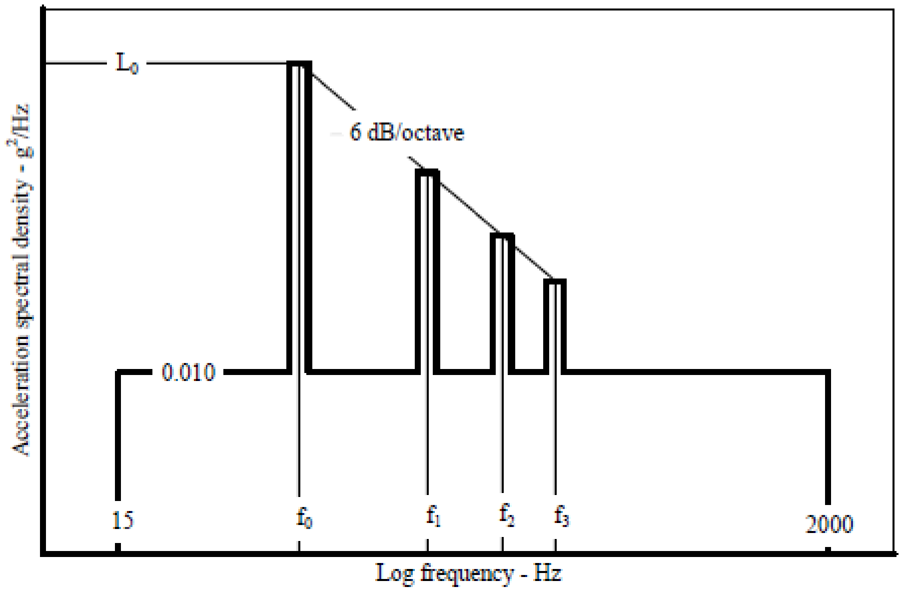
V_r - Reference exhaust velocity, m/sec (ft/sec).
= 564 m/sec

V_c - Engine core exhaust velocity Engine core exhaust velocity (without afterburner), m/sec (ft/sec).

V_f - Engine fan exhaust velocity (without afterburner), m/sec (ft/sec).</td></tr>
<tr><td colspan="2">**If Dimensions are in feet and pounds then:**
a = 1.0 for materiel mounted on vibration isolators (shock mounts) and materiel weighing less than 80 lb.
= $1.0 \times 10^{(0.60 - 0.0075 W)}$ for materiel weighing between 80 and 160 lb.
= 0.25 for materiel weighing 160 lb. or more.
b = 6.78×10^{-9}, 2.70×10^{-8}, or 1.40×10^{-7} in the order listed above.
V_r = 1850 feet / second</td></tr>
</table>

Jet aircraft vibration exposure, taken directly from MIL-STD-810F, Table 514.5C-III

Figure 9



Propeller aircraft vibration exposure, taken directly from MIL-STD-810F, Figure 514.5C-9

Table 7

MATERIEL LOCATION 1/, 2/, 3/, 4/	VIBRATION LEVEL L_o (g^2/Hz)
In fuselage or wing forward of propeller	0.10
Within one propeller blade radius of propeller passage plane	1.20
In fuselage or wing aft of propeller	0.30
In engine compartment, empennage, or pylons	0.60

1/ For Materiel mounted to external skin, increase level by 3 dB.

2/ f_0 = blade passage frequency (propeller rpm times number of blades) (Hz).

$f_1 = 2 \times f_0$ $f_2 = 3 \times f_0$ $f_3 = 4 \times f_0$

3/ Spike bandwidths are ± 5% of center frequency.

4/ C-130 Aircraft

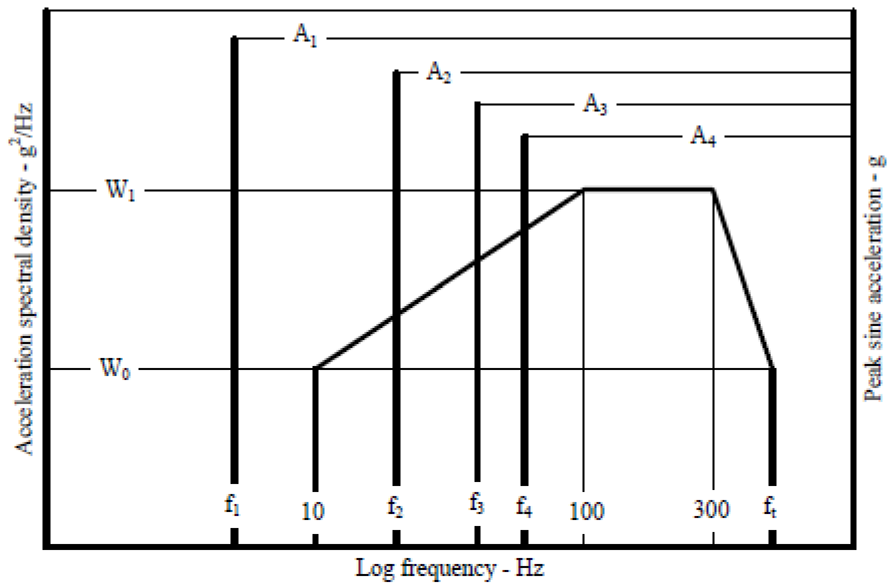
3 blade propeller - f_0 = 51 Hz

4 blade propeller - f_0 = 68 Hz

6 blade propeller - f_0 = 102 Hz (C-130J)

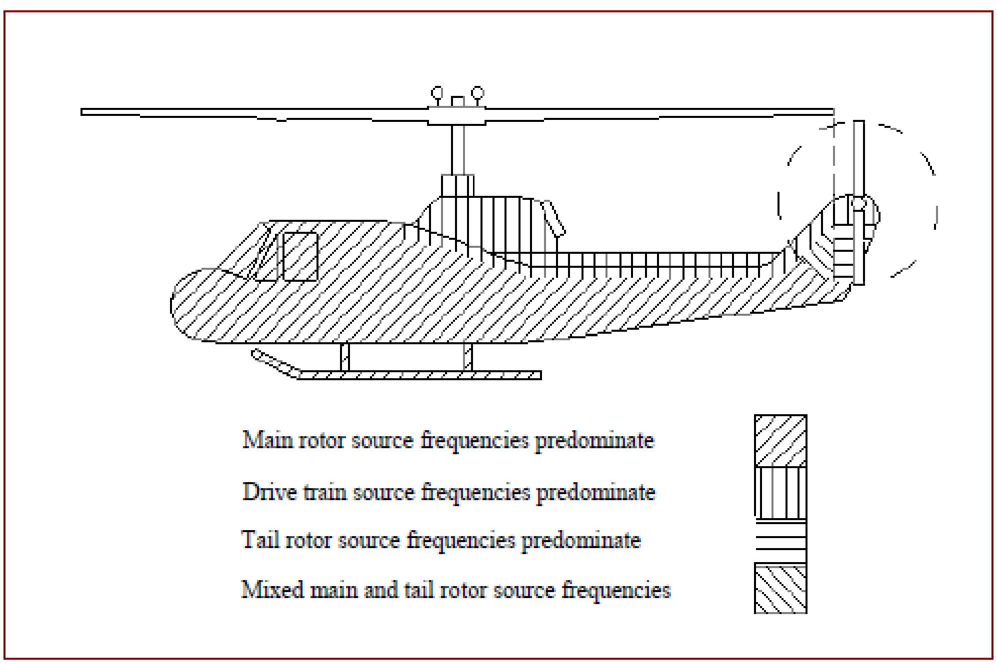
Propeller aircraft vibration exposure, taken directly from MIL-STD-810F, Table 514.5C-II

Figure 10



Helicopter vibration exposure, taken directly from MIL-STD-810F, Figure 514.5C-10

