National Aeronautics and
Space Administration

Lyndon B. Johnson Space Center
Houston, Texas 77058

October 2001

Acoustic Noise Control and Analysis Plan
for
Human Research Facility
Payloads and Racks

LS-71011A

PROJECT DOCUMENT APPROVAL SHEET		
DOCUMENT NUMBER LS-71011A	**DATE** 10/26/01	**NO. OF PAGES** 59

TITLE:

Acoustic Noise Control and Analysis Plan for Human Research Facility Payloads and Racks

APPROVED:	EB/E. Strong HRF Rack Integration Manager	Original Signature on File	10/26/01

DATE	PREPARED BY	CHANGE APPROVALS	CHANGE NUMBER

Report Number LS-71011A	**Date** 10/19/01

Acoustic Noise Control and Analysis Plan
for
Human Research Facility
Payloads and Racks

Prepared by: Original Signature on File 10/12/01
E. Azzi Date
Project Manager

Approved: Original Signature on File 10/12/01
A. Milliken Date
Manager, Thermal Analysis Section

Approved: Original Signature on File 10/16/01
L. Kimmel Date
Technical Coordinator

Approved: Original Signature on File 10/17/01
J. Thompson Date
Subtask Manager

Approved: Original Signature on File 10/19/01
J. McDonald Date
Technical Work Plan Manager

Prepared by:

Lockheed Martin Space Operations
Houston, Texas
for
National Aeronautics and Space Administration
Johnson Space Center

REVISION/CHANGE APPROVALS

Date	Change Number	Prepared by	Approved by:		
			Unit Manager	SR&QA Manager	Projects Manager

DOCUMENT NUMBER LS-71011	**DOCUMENT CHANGE/ REVISION LOG**	PAGE 1 OF 1

CHANGE/ REVISION	DATE	DESCRIPTION OF CHANGE	PAGES AFFECTED
Basic	2/22/99	Baseline Issue	
A	10/26/01	Complete Revisions. Revisions includes added explanation of sound power level measurement and description of HRF Racks acoustic abatement hardware.	All

Altered pages must be typed and distributed for insertion.

CONTENTS

CONTENTS (Cont'd)

LIST OF TABLES

LIST OF FIGURES

ACRONYMS AND ABBREVIATIONS

ANSI	American National Standards Institute
3-D	Three Dimensional
CSD	Cooling Stowage Drawer
dB	Decibel
dBA	A-weighted Decibel
EMI	Electromagnetic Interface
ft	foot/feet
GFE	Government Furnished Equipment
HRF	Human Research Facility
Hz	Hertz (cycles per second)
ISO	International Organization for Standardization
ISS	International Space Station
LPM	Lines Per Minute
m	meter
MSE	Modal Strain Energy
NC	Noise Criteria Curve
RIC	Rack Interface Controller
SPL	Sound Pressure Level
SSPCM	Solid State Power Control Module
TLD	Tuned Liquid Dampers
TMD	Tuned Mass Dampers
VEM	Viscoelastic Materials
W	Watt (s)

REFERENCES

1. Vibration Damping, A. D. Nashif, D. I. G. Jones, and J. P. Henderson, John Wiley and Sons, New York, 1985.

2. Damping in Spacecraft Structures - A Review of Research and Hardware Development in European Space Programs, The Role of Damping in Vibration and Noise Control, DE-Volume 5, I. Kolsch, H. Baier, and E. Hilbrandt, ASME Publication, 1987.

3. Shock and Vibration Handbook, Second Edition, C. M. Harris and C. E. Crede, McGraw-Hill Book Company, New York, 1976.

4. Products to Control Shock, Vibration and Noise, Barry Controls, Bulletin 10EM1, 1991.

5. Use of Tuned Viscoelastic Dampers in Aerospace Structures, Proc., D. I. G. Jones, J. P. Henderson, G. H. Bruns, 13th Annual Air Force Science and Engineering Symposium, University of Tennessee, September, 1966.

6. Tuned Liquid Dampers for Suppressing Horizontal Motion of Structures, Proceedings of Damping, Volume II, Y. Fujino, B. M. Pacheco, L. Sun, and P. Chaiseri, Flight Dynamics Laboratory, AFWAL, Wright-Patterson Air Force Base, Ohio, 1989.

7. A New Class of Fluid-Loop Dampers, Proceedings of Damping, Volume II, A. Baz, L. Gunusel, and J. Fedor, Flight Dynamics Laboratory, AFWAL, Wright-Patterson Air Force Base, Ohio, 1989.

8. The Design and Effectiveness of Impact Dampers for Space Applications, The Role of Damping in Vibration and Noise Control, DE-Volume 5, P. J. Torvik and W. Gibson, ASME Publication, 1987.

9. Application of the Component Element Method to the Impact Damped Simple Harmonic Oscillation, Proceedings of Damping, Volume III, C. M. North and R. E. Jones, Flight Dynamics Laboratory, AFWAL, Wright-Patterson Air Force Base, Ohio, 1989.

10. The Investigation of Large Space Structure Passive Electrodynamics Dampers, Proceedings of Damping, Volume II, R. Stettner and P. Mlakar, Flight Dynamics Laboratory, AFWAL, Wright-Patterson Air Force Base, Ohio, 1991.

11. Current-Based Vibration Damping for Aerospace Structures, Proceeding of Damping, Volume II, J. H. Goldie, Flight Dynamics Laboratory, AFWAL, Wright-Patterson Air Force Base, Ohio, 1991.

12. Damping of Structural Vibration with Piezoelectric Materials and Passive Electrical Networks, Proceedings of Dampings, Volume III, N. W. Hagwood and A. Flotow, Flight Dynamics Laboratory, AFWAL, Wright-Patterson Air Force Base, Ohio, 1989.

13. An Analytical Approach to Designing Friction Dampers in Turbomachinery Blading, Proceedings of Damping, Volume III, J. Panovsky, D. G. Handley, and R. A. McKay, Flight Dynamics Laboratory, AFWAL, Wright-Patterson Air Force Base, Ohio, 1991.

14. An Integrated Approach to Friction Damper Design, The Role of Damping in Vibration and Noise Control, DE-Volume 5, T. M. Cameron, J. H. Griffin, R. E. Kielb, and T. M. Hoosac, ASME Publication, 1987.

15. Theory and Practice of Fluid Dampers in Machine Tools, Advances in Machine Tool Design and Research, J. Peters and P. VanHerck, Proceeding of 100th International M.T.D.R. Conference, University of Manchester, September, 1969.

16. Passive Damping Impact on Air Force System Maintenance Costs, The Role of Damping in Vibration and Noise Control, DE-Volume 5, M. L. Drake, ASME Publication, 1987.

17. Practical Stand-Off Damping Treatment for Sheet Metal, Proceeding of Damping, Volume II, M. L. Parin, L. Rogers, Y. I. Moon, and M. Falugi, Flight Dynamics Laboratory, AFWAL, Wright-Patterson Air Force Base, Ohio, 1989.

18. Improving Single-Constraining Layer Damping Treatment by Sectioning the Constraining Layer, The Role of Damping in Vibration and Noise Control, DE-Volume 5, G. Kress, ASME Publication, 1987.

19. Design of Constrained Layer Treatment for Broad Temperatures Damping, Shock and Vibration Bulletin, Number 44, Part 5, D. I. G. Jones, August, 1975.

20. Multiple Band Surface Treatments for High Damping, Air Force Materials Laboratory Report, AFML-TR-65-269, B. J. Lazan, A. F. Metherell, and G. Sokol, Wright-Patterson Air Force Base, Ohio, 1965.

21. Spacelab Life Sciences 2 Acoustic Analysis Report, JSC-26218, E. Azzi, Martin Marietta, Houston, Texas, 5/93.

22. Program Requirements Document for The Human Research Facility, LS-71000, NASA, 9/95.

1.0 INTRODUCTION

The Human Research Facility (HRF) is a facility class payload that consists of a suite of generic human life sciences hardware needed to support a multidisciplinary research program that encompasses basic, applied, and operations research. The HRF will include equipment to support research to understand the effects of weightlessness and the space environment on human systems and to develop, where appropriate, methods to counteract these effects to ensure safe and efficient crew operations.

Basic research and clinical investigations from both the intramural and extramural communities, as well as investigations from other federal agencies, and the international community will all be conducted using the HRF. All hardware elements to be used during the conduct of human research on International Space Station (ISS) may not necessarily be included in the HRF racks. The ability to conduct thorough, multidisciplinary investigations will depend on the interaction of the HRF with ISS systems, the Crew Health Care System program, and the Space Station Biological Research Project including the Centrifuge Facility, as well as other hardware provided by the international partners. In addition, the HRF subsystems and experiment packages will be modular in design so that the HRF can be configured to meet many sets of research objectives for the duration of the ISS.

There has been increased emphasis on the acoustic assessment of the individual payloads and the ISS habitable environment. The acoustic environment prevailing in the Space Station has significant impact on crew comfort and crew performance.

Historically some payload developers have disregarded noise specifications to the point of simply measuring the resulting product noise and then requesting a waiver. Traditionally this happens after the flight hardware has been built and tested and when all the money and schedule have been depleted. At this point there are only two likely options: demanifest the payload or approve a waiver. The NASA has recognized that quieter hardware will result if the hardware developer takes the following steps: be attentive to acoustic requirements throughout the length of the program; be selective in choosing quiet prime movers (fans, pumps, etc.), establish and implement noise control plans; and do noise testing from the onset of design and continue it periodically throughout the development stages.

The noise level generated by each payload and the integrated payload rack will be measured under controlled conditions. The total Sound Pressure Level (SPL) of the HRF rack (from any set of noise sources that will be operated simultaneously in flight) shall not exceed the continuous and/or intermittent requirements when measured at 0.6 meter from any of the equipment.

The composite noise level in the HRF EXPRESS Rack will be determined based on measured equipment noise data, equipment timeline, and equipment location within the rack. An analysis software will be used to generate this noise data including the direct, reverberant, and subsystem generated noise levels for each frequency band. This document provides a summary of the general analytical approach. This approach has been successfully used in the Spacelab program and excellent correlation has been found between its analytical procedure and measured data.

This document provides a general outline of the acoustic control plan and analysis approach. Section 2 describes the ISS acoustic design requirement. Section 3, introduces the general theory of sound propagation and analytical formulations to

compute SPLs. Section 4 describes the acoustic analysis approach which will be used to perform the rack level, component level analysis process, and component level sub-allocation procedure. Section 5 investigates the methods of noise control. A summary is provided in Section 6. Appendix A describes the noise measurement procedures for ISS Government Furnished Equipment (GFE) equipment and payloads. This appendix was provided by JSC-SP3. The
document, JSC-26218, Spacelab Life Sciences 2 Acoustic Analysis Report [21], is an example of a typical acoustics analysis involving several sources located in different locations in an enclosed volume operated intermittently or continuously according to a timeline.

In case of conflicts between this document and LS-71000, HRF Program Requirements Document [22], LS-71000 takes precedence.

2.0 INTERNATIONAL SPACE STATION (ISS) ACOUSTIC DESIGN REQUIREMENTS FOR PAYLOADS (P/Ls)

The following are specific requirements that shall be implemented for ISS Payloads.

2.1 ACOUSTIC NOISE DEFINITIONS

2.1.1 Significant Noise Source

A significant noise source is any individual item of equipment, or group of equipment items which collectively function as an operating system, that generates an A-weighted SPL equal to or in excess of 37 decibels (dBs), A-weighted (dBA), measured at 0.6 meter distance from the noisiest part of the equipment.

2.1.2 Continuous Noise Source

A significant noise source which exists for a cumulative total of 8 hours or more in any 24-hour period is considered a continuous noise source.

2.1.3 Intermittent Noise Source

A significant noise source which exists for a cumulative total of less than 8 hours in a 24-hour period is considered an intermittent noise source.

2.1.4 Acoustic Reference

All SPLs in decibels are referenced to 20 micropascals.

2.2 ACOUSTIC NOISE LIMITS

The acoustic limits that shall be utilized are provided in the tables which follow. The limits apply to both an integrated payload rack of equipment or to a payload that is independently operated outside a rack. The integrated rack configuration limits include any adjunct equipment such as external computers or fans which are added in support of the rack system. These limits apply to measurements taken at 0.6 meters distance from the loudest part of the individual equipment. Rack level tests shall be performed at the loudest location 0.6 meters inboard from the rack surface. In areas where the rack surfaces are exposed directly to the habitable volume (other than the inboard face), measurements shall be taken at these areas to ensure they are included in determining the loudest location. An example of this would be where the side of a rack is adjacent to a passageway or a window. This assumes that noise radiating from the rack is sufficiently contained by the rack structure or other close-out panels. Actual flight equipment (each serialized unit) shall be used for flight acceptance testing, even though prototype or qualification units may have been tested earlier. These levels shall not be exceeded for the following conditions: when the equipment is operating in the loudest mode of operation that can occur on orbit under nominal crew or hardware operation circumstances: during payload setup operations, or during operations where doors/panels are opened or removed.

2.2.1 Continuous Noise Limits

A payload facility/system or group of payloads manifested at a rack level which generates continuous noise levels, shall not exceed the limits provided in Table 2.1, for all octave bands (NC-40 equivalent). These levels apply to an integrated rack (within or interfacing with the crew habitable volume) that is operated in the noisiest configuration or operating mode. Individual items of active hardware contained within the rack are apportioned in Section 4.0 to ensure the limits in the tables are not exceeded by the whole rack.

TABLE 2.1. NOISE LIMITS FOR CONTINUOUS PAYLOADS

Frequency Band Hz	63	125	250	500	1000	2000	4000	8000
Total Rack or External Payload SPL dB	64	56	50	45	41	39	38	37

Measurements shall be taken at 0.6 meters.

2.2.2 Intermittent Noise

If a rack is classified as an intermittent noise source (see Paragraph 2.1.3), then the rack shall comply with the limits provided in Table 2.2. For design or manifesting cases where the rack contains multiple noise sources, the rack must meet limits in Table 2.2 as an integrated rack. The allowable noise for each source in the rack is sub-allocated in Section 4.0 to ensure total rack compliance.

The maximum A-weighted SPL emitted by any independently operated intermittent noise source outside of a payload rack that is within or interfacing with the crew habitable volume shall not exceed the limits specified in Table 2.2.

TABLE 2.2. NOISE LIMITS FOR INTERMITTENT PAYLOADS

Rack Noise Limits*	
Maximum Rack Noise Duration†	Total Rack A-weighted SPL (dBA)
8 Hours	≤ 49
7 Hours	≤ 50
6 Hours	≤ 51
5 Hours	≤ 52
4 Hours	≤ 54
3 Hours	≤ 57
2 Hours	≤ 60
1 Hour	≤ 65
30 Minutes	≤ 69
15 Minutes	≤ 72
5 Minutes	≤ 76
2 Minutes	≤ 78
1 Minute	≤ 79
Not Allowed	80 or above

*Measurements shall be taken at 0.6 meters, per above reference.
†Per 24-hour Period

2.2.3 Continuous Noise Sources with Intermittent Noise Features

Continuous noise sources which exhibit intermittent acoustical characteristics must meet both the continuous noise specification and the intermittent limits of Paragraphs 2.2.1 and 2.2.2. The intermittent noise characteristics must be quantified in terms of (1) when the intermittent sound occurs, (2) duration, (3) a projected mission timeline(s) for items (1) and (2), and (4) maximum A-weighted SPL measured at 0.6 meter distance from the loudest part of the equipment. These data shall be submitted to the ISS Acoustic Working Group.

2.4 ACOUSTICAL VERIFICATION MEASUREMENTS

Acoustic measurements of continuous and intermittent equipment shall be obtained and reported in accordance with procedures specified in Appendix A.

2.5 SOUND POWER READINGS ON PAYLOADS

The requirements which have been previously designated have been stated in terms of SPL. Payloads (P/Ls) shall comply with these SPLs, but additional acoustic measurement information is required for mission planning and overall acoustic analysis purposes. Sound power measurements are required when continuous noise levels of any octave band determined by sound pressure exceeds the NC-40 limit, and the extra data will be used to process proposed exceptions/deviations/waivers. Sound power measurements shall be performed in accordance with the appropriate American National Standards Institute (ANSI) S1 Standards
on Acoustics. Reference the following ANSI Standards:

ISO 9614-2, Acoustics – Determination of Sound Power Levels of Noise Sources using Sound Intensity – Part 2: Measuring by Scanning, (1996).

ANSI S1.4, Specification for Sound Level Meters Amendment S1.4A-1985 ASA 47 R (1994).

ANSI S1.11, Specification for Octave-Band and Fractional-Octave-Band Analog and Digital Filters; ASA 65-1986 R (1993).

ANSI S12.12 - 1992, Engineering Method for the Determination of Sound Power Levels of Noise Sources using Sound Intensity ASA 104.

ANSI S12.23 -1989, (R1996), Method for the Designation of Sound Power Emitted by Machinery and Equipment.

ANSI S12.31 - 1990 (R1996), Precision Methods for Determination of Sound Power Levels of Broad-band Noise Sources in Reverberation Rooms.

ANSI S12.32- 1990 (R1996), Precision Methods for the Determination of Sound Power Levels of Discrete Frequency and Narrow-band Noise Sources in Reverberation Rooms.

ANSI S12.33 -1990, Engineering Methods for the Determination of Sound Power Levels of Noise Sources in a Special Reverberation Test Room.

ANSI S12.34 - 1988 (R1993), Engineering Methods for the Determination Sound Power Levels of Noise Sources for Essentially Free-field Conditions over a Reflecting Plane.

ANSI S12.35 - 1990 (R1996), Precision Methods for the Determination of Sound Power Levels of Noise Sources in Anechoic and Hemi-anechoic Rooms.

ANSI S12.36 - 1990, Survey Methods for the Determination of Sound Power Levels of Noise Sources.

Equipment or facilities associated with obtaining this data is available at some NASA sites or from noise consultants.

3.0 THEORY

This section describes the general theory of sound propagation and analytical formulations to compute SPLs. The first sound waves to reach the receiver are those that are directly radiated when the sound source is turned on. Then, the reflected waves start reaching the receiver (Figure 3-1). These reflected waves reach the receiver at different times because they travel many paths with different path lengths. This results in the period required for the sound level to reach its steady state value.

When a sound wave encounters an object, part of the sound energy is reflected and another is absorbed. To reduce the sound levels caused by a reverberant sound field in an enclosed area, sound-absorbing materials are used. These materials convert some of the acoustical energy into a different form of energy, usually heat, which is absorbed by the surface.

