Reverberant Acoustic Test Facility (RATF)
Structural Design for Vibroacoustic Loads

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Structural Engineering Association of Ohio,
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April 25, 2012
Presentation Outline

• Introduction

• Design Requirements

• Structural Design for Vibroacoustic Loads:
  ➢ Chamber Wall Flexural Design
  ➢ Horn Room Piping Repair

• Construction Photos
Introduction
Introduction

• To support NASA’s developing space exploration program, the NASA Space Environmental Test (SET) Project was tasked to develop new test facilities, known as the Vibroacoustic Test Capability (VTC).
  – The Space Power Facility (SPF), located at the NASA Glenn Research Center’s Plum Brook Station in Sandusky, OH, USA is already the home of the world’s largest thermal vacuum chamber.
  – The new test facilities provides one-stop testing for a suite of space environmental testing. SPF has been augmented through the NASA Space Environmental Testing Project Office with new reverberant acoustic, mechanical vibration, modal, and electromagnetic environmental effects test facilities.
Space Power Facility, NASA Plum Brook Station
Sandusky, Ohio (50 miles west of Cleveland)
In August 2007, SAIC-Benham won the NASA prime contract to design and construct the acoustic, vibration and modal test facilities, as well as to provide the high speed data acquisition system to support these facilities.

- SAIC-Benham contracted with Aiolos Engineering Corporation to provide the acoustic design of the Reverberant Acoustic Test Facility (RATF).

Construction was completed in February 2011.

Acoustic verification testing to 161 dB overall sound pressure level (OASPL) was successfully completed in September 2011.
Vibroacoustic Test Capability (VTC)
Vibro-Acoustic Highbay Construction Photo
(taken mid-December 2010)
Design Requirements
RATF Design Requirements

• The RATF shall be as large as possible within the given space constraints of the SPF Vibro-Acoustic Highbay.
• The RATF’s test chamber shall be properly sized to acoustically test four space vehicle configurations, encompassing an 18-ft diameter test article, and a 47-ft tall test article.
• The RATF’s test chamber shall physically allow a 32.8-ft diameter test article weighing up to 120,000 pounds.
• The RATF shall generate the empty chamber acoustic test spectra shown in Figure 1, for continuous test duration of 10 minutes. These eight (8) “C” spectra represent a wide range of current and future NASA missions, including (5) spectra with a 163 dB overall sound pressure level (OASPL).
• The RATF acoustic control system shall control the noise sources in Fig. 1 within the following tolerances:
  - ±5 dB below the 50 Hz one-third octave bands (OTOB)
  - ±3 dB covering 50 Hz - 2KHz OTOB's
  - ±5 dB above 2KHz OTOB's
  - ±1.5 dB on OASPL
The C1-C8 test spectra provide a wide range of test curves, each providing a unique spectral control challenge. C2 has the highest low frequency SPL value.
RATF Design Summary

• SAIC-Benham and Aiolos designed the reverberant acoustic test chamber with the following dimensions: 47.5-ft long x 37.5-ft wide x 57-ft high. The chamber volume is ~ 101,000 cubic ft.

• The overall layout and key properties of the RATF chamber and horn room are illustrated in Figure 2. There will be a total of 36 modulators and 36 horns to produce the acoustic power to meet the RATF requirements. The RATF design (see Figures 3 - 7) has:
  - Eleven (11) MK-VII modulators distributed on the 25, 35, 50 and 80 Hz horns
  - Twelve (12) MK-VI modulators distributed on the 100 and 160 Hz horns
  - Thirteen (13) WAS5000 modulators on the 250 Hz horns

• The gaseous nitrogen (GN2) generation system (see Figure 8) is designed to meet the flow needs of RATF.
  - Water bath vaporizer capable of GN2 flow rate of 72,000 SCFM (standard cubic feet per minute)
  - One (1) 6,000 gallon liquid nitrogen (LN2) pusher tank
  - Two (2) 9,000 gallon liquid nitrogen (LN2) high pressure storage tanks
Figure 2. RATF Acoustic Design