Figure 11



Helicopter vibration zones, taken directly from MIL-STD-810F, Figure 514.5C-11

Table 8

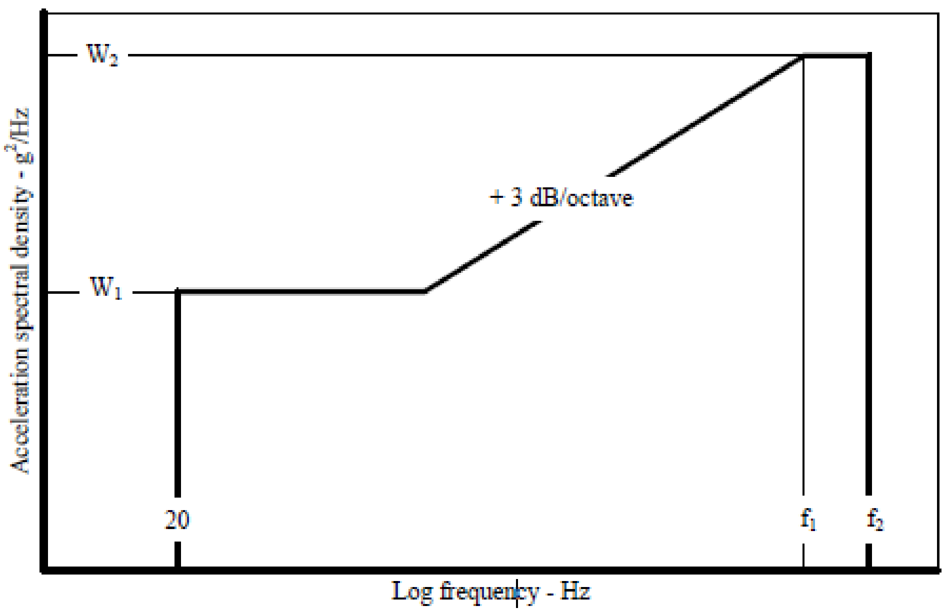
MATERIEL LOCATION	RANDOM LEVELS	SOURCE FREQUENCY (f_x) RANGE (Hz)	PEAK ACCELERATION (A) at f_x (GRAVITY UNITS (g))
General	$W_0 = 0.0010\ g^2/Hz$ $W_1 = 0.010\ g^2/Hz$ $f_t = 500$ Hz	3 to 10 10 to 25 25 to 40 40 to 50 50 to 500	$0.70 / (10.70 - f_x)$ $0.10 \times f_x$ 2.50 $6.50 - 010 \times f_x$ 1.50
Instrument Panel	$W_0 = 0.0010\ g^2/Hz$ $W_1 = 0.010\ g^2/Hz$ $f_t = 500$ Hz	3 to 10 10 to 25 25 to 40 40 to 50 50 to 500	$0.70 / (10.70 - f_x)$ $0.070 \times f_x$ 1.750 $4.550 - 0.070 \times f_x$ 1.050
External Stores	$W_0 = 0.0020\ g^2/Hz$ $W_1 = 0.020\ g^2/Hz$ $f_t = 500$ Hz	3 to 10 10 to 25 25 to 40 40 to 50 50 to 500	$0.70 / (10.70 - f_x)$ $0.150 \times f_x$ 3.750 $9.750 - 0.150 \times f_x$ 2.250
On/Near Drive System Elements	$W_0 = 0.0020\ g^2/Hz$ $W_1 = 0.020\ g^2/Hz$ $f_t = 2000$ Hz	5 to 50 50 to 2000	$0.10 \times f_x$ $5.0 + 0.010 \times f_x$

Main or Tail Rotor Frequencies (Hz) Determine 1P and 1T from Specific Helicopter or from Table (below).			Drive Train Component Rotation Frequency (Hz) Determine 1S from Specific Helicopter and Component.	
$f_1 = 1P$	$f = 1T$	fundamental	$f = 1S$	fundamental
$f = n \times 1P$	$f = m \times 1T$	blade passage	$f = 2 \times 1S$	1st harmonic
$f = 2 \times n \times 1P$	$f = 2 \times m \times 1T$	1st harmonic	$f = 3 \times 1S$	2nd harmonic
$f = 3 \times n \times 1P$	$f = 3 \times m \times 1T$	2nd harmonic	$f = 4 \times 1S$	3rd harmonic

Helicopter	MAIN ROTOR		TAIL ROTOR	
	Rotation Speed 1P (Hz)	Number of Blades n	Rotation Speed 1T (Hz)	Number of Blades m
AH-1	5.40	2	27.7	2
AH-6J	7.80	5	47.5	2
AH-64(early)	4.82	4	23.4	4
AH-64(late)	4.86	4	23.6	4
CH-47D	3.75	3	2 main rotors and no tail rotor	
MH-6H	7.80	5	47.5	2
OH-6A	8.10	4	51.8	2
OH-58A/C	5.90	2	43.8	2
OH-58D	6.60	4	39.7	2
UH-1	5.40	2	27.7	2
UH-60	4.30	4	19.8	4

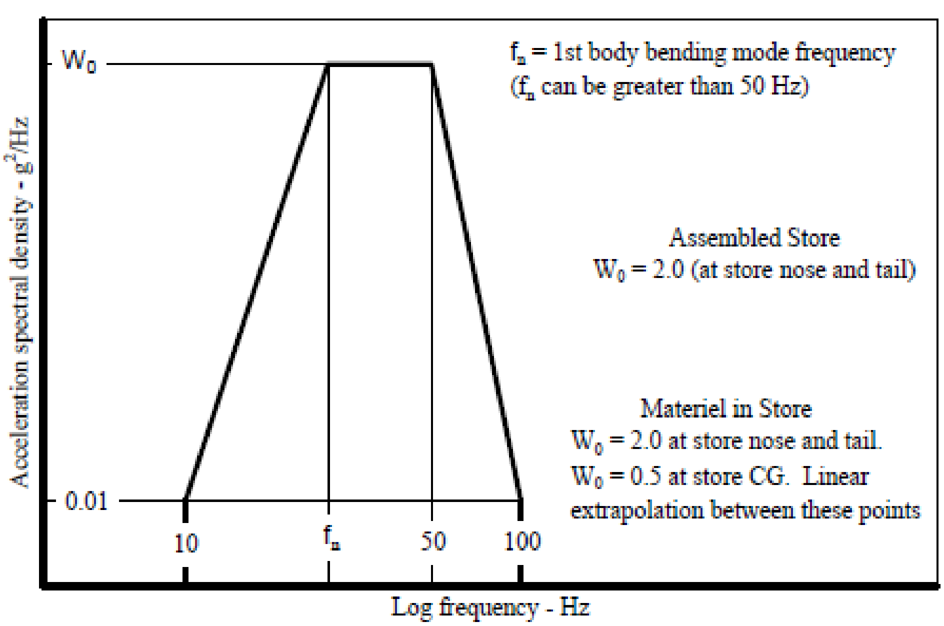
Helicopter vibration exposure, taken directly from MIL-STD-810F, Table 514.5C-IV

Figure 12



Jet aircraft store vibration response, taken directly from MIL-STD-810F, Figure 514.5C-12

Figure 13



Jet aircraft store buffet response, taken directly from MIL-STD-810F, Figure 514.5C-13

Table 9

$$W_1 = 5 \times 10^{-3} \times K \times A_1 \times B_1 \times C_1 \times D_1 \times E_1; \quad (g^2/Hz) \quad \underline{1/}$$

$$W_2 = H \times (q/\rho)^2 \times K \times A_2 \times B_2 \times C_2 \times D_2 \times E_2; \quad (g^2/Hz) \quad \underline{1/}$$

$$M \le 0.90,\ K = 1.0; \quad 0.90 \le M \le 1.0,\ K = -4.8 \times M + 5.32; \quad M \ge 1.0,\ K = 0.52 \quad \underline{2/}$$

$$f_1 = C(t/R^2),\ (Hz) \quad \underline{3/}, \underline{4/}, \underline{5/} \qquad f_2 = f_1 + 1000,\ (Hz) \quad \underline{3/}$$

$$f_0 = f_1 + 100,\ (Hz) \quad \underline{6/}, \underline{7/}$$

Configuration	Factors	
Aerodynamically clean	A_1	A_2
Single store	1	1
Side by side stores	1	2
Behind other store(s)	2	4
Aerodynamically dirty 8/	C_1	C_2
Single and side by side	2	4
Behind other store(s)	1	2
Other stores	1	1
	E_1	E_2
Jelly filled firebombs	1/2	1/4
Other stores	1	1

Configuration	Factors	
	B_1	B_2
Powered missile, aft half	1	4
Other stores, aft half	1	2
All stores, forward half	1	1
	D_1	D_2
Field assembled sheet metal fin / tailcone unit	8	16
Powered missile	1	1
Other stores	4	4

M – Mach number.

H – Constant = 5.59 (metric units) ($= 5 \times 10^{-5}$ English units).

C – Constant = 2.54×10^3 (metric units) ($= 1.0 \times 10^5$ English units).

q – Flight dynamic pressure (see table 514.5C-VI) – kN/m^2 (lb/ft^2).

ρ – Store weight density (weight/volume) - kg/m^3 (lb/ft^3).
Limit values of ρ to $641 \le \rho \le 2403\ kg/m^3$ ($40 \le \rho \le 150\ lb/ft^3$).

t – Average thickness of structural (load carrying) skin - m (in).

R – Store characteristic (structural) radius m (in) (Average over store length).
= Store radius for circular cross sections.
= Half or major and minor diameters for elliptical cross section.
= Half or longest inscribed chord for irregular cross sections.

1/ – When store parameters fall outside limits given, consult references.