The sound absorption coefficient, α, of the surface is defined as $\alpha = \frac{W_a}{W_i}$ where W_a is the sound energy absorbed and W_i is the sound energy incident upon the surface. When an area is composed of surfaces with different absorption coefficients, the average absorption coefficient $\overline{\alpha}$ is obtained from

$$\overline{\alpha} = \frac{\alpha_1 S_1 + \alpha_2 S_2 + \alpha_3 S_3 + ... + \alpha_n S_n}{S_1 + S_2 + S_3 + ... + S_n}$$

where α_1, α_2 ...α_n are the absorption coefficients for the respective surface areas S_1, S_2, ... S_n.

The SPL is determined from the following relationship:

$$L_p = L_w + 10 \log_{10}\left(\frac{4}{R} + \frac{Q}{4\pi r^2}\right) + 10 \log^{10}\left(\frac{\rho_o C W_{ref}}{P^2{}_{ref}}\right) \quad (3.1)$$

Where,

P_{ref} = 20×10^{-6} P_a (reference pressure)

W_{ref} = 10^{-12} Watt (reference power)

ρ_o = reference air density

C = reference speed of sound

Q = directivity factor (Figure 3-2)

Q = 1 for whole space
Q = 2 for half space
Q = 4 for quarter space
Q = 8 for eight space

L_w = sound power level of source defined as $10 \log_{10} \frac{W}{Wref}$

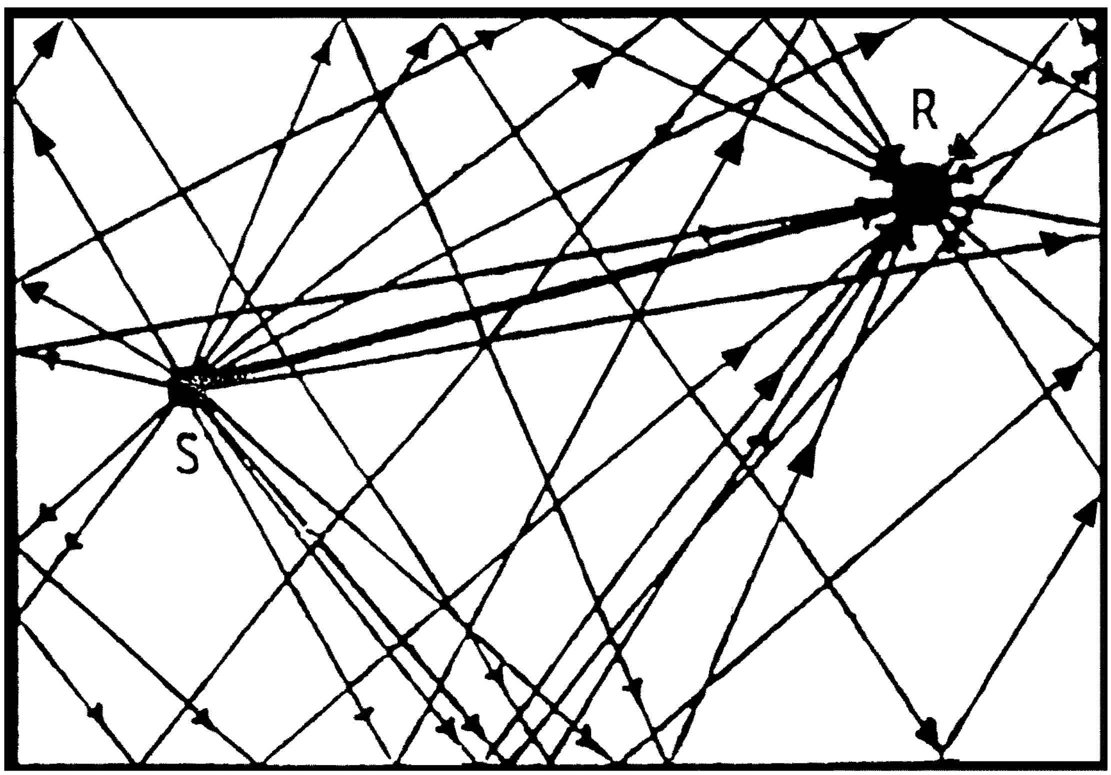
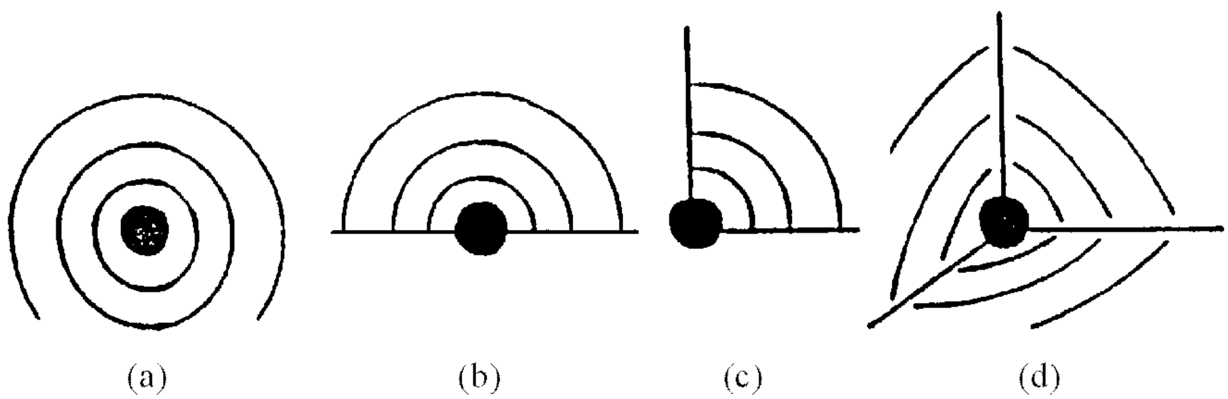


Figure 3-1. Direct and Reverberant Sound Fields in an Enclosed Area



a) Whole space, b) Half space,
c) Quarter space, d) Eighth space

Q	=	1	for whole space
Q	=	2	for half space
Q	=	3	for quarter space
Q	=	4	for eighth space

Figure 3-2. Directivity Factor Q

Where,

W = sound power

R = room constant defined as $R = \dfrac{S\bar{a}}{1-\bar{a}}$

Where

S = surface area

$\bar{\alpha}$ = average surface absorption coefficient

r = distance from center of sound source

Direct and reverberant fields in Equation 3.1 are represented by:

$$\text{Direct} = \frac{Q}{4\pi r^2}$$

$$\text{Reverberant} = \frac{4}{R}$$

Sound level meter is usually used to measure the SPL. Therefore, L_w could be determined by using equation:

$$L_w = L_{p_M} - 10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi r^2}\right)_T - 10\log_{10}\left(\frac{\rho_o C W_{ref}}{P^2_{ref}}\right) \tag{3.2}$$

Where

L_{p_M} = measured SPL at fixed distance

$(\ )_T$ = parameter for the testing room

By combining Equations 3.1 and 3.2, we obtain:

$$Lp = L_{p_M} + 10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi r^2}\right) -10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi r^2}\right)_T \tag{3.3}$$

The combined noise level is given by:

$$\text{Total noise} = 10[\log_{10}(10^{L_{P1}}/10 + 10^{L_{P2}}/10 + 10^{L_{P3}}/10 + \ldots 10^{L_{Pn}}/10), \tag{3.4}$$

where L_{p_1}, L_{p_2}, ... L_{p_n} are individual SPLs at the same frequency.

The overall A weighted level is calculated as follows:

$$\text{A-wtd} = 10 \log_{10} \big(10^{((L63 - 26.2)/10)} + 10^{((L125 - 16.1)/10)} + 10^{((L250 - 8.6)/10)} + 10^{((L500 - 3.2)/10)} + 10^{(L1000/10)} + 10^{((L2000 + 1.2)/10)} + 10^{((L4000 + 1)/10)} + 10^{((L8000 - 1.1)/10)} \big)$$

It can be noted that, within regions very close to noise sources in an enclosure, the direct sound field, $\frac{Q}{4\pi r^2}$, dominates. The only way to significantly reduce the SPL in this area is to reduce the source sound power. The reverberant sound field, $\frac{4}{R}$, dominates in a region far from the noise source in an enclosed area. The SPL can be altered by changing the enclosure constant. This can be achieved by adding or taking away sound absorption in the enclosure.

However, in a room containing noise sources such as machinery, the treatment of the walls with acoustics material often does not reduce the SPL significantly; particularly when the direct sound field predominates over the reverberant sound field.

4.0 ACOUSTIC ANALYSIS APPROACH

Acoustics analyses are critical to the design process. Designs are continually analyzed to verify compliance with the ISS requirements and constraints. Payload and rack level system analyses are conducted in conjunction with the hardware development to verify compliances. Analysis software will be used to perform the acoustical analysis. This software has been successfully used in the Spacelab and HRF Rack 1 rack program, and excellent correlation has been found between its analytical results and measured data. The analyses are carried out at the component and integrated rack level.

4.1 RACK LEVEL ACOUSTIC ANALYSIS

The composite noise level in the HRF EXPRESS Rack will be determined based on measured equipment noise data, equipment timeline, and equipment location within the EXPRESS Rack. The analyses will include direct and reverberant fields. Spatial distribution of the noise levels will be represented. The purpose of the analysis is to verify The total SPL of the HRF rack (from any set of noise sources that will be operated simultaneously in flight) shall not exceed the requirements specified in Section 2 when measured at 0.6 meter in front of the unit. One of the design options that can be considered for the rack acoustical management system is to develop the optimum configurations for the EXPRESS Rack by performing trade studies. These studies involve determining the best location for each hardware in the EXPRESS Rack based on their SPL.

4.2 COMPONENT LEVEL SUB ALLOCATION

The total SPL of the HRF rack (from any set of noise sources that will be operated simultaneously in flight) shall not exceed the requirements specified in Section 2 when measured at 0.6 meter from any of the equipment. This requirement also applies for each of the deployed payloads. The SPL meter can be placed anywhere at 0.6 meter in front of the rack. The meter measures the combined SPL of the operating payloads in the rack. Contribution of each payload to the combined SPL depends on its location and its SPL. Limitations on each individual noise source are defined to provide a guideline for the Project Engineer Developer. The maximum allowable SPL generated by individual noise sources can be determined based on the overall rack noise limitations, number of operating units, and location of the units in the rack. The procedures that can be used to determine the maximum allowable SPL of each unit is described as follows.

The combined noise level is given as:

$$\text{Total noise} = Lp_T = 10\left[\log_{10}\left(10^{(Lp_1/10)} + 10^{(Lp_2/10)}...+10^{(Lp_n/10)}\right)\right] \qquad (4.1)$$

Where,

$Lp_1, Lp_2, ...Lp_n$ are the individual SPLs of various units at the same frequency.

The SPL is determined from the following equation.

$$Lp = Lp_M + 10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi r^2}\right)_M - 10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi r^2}\right)_T \qquad (4.2)$$

Variables are defined in Section 2.0.

Using Equation 4.2:

$$Lp_1 = Lp_{M1} + 10 \log_{10}\left(\frac{4}{R_1} + \frac{Q_1}{4\pi r^2{}_1}\right)_M - 10 \log_{10}\left(\frac{4}{R_1} + \frac{Q_1}{4\pi r^2{}_1}\right)_T$$

$$Lp_2 = Lp_{M2} + 10 \log_{10}\left(\frac{4}{R_2} + \frac{Q_2}{4\pi r^2{}_2}\right)_M - 10 \log_{10}\left(\frac{4}{R_2} + \frac{Q_2}{4\pi r^2{}_2}\right)_T$$

through

$$Lp_n = Lp_{Mn} + 10 \log_{10}\left(\frac{4}{R_n} + \frac{Q_n}{4\pi r^2{}_n}\right)_M - 10 \log_{10}\left(\frac{4}{R_n} + \frac{Q_n}{4\pi r^2{}_n}\right)_T$$

Where,

$Lp_1, Lp_2, \ldots Lp_n$ are the calculated SPLs of each unit in one operational environment (at the sound meter that is measuring the SPL of the rack at 0.6 meter from the front surface of the rack).

$Lp_{M1}, Lp_{M2}, Lp_{M3}, \ldots Lp_{Mn}$ are the Measured SPLs of each unit at a distance of 0.6 meter from the front surface of the unit in the tested environment.

$(\ )_T$ = parameter of the room where the SPLs ($Lp_{M1}, Lp_{M2}, \ldots Lp_{Mn}$) of each unit are tested.

$(\ )_M$ = parameter of the room where the SPLs ($Lp_1, Lp_2, \ldots Lp_n$) of the integrated rack are calculated.

Assume the following:

- The SPL allocations for each individual noise source are evenly distributed. Therefore,

$$Lp_{M1} = Lp_{M2} = Lp_{M3} = \ldots.\ Lp_{Mn} = Lp_M$$

- SPLs of all units are tested in the same environment, therefore,

$$\left(\frac{4}{R_1} + \frac{Q_1}{4\pi r^2{}_1}\right)_T = \left(\frac{4}{R_2} + \frac{Q_2}{4\pi r^2{}_2}\right)_T = \ldots \left(\frac{4}{R_n} + \frac{Q_n}{4\pi r^2{}_n}\right) = \left(\frac{4}{R} + \frac{Q}{4\pi r^2}\right)$$

The room constant, R, depends on the surface area of the room and absorption coefficient of the walls. The directivity factor, Q, depends on the position of the noise source. Assume all units are measured in similar environment. Therefore,

$$\left(\frac{4}{R_1} + \frac{Q_1}{4\pi}\right)_M = \left(\frac{4}{R_2} + \frac{Q_2}{4\pi}\right)_M = \ldots \left(\frac{4}{R_n} + \frac{Q}{4\pi}\right)_M = \left(\frac{4}{R} + \frac{Q}{4\pi}\right)$$

Based on the above assumptions:

$$\left.\begin{aligned}
Lp_1 &= Lp_M + 10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi {r^2}_1}\right)_M - 10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi r^2}\right)_T \\
Lp_2 &= Lp_M + 10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi {r^2}_2}\right)_M - 10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi r^2}\right)_T \\
&\text{through} \\
Lp_n &= Lp_M + 10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi {r^2}_n}\right)_M - 10\log_{10}\left(\frac{4}{R}+\frac{Q}{4\pi r^2}\right)_T
\end{aligned}\right\} \qquad (4.3)$$

The following parameters are defined based on room conditions and the position of the noise sources.

$$\left(\frac{4}{R}+\frac{Q}{4pr^2_j}\right)_M \; ; \; \left(\frac{4}{R}+\frac{Q}{4\pi r^2}\right)_T$$

where,

J = 1, 2, ... n

- $(r_1, r_2, ..., r_n)_M$ are the distances from the units to the SPL meter that is measuring the total SPL of the payload rack at 0.6 meter from the noisiest front surface.
- r is equal to 0.6 meter. This is the distance that will be used to determine the SPL of each unit, Lp_M.

Equations 4.1 and 4.3 are used to determine the allocated SPLs, Lp_M, generated by each noise source at a distance r (r = 0.6 meter). These SPLs will be the maximum allowable generated by the individual noise source.

Since the HRF payloads and their locations in the rack are likely to change, an approximation can be used to determine the maximum allowable SPL generated by individual noise sources. This approximation is based on the assumption that the SPLs of all the units are equally contributing to the overall SPL of the payload rack. The following procedure is used to determine the SPL allocated for each individual noise source at a distance of 0.6 meter from the front surface. These SPLs are based on the NC-40 constraint for the rack and the number of noise sources in the rack (n = 10, 8, and 5).

The combined noise level is given by:

$$\text{Total noise} = Lp_T = 10\left[\log_{10}\left(10^{(Lp_1/10)} + 10^{(Lp_2/10)} ...+10^{(Lp_n/10)}\right)\right]$$

Assume n units generate equal SPLs.

n = number of noise sources in the rack

$Lp_1 = Lp_2 = Lp_3 = \ldots Lp_n = Lp$ (assuming the same SPL limitations are imposed on each individual noise source)

therefore, using equation 3.1:

$$Lp_T = 10\left[\mathrm{Log}_{10}\left(n * 10^{Lp/10}\right)\right]$$

$$\frac{Lp_T}{10} = \mathrm{Log}_{10}\, n + \mathrm{Log}_{10}\, 10^{Lp/10}$$

$$\frac{Lp_T}{10} = \mathrm{Log}_{10}\, n + \frac{Lp}{10} \implies$$

$$Lp = Lp_T - 10\,\mathrm{Log}_{10}\, n \tag{4.3a}$$

$$n = 10 \implies Lp = Lp_T - 10$$

$$n = 8 \implies Lp = Lp_T - 9$$

$$n = 5 \implies Lp = Lp_T - 7$$

Tables 4.1 and 4.2 show the continuous and intermittent maximum allowable SPL generated by the individual noise sources in the rack at a distance of 0.6 meter from the noisiest surface for the external payload and 0.6 meter from the front surface of the rack mounted equipment.

4.3 COMPONENT LEVEL ANALYSIS

A component level analysis will be performed at the early stage of the design. At this stage, the components of the hardware are defined but not assembled. The purpose of this analysis is to perform an assessment of the SPLs outside the enclosure of the unit. The components of the unit generate acoustic energy which is transmitted through the material. When sound waves strike a surface, the acoustic energy is partially reflected and partially absorbed. The energy absorbed by the surface is partially transmitted through the material and partially dissipated within the material. In a steady state operation, all the power radiated from the noise source is absorbed and/or transmitted through the enclosure surfaces.

TABLE 4.1. MAXIMUM ALLOWABLE SPLs GENERATED BY INDIVIDUAL CONTINUOUS NOISE SOURCE AT A DISTANCE OF 2 FT FROM THE NOISIEST SURFACE FOR EXTERNAL PAYLOAD AND 2 FT FROM THE FRONT SURFACE OF THE RACK MOUNTED UNITS

Requirements		Guidelines								
Rack Noise Limits*		Recommended Maximum Design Levels for Active Hardware Items								
A	B	C	D	E	F	G	H	I	J	K
Frequency Band Hz	Total Rack or Single Item (NC 20 Curve) SPL	2 Items dB	3 Items dB	4 Items dB	5 Items dB	6 Items dB	7-8 Items dB	9-10 Items dB	11-12 Items dB	13-16 Items dB
63	64	61	59	58	57	56	55	54	53	52
125	56	53	51	50	49	48	47	46	45	44
250	50	47	45	44	43	42	41	40	39	38
500	45	42	40	39	38	37	36	35	34	33
1000	41	38	36	35	34	33	32	31	30	29
2000	39	36	34	33	32	31	30	29	28	27
4000	38	35	33	32	31	30	29	28	27	26
8000	37	34	32	31	30	29	28	27	26	25

*Measurements shall be taken at 0.6 meters, per above reference.