<table>
<thead>
<tr>
<th>Chamber Properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chamber Size</td>
</tr>
<tr>
<td>Chamber Volume</td>
</tr>
<tr>
<td>Acoustic Modulators</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Horns</td>
</tr>
<tr>
<td>Maximum GN₂ flow rate</td>
</tr>
<tr>
<td>Main Door Opening</td>
</tr>
<tr>
<td>Number of Main Doors</td>
</tr>
<tr>
<td>Door Type</td>
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<td>OASPL, empty</td>
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Figure 3. Modulator/Horn Pairings

<table>
<thead>
<tr>
<th>Horn</th>
<th>25 Hz</th>
<th>35 Hz</th>
<th>50 Hz</th>
<th>80 Hz</th>
<th>100 Hz</th>
<th>160 Hz</th>
<th>250 Hz</th>
<th>TOTAL</th>
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<tr>
<td>Modulator</td>
<td>MKVII</td>
<td>MKVII</td>
<td>MKVII</td>
<td>MKVII</td>
<td>MKVI</td>
<td>MKVI</td>
<td>WAS5000</td>
<td>36</td>
</tr>
<tr>
<td>Final Design Count</td>
<td>2</td>
<td>2</td>
<td>4</td>
<td>3</td>
<td>4</td>
<td>8</td>
<td>13</td>
<td></td>
</tr>
</tbody>
</table>
Figure 4. Construction photo showing the installation of the final RATF horn (25 Hz)
Figure 5. RATF Horn Layout
Figure 6. RATF Construction Photo
(taken September 2010)
Figure 7. Construction photo of the RATF horn room (level 5) platform and modulators
Figure 8. Construction photo showing the RATF nitrogen generation system, including the water-bath vaporizer and the liquid nitrogen tanks and vaporizers.

The two 9,000 gal LN2 supply tanks that feed the water-bath vaporizer.

The 6,000 gal LN2 tank that feeds the two head-pressure vaporizers.

The (re-circulating) vaporizer that maintains head pressure on the 6,000 gal LN2 tank.

The vaporizers to maintain head-pressure on 9,000 gal LN2 tanks.
Structural Design for Vibroacoustic Loads:
  ➢ Chamber Wall Flexural Design
RATF Structural Design Methodology for Acoustic Loads

• The RATF wall structural design uses ACI 318-02 (American Concrete Institute) strength based design code (Load Resistance Factor Design – LRFD).

\[
\text{Factored Resistance} \geq \text{Factored Load}
\]

• ACI 318, Section 9.2 provides factored load combinations for various dead load and live load conditions.

\[
\text{Example: } U = 1.2 \, D + 1.6 \, L
\]

• ACI 318 does not provide load combination guidance for the RATF acoustic test live load.

• NASA GRC collaborated with Dr. Arthur A. Huckelbridge, a structural engineering professor at Case Western Reserve University and registered professional engineer, to determine the appropriate live load factor for RATF wall flexural design.
RATF Wall Design due to Acoustic Loading
3-Step Process

**Step 1)** Define the RATF chamber acoustic test excitation using the “enveloping case” in units of Sound Pressure Level (SPL) versus 1/3 octave band frequency (Hz). Convert the SPL to an acoustic Power Spectral Density (PSD) spectrum.

\[
\text{Acoustic PSD } \left( \frac{Pa^2}{Hz} \right) = \left[ P_{ref}^2 \times 10\left( \frac{\text{SPL}}{10} \right) \right] \left/ \left[ \frac{1}{3} \text{octave bandwidth} \right] \right.
\]

where \( P_{ref} = 20 \times 10^{-6} \) pascals (Pa)

**Step 2)** Apply the acoustic PSD (from Step 1) to excite the RATF finite element structural model (SAP 2000). The chamber structure has 95% cumulative modal effective mass fraction (or greater) in each translational direction below 50 Hz, so the acoustic excitation is applied between 2-50 Hz. Bending moments \((M_u)\) are computed for each interior chamber surface.

