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

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April 25, 2012

Presentation Outline

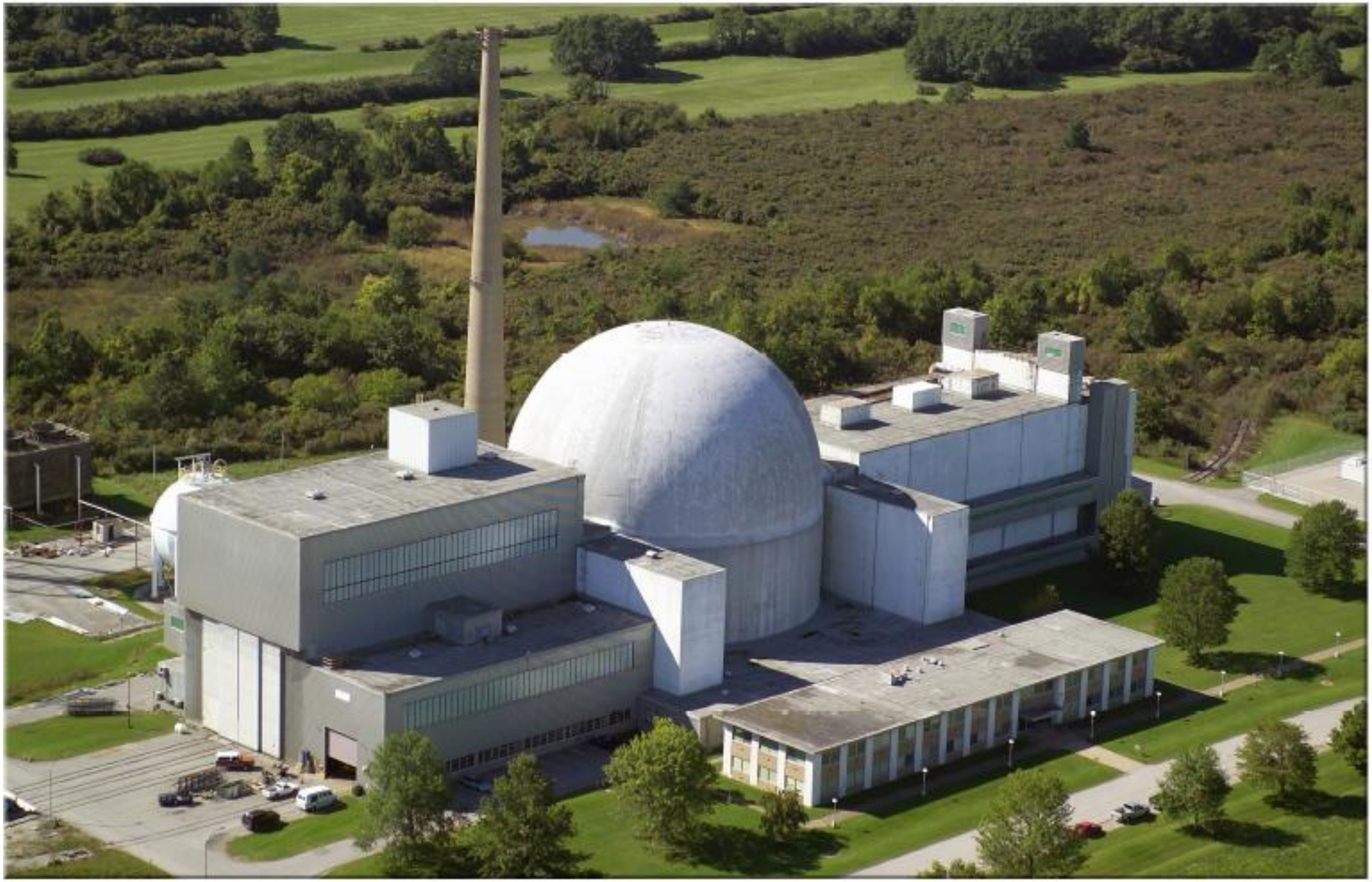
- Introduction
- Design Requirements
- Structural Design for Vibroacoustic Loads:
  - Chamber Wall Flexural Design
  - Horn Room Piping Repair
- Construction Photos

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.

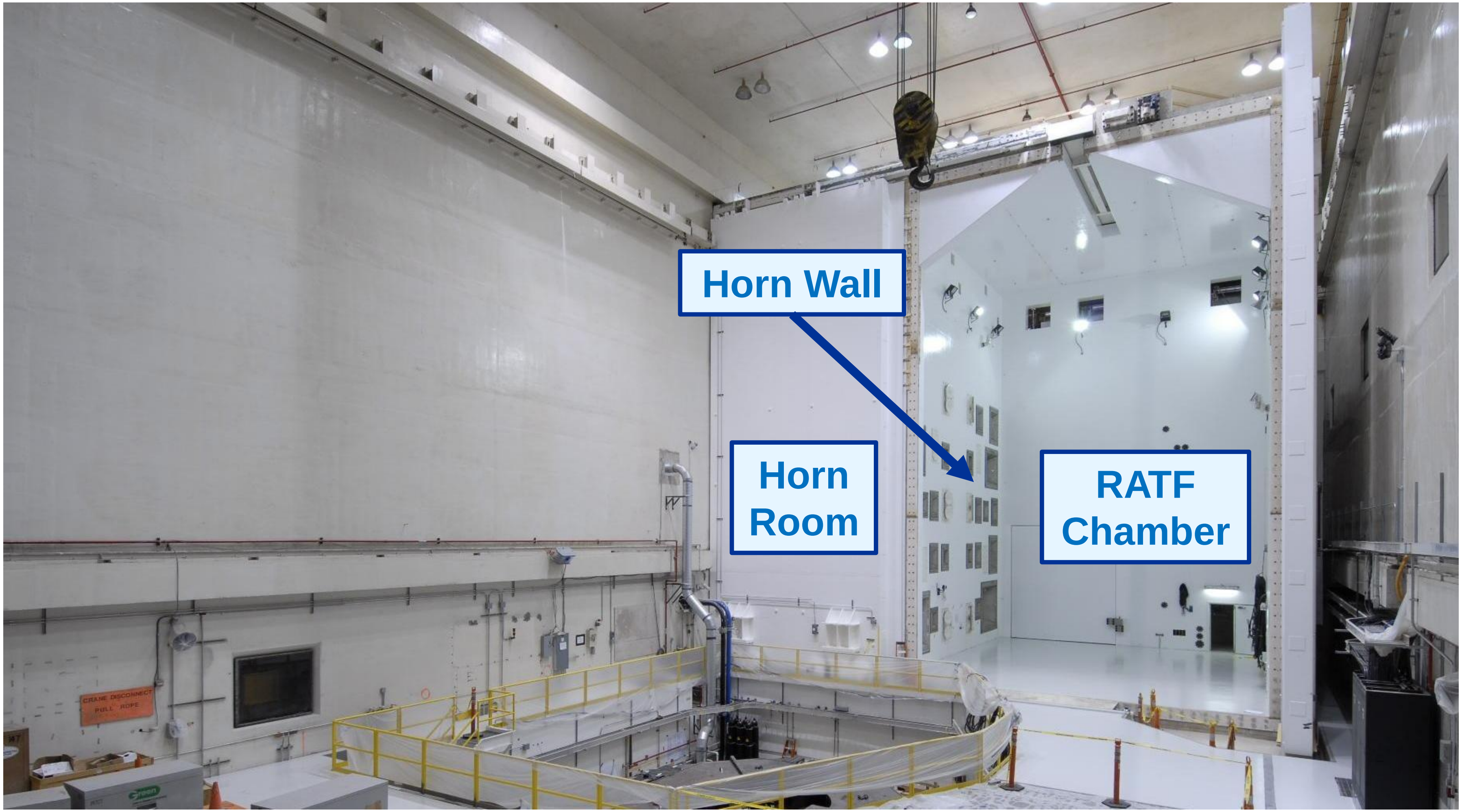
Space Power Facility, NASA Plum Brook Station Sandusky, Ohio (50 miles west of Cleveland)



Introduction
(continued)

- 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)