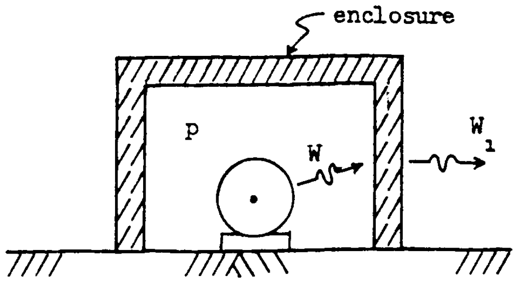
TABLE 4.2. MAXIMUM ALLOWABLE SPLs GENERATED BY INDIVIDUAL INTERMITTENT NOISE SOURCE AT A DISTANCE OF 2 FT FROM THE NOISIEST SURFACE FOR EXTERNAL PAYLOAD AND 2 FT FROM THE FRONT SURFACE OF THE RACK MOUNTED UNITS

Requirements		Guidelines								
Rack Noise Limits*		Recommended Maximum Design Levels for Individual Sub-rack Items								
A	B	C	D	E	F	G	H	I	J	K
Maximum Rack Noise Duration†	Total Rack A-weighted SPL (dBA)	2 Items dBA	3 Items dBA	4 Items dBA	5 Items dBA	6 Items dBA	7-8 Items dBA	9-10 Items dBA	11-12 Items dBA	13-16 Items dBA
8 Hours	≤ 49	46	44	43	42	41	40	39	38	37
7 Hours	≤ 50	47	45	44	43	42	41	40	39	38
6 Hours	≤ 51	48	46	45	44	43	42	41	40	39
5 Hours	≤ 52	49	47	46	45	44	43	42	41	40
4 Hours	≤ 54	51	49	48	47	46	45	44	43	42
3 Hours	≤ 57	54	52	51	50	49	48	47	46	45
2 Hours	≤ 60	57	55	54	53	52	51	50	49	48
1 Hour	≤ 65	62	60	59	58	57	56	55	54	53
30 Minutes	≤ 69	66	64	63	62	61	60	59	58	57
15 Minutes	≤ 72	69	67	66	65	64	63	62	61	60
5 Minutes	≤ 76	73	71	70	69	68	67	66	65	64
2 Minutes	≤ 78	75	73	72	71	70	69	68	67	66
1 Minute	≤ 79	76	74	73	72	71	70	69	68	67
Not Allowed	80 or above									

*Measurements shall be taken at 0.6 meters, per above reference.
†Per 24-hour Period

<u>NOTE</u>: Columns A and B contain the specifications for the rack. Columns C through K are suggestions or a reasonable way to sub-allocate the noise per payload to help ensure total rack compliance. For example, if a rack was being designed to contain four active (noise producing) equipment items and the total duration for all four items was 100 minutes, then each equipment item should be engineered to produce no more than 54 dBA at a distance of 0.6 meters from the front of the rack. Then the total noise from all four equipment items operating at the same time should meet the 60 dBA rack requirement for a two-hour duration.

A typical hardware unit is shown in following figure. Consider the enclosure, which encloses a noise source which radiates noise at an acoustic power W. The outer surface of the enclosure radiates noise at an acoustic power of W_1.



$$W = \frac{D\,V\,n\,\bar{\alpha}}{t} \tag{4.4}$$

where,

$D = \dfrac{p^2}{\rho_o c^2}$ = acoustic energy density

p = pressure

ρ = density

c = speed of sound

V = internal volume of the enclosure

n/t = number of reflections per unit time of the sound within the enclosure

$\bar{\alpha} = \dfrac{\sum \alpha_i S_i}{S}$ = average surface absorption coefficient for the interior of the enclosure

From the study of architectural acoustics, the number of reflections per unit time is given by:

$$n/t = c\,S/4V$$

where,

S = surface area

Making these substitutions, we find the following relationship for the acoustic power radiated from the machine or noise source.

$$W = \frac{p^2 \bar{\alpha} S}{4 \rho_o c} \tag{4.5}$$

The power transmitted through the walls of the enclosure is related to the power incident upon the surface of the enclosure and the sound power transmission coefficient $\bar{a}_t$ for the enclosure walls.

$$W_1 = \bar{a}_t W_i = \frac{\bar{a}_t S p^2}{4 \rho_o c} \tag{4.6}$$

where,

I = intensity

and the average sound power transmission coefficient is:

$$\bar{a}_t = \frac{\sum a_{t;i} S_i}{S}$$

Note that the sound power transmission coefficient is related to the transmission loss, TL, by:

$$TL = 10 \log_{10} (1/a_t)$$

or

$$a_t = 10^{-TL/10}$$

Dividing the power for Equation 4.6 by the power from Equation 4.5, we obtain the governing equation for enclosure design.

$$\frac{W_1}{W} = \frac{\bar{a}_t}{\bar{\alpha}} = \frac{\sum a_{t;i} S_i}{\sum \alpha_i S_i} \tag{4.7}$$

By taking logs of both sides of Equation 4.7 and multiplying by 10, we obtain the corresponding expression for the sound power level for the sound radiated from the enclosure, in terms of the sound power level of the machine.

$$L_{W_1} = L_W - 10 \log_{10} \left[\frac{\sum \alpha_i S_i}{\sum a_{t;i} S_i} \right] \tag{4.8}$$

The resulting SPL with the enclosure in place and for normal temperature and atmospheric conditions may be found from the following:

$$L_p = L_{W_1} + 10 \log_{10}\left(\frac{4}{R} + \frac{Q}{4\pi r^2}\right) + 10 \log_{10} \frac{W_{ref}\,(\rho_o C_o)_{ref}}{P_{ref}^2} \qquad (4.9)$$

For partial enclosures or enclosures with openings, the absorptivities and sound power transmission coefficients for the openings are required.

Although the sound power transmission coefficient for an opening would be equal to 1, there is also a directional effect for the opening, as far as the crew member is concerned. In this case, the transmission coefficient must be modified for the directivity and diffraction effects of the opening. The following values are recommended for simple openings in an enclosure.

a. Front opening $a_t = 1$

b. Side or top opening

no reflective surfaces nearby $a_t = 1/3$
with reflective surfaces nearby $= 2/3$

c. Back opening

no reflective surfaces nearby. $a_t = 1/6$
with reflective surfaces nearby $= 1/3$

The sound power transmission coefficient for other opening covers is generally dependent upon the frequency of the sound.

4.4 NOISE TEST APPROACH

4.4.1 Measurement of Sound Pressure Level

The purpose of the acoustical test is to measure the sound pressure levels generated by the payloads and verify that they meet the criteria established by ISS verification documents. The SPL is a meaningful description of the noise source only if the location of test article and microphones, and a description of the acoustical environment of the test room are identified. The environmental conditions and orientation of the test article should be reported along with the measured SPLs of the hardware. This is necessary because the environmental conditions inside ISS and the testing environment would be different.

The approach for performing the tests is explained in SSP 57010C. The guidelines for performing the test are briefly described in this section. They are explained in detail in Appendix A (Noise Measurement Procedure for ISS GFE Equipment and Payloads). Some of the criteria that has to be considered during measurement are:

- Test Area: The ideal place for the test would be an anechoic chamber/room. These rooms have very high absorption. Otherwise, the room dimensions should be as large as possible and the inner surfaces of the walls, floors, and ceiling should be acoustically absorbent as much as possible. The rack (payload) must be located as far as possible from any reflective surfaces such as walls, filing cabinets, and bookcases. The test room must be isolated from other noise sources.

- Orientation and placement of the test article: SPL depends on orientation and placement of the test article (directivity factor). In many cases, a sound source may be sitting on a hard floor, radiating sound into half-space; or in a corner where two walls and a floor meet, radiating into eighth spaces as shown in Figure 3-2. The test article to be measured has to be placed on a table or stand and its location has to be recorded.

- Test Equipment and Calibration: A precision calibrated Type 1 instrument must be used for making SPL measurements. The measurements must be taken at 0.6 m from the outer surfaces of the payload. The sound pressure level at the noisiest location must be recorded. Acoustic data acquisition shall be performed by a person who is familiar with the basic techniques used for testing, knows how to make meaningful background noise measurements, and understands at least some of the fundamentals of physical acoustics.

- Background Noise: The accuracy of the noise measurements greatly depends on achieving "quiet" room characteristics. The background noise should preferably be at least 15 dB below the noise limit specified for the test article. If this condition cannot be achieved, the noise level of the test article should be at least 3 dB higher than the background in each of the eight octave bands between 63 Hz to 8000 Hz.

The approach for performing the test is explained here. The test article should be configured as much as practical to be representative of how it will be mounted on ISS. The test article should be operated in the mode or setting that will occur on-orbit and that produces maximum noise. The Type 1 instrument, such as a Bruel and Kjaer model 2825 PULSE front-end system with several channels (microphones) of simultaneous data acquisition could be used.

At first, the background noise level has to be measured. The Sound Pressure Levels at each of the eight band levels (between 63 Hz and 8000 Hz) and the A-weighted Sound Pressure Levels must be checked before making any noise measurements from the test article. The noise generated by air handling systems and ancillary power equipments must be eliminated or minimized. It has to be made sure that the background levels are at acceptable levels.

The SPL measurements must be taken at 0.6 m from the outer surfaces of the test article. For an integrated rack, the measurement can be taken at 0.6 m from the surfaces adjacent to the crew environment. The noisiest location has to be identified by measuring the SPLs at several locations. The location of the highest noise level and the SPL at that location has to be recorded for each operational mode. It has to be noted that this location differs for each operational mode. The Sound Pressure Level at the eight frequency bands and the overall A-weighted noise levels must be recorded at the measured locations and at the location of the highest noise level. As the location of the highest noise level differs for each operational scenario of the rack, the procedure of searching for this location must be repeated for each scenario.

The test approach must follow the guidelines outlined, and the following information must be documented during the test:

1. Test Area: Description of testing room and test setup, test room dimensions and absorption coefficient of the room surfaces.

2. Locations of measurement including the location of highest noise level.

3. Acoustic Noise Data: SPL at the following either octave band frequencies: 63 Hz, 125 Hz, 250 Hz, 1000 Hz, 2000 Hz, 4000 Hz, and 8000 Hz, Overall SPL, Overall A-weighted SPL.

4. Background Noise Data: SPL at the eight octave band frequencies from 63 Hz to 8000 Hz, Overall SPL, Overall A-Weighted SPL. This has to be recorded at all locations, including the locations of the highest noise level for all operating scenarios.

5. Directivity Factor of sound source in the Testing Room (location of the noise sources in the test room).

After the data is obtained, the following calculations must be made:

For consistency check, calculate the A-weighted SPL using the measured octave band SPL data and compare with the measured data.

Compute the background corrected SPL at the eight octave band frequencies 63 Hz to 8000 Hz, Overall SPL, and Overall A-Weighted SPL. The background corrected data is used to compare with the Noise Limits.

4.4.2 Measurement of Sound Power Level

The sound power measurement is required if the sound pressure level requirement is violated. The sound power measurements shall be performed in accordance with the appropriate ANSI S1 and International Organization for Standardization (ISO) standards on acoustics, depending on the method selected for measurement.

One of the approaches for determining the Sound Power Level is by measuring the sound intensity using a sound intensity-scanning probe. This method was used for the HRF Rack 1. For measuring the sound intensity, surfaces enclosing the rack have to be determined. The sound intensity must be measured on all surfaces enclosing the rack. This surface on which intensity measurements are made must completely enclose the noise source, or in conjunction with a rigid wall, enclose the noise source. The intensity probe must be placed normal to these surfaces while measuring. The sound intensity at all the frequency bands between 63 Hz and 8000 Hz must be recorded. The dimensions of the enclosing surface must be recorded. The intensity data on each surface is used to compute the Sound Power Level. The guidelines provided in ISO 9614-2 and ANSI standard S12.12 must be carefully followed for calculating the Sound Power Level using the measured sound intensity data.

The results from the measurement must clearly document the sound intensity on each surface. The sound power must be calculated using the intensity data and the surface area of each surface. The sign convention shown in ISO 9614-2 must be used to represent these quantities. The Sound Power Level at each frequency band must be calculated. The overall Sound Power Level and the A-weighted Sound Power Level must be also recorded.

4.5 INTEGRATED RACKS WHOSE SUB-RACK EQUIPMENT WILL BE CHANGED OUT

A test correlated analytical model will be used to verify compliance of one rack with the acoustics requirement during equipment change out. The analytical model will include system noise sources and anticipated sub-rack payload complement noise sources. The test correlated model process is shown in Figure 4.5-1.

Since the rack is available for hardware acoustics testing, a test correlated analytical model will be generated for the rack mounted hardware. Based on the tests results, a test correlated model will be developed. The process is show in Figure 4.5-2.

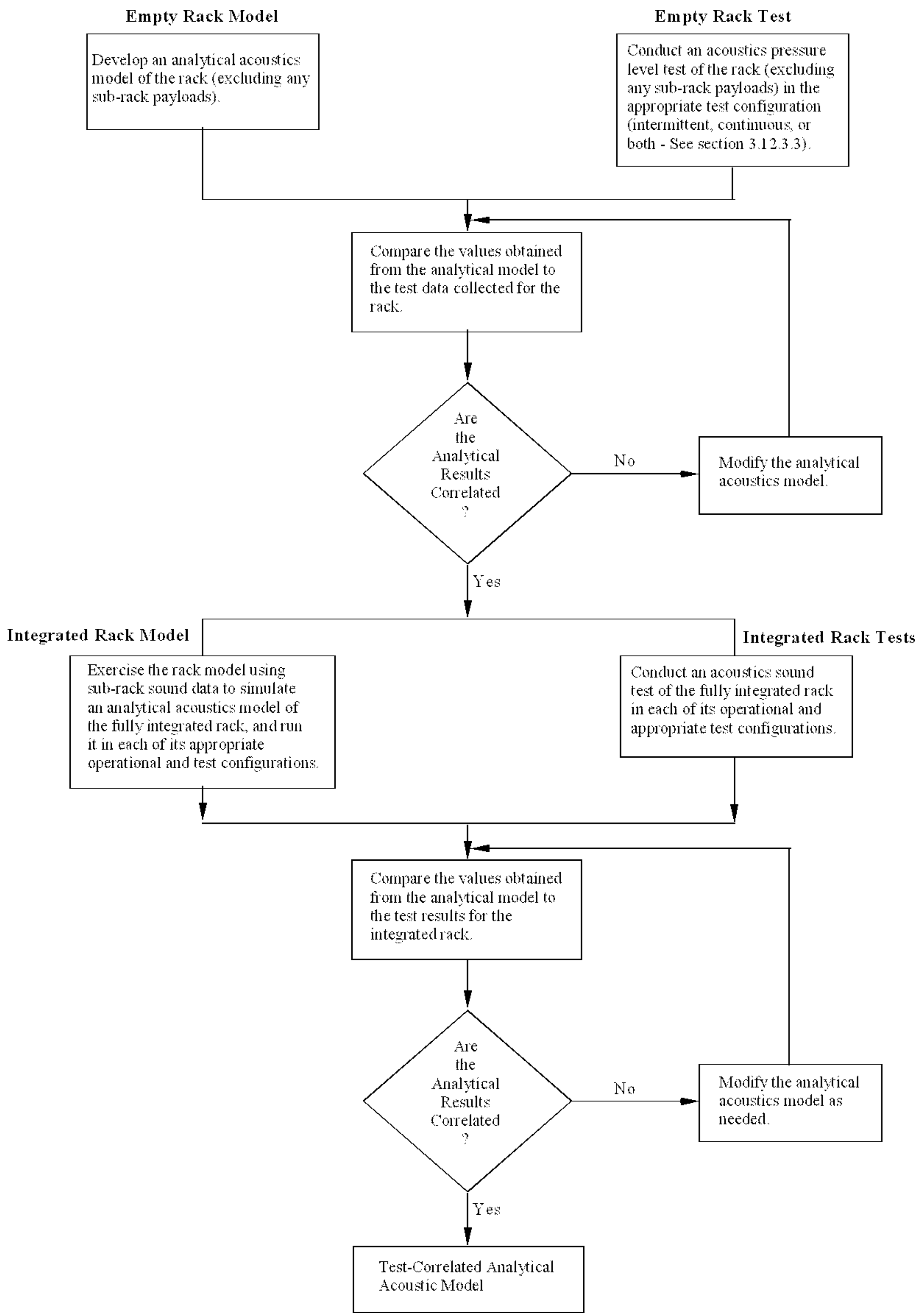


Figure 4.5-1. Test-Correlated Process for the Rack Model

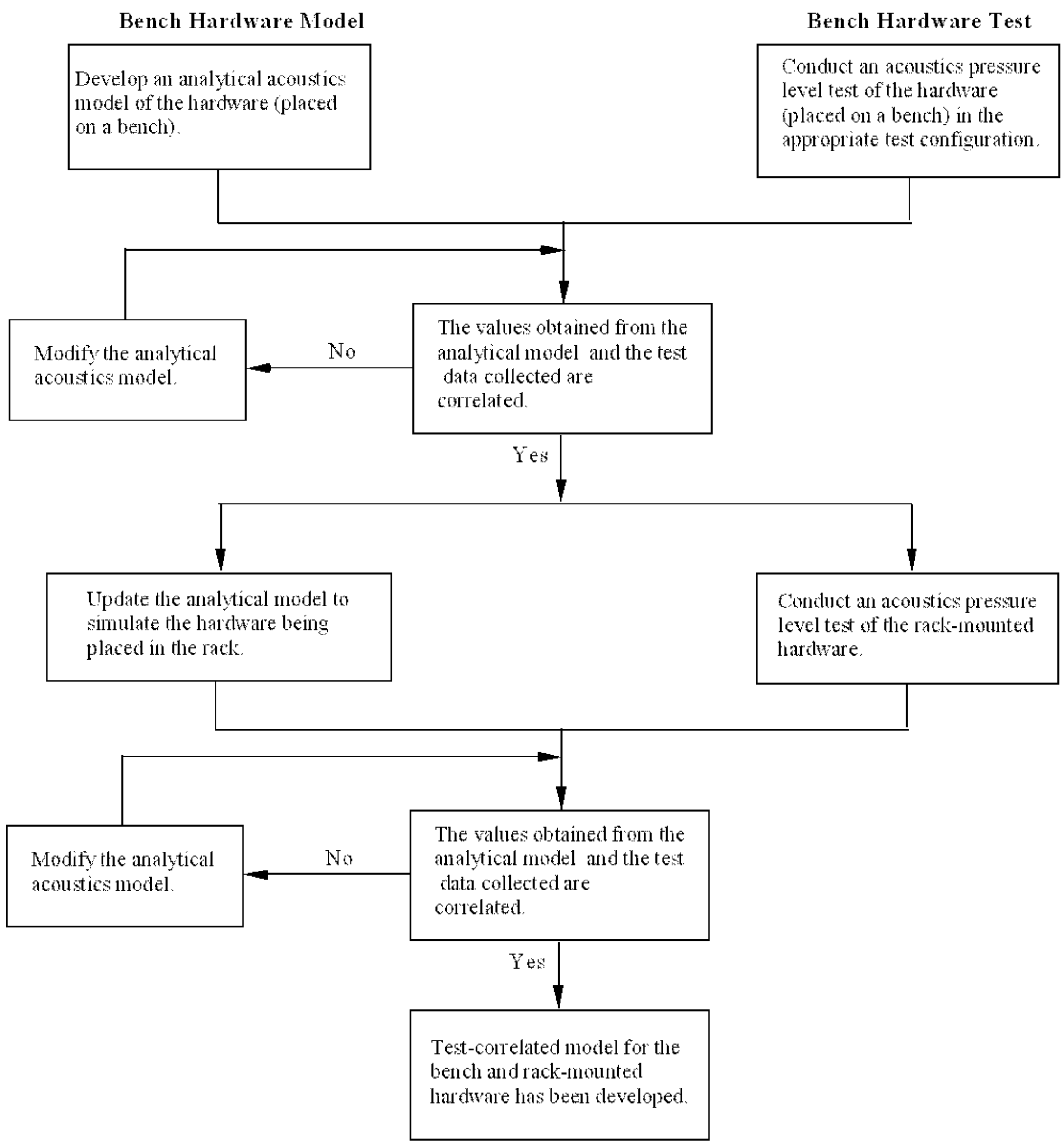


Figure 4.5-2. Test-Correlated Model for the Hardware

5.0 ACOUSTIC NOISE CONTROL

The acoustics can be classified as structure-borne or air-borne. The methods utilized to control these are generally different and an effective noise control strategy should employ a combination of the two.