2/ – Mach number correction (see Annex B, 2.6)

3/ – Limit f_1 to $100 \le f_1 \le 2000$ Hz

4/ – Free fall stores with tail fins, $f_1 = 125$ Hz

5/ – Limit (t/R^2) to: $0.0010 \le (t/R^2) \le 0.020$

6/ – $f_o = 500$ Hz for cross sections not circular or elliptical

7/ – If $f_o > 200$ Hz then use $f_o = 2000$ Hz

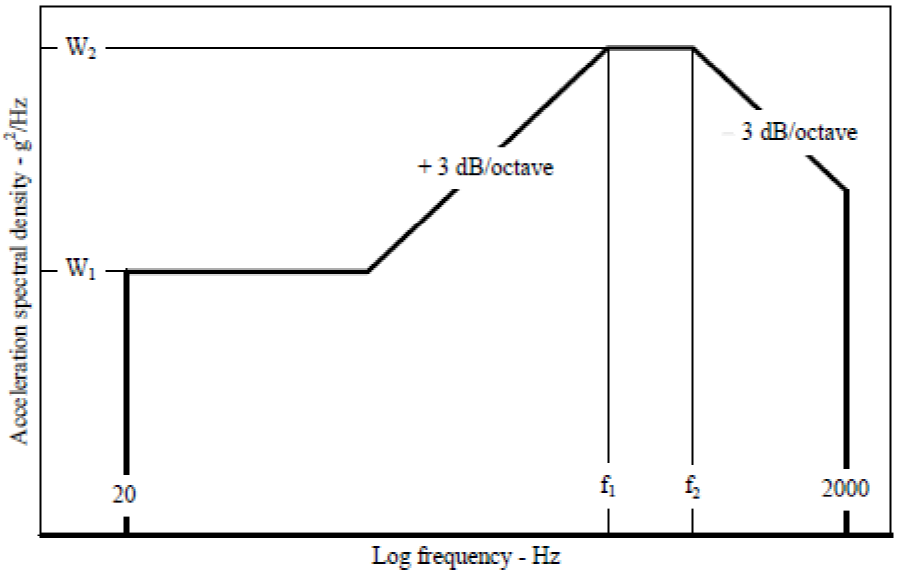
8/ – Configurations with separated aerodynamic flow within the first ¼ of the store length. Blunt noses, optical flats, sharp corners, and open cavities are some potential sources of separation. Any nose other than smooth, rounded, and gently tapered is suspect. Aerodynamics engineers should make this judgment.

Representative parameter values

Store type	Max q		ρ		f_1	f_2
	kN/m^2	(lb/ft^2)	kg/m^3	(lb/ft^3)	Hz	Hz
Missile, air to ground	76.61	(1600)	1602	(100)	500	1500
Missile, air to air	76.61	(1600)	1602	(100)	500	1500
Instrument pod	86.19	(1800)	801	(50)	500	1500
Dispenser (reusable)	57.46	(1200)	801	(50)	200	1200
Demolition bomb	57.46	(1200)	1922	(120)	125	1100
Fire bomb	57.46	(1200)	641	(40)	100	1100

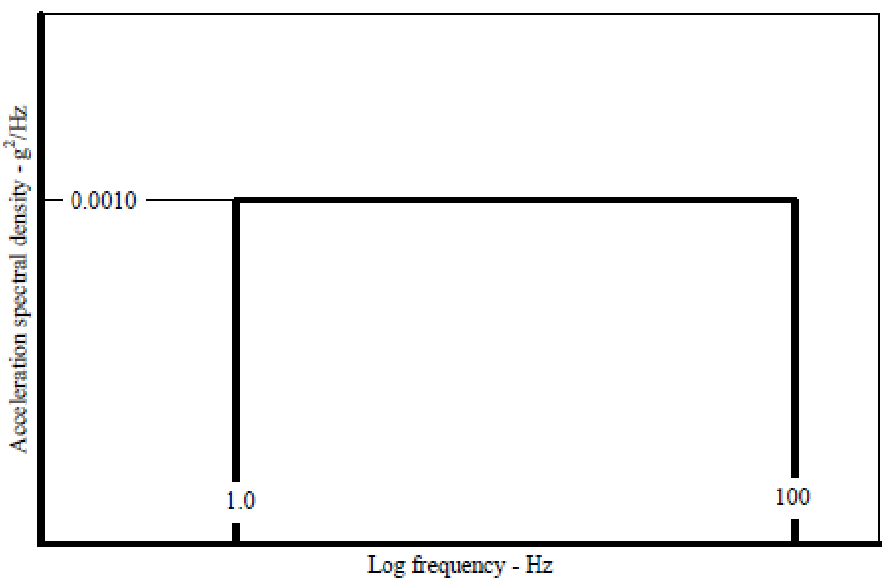
Jet aircraft external store vibration exposure, taken directly from MIL-STD-810F, Table 514.5C-V

Figure 14



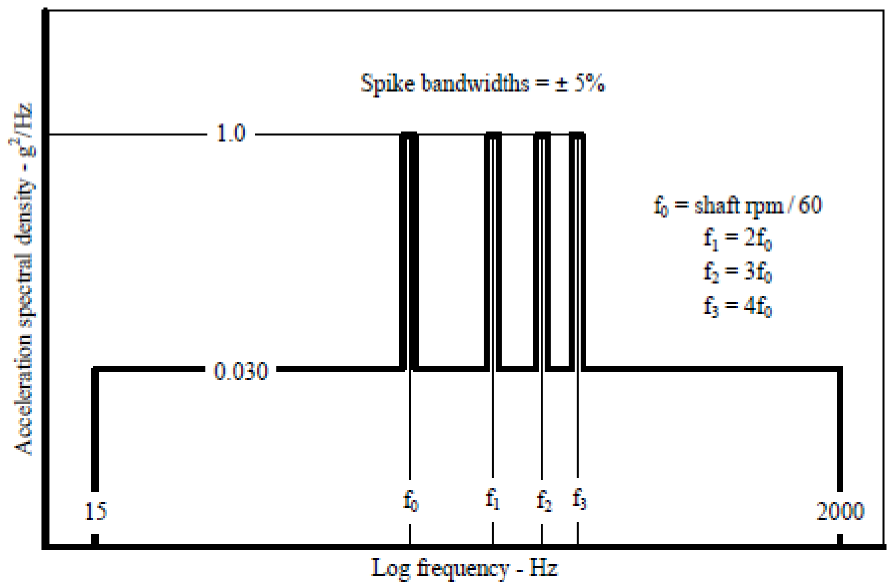
Jet aircraft store equipment vibration exposure, taken directly from MIL-STD-810F, Figure 514.5C-14

Figure 15



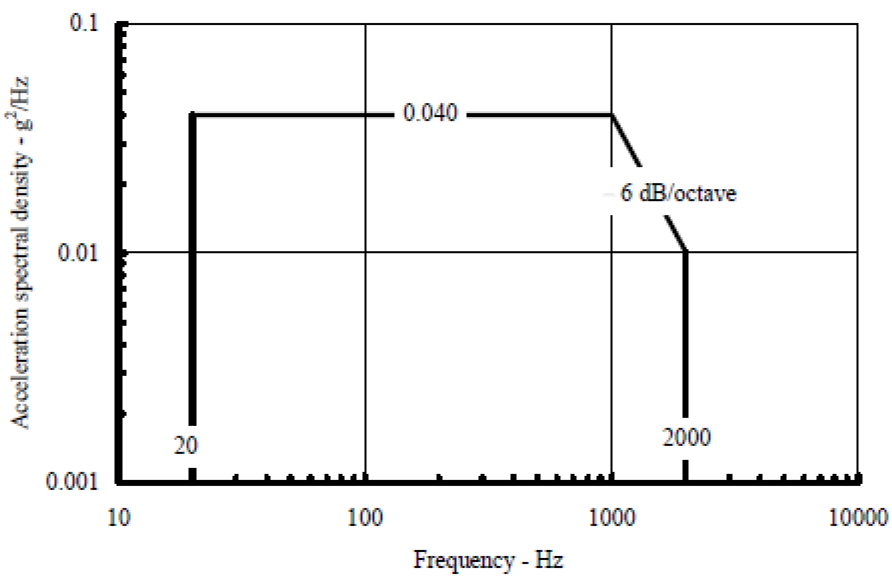
Shipboard random vibration exposure, taken directly from MIL-STD-810F, Figure 514.5C-15

Figure 16



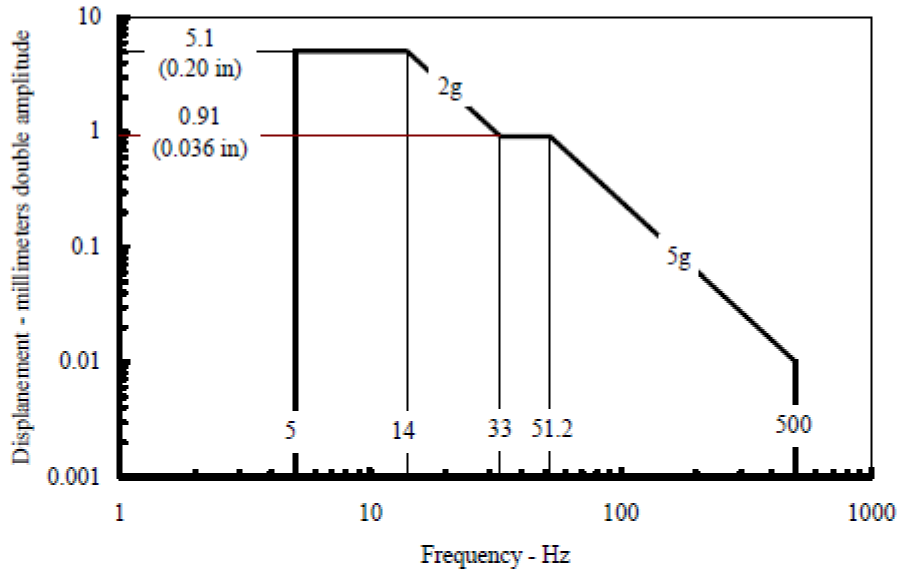
Turbine engine vibration exposure, taken directly from MIL-STD-810F, Figure 514.5C-16