**Step 3)** Use the bending moments \((M_u)\) from Step 2 to size the rebar necessary for flexural design of each interior chamber surface.

\[
\phi M_n \geq M_u
\]
The C2 test spectrum has the highest SPL value in the low frequencies.
Load Resistance Factor Design (LRFD)

Assume $R$ represents structural resistance (strength)
Assume $R$ is a normally distributed random variable with mean $R^*$ and std dev $\sigma_R$

Assume $S$ represents structural load effect
Assume $S$ is a normally distributed random variable with mean $S^*$ and std dev $\sigma_S$

Define $Z = R - S$
$Z$ will be a normally distributed random variable with mean:

$$Z^* = R^* - S^* \quad \text{and std dev } \sigma_Z = [\sigma_R^2 + \sigma_S^2]^{0.5}$$

A structural failure will occur if $Z < 0$
\( \beta \) (safety index) represents the degree of conservatism desired or acceptable. For “satisfactory” structural performance (no failure): \( Z^* > \beta \sigma_Z \)
Load Resistance Factor Design (LRFD)

- Separate combined uncertainty into the resistance and load contributions:

$$\sigma_Z = \sqrt{\sigma_R^2 + \sigma_S^2} \approx 0.7 (\sigma_R + \sigma_S)$$

(Pythagorean theorem for isosceles right triangle; good if \(\sigma_R\) and \(\sigma_S\) not TOO different)

$$Z^* > \beta \sigma_Z \Rightarrow R^* - S^* > 0.7 \beta (\sigma_R + \sigma_S) \Rightarrow R^* - 0.7 \beta \sigma_R > S^* + 0.7 \beta \sigma_S \Rightarrow$$

$$R^*(1 - 0.7 \beta V_R) > S^*(1 + 0.7 \beta V_S) \text{ where } V_R = \frac{\sigma_R}{R^*} \text{ and } V_S = \frac{\sigma_S}{S^*}$$

**1 - 0.7 \beta V_R = resistance factor** and **1 + 0.7 \beta V_S = load factor**

- Distinct load and resistance factors must be developed for different resistance mechanisms (flexure, shear, torsion, stability, etc.) as well as different load sources and load combinations (dead, live, wind, seismic, blast, etc.).

Load Resistance Factor Design (LRFD)  
Factored Resistance $\geq$ Factored Load

$$R^* \ [1.0 - 0.7 \ \beta \ V_R] \geq S^* \ [1.0 + 0.7 \ \beta \ V_S]$$

where:

Resistance Factor $= [1.0 - 0.7 * \beta * V_R] = 0.9$ (ACI 318 code for flexural design)

Load Factor $= [1.0 + 0.7 * \beta * V_S]$

$R^*$ = mean structural resistance (capacity)

$S^*$ = RMS acoustic test load

$\beta$ = safety index (historically 2.5 – 3.0 for civil structures)

$V_R$ = coefficient of variation for the structural capacity $= \sigma_R / R^*$

$V_S$ = coefficient of variation for the load $= \sigma_S / S^*$

Coefficient of Variation = \textit{ratio of the standard deviation of the mean square pressure to the space-averaged value of the mean square sound pressure}

ACI 318 does not prescribe an “Acoustic Testing Live Load Factor.”  
The following slides develop the computation of this load factor.
Statistics of the Acoustic Sound Pressure Field
- Schroeder Frequency

• Statistical analysis of the chamber sound pressure field at locations away from the chamber walls can be divided into three frequency ranges – low, mid and high – with the Schroeder frequency, \( f_s \), as the crossover frequency between low and high frequencies.

• The Schroeder frequency is defined as:

\[
f_s = 2000 \sqrt{\frac{T_{60}}{V}} \text{ Hz in mks units}
\]

where \( T_{60} \) = chamber reverberation time (seconds)
\( V \) = chamber volume (m\(^3\))

• At frequencies above \( f_s \), the sound pressures for bands of noise (e.g. 1/3 octave bands) in the chamber are approximately uniform. At lower frequencies, the wide-band sound field in the chamber can show several peaks corresponding to individual room modes.