Before we investigate the methods of noise control, it is necessary to understand the noise sources.

5.1 NOISE SOURCES

Before developing a noise control strategy, it is necessary to evaluate the expected noise sources. The noise sources within the HRF EXPRESS Rack will consist of the following:

Fan Noise

There are several paths through which noise may be radiated from a fan, including:

(a) Sound radiated directly from the fan outlet and/or inlet if there are no ducts connected to the fan.

(b) Sound radiated through the fan housing, and

(c) Sound induced by vibrations transmitted from the fan through the supports to the adjoining structure.

The noise generated by fans is composed of two components:

(a) Broad-band noise generated by vortex shedding from the fan blades, and

(b) A discrete tone (blade tone), produced as the blade passes by the inlet or outlet opening.

Standard techniques are available to predict the noise produced by fans. To calculate the total noise generated by the fan, it will be necessary to add the noise generated by the drive motor.

Electric Motor Noise

The noise generated by an electric motor is fairly complex and is a result of the following factors:

(a) Motor rotor-slot noise caused by rotating open slots. This noise is a tonal noise with a frequency equal to the product of the rotational speed and the number of slots. This noise may be reduced by filling slots with epoxy or other filler material.

(b) Electrical noise contribution due to rotor and stator slot magnetomotive force interaction.

(c) Noise due to high magnetic flux density. This component has a frequency equal to twice the line frequency.

(d) Noise due to any dynamic unbalance of the rotating elements.

(e) Bearing noise

(f) Noise from vibrations caused by unbalanced line currents in a 3-phase power supply.

Pump Noise

Standard-line pumps are not major noise sources in ordinary situations; however, if several pumps are operated in an enclosed volume, a noise problem can result. Often the motor that drives the pump may make more noise than the pump itself, unless the pump is operating on a severely cavitating regime.

Noise in a hydraulic pump is induced by such actions as recombining of fluid streams at the pump outlet, cavitation within the pump, pressure ripples, impact of internal parts, and unbalance of the rotor.

Transformer Noise

Electric transformers emit a continuous “hum” which can be quite annoying. In addition, the cooling fan (if used) on the transformer generates additional noise.

Transformers are wound on cores built up of laminations and the noise generated by the transformer depends upon the magnetic field variations in these laminations. Because of magnetostriction effects, the laminations change in length as the magnetic field changes. The total change in the length may be on the order of micrometers, but this small dimension change is enough to be the major contribution to transformer hum. The magnetostriction effect is independent of the direction of magnetization, so a complete cycle of dimensional change occurs during each half-cycle of the alternating current supply. The fundamental acoustic frequency is twice the electrical frequency; therefore, the peak in the noise spectrum from the transformer occurs at (60 Hz) (2) = 120 Hz, or in the 125 Hz octave band.

Noise From Gas Vents Vacuum Exhaust System

One of the more severe problems in the rack cooling system is noise from the discharge of air into the atmosphere. The noise generated by the jet of gas discharged through vents is a result of turbulent mixing in a high-shearing region near the exit plane of the vent. In this region, turbulent eddies are quite small and the noise radiated from these eddies is predominantly high-frequency noise. Sound is also radiated from the fluid stream further from the jet as a result of larger turbulent eddies in this region of the jet. Lower-frequency noise is radiated from this region.

5.2 CONTROL OF NOISE

The most effective means of achieving quiet integrated hardware is by choosing quiet components at the beginning, especially the noisy components such as fans, pumps, and motor. Attention must be paid to assure that by selecting the quiet devices, the performance of the fan or pump needed to provide the required cooling to the payload is not compromised. However, effective control of noise can be implemented if needed. Effective control of noise will require management of air-borne and structure-borne noise components. Several approaches that are available for effective noise control that can be employed for the HRF payload and integrated racks are summarized below.

5.2.1 Airborne Sound Attenuation

The noise level in the space occupied by a listener will depend on the sound power emitted by the source, sound absorption in the room/volume occupied by the listener, and the sound volumes occupied by the intervening structure. Air-borne noise radiated into and around the sources may be reduced by interposing one or more partitions between the source space and the occupied room. Sound absorbing materials can be used to reduce the sound levels caused by the presence of reverberant sound field in enclosures by preventing unwanted sound reflections from hard surfaces.

For most of the HRF payloads, the primary noise generators will be the cooling fans. Proper selection of the fan and design of the acoustic cavity by adding partitions and sound absorbing materials will be the primary technique for air-borne noise control. Attention will be paid to assure that addition of the acoustic devices will not compromise the fan performance needed to provide the required cooling to the payload.

Enclosures and Barriers

From the point of view of noise control, a heavy, stiff enclosure placed completely around an offending machine or structure is used quite frequently in stationary industrial applications where weight is of no special concern. It is also used occasionally in non-stationary applications, such as diesel engines on trucks, where the weight penalty is not always considered unacceptable. However, for aerospace applications, this is not the case and barriers are not particularly effective on a combined weight, convenience, accessibility, and economic impact basis. If the enclosure is made lighter and more flexible, and hence cheaper, the surfaces of the barrier will vibrate under the excitation from the radiated noise of the structure and will in turn transmit some of the noise, hence reducing its effectiveness.

Similarly, walls or barriers can be effective in blocking some of the noise radiating from a source. However, gaps around the barrier can lead to reflection or refraction of some noise around the sides of barriers, again reducing their efficiency. The enclosure and barrier approaches are certainly effective ways of controlling noise transmission in many cases and are recommended for consideration when conditions merit this approach. However, noise control at the source, isolation of the structure, and introduction of damping into the structure must all be given equal attention until evidence available from examination of each specific problem indicates the most cost effective approach. It is terribly inefficient to put a barrier around a noisy machine when, for example, a set of simple isolators placed under it can reduce the noise by the required amount, or damping of a single excessively vibrating local mode can do the same.

5.2.2 Structure-Borne Noise Control

Structural vibrations, their causes, effects, and methods of control are important topics of study in many engineering disciplines. Mechanical vibrations caused by rotating machine elements, vibrations induced in buildings and ocean structures by wind, wave, and seismic loads, and the response of aerospace structures to the vibroacoustic environment generated by engine exhaust and aerodynamic disturbances are some examples of their manifestation. The effects of structural vibration can range from noise pollution to health hazards to equipment malfunctions and catastrophic structural failures. Techniques to reduce structural vibrations are required to mitigate their ill-effects. Vibration reduction techniques may be grouped under three broad categories: reduction at the source, isolation, and reduction of the response. This categorization emphasizes the importance of considering the nature of the excitation, the modes of transmission, and the structural properties that govern the response for developing effective vibration control strategies.

The dynamic and vibration transmission characteristics of structures are governed by three inherent properties: mass, stiffness, and damping. These properties determine the natural frequencies, mode shapes, and modal damping of the structure. Traditionally, one of the popular techniques of controlling vibration is to increase the stiffness of the structure or change its mass characteristics so that the system resonance frequencies are shifted away from the zone of peak excitation frequencies. However, when the system response is dominated by multiple resonances or the excitation is very broad banded, such structural detuning may not be feasible. In such cases, increasing damping in the dominant response modes can dramatically cut down response levels. The structural response for resonant or near-resonant excitation is controlled by damping as the forces due to inertia and stiffness effects essentially cancel each other. For non-resonant excitation, damping will have negligible influence on the response levels.

A vibrating structure contains kinetic and potential (strain) energy associated with its modal mass and stiffness values. As the vibration cycle proceeds, this energy is converted from one form to the other. Realistic behavior involves some energy dissipation as well, usually in the form of heat. This non-conservative nature of energy conversion is what we call damping. Unlike mass, which is a single physical phenomenon, and stiffness, which results from very few physical effects, damping may be caused by a great variety of phenomena. These include mechanical hysteresis (also called material damping or internal friction), electromagnetic effects (eddy current, magnetic hysteresis), friction due to motion relative to fluids or solid surfaces (interface friction, fluid viscosity), and energy transport to adjacent structural components or fluids (impact damping, acoustic radiation, turbulence). This great variety of phenomena makes damping difficult to measure, model, and modify, but enables one to conceive of a variety of means for increasing it.

The primary effects of increased damping are reduction of vibration amplitude at system resonances, more rapid decay of free vibration amplitudes, and decreased spatial conduction of vibration. The practical consequences of these effects are the reasons why one is interested in techniques or devices that increase damping. Reduction in response levels result in decreased oscillatory stresses and in attendant increases in the fatigue life of structures and the reliability of mechanical devices. Decreased spatial conduction implies increased system impedance which improves the effectiveness of vibration isolation. More rapid decay of free vibration reduces noise, structural fatigue, and settling times, which is very important for space-based optical devices and microgravity payloads.

Overview of Noise and Vibration Control Techniques

Although noise and vibration problems can vary in their nature and severity, there exist only a limited number of ways to solve them. The approaches used for solving these problems are similar because structure-borne noise problems are concerned with the noise resulting from vibration of the structure. Techniques that are currently in use can be classified under two major categories: active and passive techniques (Figure 5-1). In active techniques, the structural parameters are continuously changed in response to varying excitation characteristics to control the vibratory motion. These techniques require an actuator to generate the necessary control forces to counteract the disturbing forces. They provide superior isolation or damping performance over a wide range of excitation frequencies. Active control techniques require an external energy supply for the actuator, complex feedback control devices, and sensors to control or modify the structural motion. The general use of these techniques have been severely limited due to the associated high costs, complexities, and poor reliability.

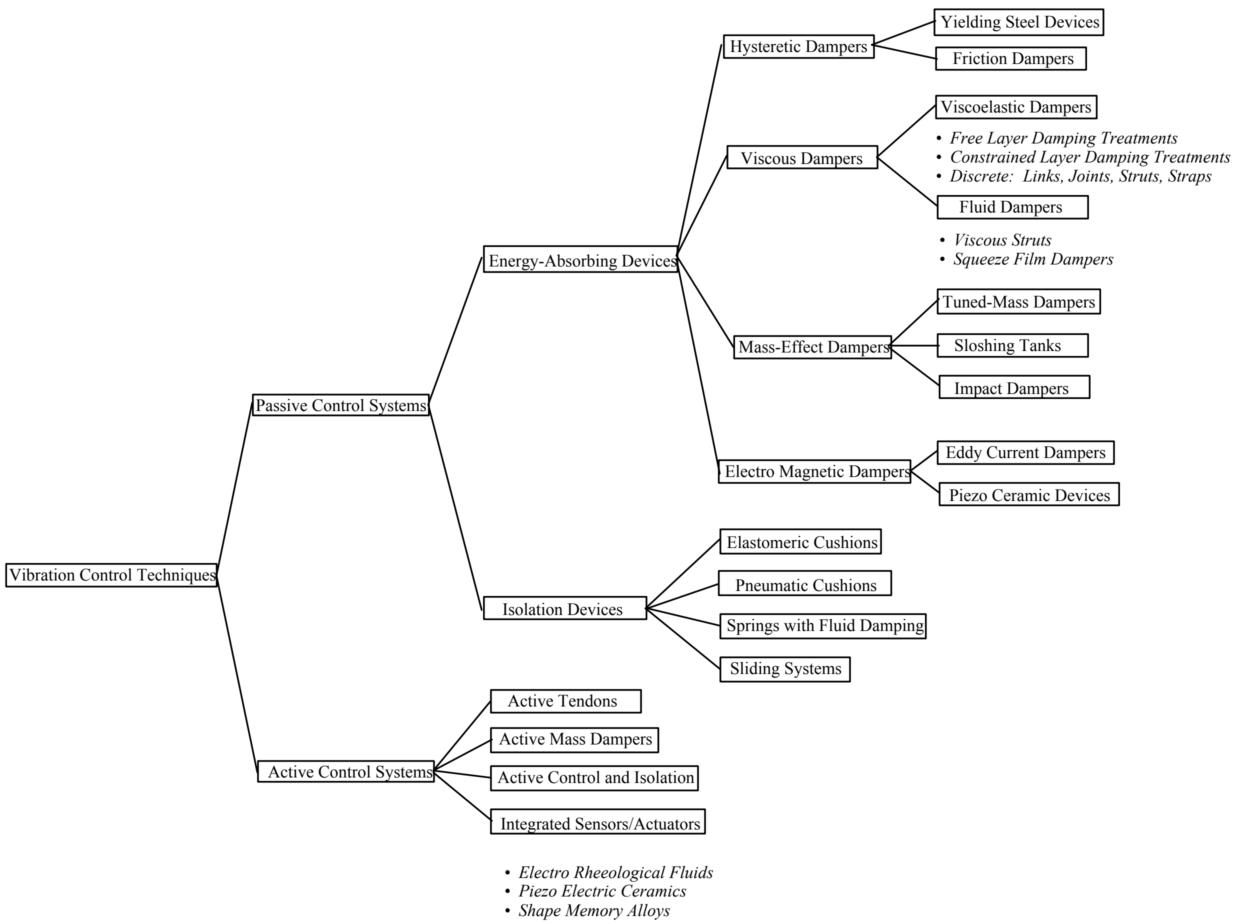


Figure 5-1. Classification of Vibration Control Techniques

Active control techniques have also been used for air borne noise reduction. Spurred by the availability of inexpensive microprocessors, such techniques are finding increasing application, especially in the automobile industry. Their operating principle is simple: the offending noise is picked up by microphones and processed by digital signal processors which produce an 180-degree out-of-phase antinoise that is played through loud speakers to cancel out the offending sound. Because of the system's selectivity, desired noises such as speech and warning sirens are not affected. Although simple in principle, practical implementations can be complex mainly due to feedback from the antinoise source to the input signal processor, the time delay between the antinoise output and the error signal input, and sound wave reflections from the environment. However, recent advances in computing sound signals and determining the optimal position for outputting the antinoise in addition to more robust design of microphones and loudspeakers, have increased the feasibility of active noise cancellation technology.

Passive vibration control techniques rely on fixed change in the physical parameters of the structure to attenuate vibration. They are more robust and relatively less expensive. They have some inherent performance limitations at very low frequency ranges which make them unsuitable for certain applications. A comparison of the relative merits of active and passive techniques is shown in Table 5.1. In consideration of their relative simplicity and overall cost effectiveness, only passive vibration control technologies will be considered for the HRF. The implementation of passive techniques usually involves a permanent change in the structural parameters which would cause the energy associated with the vibrations to be prevented from being transmitted into or out of the structure (isolation devices, enclosures, and barriers) or to be removed from the structure (absorption and dissipation devices). In either case, they cut down the mechanically transmitted vibrations and reduce the associated noise. The various types of passive control techniques are described in Figure 5-1.

TABLE 5.1 COMPARISON BETWEEN PASSIVE AND ACTIVE TECHNIQUES [2]

Criteria	Active Damping and Isolation	Passive Damping and Isolation
Type	Local	Local and Distributed
Tolerable Loads	Relatively Low	Relatively High
Frequency Range	Applicable to even very low frequencies, problems with the control system for very high frequencies	Lower bound by the required restoring forces; problems for very slow vibrations
Effectiveness	Nearly Complete	Less Effective
Reliability	Electronics very reliable but single error might cause system failure; backup required	Not precisely predictable but error will not cause system failure; no backup required
Required Effort	Additional power supply and data processing required; electromagnetic protection, etc.	--
Costs	Comparatively High	Comparatively Low

Isolation Techniques

An isolator is basically a resilient element (weak spring) that is introduced between the source of excitation and the structure in order to reduce the transmittibility of dynamic motion or force. Because of its resilience, the isolator stores the incoming energy and releases it into the structural system at a time interval which affords a reduction of the magnitude of motion. In terms of frequency, the isolator causes the natural frequencies associated with the rigid body motion of the structural system on its isolator mounts to be much lower than the excitation frequency, thereby reducing the force or motion transmittibility.