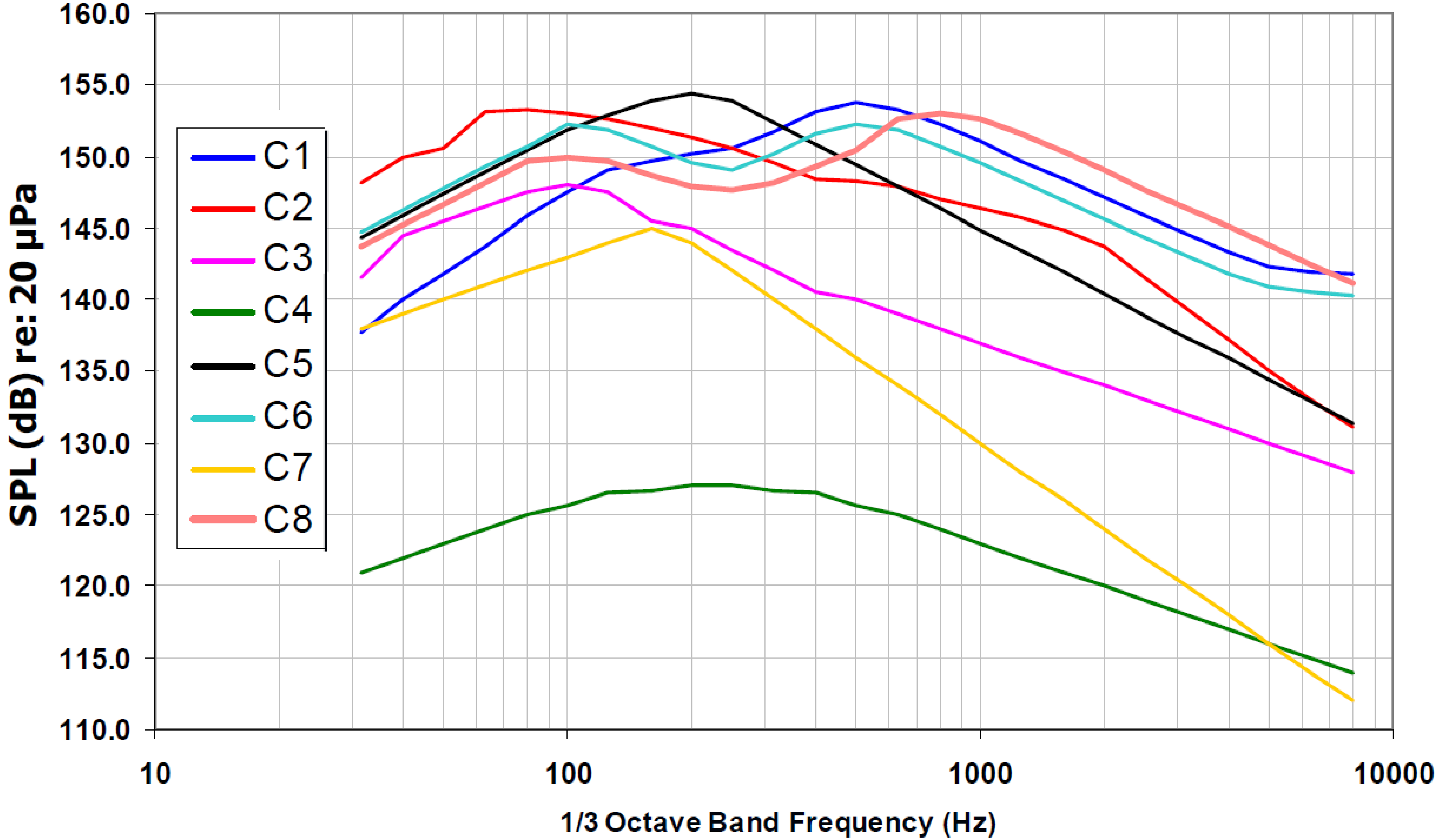


Design Requirements

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

Figure 1. RATF Acoustic Test Spectral Design Requirements



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.

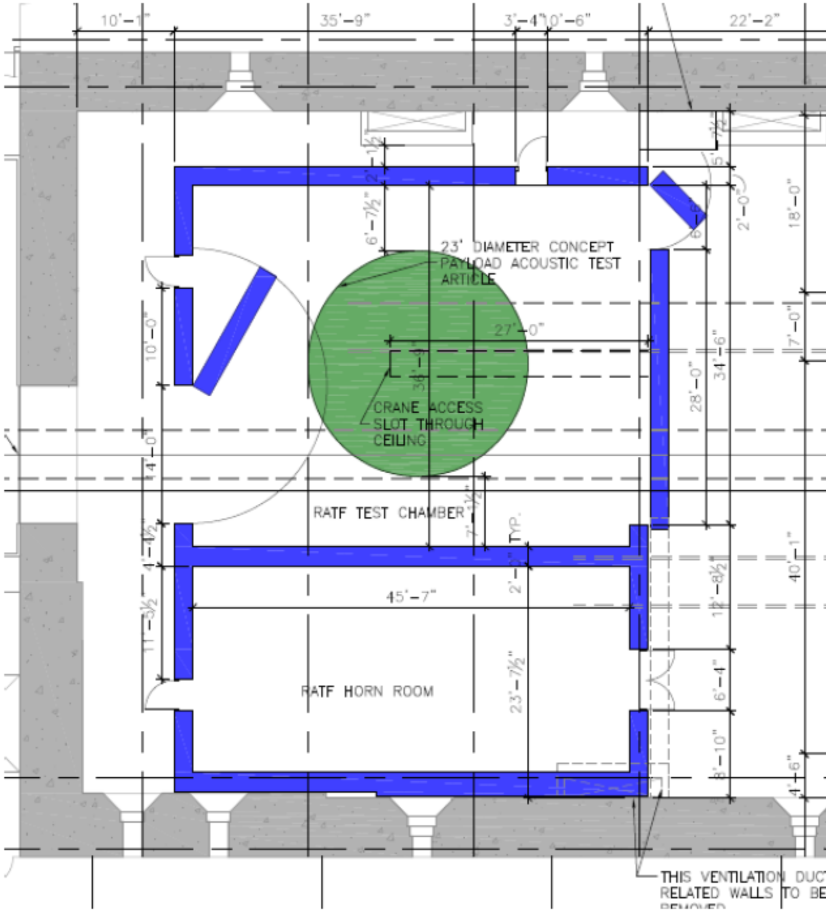
RATF Design Summary

- 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

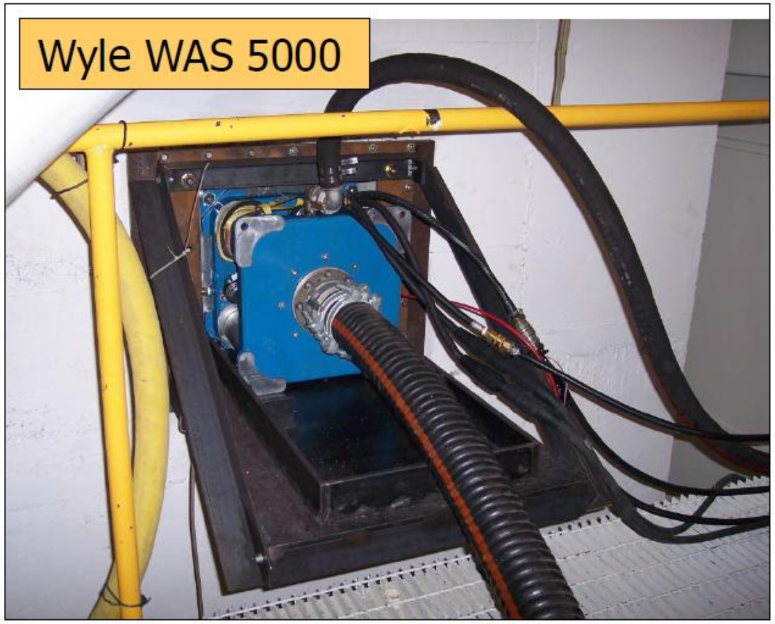
- The gaseous nitrogen (GN2) generation system (see Figure 8) is designed to meet the flow needs of RATF.
  - Water bath vaporizer capable of GN2 flow rate of 72,000 SCFM (standard cubic feet per minute)
  - One (1) 6,000 gallon liquid nitrogen (LN2) pusher tank
  - Two (2) 9,000 gallon liquid nitrogen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

Figure 2. RATF Acoustic Design



	Chamber Properties
Chamber Size	47.5 ft L x 37.5 ft W x 57 ft H
Chamber Volume	101,189 ft^3
Acoustic Modulators	23 TEAM Modulators & 13 WAS 5000 Modulators
Horns	36 (grouped at 7 different horn cut-off frequencies)
Maximum GN_2 flow rate	72,000 scfm
Main Door Opening	34.5 ft wide
Number of Main Doors	2
Door Type	Sliding and hinged
OASPL, empty	163 dB OASPL