Statistics of the Acoustic Sound Pressure Field
- Normalized Variance

- The normalized variance, $\nu^2$, is defined as the variance $\sigma^2$ of the mean square pressure normalized with respect to the square of the space-averaged value of the mean square pressure:

$$\nu^2 \langle \bar{p}^2 \rangle = \frac{\sigma^2 \langle \bar{p}^2 \rangle}{\langle \bar{p}^2 \rangle^2}$$

where $\langle \bar{p}^2 \rangle$ denotes the space-averaged value of the mean square pressure.

- The coefficient of variation (COV) is the square root of the normalized variance:

$$\nu = \text{COV} = \frac{\sigma}{\langle \bar{p}^2 \rangle}$$

The normalized variance, $\nu_L^2$, of the sound field at low frequencies is defined as:

\[
\nu_L^2 = \left[ 1 + \frac{Bn}{\pi} \right]^{-1}
\]

where:

- $B = \text{frequency bandwidth} = 0.23 f_c$ for 1/3 octave bands
- $f_c = \text{band center frequency}$
- $N = \text{modal density}$

\[
N = \frac{4\pi f^2 V}{c^3} + \frac{\pi f S}{2c^2} + \frac{P}{8c}
\]

- $V = \text{chamber volume}$
- $S = \text{total area of chamber walls, floor, and ceiling}$
- $P = \text{total length of all edges}$

Statistics of the Acoustic Sound Pressure Field
- Acoustic Live Load Factor

• Based on a statistical review of the microphone pressure time histories from the TEAM modulator characterization testing at the U.S. Army Redstone Technical Test Center (RTTC) in Huntsville, Alabama and the National Research Council (NRC) in Ottawa, Canada, a $V_S = 0.75$ was calculated.

Assuming:
$\beta = 3.0$ (safety index, historically 2.5 – 3.0 for civil structures)
$V_S = 0.75$

**Acoustic Testing Live Load Factor** = $[1.0 + 0.7 * \beta * V_S] = 2.6$

<table>
<thead>
<tr>
<th>Statistical Analysis of Microphone Test Data from NRC (Positive Valued Pressure)</th>
</tr>
</thead>
<tbody>
<tr>
<td>mean</td>
</tr>
<tr>
<td>max</td>
</tr>
<tr>
<td>min</td>
</tr>
<tr>
<td>stdev</td>
</tr>
<tr>
<td>COV = $V_S$</td>
</tr>
</tbody>
</table>

Acoustic Live Load Factor = 2.6 was used for the RATF wall design. The 2-way slab design is 2 feet thick concrete reinforced with #8 rebar to resist bending moments.
Statistics of the Acoustic Sound Pressure Field
- Normal Distribution Evaluation
The microphone time history from the TEAM MK- VII modulator data on the 25 Hz horn is normally distributed. For a normal distribution of 2.6 \( \sigma \) above the mean, the corresponding load non-exceedance probability is \(~0.9953\).
RATF acoustic verification testing achieved 161 dB OASPL using the “C5 – 2dB” design test spectrum.
Measured versus Predicted "Noise Reduction" from inside Chamber to Mezzanine
("C5 - 2dB" Acoustic Verification Test)

- Measured
- Predicted (Statistical Energy Analysis)

RATF Wall Critical Frequency = 31 Hz

Measured “Noise Reduction” is less than predicted at frequencies greater than 160 Hz OTOB.
Structural Design for Vibroacoustic Loads:

- Horn Room Piping Repair
RATF Horn Room Illustration
Cutaway View of 5 Levels
RATF Horn Room Piping System

10 inch riser
12 inch riser
4 inch connector
LEVEL 1
LEVEL 2
LEVEL 3
LEVEL 4
LEVEL 5
Typical "T-Junction"
Typical TEAM modulator
Typical WAS 5000
4 inch connector
T-Junction Failures

• Detailed structural dynamic modeling of the RATF Horn Room piping system was initiated due to the vibration failure of T-junction near the TEAM modulator on 35 Hz horn. The piping system is constructed of Schedule 10 stainless steel piping.
T-Junction Failures *(in red)* from initial Acoustic Checkout Testing
RATF Horn Room Piping System

- TEAM Modulator (acoustic noise source)
- 35 Hz Horn
- 4” GN2 Piping
- 12” GN2 Piping
- Failed T-Junction
- Catwalk
Analytically Assess Piping System

• The objective of the structural dynamic analysis was to characterize the piping system modes and how they dynamically couple to the RATF building\(^1\) (<20 Hz) and catwalk\(^2\) (<17 Hz) structure modes.