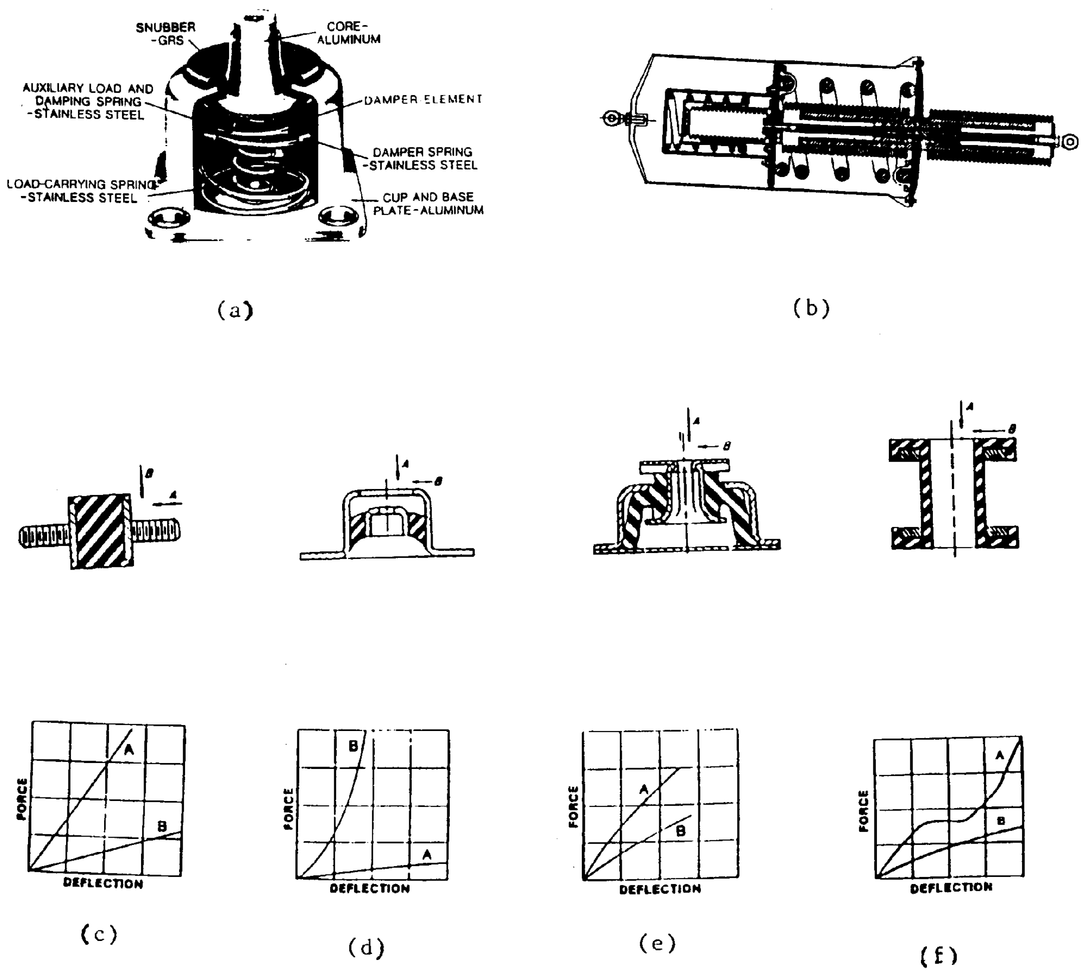
In addition to stiffness adapted to isolate vibration, most isolators also have substantial damping properties. This is especially important when the excitation covers a wide range of frequencies and during the excitation build-up phase when the isolator may be required to operate at resonance momentarily. Fluid (viscous), hysteretic (material), and Coulomb (dry friction) damping mechanisms are most often used to incorporate damping in isolators.

There are many other design considerations, besides stiffness and damping, that must be taken into account when selecting isolators for vibration control [3]. The load carrying capacity of the isolators, their static deflection under the dead-weight of the supported structure, their ability to provide fail-safe attachment, weight and space limitations for isolator installation, extremes of temperature and other environmental conditions to which the isolator will be subjected, the stiffness of the isolator in lateral directions relative to its stiffness in normal direction, and non-linear stiffness characteristics are some of the important considerations that determine the type of resilient material to be used in the isolator.

Isolators, made from a wide variety of resilient media having diverse characteristics, are commercially available. Each type of isolator has characteristic properties which make them particularly suited for specialized applications. The simplest form of isolator is a metal coil spring which may be loaded in tension or compression. It is easy to obtain the necessary stiffness characteristics with such isolators and they are relatively free from drift or creep under applied loads. However, metal springs possess practically no damping, so transmittibility at resonance is very high. To overcome this disadvantage, additional damping can be designed into the isolator, in parallel with the load carrying spring. Figure 5-2(a) shows a spring-friction damper in which a plastic damper slides along the walls of a cup housing providing damping for vertical vibrations while a damper attached to the bottom provides damping for horizontal vibrations. Such sliding or sliding/bearing systems with metal springs and friction damping are commonly used in applications where excitation frequencies are low to moderate, static deflections are high, temperature and environmental conditions are harsh, and low-cost isolation is a priority.

Another method of adding damping to an isolator spring is through the use of fluid (viscous) damping. An example of this type is shown in Figure 5-2(b). These mechanical devices usually have built-in damping chambers containing air, oil, or some other damping fluid. Vibrations produce relative displacement across the damping chamber which forces the fluid through a small diameter orifice or a constricted passage, thereby causing shear flow in the fluid. This produces velocity-dependent damping which can be increased or decreased by changing the orifice area.

Pneumatic cushions are utilized as isolators in cases where the excitation frequencies are very low (of the order of 1 Hz or lower). In such cases, conventional spring isolators are



Figures 5-2. (a) Spring-Damper (b) Viscous Isolator (c-f) Elastomer Isolators

not suitable because the low stiffness values required for effective isolation will cause very large static deflection in the spring element. The air spring in the pneumatic isolator will enable the system to have small deflections while providing the necessary low stiffness. A potential limitation with this type of isolator is its inability to provide fail-safe attachment.

Elastometric isolators have found wide applications in vibration control because of the advantages that elastomers possess as a resilient medium. Elastomers can be conveniently molded to any desired shape and stiffness, embody more internal hysteresis than metal springs, require minimum space and weight, and can be bonded to metallic inserts adapted for convenient attachment to isolated structures. They also afford high frequency isolation. The shape flexibility of elastomers makes it possible to vary stiffness within wide limits and to attain any degree of linearity or nonlinearity in stiffness characteristics. This is evident from Figure 5-2(c) which shows some typical configurations and the corresponding applications where the space available for full travel of the isolator is limited or the force in the isolator has to be limited. Elastomeric isolators with hardening or softening spring isolators tend to drift or creep when subjected to large strains over prolonged periods. Their temperature range is also somewhat limited when compared to metallic springs.

Energy Absorption Devices

The use of energy absorption devices to improve the dynamic behavior of structural systems is well-established. Unlike isolation devices, they function by absorbing and removing the vibration energy by converting it to other forms like heat or electricity, transferring it to connected structures or ambient media, or channeling it to other structural motions than those of immediate concern.

Auxiliary masses are frequently attached to vibrating structures by means of springs and damping devices to control vibrations. When the auxiliary mass system has little or no damping, it is called a dynamic absorber. Such devices are used to eliminate sharp resonance peaks at specific excitation frequencies. By adjusting the auxiliary mass and attachment spring values, the dynamic absorber is tuned to the specific frequency that has to be eliminated. When attached to the structure, the dynamic absorber splits the single resonant frequency into two system frequencies, one of which is lower and the other higher than the initial frequency. If the two new system frequencies are outside the frequency range of input excitation, the dynamic absorber will be effective in reducing structural response.

Dynamic absorbers are not practical for vibration control problems where the excitation is broad banded because eliminating the system resonance at one frequency may introduce resonances at one or more other frequencies. Tuned Mass Dampers (TMD), or damped absorbers, which incorporate damping in the auxiliary mass system can be effective in controlling the resonant response in such situations. Unlike dynamic absorbers which function as an energy transfer device at the tuned frequency, the tuned mass damper dissipates the mechanical energy of vibration in the damping element, thereby contributing to system damping. Dampers relying on Coulomb friction, viscous damping, and material damping (in viscoelastic elements) have been used in TMD devices. TMDs using viscoelastic damping elements have been used to control vibrations in many aerospace applications [1][5]. Some possible configurations of tuned viscoelastic dampers are shown in Figure 5-3. The effectiveness of TMDs in controlling vibrations is determined by three basic design parameters; the ratio of the auxiliary mass to the mass of the system, the resonant frequency of the auxiliary mass to the resonant frequency of the

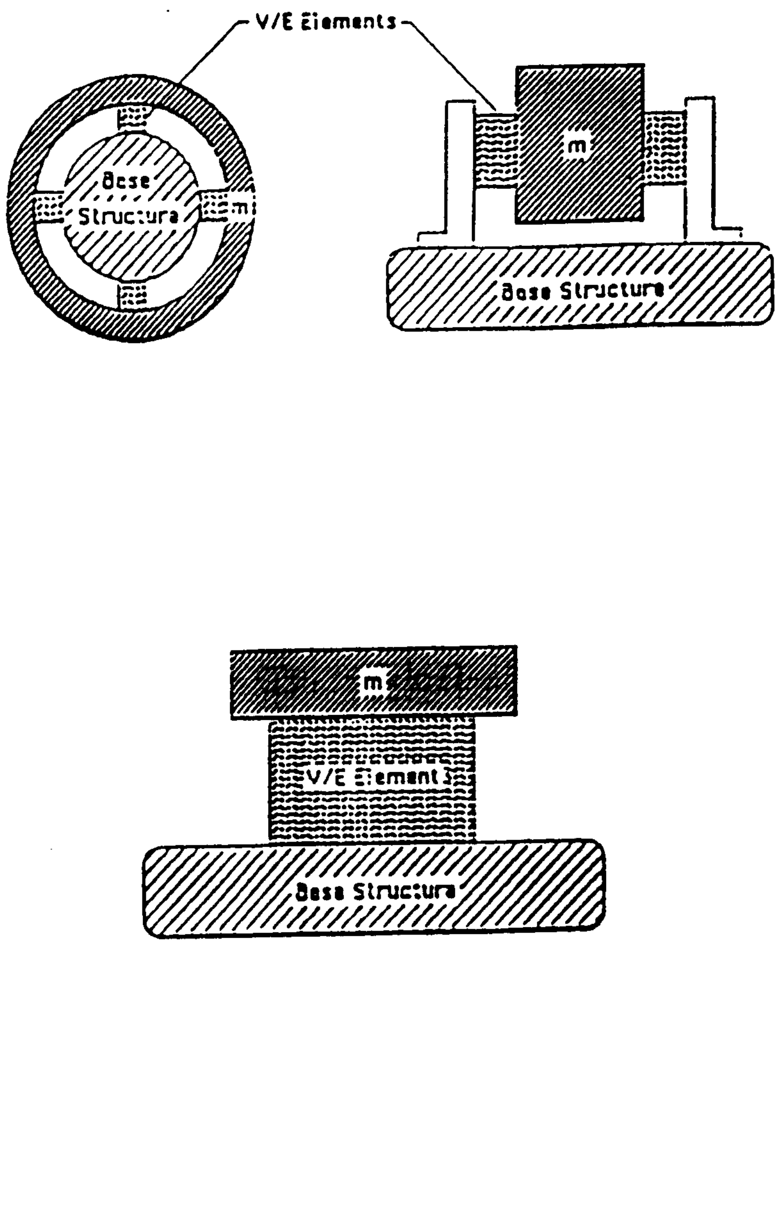


Figure 5-3. Tuned Viscoelastic Dampers

system, and the energy dissipation as measured by the loss factor. Energy dissipation in TMDs can be maximized at points of high amplitude response in the structure. It is important to note that energy dissipater in TMDs depend on local displacements of the structure rather than surface strains as in the case of layered damping treatments. Thus, for structures with low vibratory surface strains involving non-platelike behavior such as space-frame type structures or highly curved elements, TMDs can be more attractive than other forms of damping treatments.

Tuned Liquid Dampers (TLD) or sloshing tanks have been used to control vibrations in space satellites, marine vessels, and civil structures [6][7]. Such dampers rely on the motion of liquids inside a rigid container to absorb and dissipate the energy associated with structural vibrations. The dissipation is primarily through viscous friction resistance between the liquid and the container walls as well as between adjacent liquid layers. As in the case of TMDs, the mass ratio, the frequency ratio, and liquid damping are the controllable parameters in the design of TLDs. Such devices are relatively cheap and have few maintenance requirements.

Impact dampers, also known as rattle dampers or acceleration dampers, have been studied as a means of passive damping for space applications [8][9]. They operate by allowing a series of collisions between the vibrating structure and a secondary impacting mass which is carried in or on the primary structure. Each impact causes momentum exchange between the primary and secondary masses and dissipates some of the kinetic energy of the system as heat, noise and elastic waves. The mass of the impactor, the coefficient of restitution, and the distance for free travel between impacting surfaces are some of the design parameters that determine the effectiveness of impact dampers. Impact dampers are simple and reliable devices but they add weight to the structure.

Passive electromagnetic damping devices are based on the principle that a conductor moving in a magnetic field will experience a drag force. As relative motion occurs due to vibration, eddy currents are induced in the conductor which interacts with the magnetic field to resist the motion. The net result is that vibratory energy is dissipated through ohmic heating in the conductor. Such devices, known as eddy current dampers, are well suited for aerospace applications because they are non-outgassing, non-contacting, stable with respect to temperature variations, and have low wear and high reliability [10]. Figure 5-4 shows the schematic of an eddy current damper which includes a flexure mechanism to transmit and amplify axial deformations into rotations of a conductor disc [11]. They can be readily incorporated as a structural element to increase system damping or used as the damping element in a TMD.

Piezoelectrics possess the property of producing an electrical voltage when they are strained. They have the ability to convert mechanical energy to electrical energy and this transformation ability makes them useful as structural dampers. In passive damping applications, the piezoelectric element is embedded in the vibrating structure and its electrodes are shunted through an external passive electrical circuit. The electrical circuit may include only a resistor, or a resistor and inductor; its electrical impedance is designed to dissipate electrical energy which has been converted from mechanical vibratory energy by the piezoelectric element. For resistive shunting, the piezoelectric element has frequency-dependent damping and stiffness, much like Viscoelastic Materials (VEMs), but with better temperature stability. Shunting with a resistor and inductor introduces an electrical resonance, which can be optimally tuned to structural resonances in a manner analogous to a TMD [12].

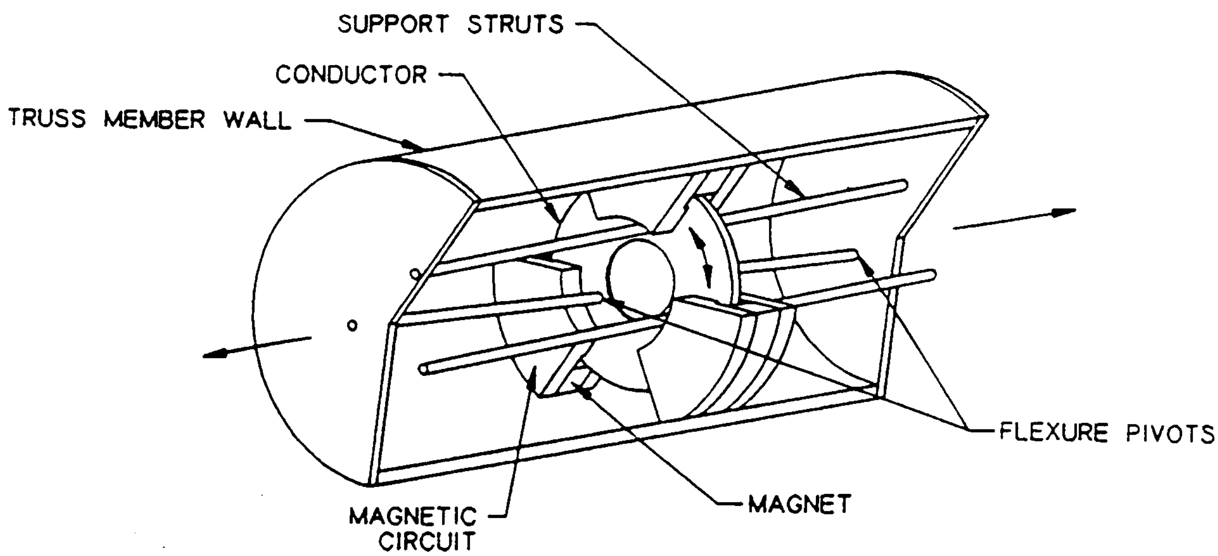


Figure 5-4. Eddy Current Damper

Friction dampers can be used to dissipate vibration energy and mitigate the structural response. Figure 5-5 shows a typical construction which consists of a cylindrical device with friction pads that slide along the inner surface of a steel casing. The magnitude of friction force that results from sliding depends on the coefficient of friction (which is a function of the surface materials and finish), the normal force between the surfaces, and the area of contact. Friction dampers have been used in turbomachinery to dissipate resonant vibratory energy and reduce stresses in turbine blades by providing sliding contact between points experiencing relative motion due to vibration [13][14].

While energy dissipation in friction dampers occurs as a result of dry sliding (Coulombic friction) between two surfaces, lubricated sliding (viscous forces) can also be used to dissipate vibratory energy. Squeeze film dampers employ this technique by interposing a thin fluid film between surfaces in relative motion [15]. The shearing of the fluid film produces a viscous drag force whose magnitude depends on the relative velocity between the mating surfaces, the coefficient of dynamic viscosity of the fluid, the area of contact, and the thickness of the fluid film. Lubricating oils, silicone oil, and glycerine are among the commonly used fluids in viscous dampers.

Techniques of structural vibration control by means of viscoelastic damping treatments have been widely applied in many engineering fields, particularly in the aerospace industry [16]. Such treatments capitalize on the inherent damping of highly dissipative VEMs to improve system level damping performance. A VEM is one that has the characteristics of an elastic solid (stiffness or energy storage capability) as well as a viscous fluid (energy dissipation capability). These characteristics are governed by the VEM's material properties, namely its elastic (E) or shear (G) modulus and its loss factor η. The material properties are highly dependent on temperature and frequency and these functional dependencies determine the proper choice of VEM for a particular damping application.

In designing a VEM treatment for a structural damping application, we seek to reduce undesirable resonant responses by increasing the structural loss factor.

$$\eta = \frac{D_s}{2\pi U_s} \tag{5.1}$$

Here, D_S is the energy dissipated per cycle and U_S is the total vibratory energy of the system. For linear VEMs, the energy dissipated in the VEM is given by

$$D_S = \pi \int_V \eta E e^2 \, dv \tag{5.2}$$

where the integral is performed over the volume of the VEM. The loss modulus, ηE in the integral, is a VEM material property and the e^2 term is a measure of the localized dynamic strain, which is a structural property. Thus, to maximize D_s, it is necessary to choose a VEM with high loss modulus under the operating environmental conditions and also to locate it on the structure such that maximum strain energy is imparted to the VEM by structural vibrations. An optimal damping treatment involves not only the proper choice of damping material but an understanding of the effects of structural geometry on the damping treatment and the deformations associated with particular modes of vibration of the structure.