Figure 17



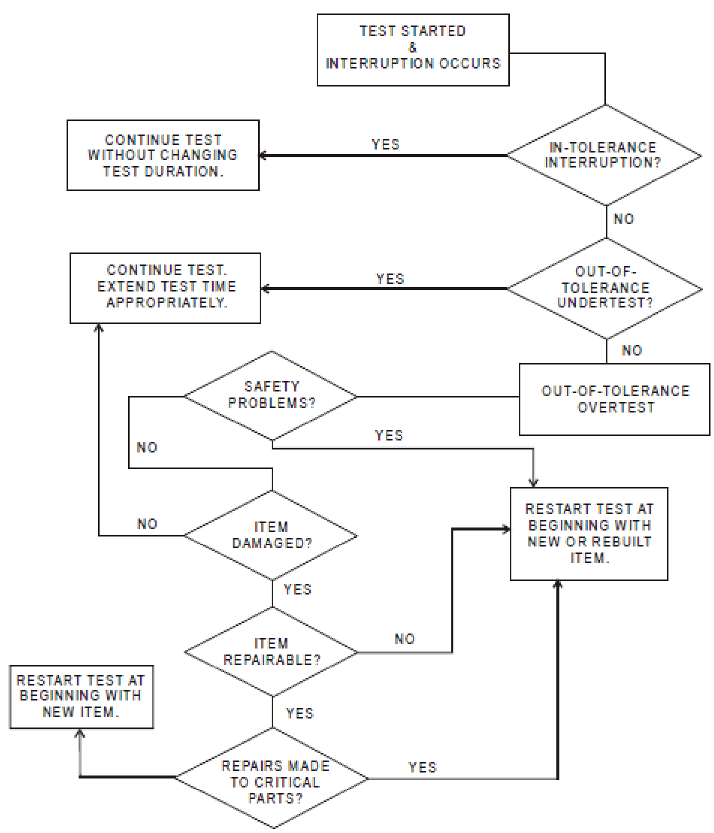
General minimum integrity exposure, taken directly from MIL-STD-810F, Figure 514.5C-17

Figure 18



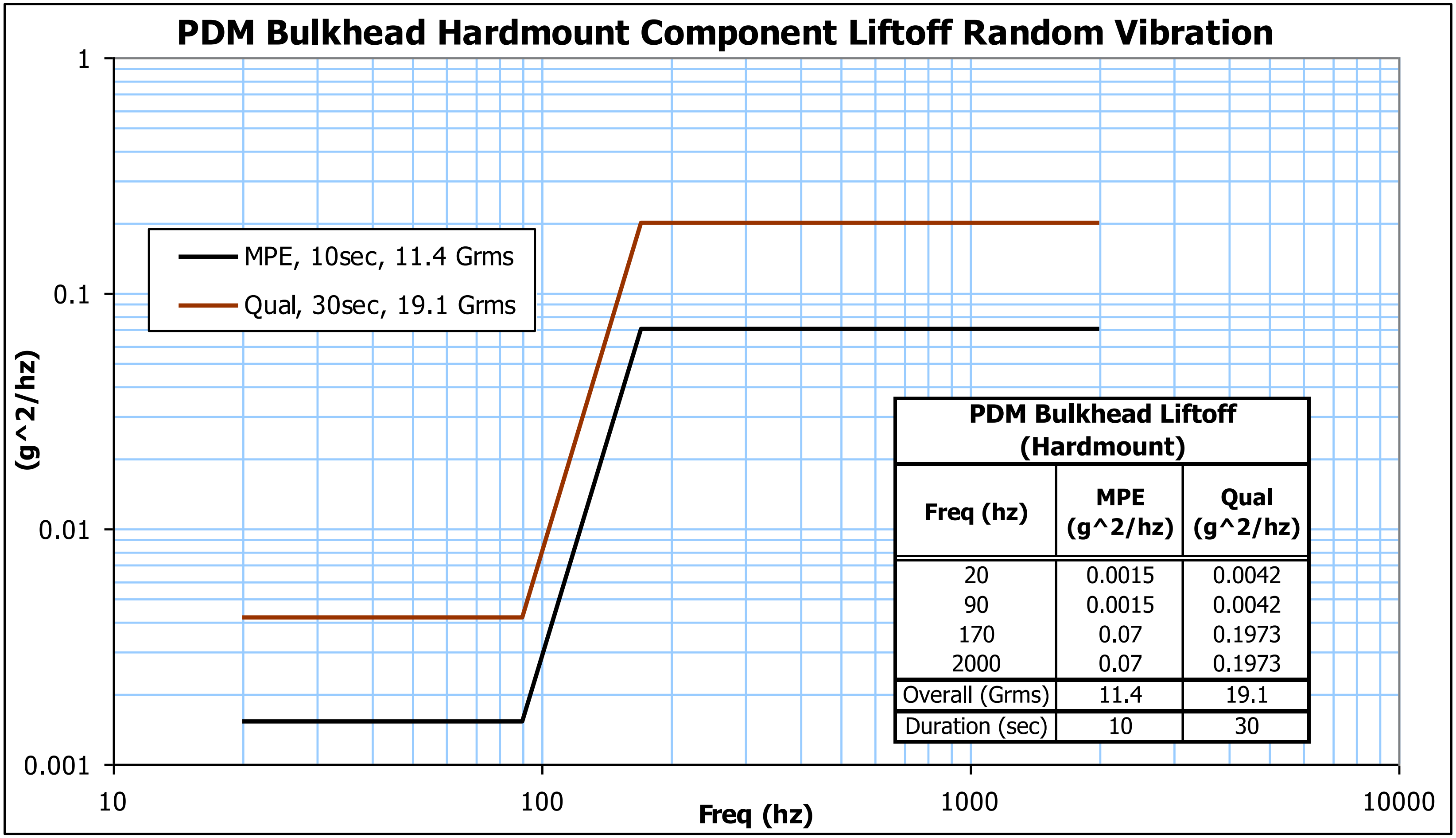
Helicopter minimum integrity exposure, taken directly from MIL-STD-810F, Figure 514.5C-18

Figure 19



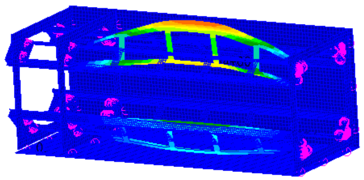
Interrupted test cycle logic, taken directly from MIL-STD-810F, Figure 5-1