• The forcing functions for the horn room are unknown (structure-borne vibration from RATF building, catwalk, modulators, and possible flow induced vibration).

• Recommendations were made to as to how best to decouple the piping system/modulator modes from the RATF building and catwalk modes. The analysis objective was to increase the piping system high effective modes to be about double the frequency of the RATF building and catwalk modes.


• **Reference 2:** “RATF Horn Room Catwalk Analysis,” by J. H. Kincaid, Benham Report, Revision 2, March 18, 2009.
Design Goal: Eliminate Dynamic Coupling

Reference: http://personal.cityu.edu.hk/~bsapplec/design2.htm

Design Goal: Increase the piping frequency high effective mass modes above 40 Hz, providing a factor of 2 separation with the RATF building and catwalk modes.
NASTRAN Dynamic Model

Mode 108, 15.65 Hz
6% Z-axis effective mass
5% Rotation-Y effective mass

Importance of Effective Mass:
Dynamic measure of global system vibration participation.
### Configurations Analyzed

#### Summary of Results

<table>
<thead>
<tr>
<th>Configuration Analyzed</th>
<th>TEAM Modulator Piping Modes</th>
<th>WAS 5000 Modulator Piping Modes</th>
<th>Piping System High Effective Mass Modes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Baseline Configuration</td>
<td>3.00-50.26 Hz</td>
<td>3.91-49.37 Hz</td>
<td><strong>10.66 Hz, 13.68 Hz, 15.65 Hz, 15.72 Hz, 47.20 Hz</strong></td>
</tr>
<tr>
<td>2. Adding lateral constraints to TEAM modulators</td>
<td>6.92-50.26 Hz</td>
<td>3.91-49.37 Hz</td>
<td><strong>10.44 Hz, 15.75 Hz, 13.29 Hz, 47.20 Hz</strong></td>
</tr>
<tr>
<td>3. Removing all constraints from the TEAM modulators</td>
<td>2.52-50.27 Hz</td>
<td>3.91-49.37 Hz</td>
<td><strong>10.63 Hz, 13.58Hz, 15.60 Hz, 15.70 Hz, 47.17 Hz</strong></td>
</tr>
<tr>
<td>4. Add 500lb mass to the base of the TEAM modulators</td>
<td>2.35-50.19 Hz</td>
<td>3.91-49.37 Hz</td>
<td><strong>10.17 Hz, 12.03 Hz, 14.87 Hz, 15.40 Hz, 47.33 Hz</strong></td>
</tr>
<tr>
<td>5. Isolate the TEAM Modulators – <strong>Gamma flex hose</strong></td>
<td>1.11-50.95 Hz</td>
<td>3.91-49.37 Hz</td>
<td><strong>10.66 Hz, 13.67 Hz, 15.63 Hz, 15.69 Hz, 47.20 Hz</strong></td>
</tr>
<tr>
<td>6. Isolate the TEAM modulators – <strong>Mason braided flex hose reoriented 90°</strong></td>
<td>2.90-50.80 Hz</td>
<td>3.91-49.37 Hz</td>
<td><strong>10.66 Hz, 13.67 Hz, 15.63 Hz, 15.69 Hz, 47.20 Hz</strong></td>
</tr>
<tr>
<td>7. Add new SAIC-Benham recommended pipe supports</td>
<td>3.05-100.14 Hz</td>
<td>3.91-100.21 Hz</td>
<td><strong>23.58 Hz, 33.51 Hz, 91.22 Hz, 94.19 Hz</strong></td>
</tr>
<tr>
<td>8. Add new SAIC-Benham and NASA recommended pipe supports</td>
<td>3.05-100.19 Hz</td>
<td>3.91-100.21 Hz</td>
<td><strong>30.96 Hz, 31.11 Hz, 91.32 Hz</strong></td>
</tr>
<tr>
<td><strong>9. Combine #6 and #8</strong>: New SAIC-Benham and NASA recommended pipe supports <strong>Mason braided flex hose reoriented 90°</strong></td>
<td>2.93-100.18 Hz</td>
<td>3.91-100.21 Hz</td>
<td><strong>30.89 Hz, 31.23 Hz, 91.31 Hz</strong></td>
</tr>
<tr>
<td><strong>10. Combine #5 and #8</strong>: New SAIC-Benham and NASA recommended pipe supports with soft connection to TEAM modulators using <strong>Gamma flex hose</strong></td>
<td>1.10-100.09 Hz</td>
<td>3.91-100.21 Hz</td>
<td><strong>1.13 Hz, 1.32 Hz, 8.85 Hz, 48.86 Hz, 90.43 Hz</strong></td>
</tr>
</tbody>
</table>