Figure 3. Modulator/Horn Pairings



Horn	25 Hz	35 Hz	50 Hz	80 Hz	100 Hz	160 Hz	250 Hz	TOTAL
Modulator	MKVII	MKVII	MKVII	MKVII	MKVI	MKVI	WAS5000	36
Final Design Count	2	2	4	3	4	8	13	

Figure 4. Construction photo showing the installation of the final RATF horn (25 Hz)

Figure 5. RATF Horn Layout

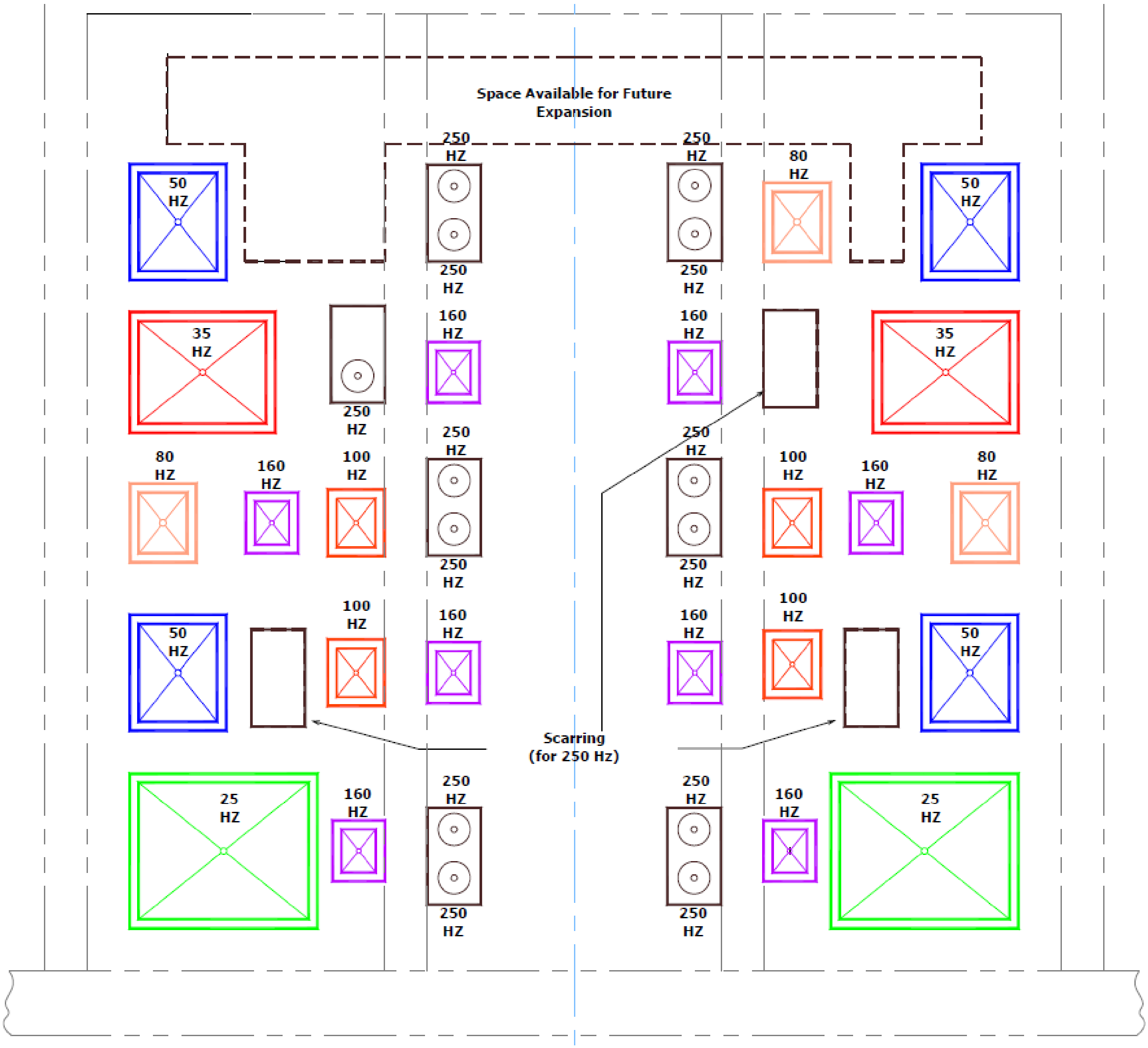


Figure 6. RATF Construction Photo

(taken September 2010)

Figure 7. Construction photo of the RATF horn room (level 5) platform and modulators