Appendix C

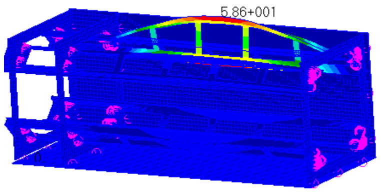


PSD Curve Provided

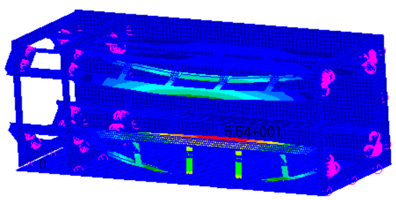
Test Fixture – Modal Analysis



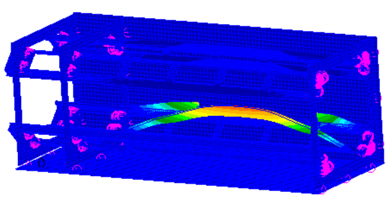
Mode 1 – 211.7 Hz



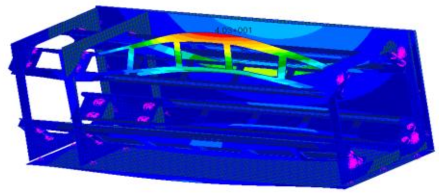
Mode 2 – 214.75 Hz



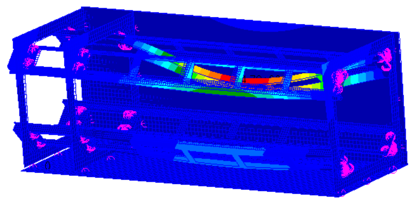
Mode 3 – 216.51 Hz



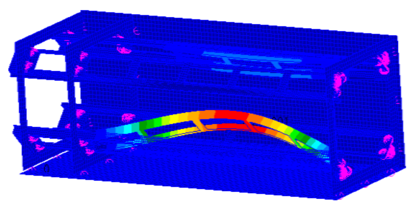
Mode 4 – 217.91 Hz



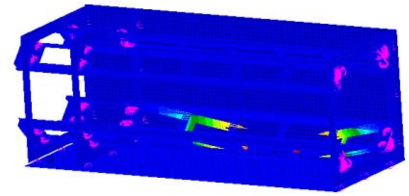
Mode 5 – 239.41 Hz



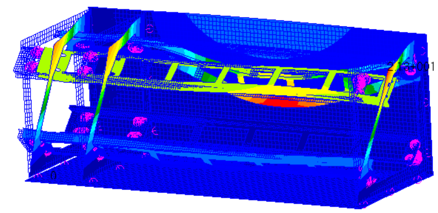
Mode 6 – 257.02 Hz



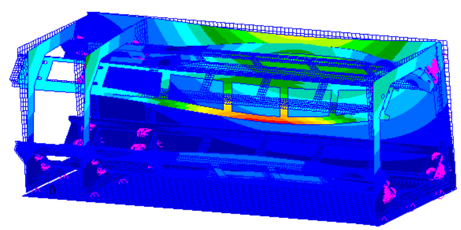
Mode 7 – 262.15 Hz



Mode 8 – 265.1 Hz



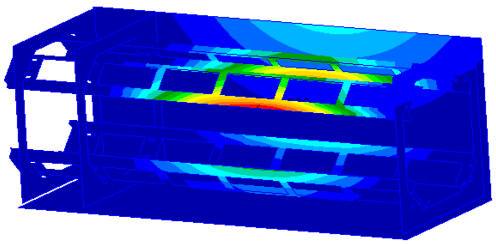
Mode 9 – 272.47 Hz



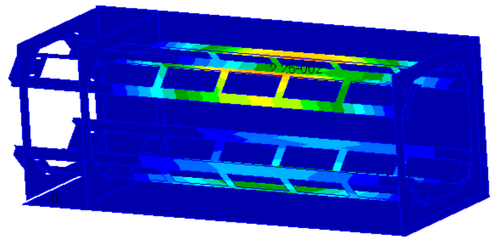
Mode 10 – 287.45 Hz

Test Fixture – Random Vibration Analysis

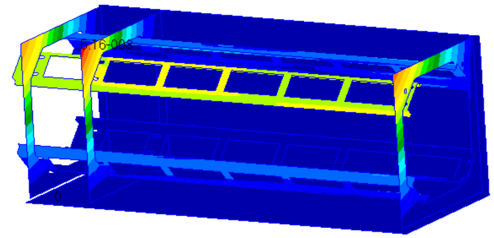
Displacement in Y Direction



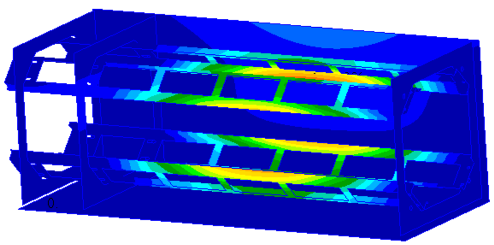
Vibration in X Direction
Displacement in X Direction



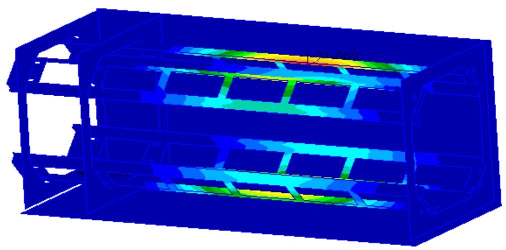
Vibration in X Direction
Displacement in Y Direction



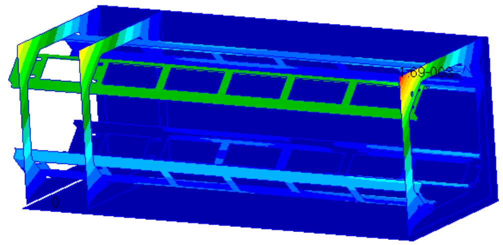
Vibration in X Direction
Displacement in Z Direction



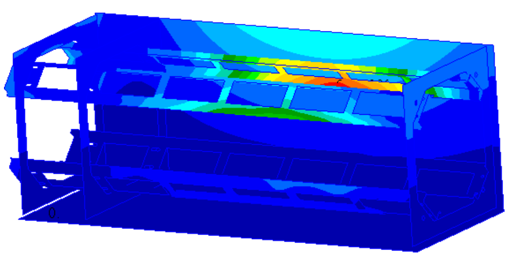
Vibration in Y Direction
Displacement in X Direction



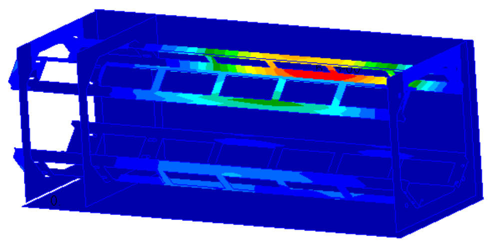
Vibration in Y Direction
Displacement in Y Direction



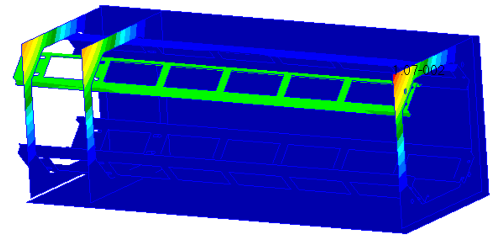
Vibration in Y Direction
Displacement in Z Direction



Vibration in Z Direction
Displacement in X Direction



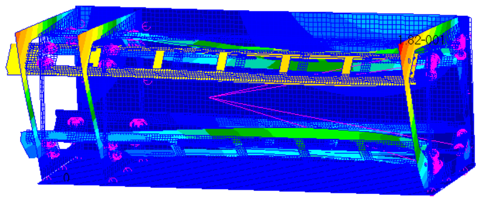
Vibration in Z Direction



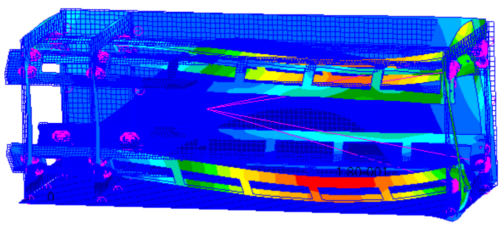
Vibration in Z Direction

Displacement in Z Direction

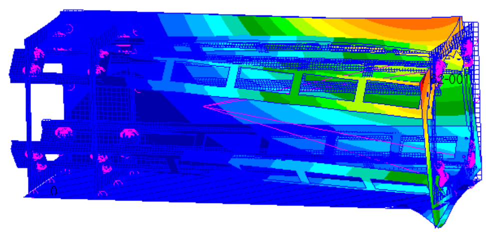
Test Fixture and Lumped Mass – Modal Analysis



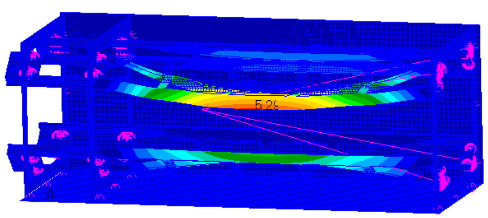
Mode 1 – 7.498 Hz



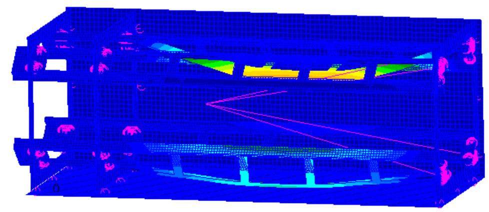
Mode 2 – 11.2 Hz



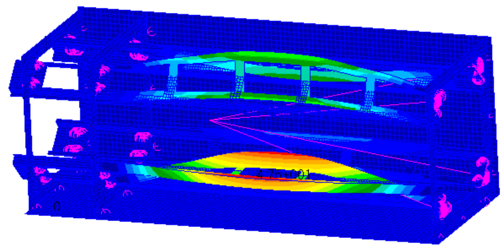
Mode 3 – 37.246 Hz



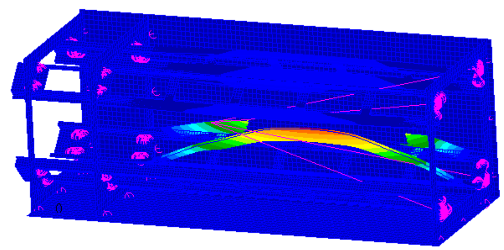
Mode 4 – 216.17 Hz



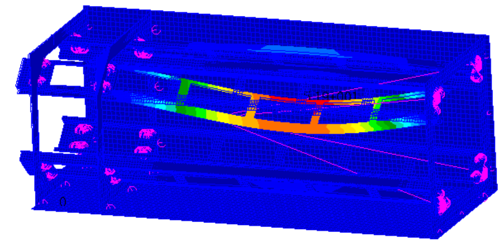
Mode 5 – 217.51 Hz



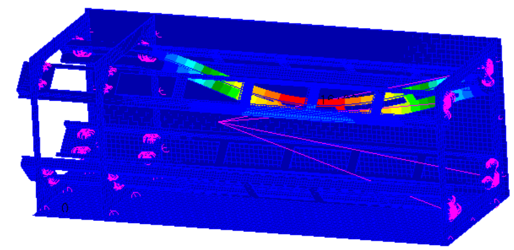
Mode 6 – 218.18 Hz



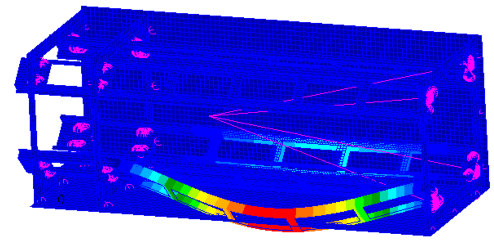
Mode 7 – 218.64 Hz



Mode 8 – 246.31 Hz

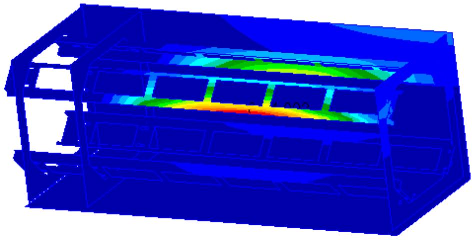


Mode 9 – 262.08 Hz

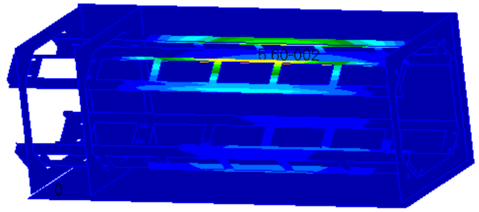


Mode 10 – 264.17 Hz

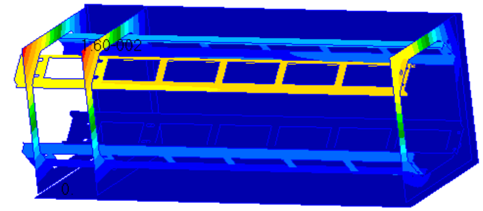
Test Fixture and Lumped Mass – Random Vibration Analysis