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Adding piping supports increases high effective mass piping modes to 90 Hz or greater, decoupling from the RATF building and catwalk modes.
T-Junction strain measurements acquired during RATF acoustic verification testing indicates resonant modes at 99 Hz and 105 Hz, validating the finite element model and redesign goal of moving the major piping system modes to greater than 90 Hz.
Configuration Analyzed

5. Isolate the TEAM modulators – Gamma flex hose

The Gamma flex hose provides a soft, flexible connection (4” bend radius) to the Wyle WAS 5000 modulators.
The as-built orientation of the Mason braided flex hose is **non-standard practice**. Need to reorient the flex hose 90° so that it is perpendicular to the modulator thrust direction to limit piping vibration fatigue.
Forced Response Analysis

- A forced response analysis was conducted at the location of the T-junction near the TEAM modulator on the 35 Hz horn.

- The forced response analysis is performed by applying a unit acceleration forcing function to the TEAM modulator thrust direction, and recovering dynamic bending moments at the T-junction.

Unit acceleration forcing function applied in the modulator thrust direction from 1-100 Hz
Forced Response Analysis
12 inch riser dynamic Y-plane bending moment

Config #1 Baseline (as-built)

Config #5 Baseline pipe supports with soft connection to TEAM Modulators (Gamma flex hose)

Config #10 Benham and NASA pipe supports with soft connection to TEAM Modulators (Gamma flex hose)
Forced Response Analysis
4 inch connector dynamic Z-plane bending moment

Config #1 Baseline (as-built)

Config #10 Benham and NASA pipe supports with soft connection to TEAM Modulators (Gamma flex hose)

Config #5 Baseline pipe supports with soft connection to TEAM Modulators (Gamma flex hose)
Forced Response Analysis
Summary of Results

• The results of the forced response analysis for Configurations #1-10 can be used to inform which configuration provides the most reduction in T-junction dynamic bending moment (corresponding to the highest TEAM modulator isolation).

• Examining the bending moment results for the 4 inch and 12 inch riser indicates that Configurations #5 and #10 provide the largest reduction in bending moment compared to Configuration #1 (baseline).

To prevent long term piping fatigue due to TEAM modulator vibrations, make a soft connection to the TEAM modulators using a Gamma flex hose. The forced response analysis indicates tremendous bending moment reduction with a soft connection.
Stress Field Analysis of T-Junction
Including SAIC-Benham Recommended Additional Pipe Supports

- For horn room health monitoring, rosette strain gauges will be placed near the high stress region of the T-Junction to measure axial, tangential, and hoop stresses.

NOTE: Actual stresses are fictitious due to the normalized mode shape vectors applied. The maximum principal stress (91.49 Hz eigenvector case) provides guidance to locate the strain gage at the high stress location.
T-Junction strain measurements acquired during RATF acoustic verification testing (C7 and C5 shaped test spectra) indicates the RATF piping system can withstand up to 165 dB OASPL for infinite fatigue life ($10^7$ alternating stress cycles). This result is dependent on the shape of the acoustic test spectrum; test spectra with larger low frequency acoustic levels could alter this conclusion.
Horn Room Piping Dynamic Analysis
Repairs Implemented