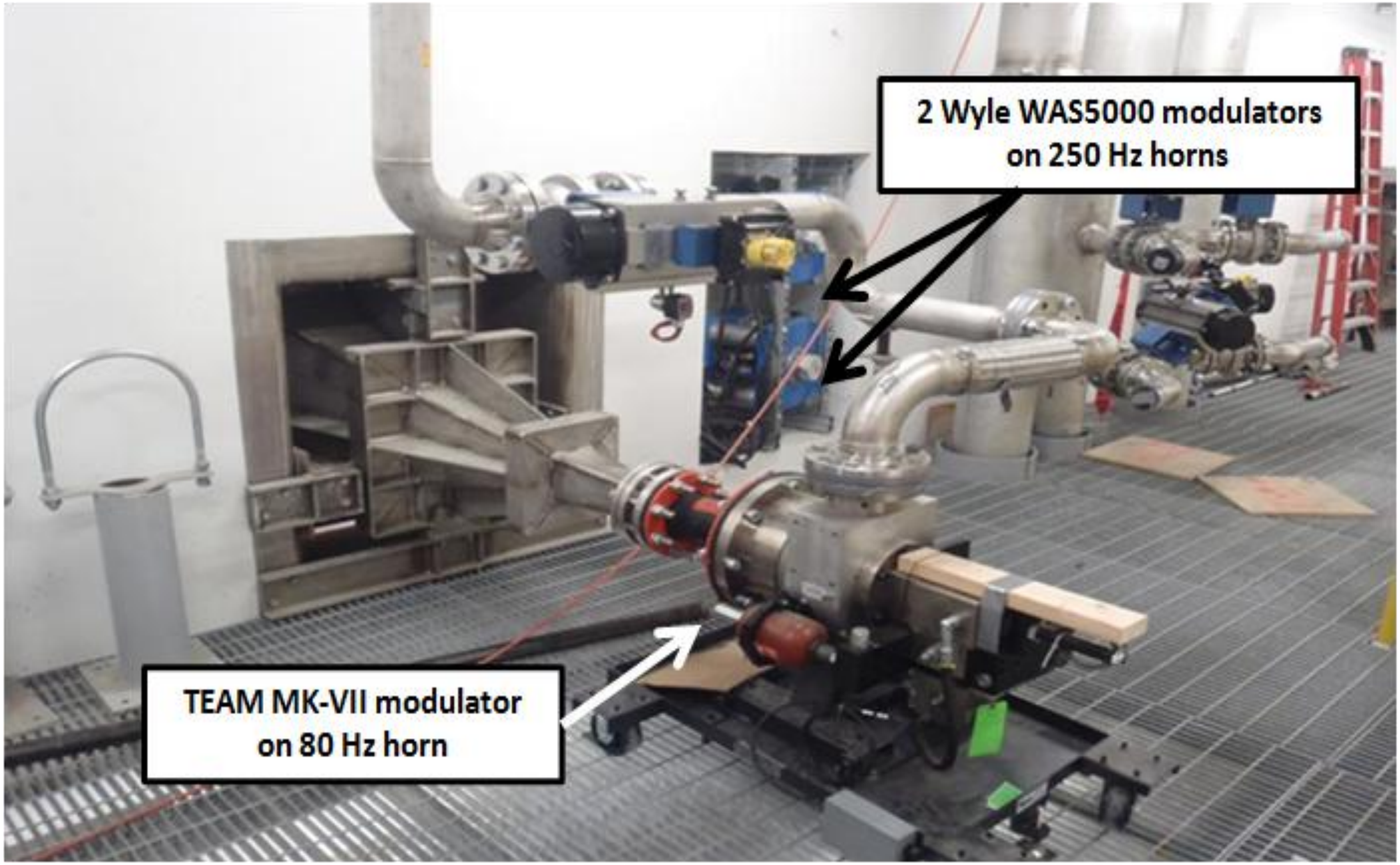
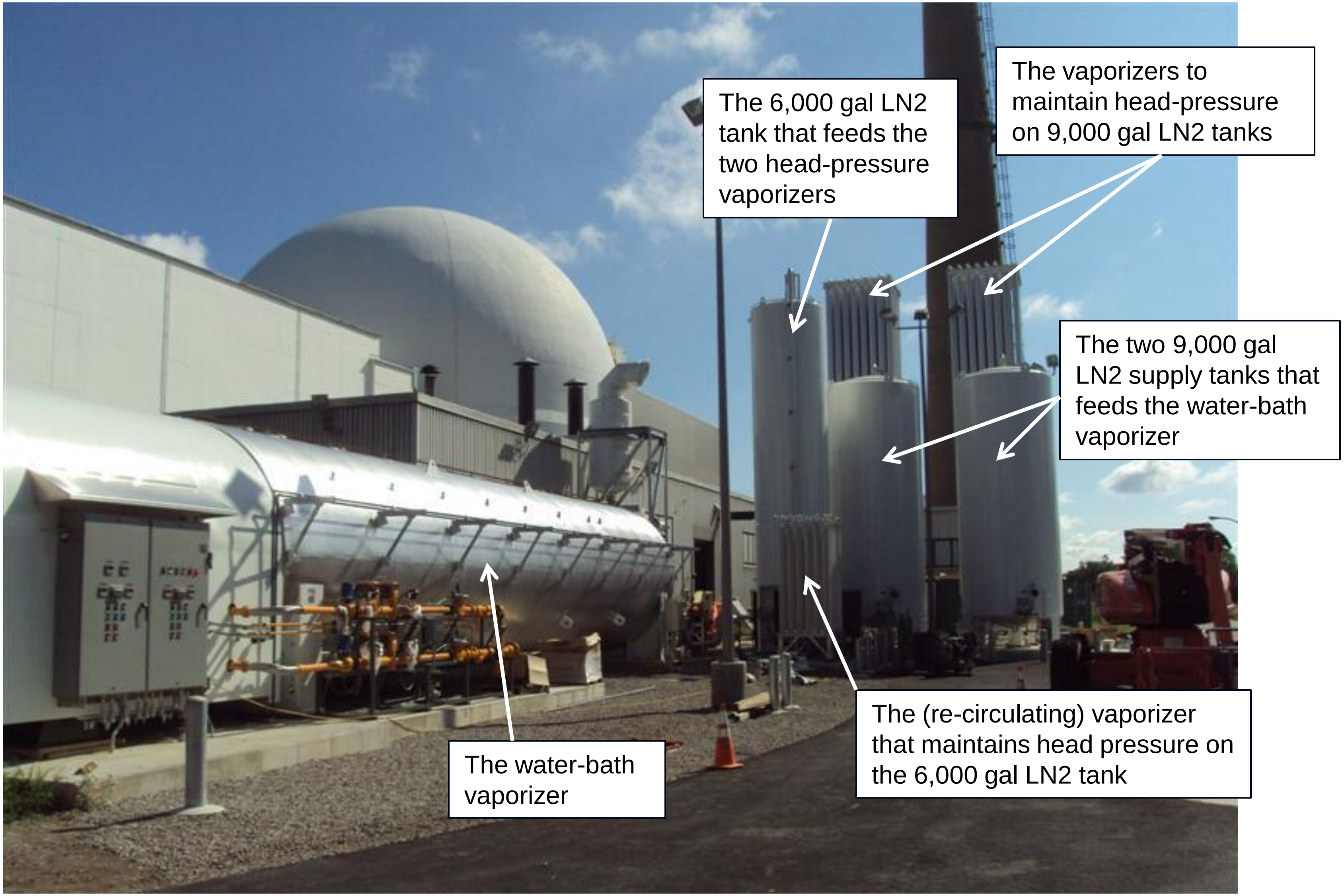


Figure 8. Construction photo showing the RATF nitrogen generation system, including the water-bath vaporizer and the liquid nitrogen tanks and vaporizers



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).

Factored Resistance ≥ Factored Load

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

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.

$$\text{Acoustic } \boldsymbol{PSD}\left(\frac{Pa^2}{Hz}\right) = \left[P_{ref}{}^2 * 10^{\left(\frac{SPL}{10}\right)}\right] \Big/ \left[\frac{1}{3}\, octave\ bandwidth\right]$$

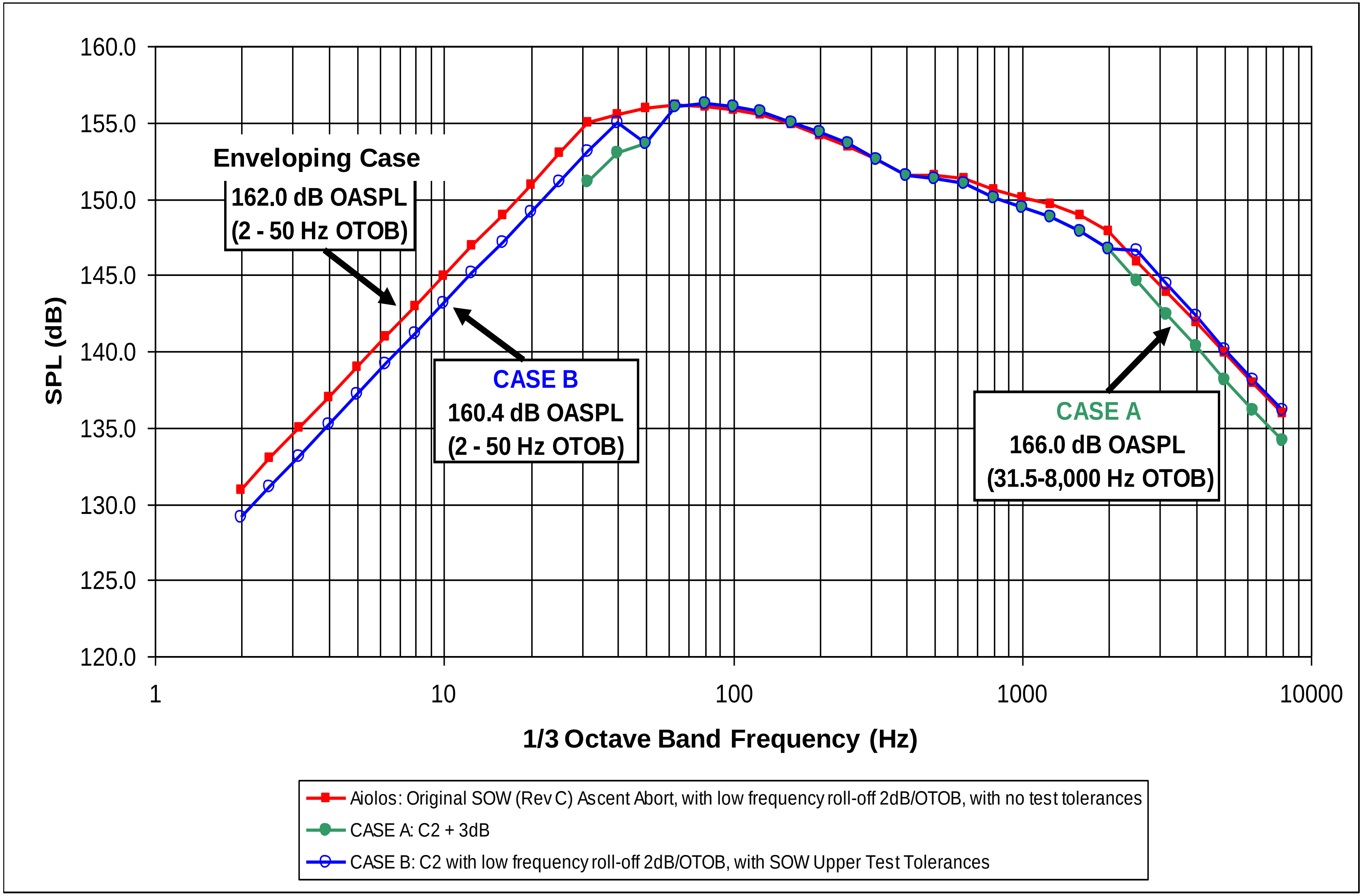
where $P_{ref} = 20\text{x}10^{-6}$ pascals (Pa)

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.