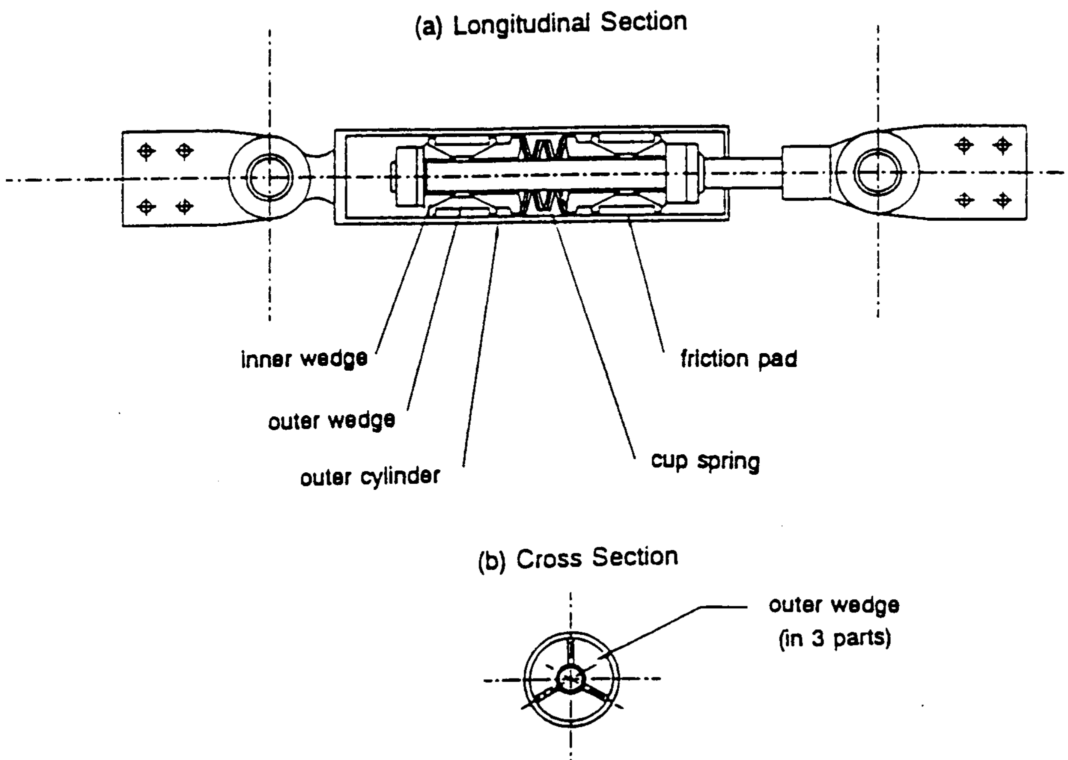


Figure 5-5. Friction Damper

There are two types of layered damping treatments: free layer treatments where only a viscoelastic polymer layer is added to a surface of the structure and constrained layer treatments where, in addition to the viscoelastic, a rigid constraining layer is also added (Figure 5-6). In the free layer treatment, the entire VEM extends and compresses with the surface of the structure as it undergoes bending deformations. The resulting strain is therefore extensional and is strictly dependent on the local curvature of the structure caused by dynamic bending and the distance of the VEM from the neutral bending plane. As the magnitude of strain induced is limited by practical thickness considerations, the designer must maximize D_S, as expressed by Equation 5.2, by picking a VEM with maximum loss modulus, ηE. Thus, free layer damping materials must be stiff and exhibit high loss factors.

In constrained layer damping treatments, the constraining layer is significantly stiffer in extension than the VEM and is not mechanically attached to the base structure (rivets, bolts, etc.) except through bonding at the viscoelastic interface. When the structure is subjected to cyclic bending, only the surface of the VEM attached to the structure extends and compresses while the other surface of the VEM is held in check by the constraining layer. This results in shear strain in the VEM, significantly improving damping efficiency. In contrast to free layer treatments, there is more design flexibility because the strain induced in the VEM can be varied by changing the stiffness of the constraining layer. In general, constrained layer treatments are more effective and weight efficient than free layer treatments, but this efficiency is balanced by greater complications in analysis and application. Several techniques have been investigated as a means of increasing the effectiveness of constrained layer treatments; the use of offsets to increase shear deformations in the VEM [17], sectioning the constraining layer to take advantage of high valued shear strains that occur at the edges of the constraining layer [8], the use of multiple layer treatment to broaden the effective temperature range [19], and schemes for alternately anchored constraining layers [20].

Viscoelastic Damping Treatments

Constrained layer damping treatments provide an attractive solution for noise and vibration problems because of their weight and cost effectiveness. However, the application of these damping treatments is a frequently misunderstood subject. Extreme care must be taken to define the problem, design the treatment, fabricate it, and apply it to the structure. This process is necessarily lengthy and comprehensive; hence, the chance for success of a hit and miss application is very small. It should also be emphasized that optimizing the treatment for only its damping performance without considering the effects of other modal properties will usually result in insufficient noise reductions. The important steps involved in the analytical design process are summarized below.

1) The first step in the design of a constrained layer treatment is to define the problem and the environment. Through this definition, the designer seeks to establish that a resonant vibration problem exists. If it is not a resonant problem, the constrained layer damping (or for that matter, any other damping enhancement technique) will be ineffective in controlling vibrations. Establishing that a vibration problem exists requires a comparison of the vibration environment, specifically its frequency content, with the dynamic characteristics of the structure. Therefore, the problem definition task must generate an accurate characterization of both the vibration equipment and structural modal parameters. In addition to this, the problem definition task also requires the specification of the thermal environment and other

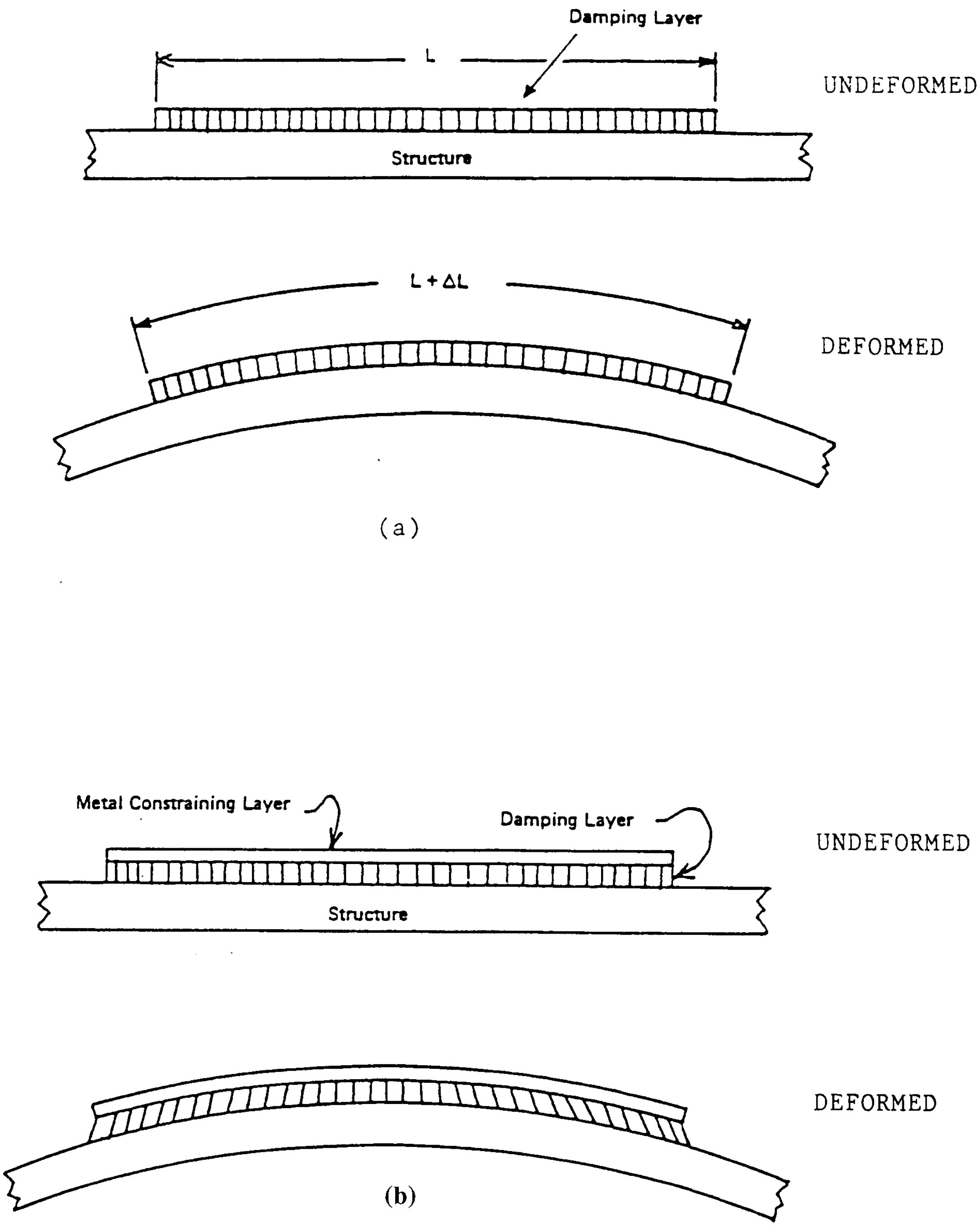


Figure 5-6. (a) Free Layer Treatment (b) Constrained Layer Treatment

habitable environmental requirements such as outgassing. These parameters are needed because damping treatments use VEMs whose properties vary with frequency and temperature. An accurate characterization of these parameters will enable the designer to pick a VEM with optimal damping and modulus under the operating conditions, and simultaneously meet the outgassing requirements. Outgassing requirements for polymeric materials used in space applications include collected volatile condensable material content and Total Mass Loss.

2) Identify prominent structural modes to be damped. This is based on excitation and structural characterization (Step 1), and by performing transient and/or random response analysis using finite element model codes such as NASTRAN.

3) Locate areas of high strain energy for each identified mode to apply the constrained layer damping treatment. Calculation of modal energy distributions is a standard option in NASTRAN. The computed strain energies are usually broken down by element type and written to an output file. Post-processing programs like I-DEAS can read this data and produce graphical output for easy visualization of the high strain areas.

4) Select the VEM based on environmental criteria established in Step 1. The VEM properties (loss factor and modulus versus frequency and temperature) are provided by the manufacturer, usually in the form of reduced frequency nomograms. Table 5.2 provides a list of space-qualified VEMs.

TABLE 5.2. VEMs RECOMMENDED FOR SPACE APPLICATIONS

VEM	Manufacturer
1) DYAD 601, 606, 609	SOUND COAT
2) SMRD 100 F90, 100B50	GE
3) ISD 110, 112, 113	3M
4) ISODAMP C-1002, 1105, 1100	EAR CORPORATION
5) HIDAMP II	BARRY CONTROLS
6) SPE/D	LORD CORPORATION
7) DENSIL 2078 I, III	DENSIL
8) VITRON RUBBER	VITRON
9) KALREZ 1058	DUPONT

5) Incorporate the VEM in the finite element model at locations identified in Step 3. To model the layered damping treatment in NASTRAN, the base and the constraining layers are typically modeled using QUAD4 and TRIA3 plate bending elements, and the viscoelastic core is modeled using HEXA and PENTA solid elements. The plate nodes are offset to one surface of the plate to coincide with the corner nodes of the adjoining solid element. This produces a coupling between the stretching and bending of the plate elements which is defined through their property cards. The

ability to account for membrane-bending coupling of plate elements through this offset feature is very important to represent proper behavior of the sandwich treatment.

6) Perform modal analysis and solve for system loss factor using Modal Strain Energy (MSE) principles. The MSE technique relates modal loss factor, η^j, of the system to the strain energy distribution and material loss factors.

$$\eta^j = \sum_{i=1}^{M} \frac{\eta i SE_i^j}{SEj}$$

where η_i is the material loss factor for VEM number i, SE_i^j is the strain energy in material *i* due to deformation in mode *j*, and SE^j is the total strain energy in all the elements due to deformation in mode *j.* It should be noted that modal analysis of systems involving VEMs must be accomplished iteratively, as the modulus (stiffness) of the VEM is a function of frequency and is not known a *priori.*

7) Perform trade studies to arrive at an optimal design which meets or exceeds the performance requirements. The parameters to be varied include VEM material properties, constraining layer stiffness, and location of the treated area.

8) Perform transient and/or random vibration analysis, with enhanced damping values achieved through constrained layer treatments to analytically demonstrate the response attenuation.

5.2.3 Acoustic Abatement Incorporated in HRF Racks for Noise Control

The acoustic abatement materials for HRF racks included Poron manufactured by Bisco Products which provides an acoustical barrier and Willtech Melamine foam manufactured by Illbruck for noise absorption. Poron used in two thickness, 0.1 inch and 0.05 inch, is placed next to the inside shell of the rack on both the sides and the back. Half-inch acoustic foam is placed on the sides of the rack adjacent to the Poron and down the centerline of the rack in between the payload drawers. Three-inch acoustic foam is placed in the back of the rack on one side, opposite of where the Rack Interface Controller (RIC) and Solid State Power Control Module (SSPCM) are located. The Poron and foam are contained in Nomex and attached to the rack with Velcro. The assembly is referred to as an acoustic pouch.

Openings between the rack and the rack-mounted payloads, and openings around the front panel handle latches allow sound to escape from the HRF racks. Elastofoam (an Electromagnetic Interface (EMI) shielding and environmental sealing gasket) manufactured by Tecknit is used to seal the openings between the rack and the rack-mounted payloads. Poron is used to cover the openings between rack-mounted payloads and around the handle latches. The Poron is covered in Nomex and mounted to the payloads with Velcro. These assemblies are referred to as acoustic closeout strips. Additional openings around the Cooling Stowage Drawer panel handle latches are covered with Delrin manufactured by

Boedeker Plastics, Inc. and held in place via ball plungers. This assembly is referred to as an acoustic closeout clip. Figures 5-7, 5-8, and 5-9 show the acoustic abatement material integrated in HRF Rack 1.