SAIC-Benham’s repair of the piping system (Configuration #8) included:

1. “T-junction” reinforced pad repair at all 23 locations
2. SAIC-Benham recommended 24 additional pipe supports
3. NASA recommended 4 additional pipe supports
4. Additional 4 inch branch pipe supports near elbows or long unsupported runs
5. Schedule 40 piping was added at the highly stressed elbows of the 4 inch branch

Although not implemented due to funding and schedule constraints, the recommended installation of the Gamma flex hose at all TEAM modulators (Configuration 10) would further reduce the dynamic bending moment.
Horn Room Piping Dynamic Analysis
Conclusions

• The implemented horn room repairs (Configuration 8) increased the piping frequency and “t-junction” strength, decoupling the piping system high effective mass modes from the RATF building (< 20 Hz) and catwalk (< 17 Hz) structure modes.

➤ **Lesson Learned:** The dynamics of the piping system, including their coupling with the structural modes of the building, must be taken into consideration when designing a piping system when dealing with high acoustic excitation levels.

➤ Installation of the Gamma flex hose at all TEAM modulators (Configuration 10) would further reduce the dynamic bending moment.

• Considering infinite life, the RATF piping system can withstand up to 165 dB OASPL based on the C7 and C5 shaped spectrum; other acoustic test spectrum shapes could alter this conclusion.
Construction Photos
RATF Foundation Construction

Foundation started in April 2008
Installation of horn frames and rebar
Concrete pour #1 completed October 2009
Concrete pour #1 completed with forms removed
Horn wall level 2 horn frame and rebar installation
Concrete pour #2 completed with forms removed
Horn Room and Chamber Wall Pour

Concrete pour of walls completed with forms
Horn Wall – Installation of Horn Frames

Legend
25 Hz  100 Hz
35 Hz  160 Hz
50 Hz  250 Hz

Space Available for Future Expansion

Scarring (for 250 Hz)
Construction photo showing the installation of the final RATF horn (25 Hz)
East Chamber Door
(September 2010)

Installation of 675,000 lb door.
RATF in the Vibro-Acoustic Highbay (mid-December 2010)
Inside RATF chamber looking at the horn room wall, 2 angles (March 2011)
RATF is the most Powerful Large Reverberant Acoustic Chamber in the World!

<table>
<thead>
<tr>
<th>(Active) Reverberant Acoustic Test Facility</th>
<th>Location</th>
<th>Volume (ft³)</th>
<th>Max. OASPL (dB) Empty Chamber</th>
<th>Year Commissioned</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lockheed Martin Missiles and Space, bldg.156, cell no.1, LVATF</td>
<td>Sunnyvale, CA</td>
<td>189,200</td>
<td>156.5</td>
<td>1973</td>
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<tr>
<td>NASA Plum Brook Station</td>
<td>Sandusky, OH</td>
<td>101,200</td>
<td>163.0</td>
<td>2011</td>
</tr>
<tr>
<td>Lockheed Martin Space Systems</td>
<td>Denver, CO</td>
<td>75,900</td>
<td>154.0</td>
<td>1985</td>
</tr>
<tr>
<td>Boeing Satellite Development Center (Boeing SDC)</td>
<td>El Segundo, CA</td>
<td>67,800</td>
<td>155.0</td>
<td>2004</td>
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<tr>
<td>Lockheed Martin Missiles and Space (LMMS), bldg.159</td>
<td>Sunnyvale, CA</td>
<td>64,000</td>
<td>157.3</td>
<td>1996</td>
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<tr>
<td>Mitsubishi Electronics</td>
<td>Kamakura, Japan</td>
<td>61,700</td>
<td>152.0</td>
<td>2002</td>
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<td>Large European Acoustic Facility (LEAF) at ESTEC</td>
<td>Noordwijk, The Netherlands</td>
<td>59,000</td>
<td>154.5</td>
<td>1990</td>
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<tr>
<td>Northrop Grumman Space Technology (NGST), LATF</td>
<td>Redondo Beach, CA</td>
<td>51,600</td>
<td>154.0</td>
<td>1996</td>
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Reference:


Contact Information:

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