$$\phi\, M_n \geq M_u$$

RATF Chamber Design Acoustic Excitation



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 σ_R

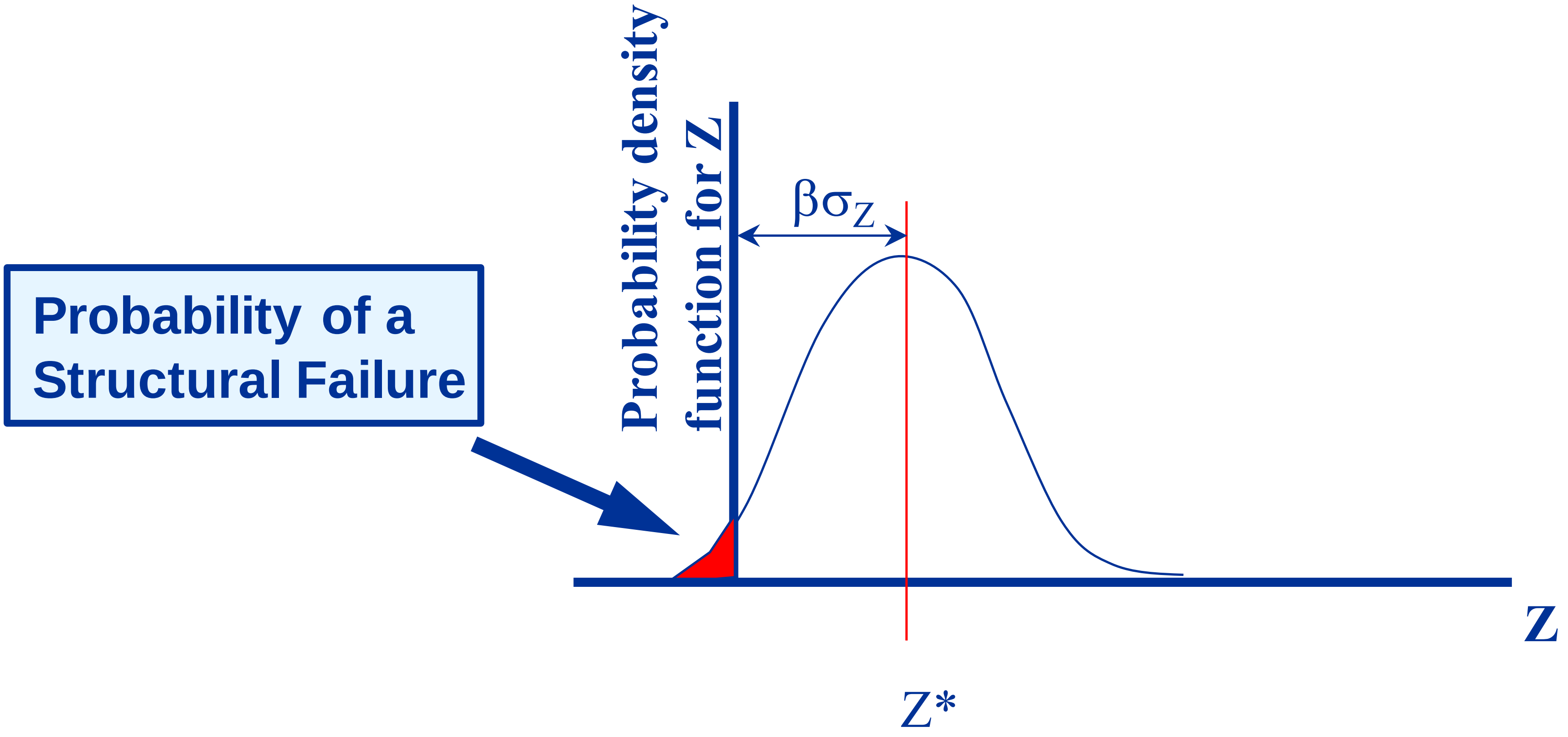
Assume **S** represents **structural load effect**
Assume S is a normally distributed random variable with mean S* and std dev σ_S

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

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

A structural failure will occur if Z < 0

Load Resistance Factor Design (LRFD)



β (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:

$$\sigma_Z = [\,\sigma_R^2 + \sigma_S^2\,]^{0.5} \cong 0.7\,(\,\sigma_R + \sigma_S\,)$$

(Pythagorean theorem for isosceles right triangle ; good if σ_R and σ_S not **TOO** different)

$$Z^* > \beta\sigma_Z \rightarrow R^* - S^* > .7\beta\,(\sigma_R + \sigma_S) \rightarrow R^* - .7\beta\sigma_R > S^* + .7\beta\sigma_S \rightarrow$$

$$R^*(\,1 - .7\beta V_R\,) > S^*(\,1 + .7\beta V_S\,) \text{ where } V_R = \sigma_R / R^* \text{ and } V_S = \sigma_S / S^*$$

$1 - .7\beta V_R$ = ***resistance factor*** and $1 + .7\beta V_S$ = ***load factor***
in Load and Resistance Factor (LRFD) design code format

- 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.).

<u>Reference:</u> "Minimum Design Loads for Buildings and Other Structures," ASCE/SEI 7-05, 2006 defines the US design load criteria.

Load Resistance Factor Design (LRFD)

Factored Resistance ≥ 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

β = safety index (historically 2.5 – 3.0 for civil structures)

V_R = coefficient of variation for the structural capacity = σ_R / R^*

V_S = coefficient of variation for the load = σ_S / S^*

Coefficient of Variation = ***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 that are well separated, corresponding to individual room modes.

<u>Reference:</u> "Some Comments on Reverberant Chamber Sound Fields," technical memorandum from John F. Wilby, Wilby Associates to William O. Hughes, NASA Glenn Research Center, October 22, 2008.

Statistics of the Acoustic Sound Pressure Field
- Normalized Variance

- The normalized variance, ν^2, is defined as the variance σ^2 of the mean square pressure normalized with respect to the square of the space-averaged value of the mean square pressure:

$$\nu^2\left(\overline{p}^2\right) = \frac{\sigma^2\left(\overline{p}^2\right)}{\left\langle \overline{p}^2 \right\rangle^2}$$

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

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

$$\nu = \text{COV} = \sigma / \left\langle \overline{p}^2 \right\rangle$$

<u>Reference:</u> "Some Comments on Reverberant Chamber Sound Fields," technical memorandum from John F. Wilby, Wilby Associates to William O. Hughes, NASA Glenn Research Center, October 22, 2008.

Statistics of the Acoustic Sound Pressure Field
- Normalized Variance in Low Frequency Range $0.2\ f_s < f < 0.5\ f_s$

- The normalized variance, ν_L^2, of the sound field at low frequencies is defined as:

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

where:

B = frequency bandwidth = 0.23 f_c for 1/3 octave bands
f_c = band center frequency
N = modal density

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

V = chamber volume
S = total area of chamber walls, floor, and ceiling
P = total length of all edges

<u>Reference:</u> "Some Comments on Reverberant Chamber Sound Fields," technical memorandum from John F. Wilby, Wilby Associates to William O. Hughes, NASA Glenn Research Center, October 22, 2008.

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

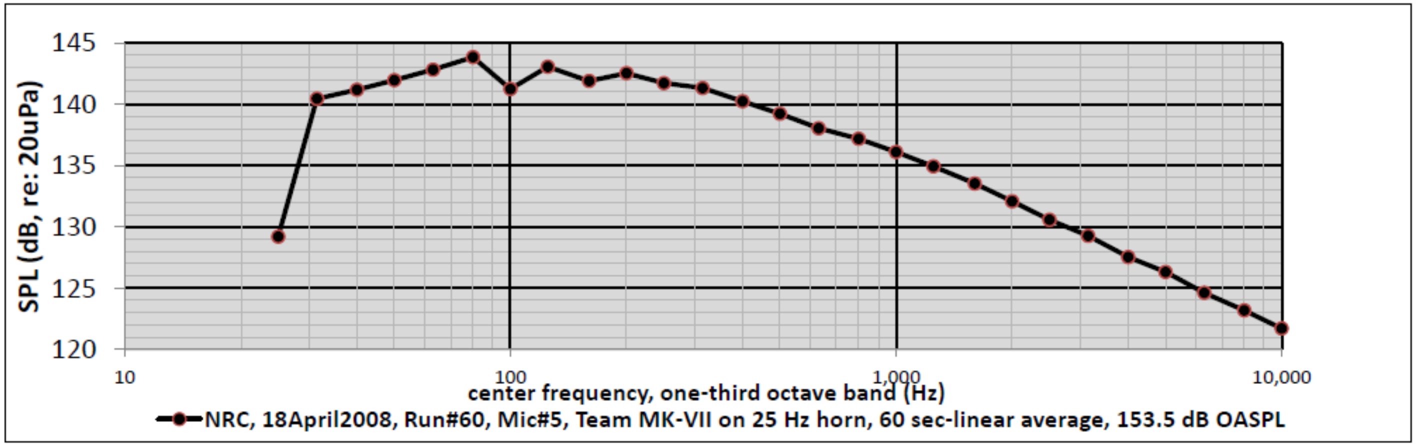
• Based on a statistical review of the microphone pressure time histories from the TEAM modulator characterization testing at the U.S. Army Redstone Technical Test Center (RTTC) in Huntsville, Alabama and the National Research Council (NRC) in Ottawa, Canada, a V_S = 0.75 was calculated.