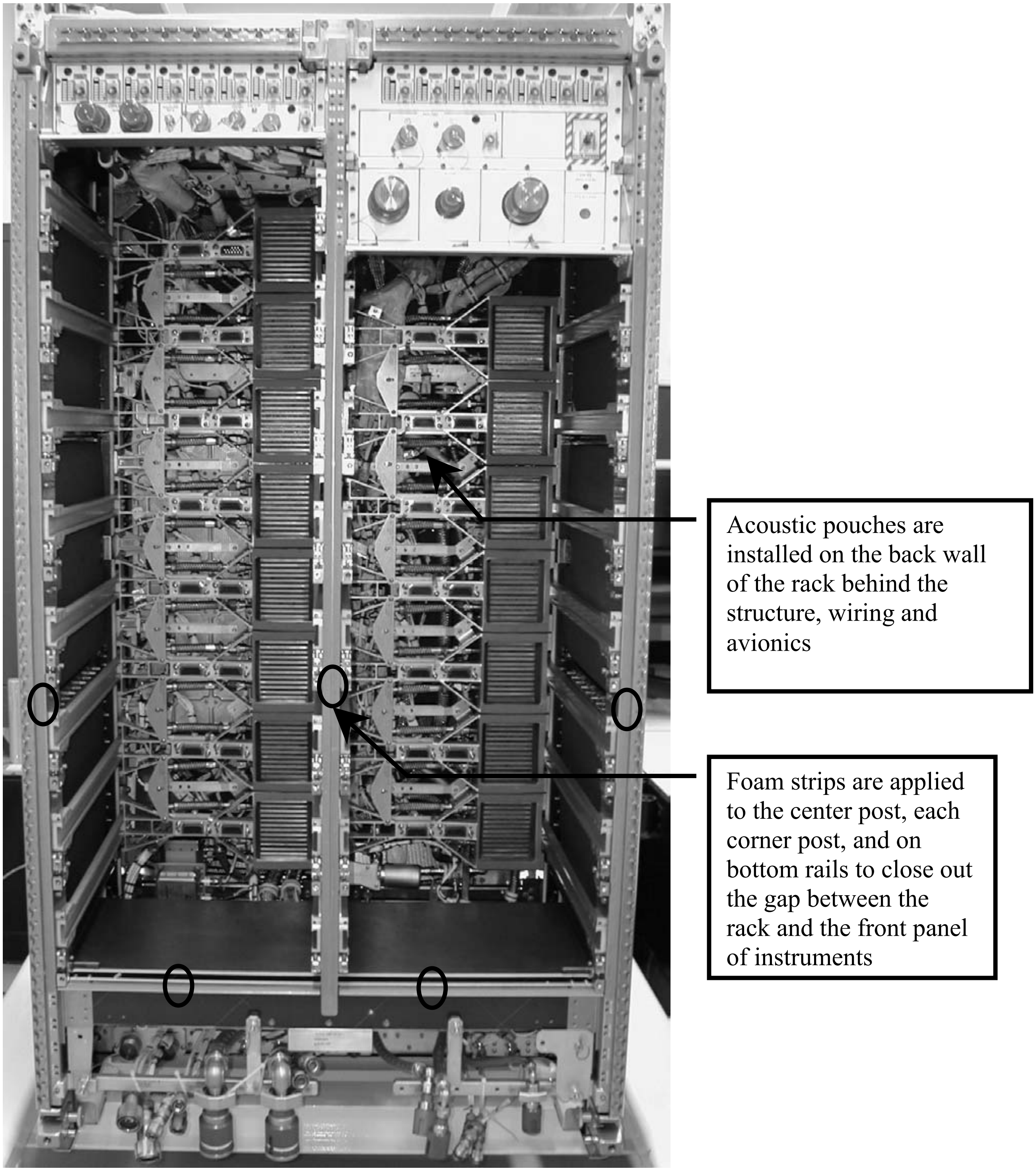


Figure 5-7. HRF Elastofoam Gasket

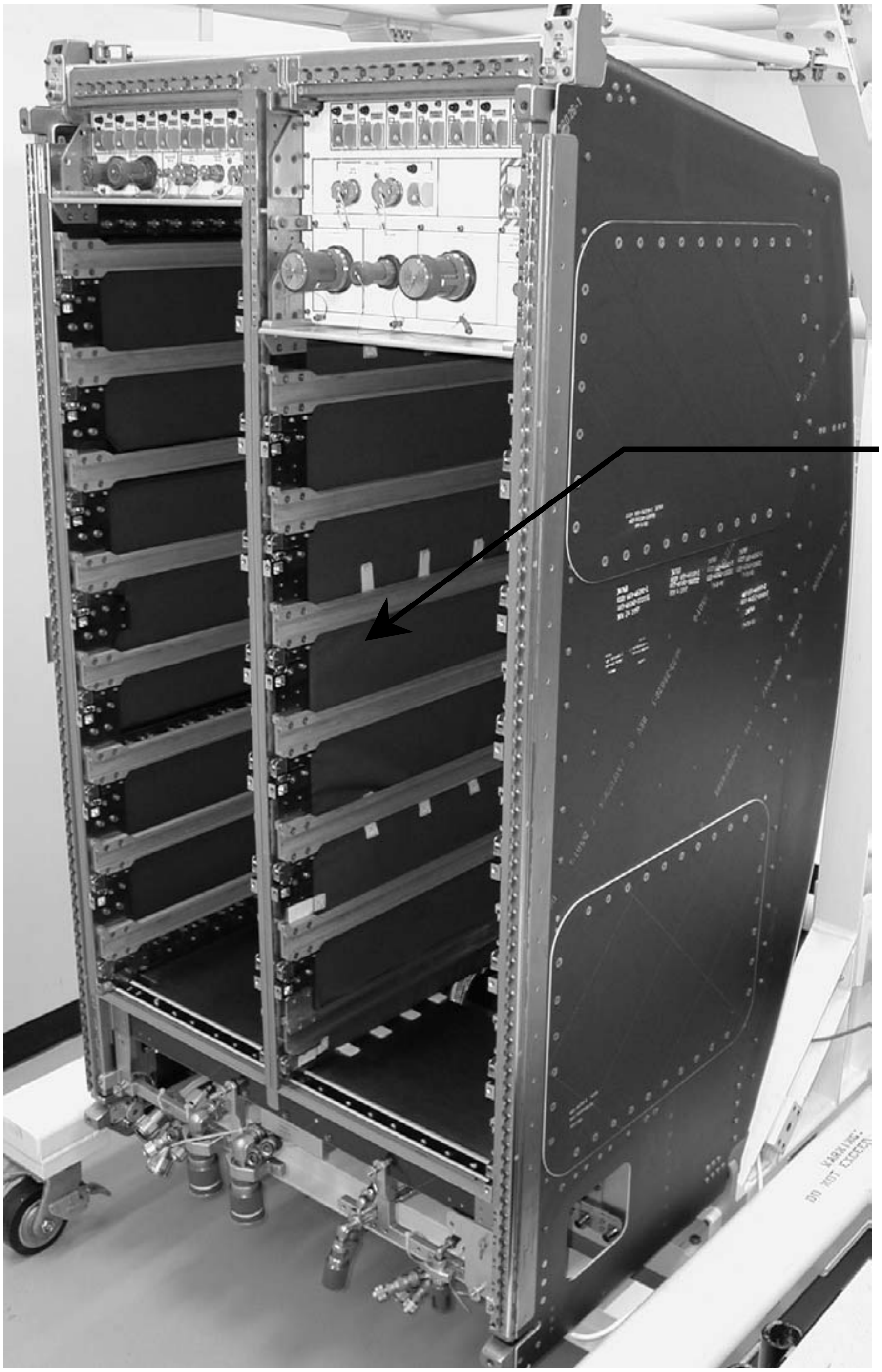


Figure 5-8. HRF Acoustic Pouches

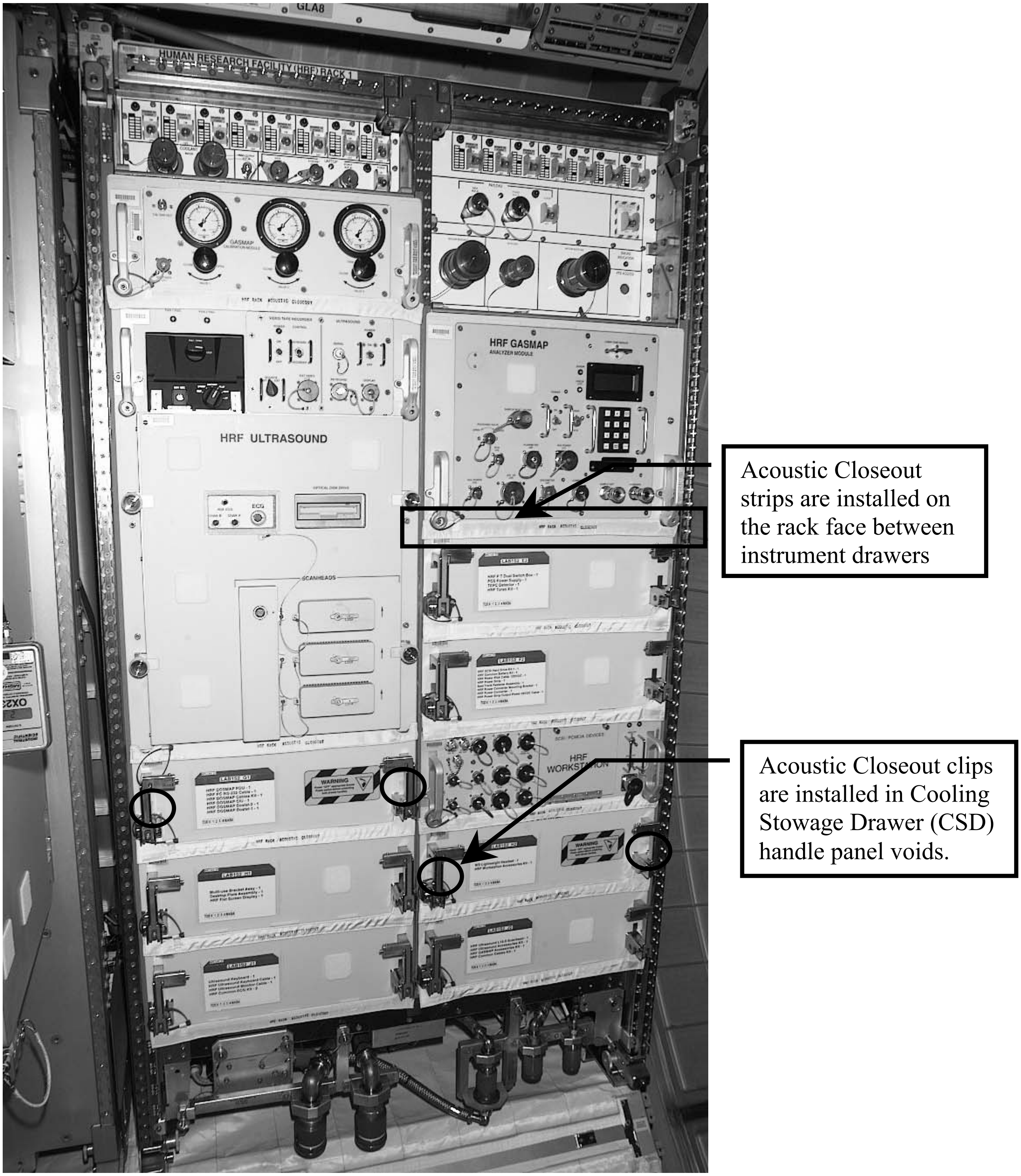


Figure 5-9 HRF Acoustic Closeout Stripes and Clips

6.0 SUMMARY

This document has presented a general plan for acoustic analysis and design of noise reduction systems for air-borne and structure-borne sound. A process flow chart for controlling acoustic noise is given in Figure 6-1.

The procedures established in this report provide a design approach for conducting acoustic analysis and test. The ISS imposes SPLs limitations on the payload rack. A procedure was established in this report to determine the maximum allowable SPLs generated by individual noise sources based on the overall rack noise limitations and number of operating units in the EXPRESS Rack. The composite noise level in the HRF express rack will be determined based on measured equipment noise data, equipment timeline, and equipment location within the EXPRESS Rack. The acoustical noise control and analysis plan is outlined as follows:

- Identify acoustical requirements for the HRF rack imposed by the ISS.
- Identify all noise source including:
  - Part number, location
  - Phase (continuous or intermittent or mixed)
- Specify SPLs limit for each source (sub-allocation process) based on the total rack limit and number of noise sources that will be operated simultaneously in flight including the EXPRESS rack subsystem.
- Determine source to listener noise paths:
  - Airborne
  - Enclosure transmission
  - Structure borne
- Each hardware must avoid exceeding its sub-allocated limit by:
  - Selecting quiet components (fans, pumps, etc.)
  - Using the process noise control discussed in Section 5.0
- Estimate noise level of each hardware component to determine compliance with specifications and make modifications as required. (Noise test requirements are not discussed in this plan in detail).
- Identify hardware/systems that exceed their sub-allocated limits and require noise control measures.
  - Evaluate severity of the exceedance and techniques to bring the unit within compliance
  - Assess cost, weight, schedule impact
  - Optimize
- Estimate combined noise level for the rack based on the measured equipment noise data, equipment timeline, and equipment location within the rack.
- Continuously update analysis to reflect hardware/rack configuration changes.

The focus has been primarily on passive noise reduction implementation, including isolators and damping devices. Survey results show that constrained layer damping treatments are a popular choice for many damping applications, mainly because of their cost and weight effectiveness. The application of viscoelastic damping of the structural attachment of the fans will be a preferred approach for the HRF payload and racks.

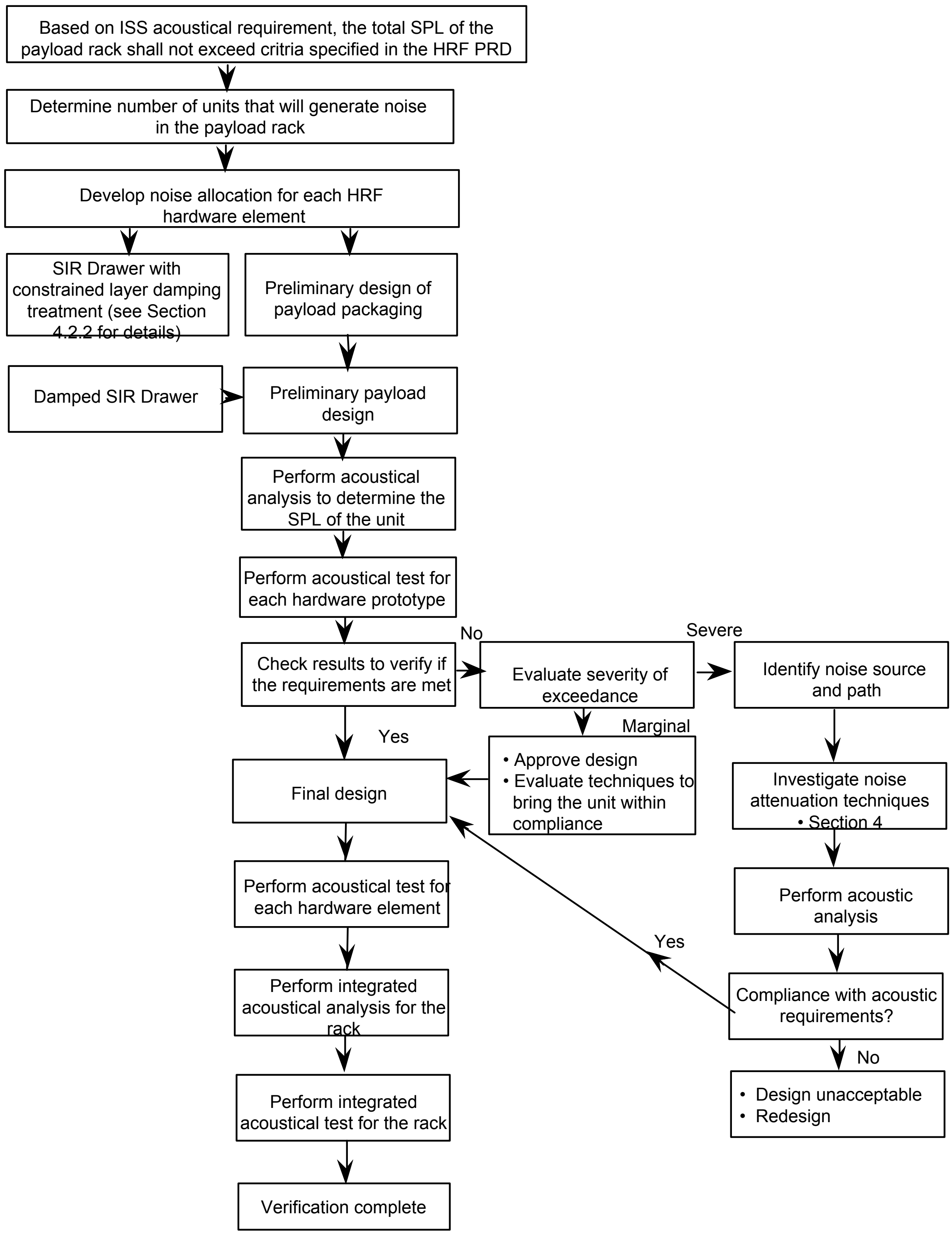


Figure 6-1. Process Flow Chart for Controlling Acoustic Noise

APPENDIX A

ISS ACOUSTIC DESIGN REQUIREMENTS FOR PAYLOADS (P/Ls)

NOISE MEASUREMENT PROCEDURES FOR ISS PAYLOADS AND PAYLOAD RACKS

I. BACKGROUND

Historically some payload developers have disregarded noise specifications to the point of simply measuring the resulting product noise and then requesting a waiver. Traditionally this happens after the flight hardware has been built and tested and when all the money and schedule have been depleted. At this point there are only two likely options: demanifest the payload or approve a waiver. The NASA has recognized that quieter hardware will result if the hardware developer takes the following steps: be attentive to acoustic requirements throughout the length of the program; be selective in choosing quiet prime movers (fans, pumps, etc.); establish and implement noise control plans; and do noise testing from the onset of design and continue it periodically throughout the development stages.

To ensure compliance with the acoustic requirements, it is recommended that rack integrators consider the acoustic benefits of vibration isolation, payload positioning, packaging, partitioning, encapsulation, and other noise control techniques within the rack. With this in mind, the ISS noise specifications strive for simplicity and commonality to help insure uniform application throughout the payload and equipment development community. The prime objective is to provide hardware that complies with the noise specifications, thereby avoiding noise problems that are deemed unacceptable and too expensive to fix after equipment is designed and manufactured.

It is necessary that ISS know beforehand what the acoustic noise will be at various locations within the ISS in order to know if the acoustic environment complies with the specification and to satisfy medical concerns related to the overall habitability of the crew's working environment. The most accurate and straightforward method of making this determination would be to install all the equipment in each habitable module and then measure the total noise from all the payloads and the ISS systems noise. Another good approach would be to measure only the noisiest equipment and payloads installed in flight configurations aboard a high-fidelity full-size mockup and add this data to measuredISS systems noise. Both of these methods will probably not be practical for most equipment developers. However, to provide support for mission planning, NASA will use the hardware suppliers' engineering assessments. Reasonable accuracy will be achieved if noise control and testing are implemented.

High-quality acoustic noise predictions can be made if payload and equipment developers make sound power measurements of their hardware. Sound power measurements do not require that testing be conducted in a high-fidelity mockup since they are independent of the measuring room environment.

II. PURPOSE

The purpose of this document is to provide guidelines for the measurement of ISS test articles. Test articles may be individual equipment items that will later be mounted in a rack with other equipment or a payload rack of several equipment items. The noise requirements contained in this procedure refer either to a stand-alone payload item or a partial or full integrated rack assembly. These noise measurements will be used to verify whether or not the integrated payload rack meets the acoustic requirements established in ISS verification documents.

III. TEST ROOM REQUIREMENTS

To measure the noise of the test article, a simple test area or room will be required. The purpose of the test room is to provide an isolated area with background noise levels sufficiently lower than the noise levels produced by the test article to be measured. Ideally the background levels from continuous sources such as air conditioning should be more than 15 dB below the maximum allowable noise levels specified for the test article. Sources of undesirable intermittent background noise such as telephones, talking, personnel traffic in the vicinity, office machines, and public address systems should be eliminated. The test article to be measured should produce at least 3 dB above the background in each octave band to be measured. If this condition can not be achieved, it is acceptable if the test article noise levels plus the background noise levels are below the maximum allowable values provided in the acoustical specification.

The room dimensions should be as large as possible and the inner surfaces of the walls, floors, and ceiling should be as acoustically-absorbent as possible. The intention of this is to reduce the strength of reflected acoustic waves. It is highly desirable that the minimum width of the room be at least six meters and in all cases at least four meters. Large acoustically-reflective articles such as bookcases, tables, filing cabinets, etc. should be removed from the room or placed more than three meters away from the test article.

IV. ORIENTATION AND PLACEMENT OF THE TEST ARTICLE

If the test article is one of the following: an independently operated payload; an individual payload that will be placed in a rack; or a loaded rack, then it should be placed on a small table or stand about one meter high near the center of the room. See Figure 1. If possible, place the surface of the test article at least two meters from the nearest wall, but do not place the test article exactly in the center of the room. Position the test article surface to be measured flush with the edge of the test stand and orient the stand so that the side surfaces of the test article are not parallel with any of the room walls.

Ancillary equipment needed to power, configure, or monitor the test equipment should be either very much quieter than the test article or placed in another room and connected by long cables through feed-throughs or under closed doors.

V. TEST EQUIPMENT AND CALIBRATION

A precision sound level meter should be used to make the noise measurements. The SLM shall comply with the Type 1 instruments described in the American National Standard Institute (ANSI) S1.4-1983 Specification for Sound Level Meters. For example, a Bruel & Kjaer Type 2230 SLM would be a suitable instrument. The SLM provides a microphone preamplifier and the A-weighting filter that is needed; however, as in the case of the B&K 2230 does not provide octave-band resolution without connecting it to another device such as an octave filter set or a real-time frequency analyzer. Octave filter sets such as the B&K Type 1625 shall meet requirements established in ANSI S1.11-1986. The SLM and filter set/real-time frequency analyzer shall have been certified by a recognized calibration laboratory within the prior 12 months. Immediately before and immediately after noise measurements are made on the test article, the SLM shall be calibrated with a calibrator or pistonphone that has also been certified by a recognized calibration lab during the prior 12 months.