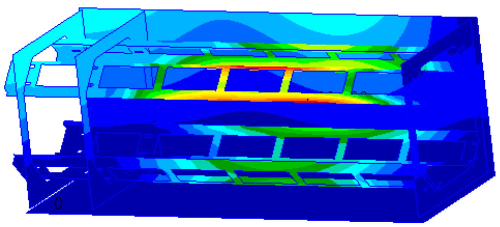
Vibration in X Direction
Displacement in X Direction



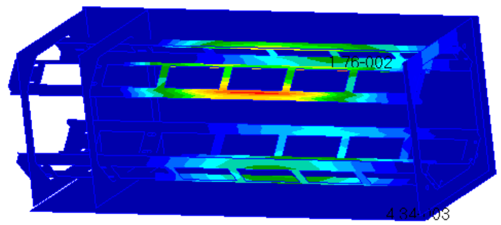
Vibration in X Direction
Displacement in Y Direction



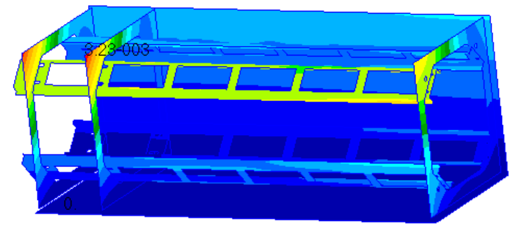
Vibration in X Direction
Displacement in Z Direction



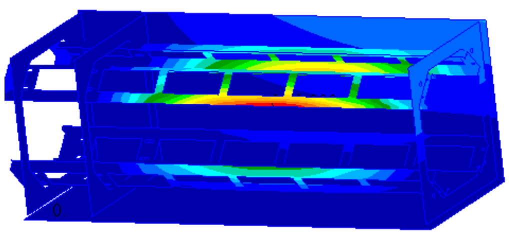
Vibration in Y Direction
Displacement in X Direction



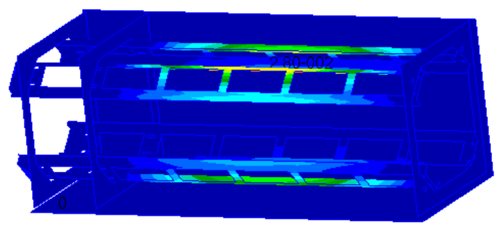
Vibration in Y Direction
Displacement in Y Direction



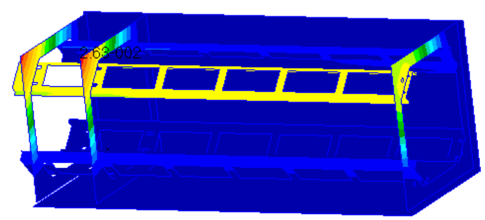
Vibration in Y Direction
Displacement in Z Direction



Vibration in Z Direction
Displacement in X Direction



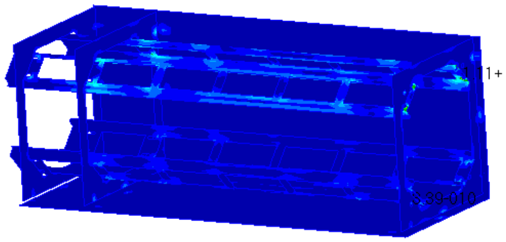
Vibration in Z Direction
Displacement in Y Direction



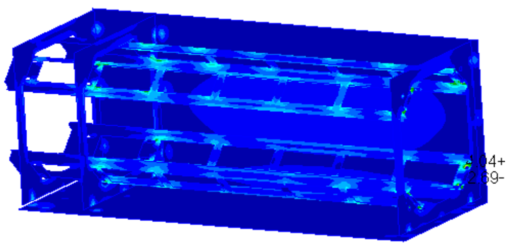
Vibration in Z Direction
Displacement in Z Direction

Von-Mises Stress

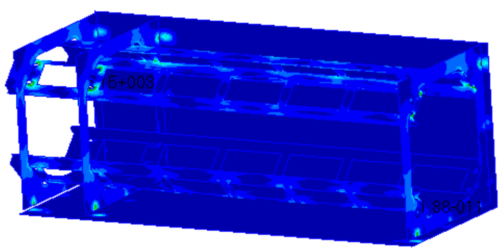
Fixture Alone



X Direction

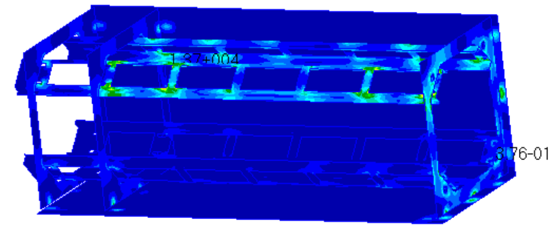


Y Direction

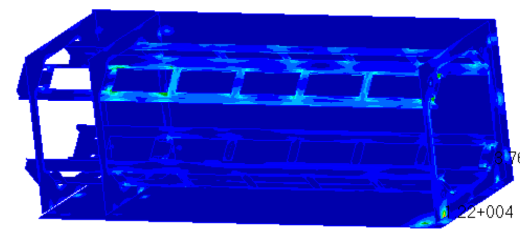


Z Direction

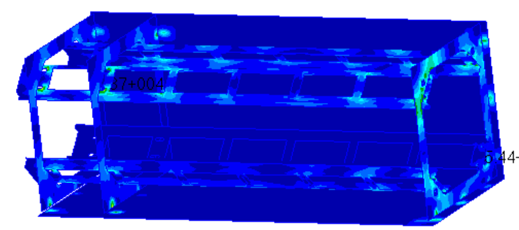
Fixture and Lumped Mass



X Direction

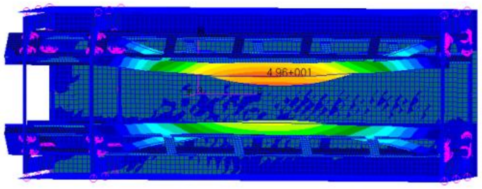


Y Direction

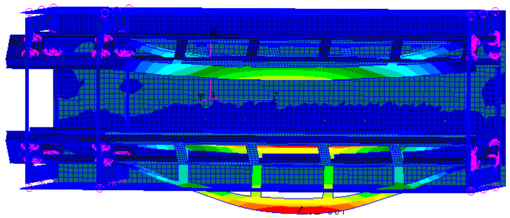


Z Direction

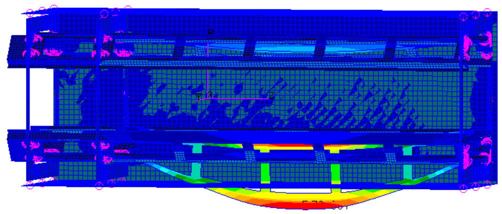
Test Fixture with Third Panel – Modal Analysis