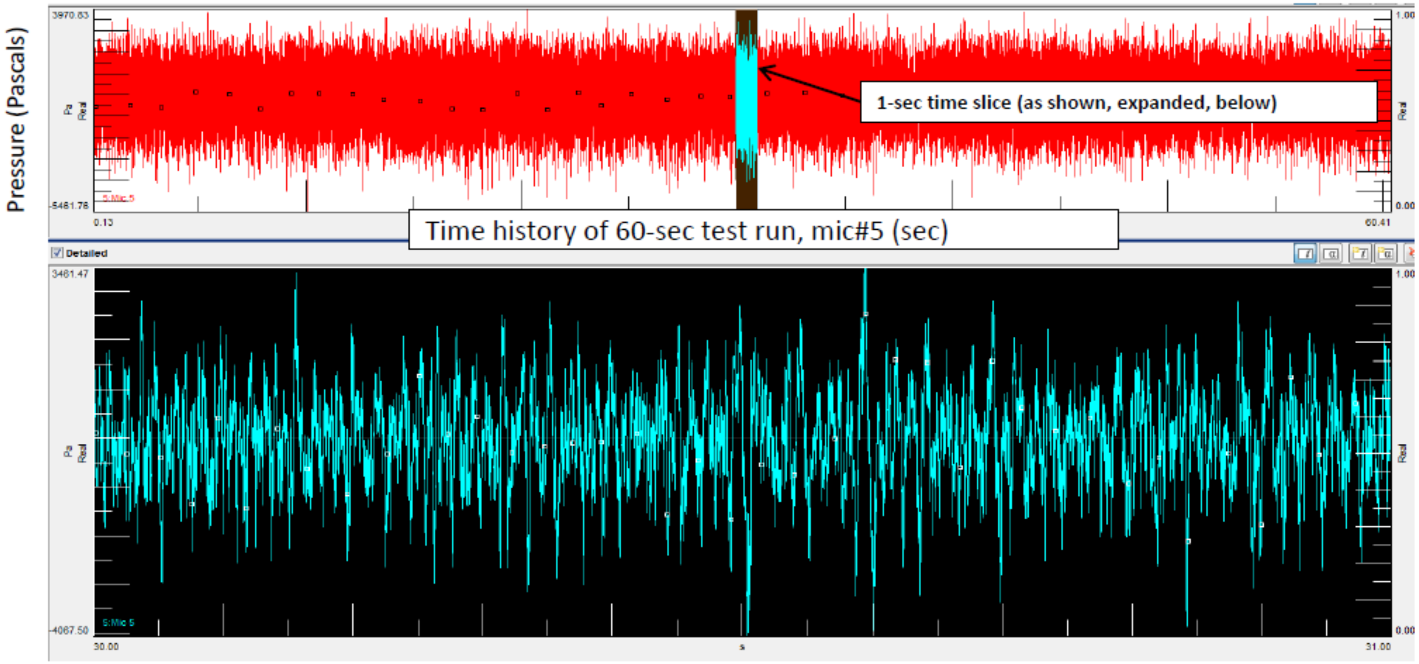
<u>Assuming:</u>
β = 3.0 (safety index, historically 2.5 – 3.0 for civil structures)
V_S= 0.75
Acoustic Testing Live Load Factor = [1.0 + 0.7 * β * V_S] = 2.6

Statistical Analysis of Microphone Test Data from NRC (Positive Valued Pressure)	
mean	841.04 Pa
max	4254.61 Pa
min	1.20 Pa
stdev	617.86 Pa
COV = V_S	COV = 0.73

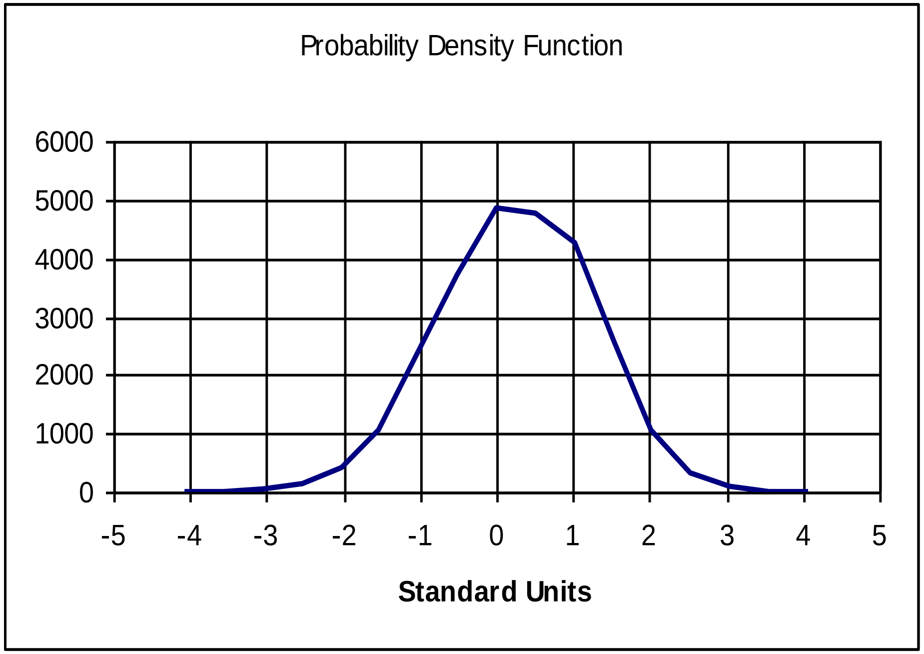
Acoustic Live Load Factor = 2.6 was used for the RATF wall design. The 2-way slab design is 2 feet thick concrete reinforced with #8 rebar to resist bending moments.

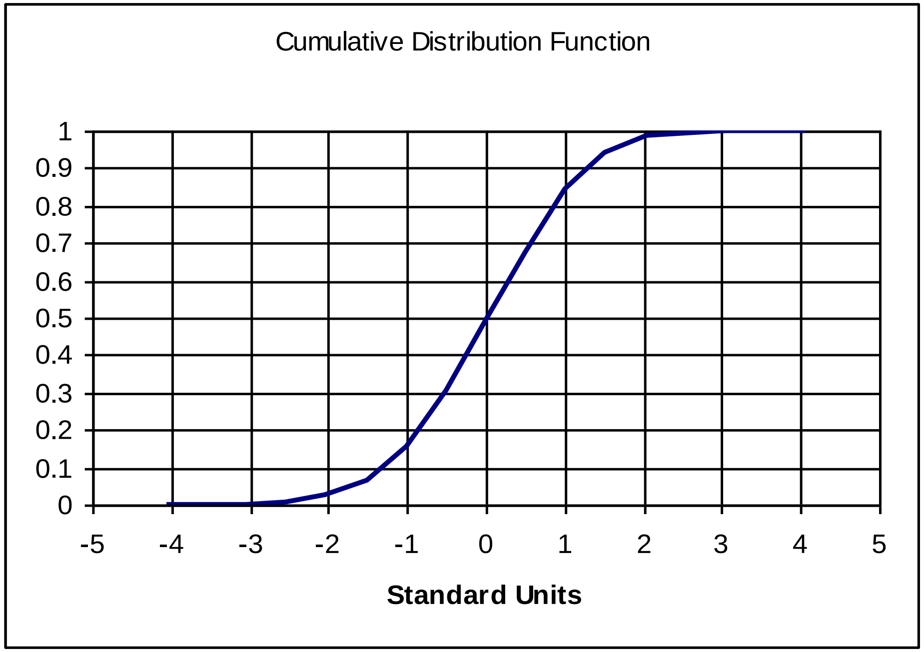
Statistics of the Acoustic Sound Pressure Field
- Normal Distribution Evaluation