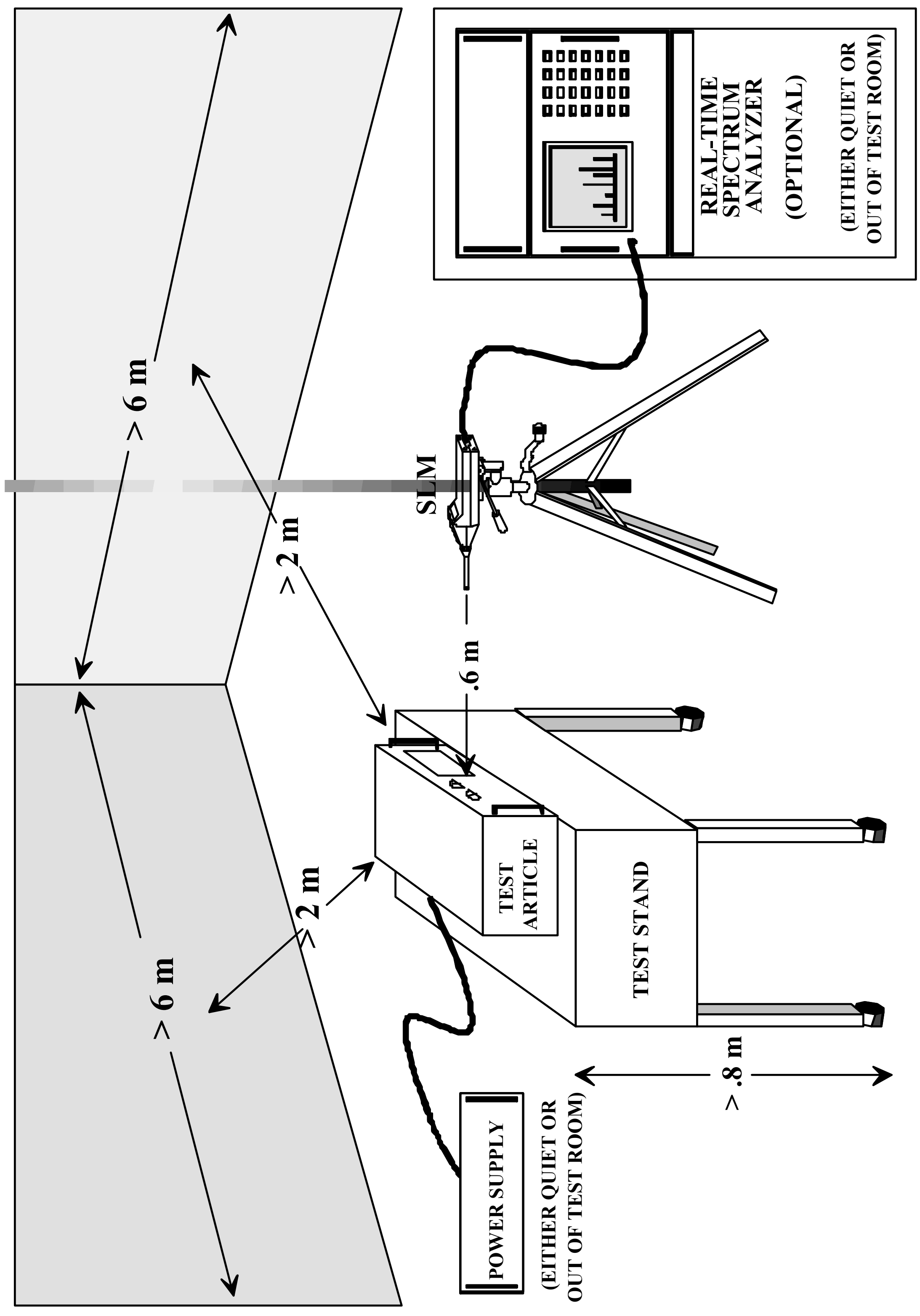


Figure 1. Typical Test Setup

Acoustic data acquisition shall be performed by a person who is familiar with basic techniques used to range the equipment to optimize signal-to-noise ratios without clipping data, knows how to make meaningful background noise measurements, and understands the fundamentals of physical acoustics.

VI. CONFIGURATION OF THE TEST ARTICLE

The test article should be configured as much as practical to be representative of how it will be mounted in the ISS, such as placement in appropriate payload racks or mockups to simulate some degree of encapsulation or being acoustically closed-out.

The test article should be operated in the mode or setting that will occur on-orbit that produces the maximum noise. If this mode is not known beforehand, it can be determined by operating the test article in several candidate modes at nominally-expected parameters and making noise measurements with a hand-held sound level meter (using A-weighting), at 0.6 meters from the front (inboard) panel of the test article or the loudest part of the test article if it is not flown in a rack.

VII. DATA ACQUISITION

Quality Assurance inspection of the testing within this procedure shall be in accordance with the standard inspection practices of the organization performing the measurements. Using a hand-held SLM in the A-weighting mode conduct a roving survey of the front of a rack mounted payload at 0.6 meters away from the surface of the test article to determine the noisiest location on the front. If the test article does not fly in a rack then determine the loudest part of the test article. These locations will usually be cooling fan inlets and outlets. In the case of measuring noise from an outlet, a microphone wind screen should be used to prevent air turbulence noise at the microphone diaphragm.

After determining the location of maximum noise, place the SLM on a camera tripod with the microphone 0.6 meters from and pointed directly at the maximum noise area of the test article and operate the test article in the mode to be measured. The SLM microphone orientation switch should be in the frontal or free-field position.

If octave-band readings are made directly from the SLM, care should be taken to insure that the octave bands are not also being A-weighted by the SLM, and the SLM operator should position himself about 0.5 meters directly behind the SLM. The preferred octave-bands 63 Hz, 125 Hz, 250 Hz, 500 Hz, 1000 Hz, 2000 Hz, 4000 Hz, and 8000 Hz should be obtained with linear (no weighting or filtering) response. The SLM should be set to read root-mean-square (rms), SPL , Slow meter response (1 sec integration). In cases where the SLM readout is consistently fluctuating over several decibels, the maximum value of the data fluctuations shall be reported. The linear overall (OA) and the A-weighted overall (A-wtd) readings should also be obtained. In all readings use the lowest full-scale range setting that does not clip any of the signal to be measured.

After recording a set of readings at 0.6 meters with the test article producing noise, switch the test article off and record the background noise measurements with the SLM ranged at the same full-scale settings as was previously used when measuring the test article. If there are other operational modes to be measured, then operate the test article in each of the other modes and obtain the acoustic data, taking care to document the operating parameters on the data logs.

The AC output from the SLM can be connected directly to a real time spectrum analyzer with octave analysis capability to greatly facilitate the data acquisition process. A real time spectrum analyzer will permit all the frequency bands to be obtained simultaneously rather than sequentially and permit printing out all the desired data in tabular form on hard copies. If a real time spectrum analyzer is to be used it will also have to calibration-lab certified within the past 12 months and be calibrated from the SLM with a known reference signal. Many Sound Level Meters such as the B&K 2230 will output a 1000 Hz reference tone for this purpose or a pistonphone may be used. The real time spectrum analyzer input range should be set as low as possible without data clipping to read the output of the SLM in the OA mode, 0.6 meters from the test article when it is producing maximum noise. The background noise measurements should also be read without reconfiguring the real time spectrum analyzer because the electronic background levels will in most cases be higher than the acoustic background levels in a quiet room if the real time spectrum analyzer has been ranged to measure a fairly loud piece of equipment.

It is not the intention to measure the acoustic background levels with accuracy but to determine how much electronic and acoustic background exist at the settings used when making measurements of the test article. The dynamic range of the instrumentation will be sufficient to adequately measure the noise at all operational modes without changing ranges on either the SLM or the real time spectrum analyzer.

VIII. DATA REPORTING

An example data packet format is provided at the end of this procedure to provide a sample to indicate typically how the data should be documented. In the example it is assumed that the one payload operates continuously and is the only significant noise source in the entire payload rack and therefore must be less than NC-40 at 60 centimeters from the front of the payload. The following type of information should be reported:

a. If the test article measures less than 37 dBA at 0.6 meter from the loudest part, part b. below is not required.

b. Adjusted octave band data (in a tabular format) measured at 0.6 from the front of the test article (or the noisiest surface). Round all SPL data to the nearest integral dB (0.5 dB shall be rounded up).

   The raw octave-band data must be adjusted because it may contain unwanted amounts of background noise and the OA and A-weighted data as read from the SLM or the real time spectrum analyzer will include energy outside the frequency range of interest where ISS acoustic requirements are concerned; therefore, adjustments have to be made to the data.

   Background adjustments for each octave band may be accomplished by employing the following equation:

$$L_{\text{test article}} = 10\ \mathrm{LOG}_{10}\ (10^{(Ltot/10)} - 10^{(Lbkg/10)})$$

   where

   Ltot = total noise measured when test article was on and where

Lbkg = measured background noise when test article was off.

Once background noise corrections have been made, new OA and A-weighted values need to be computed using the following equations:

$$O. A. = 10 \, LOG_{10} \, (10^{(L63/10)} + 10^{(L125/10)} + 10^{(L250/10)} + 10^{(L500/10)} + 10^{(L1000/10)} + 10^{(L2000/10)} + 10^{(L4000/10)} + 10^{(L8000/10)})$$

where

L63 = the SPL in the 63 Hz octave band and

$$A\text{-wtd} = 10 \, LOG_{10} \, (10^{((L63 - 26.2)/10)} + 10^{((L125 - 16.1)/10)} + 10^{((L250 - 8.6)/10)} + 10^{((L500 - 3.2)/10)} + 10^{(L1000/10)} + 10^{((L2000 + 1.2)/10)} + 10^{((L4000 + 1)/10)} + 10^{((L8000 - 1.1)/10)})$$

c. Name and telephone number of test conductor.

d. Dates of testing.

e. Payload Part Number and Serial Numbers.

f. Type of test equipment used and calibration dates.

g. Duty cycles and duration of each mode of operation of the test article that will occur on orbit.

h. Test procedure indicating Quality Assurance acceptances.

j. A sketch or figure of the test setup in the test room should be included if the test conductor thinks it would help clarify under what circumstances the data were obtained.

VATF
Acoustic Data Packet

Test Article: Payload Experiment Equipment (PXE)

Parts Number: SED39128357-301

S/N: 1001

Flight: STS-81 Mid-deck

Test: Acoustic Noise Emission

Laboratory: Acoustic Quiet Room

Facility TPS No: FA9720003

Test Date: 23 April 1997

Report Date: 24 April 1997

Test Engineer: James L. Warnix, (713) 483-6384

Vibration & Acoustic Test Facility
Structures & Mechanics Division
Johnson Space Center
Houston Texas

TASK PERFORMANCE SHEET

NASA - LYNDON B. JOHNSON SPACE CENTER

1. TYPE A — CONFIGURATION CHANGE	2. TPS NO. FA9720003		3. PAGE 01 OF 02	
PERMANENT / TEMPORARY	4. MOD SHEET(S) NUMBER(S)	5. ORG. ES4	6. SYSTEM ISS	7. PROJECT SD3
B — NONCONFIGURATION CHANGE: X				
8. PART NAME: Payload Experiment Equipment	9. PART NO./DRAWING NO.: SED39128357-301	10. SERIAL/LOT: 1001	11. TIME/CYCLE USED: YES [ ] NO [X]	
12. APPLICABLE DOCUMENTS	13. CONTRACT NO./JOB NO.: NAS9-19100	14. HAZ. TEST: YES [ ] NO [X]	15. ENG. INVEST.: YES [ ] NO [X]	

16. SHORT TITLE OF TPS

Noise Emission Test of the Payload Experiment Equipment (PXE)

OPER SEQ. NO.	17. OPERATIONS (*Print, Type, or Write Legibly*)	VERIFICATION 18. TECH.	19. QA
1.	**Calibrate a B&K Type 2230 Sound Level Meter (SLM) using a B&K 4228 pistonphone, and calibrate the Ono-Sokki SR-5300 real-time octave-band analyzer using the SLM's reference signal.**		
2.	**Turn off air handlers and other equipment as needed to reduce the background noise levels.**		
3.	**Determine the noisiest (A-weighted) location on the front 60 cm ± 1 cm from the test article front panel and record the data below:** **Front: ______ dBA**		
4.	**Range the data acquisition equipment for maximum signal-to-noise ratio with the microphone 60 cm ± 1 cm from the noisiest location with the equipment operating in its noisiest mode.**		

20. ORIGINATOR	DATE	21. FINAL ACCEPTANCE STAMP AND DATE
James L. Warnix	4/23/97	

APPROVALS (*Printed or Typed and Signed*)

22. TEST PROJECT ENGINEER	DATE	23. QUALITY ENGINEER	DATE
J. Warnix			
24.		25.	
26.		27.	

JSC Form 1225 (Rev Aug 96) (MS Word Aug 96)

TASK PERFORMANCE SHEET CONTINUATION PAGE NASA - LYNDON B. JOHNSON SPACE CENTER	TPS NO.	FA9720003
	MOD NO.	

OPER SEQ. NO.	17. OPERATIONS *(Print, Type, or Write Legibly)*	VERIFICATION 18. TECH.	19. QA
5.	**Analyze 30 seconds of noise and obtain octave-band data from 63 Hz through 8kHz at a distance of 60 cm ± 1 cm from the front of the equipment and the noisiest side if it is not the front for the following condition: All electronics ON and all fans ON.**		
6.	**Record background noise at each data measurement position.**		
7.	**Close this TPS.**		

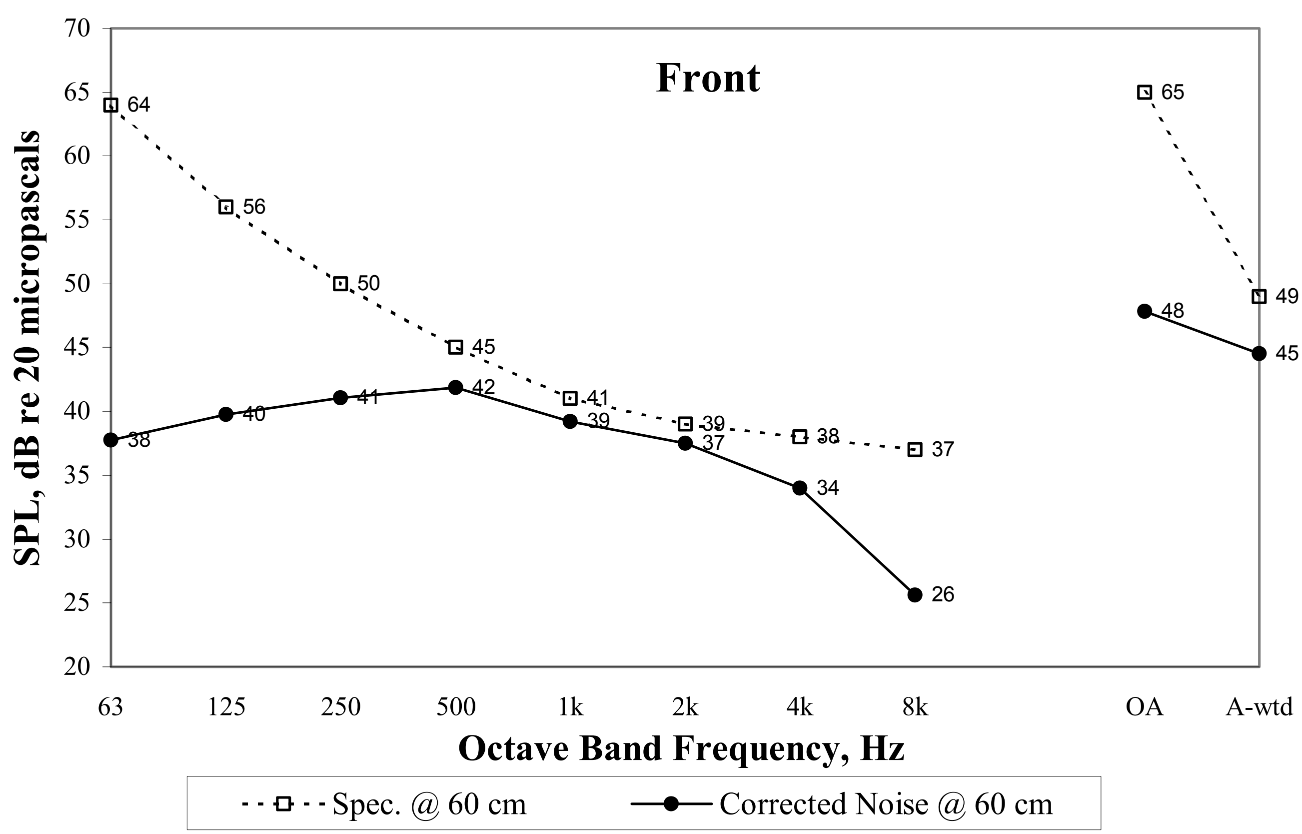
JSC Form 1225A (Rev Aug 96) (MS Word Aug 96)

Payload Experiment Equipment
Acoustic Certification Test

Part Number: 39128357-301 Serial Number: 1001
Test Location: VATF Quiet Room Test Date: 6 Jan 97

Operational Mode: ON with all fans ON

OCTAVE BAND (Hz)	60 cm. from Front of Unit			60 cm. Acoustic Requirement
	Total Noise	Background Noise	Corrected Noise	
63	45.5	44.7	38	**64**
125	41.0	34.9	40	**56**
250	41.2	25.8	41	**50**
500	41.9	20.5	42	**45**
1k	39.2	14.5	39	**41**
2k	37.5	11.3	37	**39**
4k	34.0	11.9	34	**38**
8k	25.8	12.2	26	**37**
OA	50	45	48	**65**
A-wtd	45	24	45	**49**



DISTRIBUTION LIST
FOR
LS-71011A

EB/E. Bauer
E. Strong

NT3/GFE Assurance Branch

SM/D. Grounds

SM3/C. Haven

EM2/G. Ecord

LOCKHEED MARTIN
C64/B. Henneke
C70/E. Azzi
S03/Science Payloads Library (3)
S18/G. Salinas
S22/C. Aguilar
S22/S. Bhaskaran
S22/L. Kimmel
S22/J. Thompson
S22/E. Witt
S22/M. Yalcinkaya
S361/J. McDonald
S362/TDI Center, Bldg. 36 (5)
S363/C. Amberboy