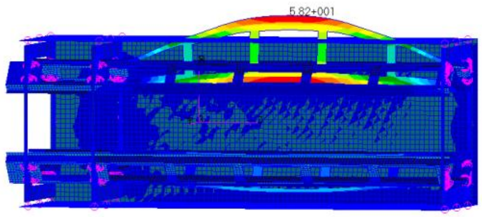
Mode 1 – 214.89 Hz



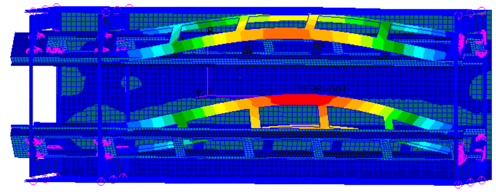
Mode 2 – 215.69 Hz



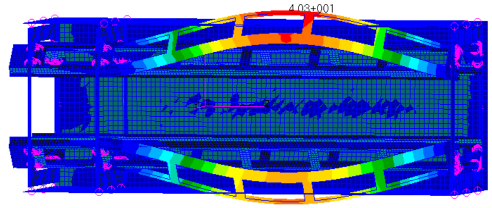
Mode 3 – 217.96 Hz



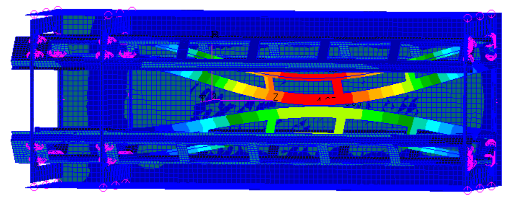
Mode 4 – 218.08 Hz



Mode 5 – 253.83 Hz

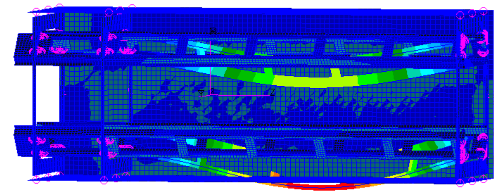


Mode 6 – 261.07 Hz

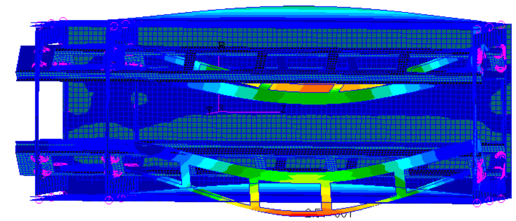


Mode 7 – 265.49 Hz

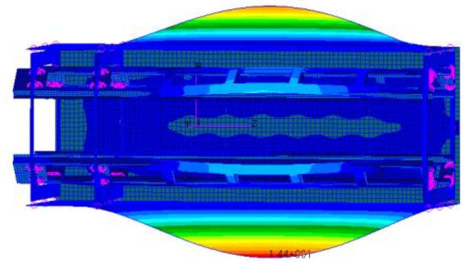
The following screenshots for the fixture with the third panel are taken from the Top View, to better illustrate the displacements.



Mode 8 – 265.54 Hz

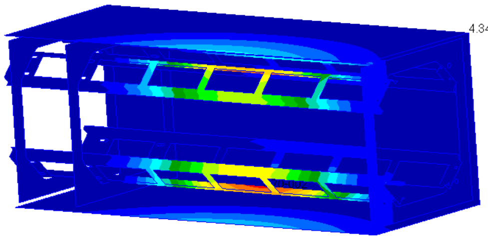


Mode 9 – 281.49 Hz

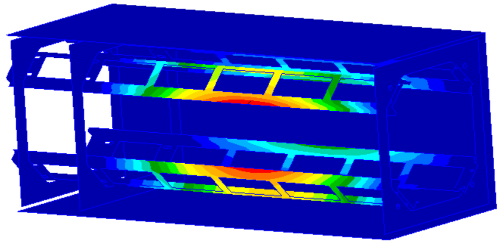


Mode 10 – 357.4 Hz

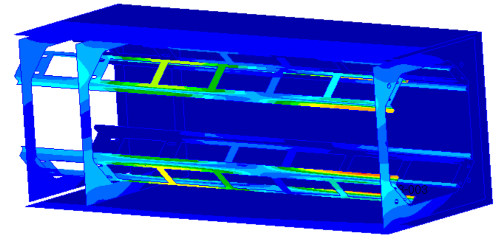
Test Fixture with Third Panel – Random Vibration Analysis



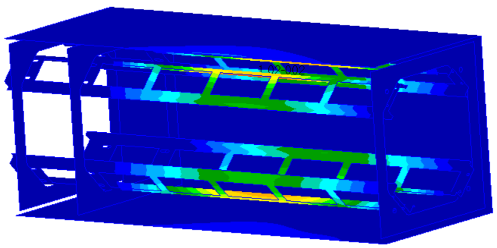
Vibration in X Direction
Displacement in X Direction



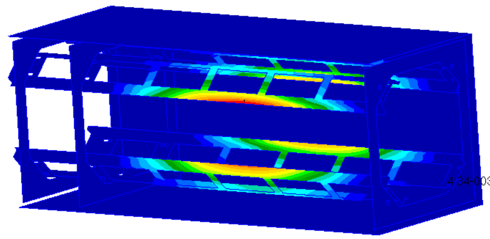
Vibration in X Direction
Displacement in Y Direction



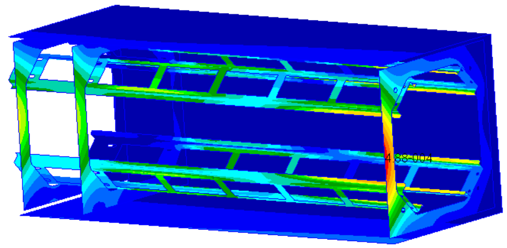
Vibration in X Direction
Displacement in Z Direction



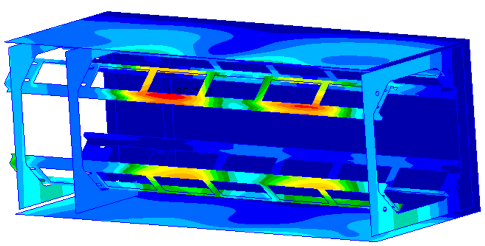
Vibration in Y Direction
Displacement in X Direction



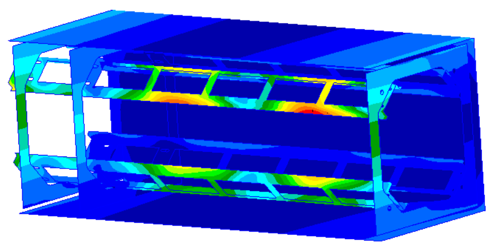
Vibration in Y Direction
Displacement in Y Direction



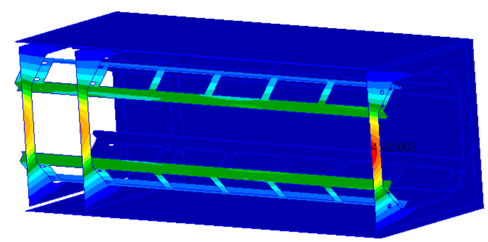
Vibration in Y Direction
Displacement in Z Direction



Vibration in Z Direction
Displacement in X Direction

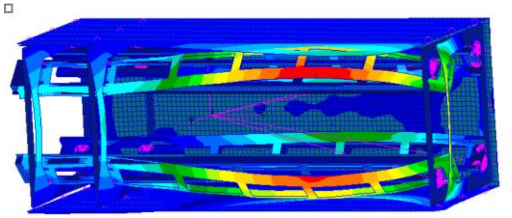


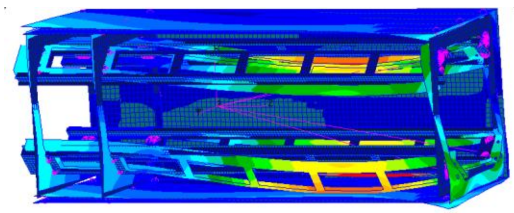
Vibration in Z Direction
Displacement in Y Direction

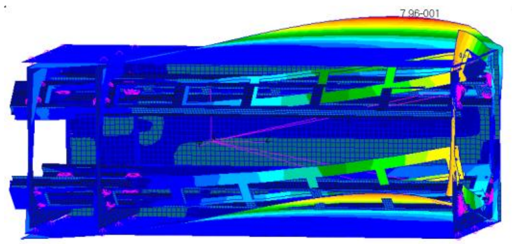


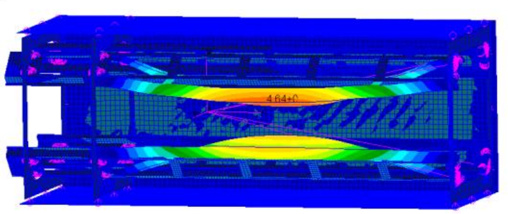
Vibration in Z Direction
Displacement in Z Direction

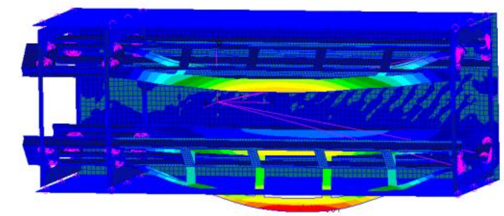
Test Fixture with Third Panel and Lumped Mass – Modal Analysis