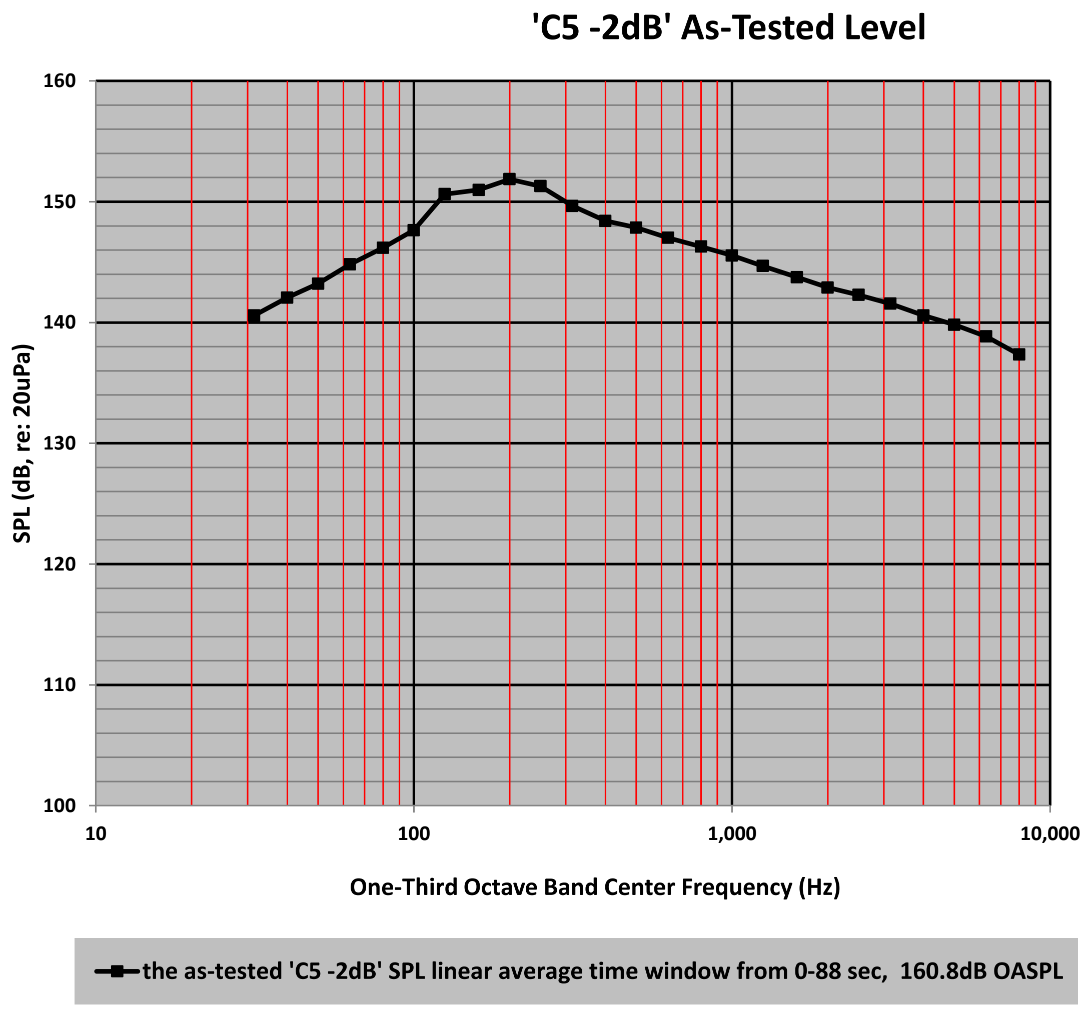
Statistics of Acoustic Sound Pressure Field
- Normal Distribution Evaluation
NRC/Run #60/1 sec time slice/Microphone #5/MK VII/ 25 Hz Horn





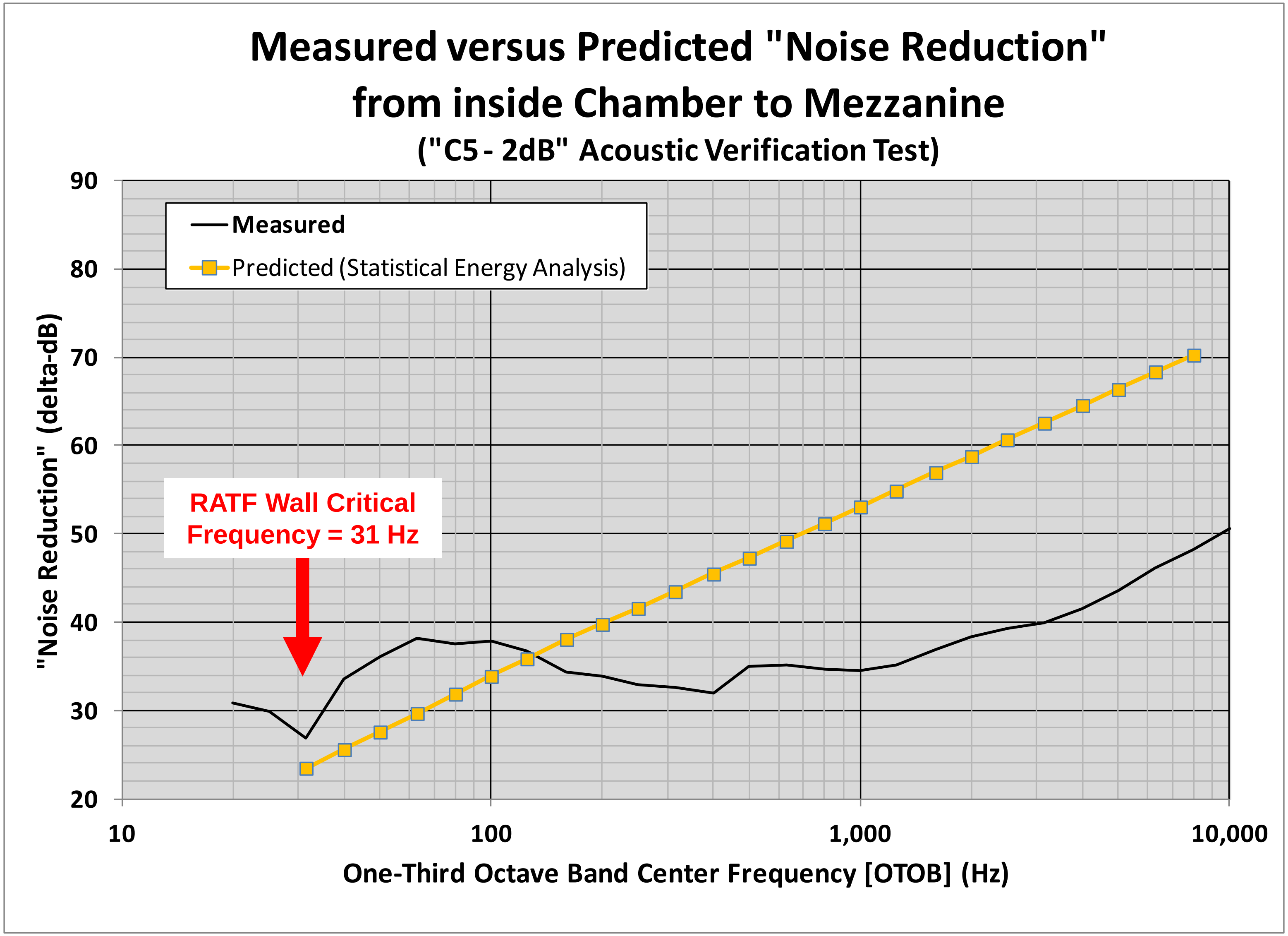
The microphone time history from the TEAM MK- VII modulator data on the 25 Hz horn is normally distributed. For a normal distribution of 2.6 σ above the mean, the corresponding load non-exceedance probability is ~0.9953.

RATF Acoustic Verification Testing



One-Third Octave Band Center Frequency (Hz)	the as-tested 'C5 -2dB' SPL linear average time window from 0-88 sec, 160.8dB OASPL
31.5	140.6
40	142.1
50	143.2
63	144.8
80	146.2
100	147.6
125	150.6
160	151.0
200	151.9
250	151.3
315	149.7
400	148.4
500	147.9
630	147.0
800	146.3
1,000	145.5
1,250	144.7
1,600	143.8
2,000	142.9
2,500	142.3
3,150	141.6
4,000	140.6
5,000	139.8
6,300	138.8
8,000	137.3
OASPL	160.8

RATF acoustic verification testing achieved 161 dB OASPL using the "C5 – 2dB" design test spectrum.