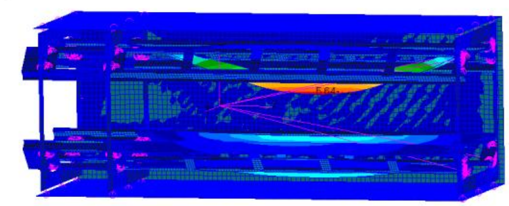

Mode 1 – 12.297 Hz

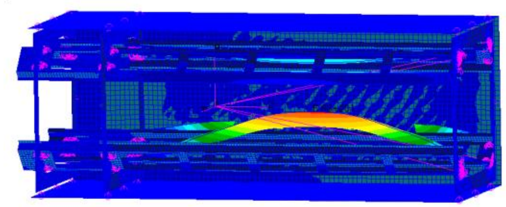

Mode 2 – 14.245 Hz


Mode 3 – 55.931 Hz

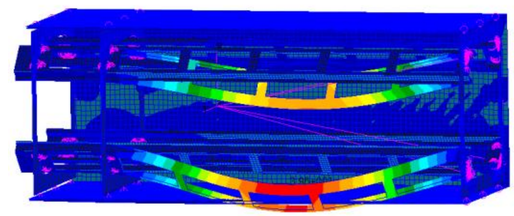

Mode 4 – 217.46 Hz

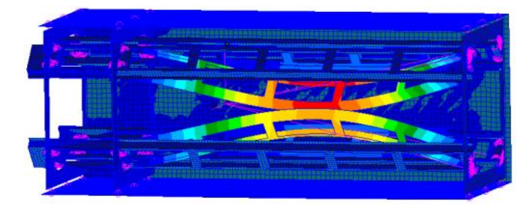

Mode 5 – 218.13 Hz

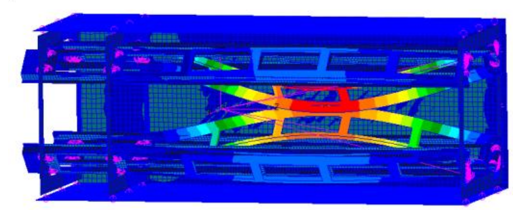

Mode 6 – 218.73 Hz


Mode 7 – 218.77 Hz

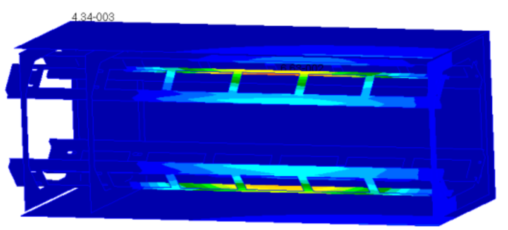
The following screenshots for the fixture with the third panel and mass are taken from the Top View, to better illustrate the displacements.


Mode 8 – 261.85 Hz

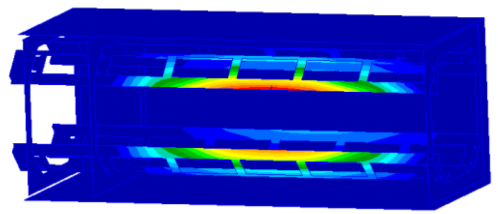

Mode 9 – 263.16 Hz


Mode 10 – 266.61 Hz

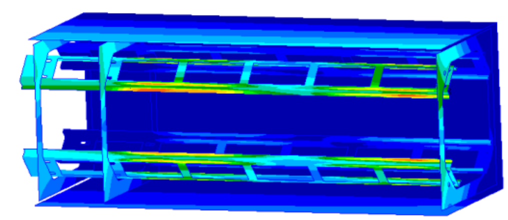
Test Fixture with Third Panel and Lumped Mass – Random Vibration Analysis



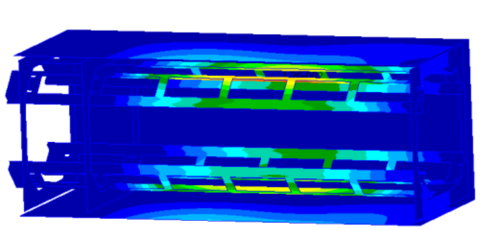
Vibration in X Direction
Displacement in X Direction



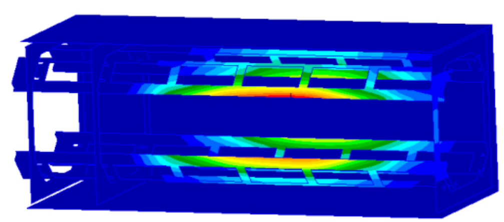
Vibration in X Direction
Displacement in Y Direction



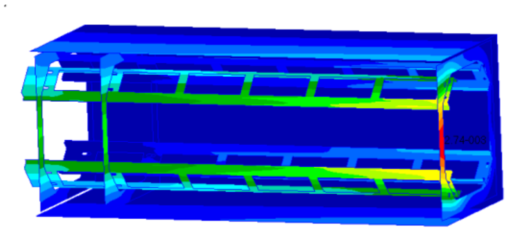
Vibration in X Direction
Displacement in Z Direction



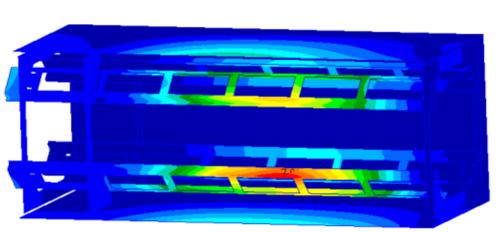
Vibration in Y Direction
Displacement in X Direction



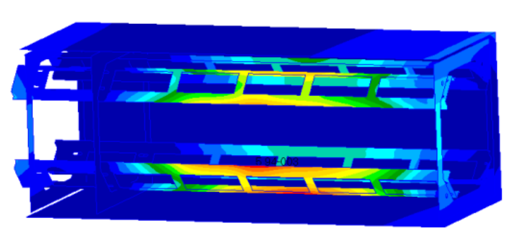
Vibration in Y Direction
Displacement in Y Direction



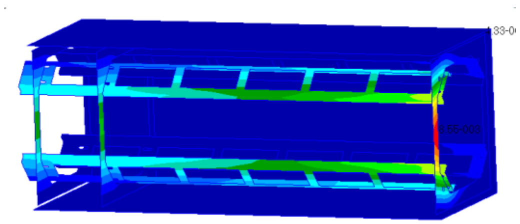
Vibration in Y Direction
Displacement in Z Direction



Vibration in Z Direction
Displacement in X Direction



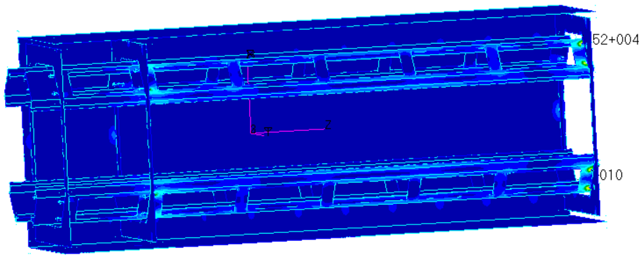
Vibration in Z Direction
Displacement in Y Direction



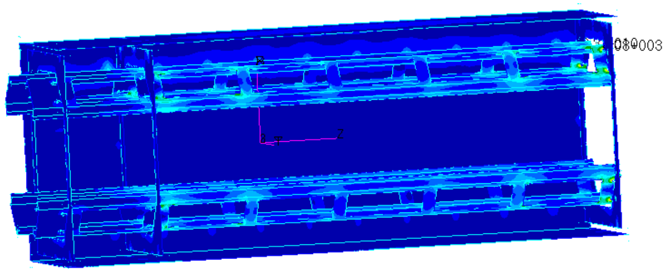
Vibration in Z Direction
Displacement in Z Direction

Von-Mises Stress

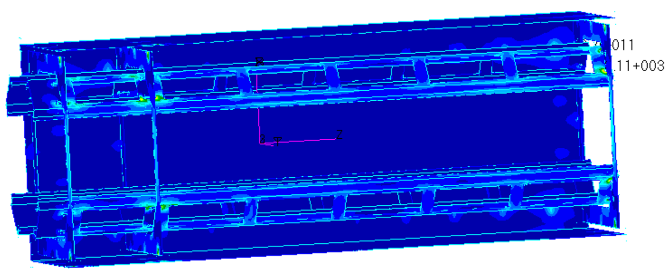
Fixture with Third Panel



X Direction

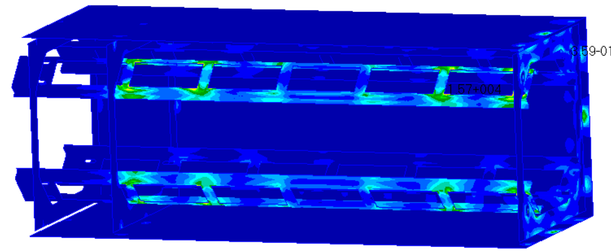


Y Direction

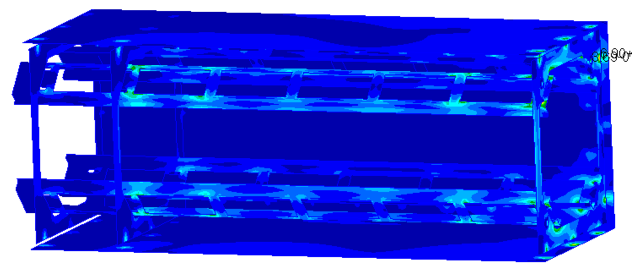


Z Direction

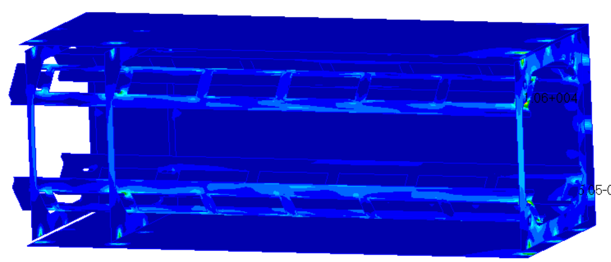
Fixture with Wall and Mass



X Direction



Y Direction



Z Direction