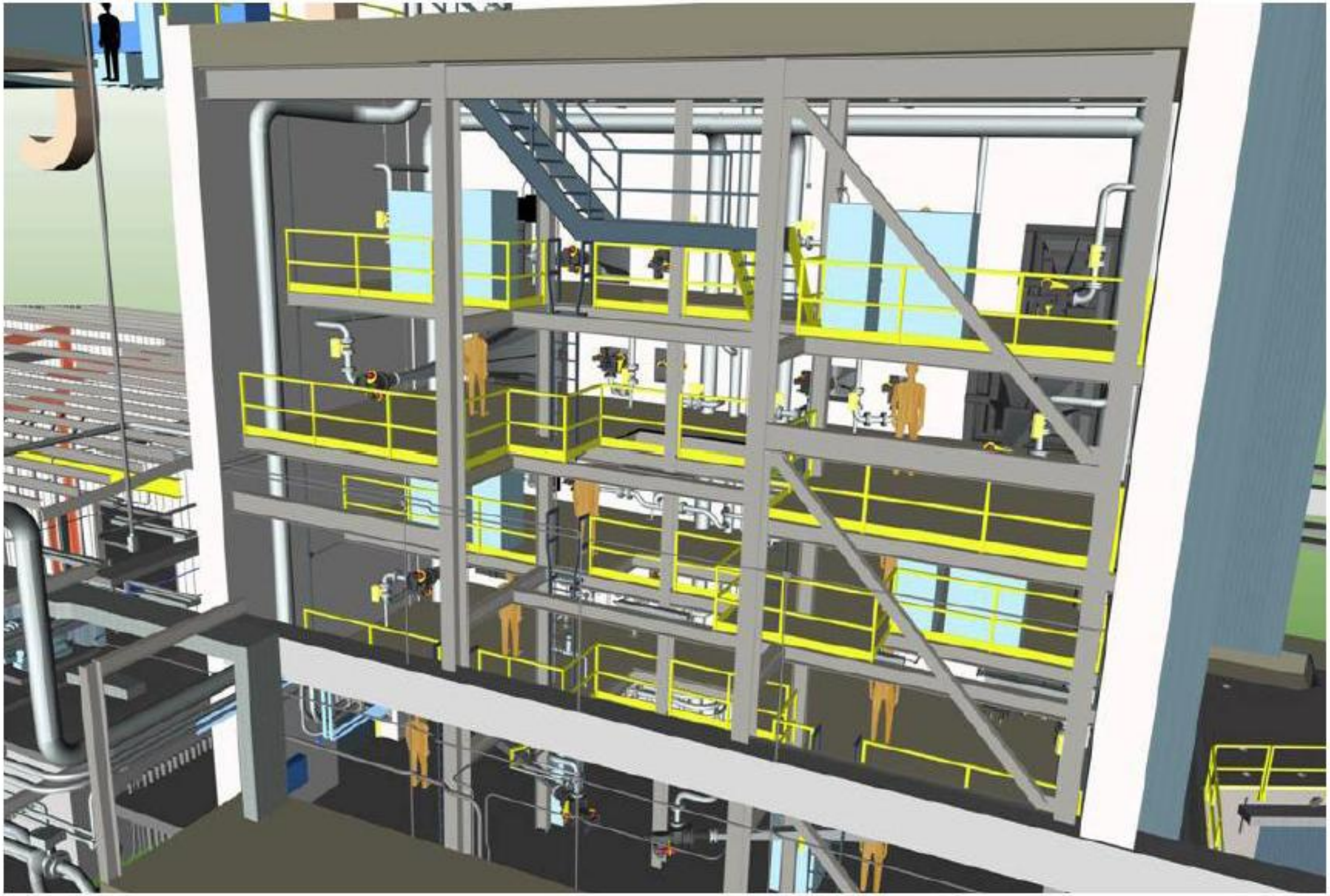
Measured "Noise Reduction" is less than predicted at frequencies greater than 160 Hz OTOB.

Plateau Method Reference: "Noise and Vibration Control Engineering," L. L. Beranek and I. L. Ver, Fig 9.24, 1992.

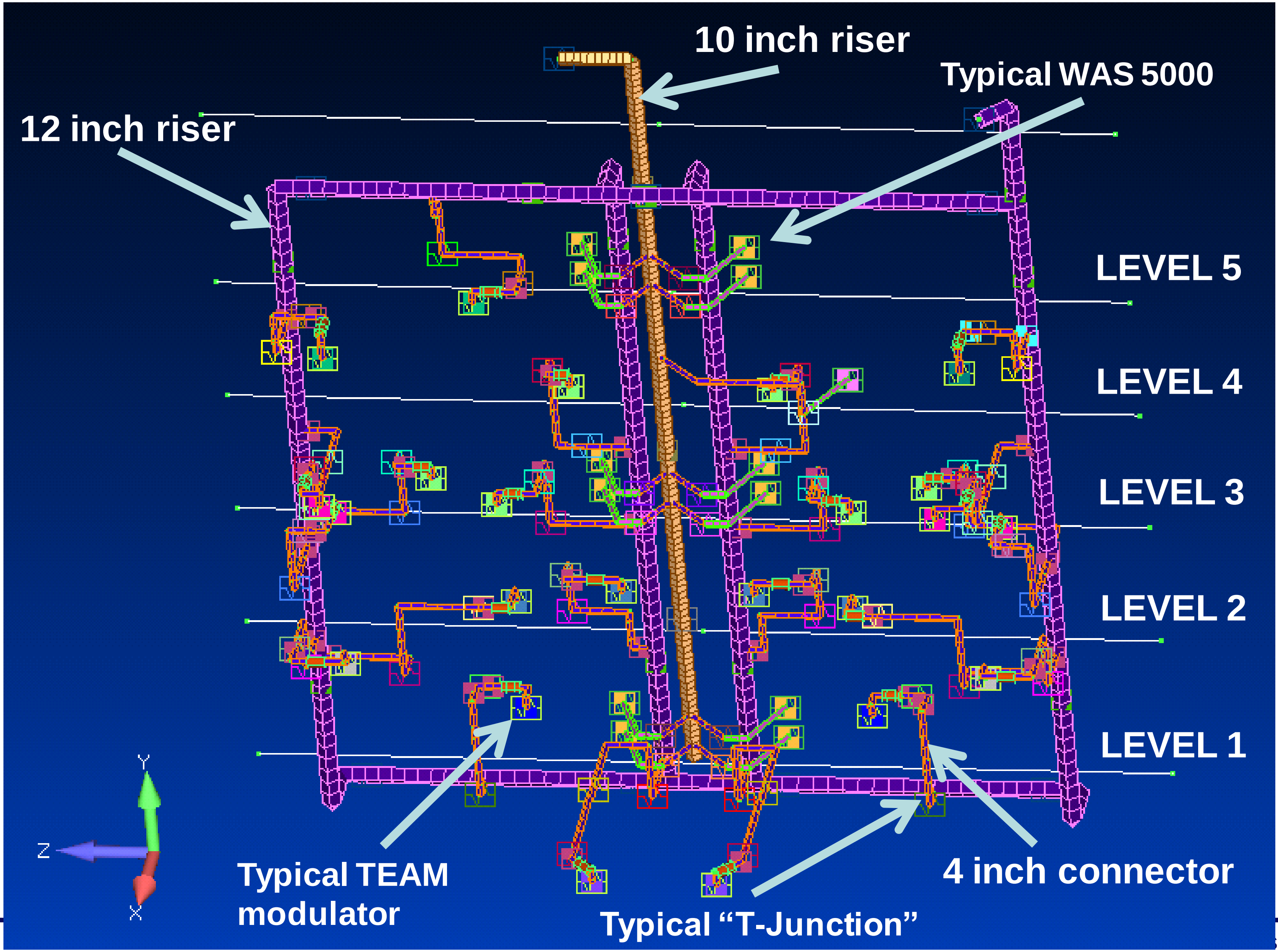
Structural Design for Vibroacoustic Loads:

- Horn Room Piping Repair

RATF Horn Room Illustration Cutaway View of 5 Levels

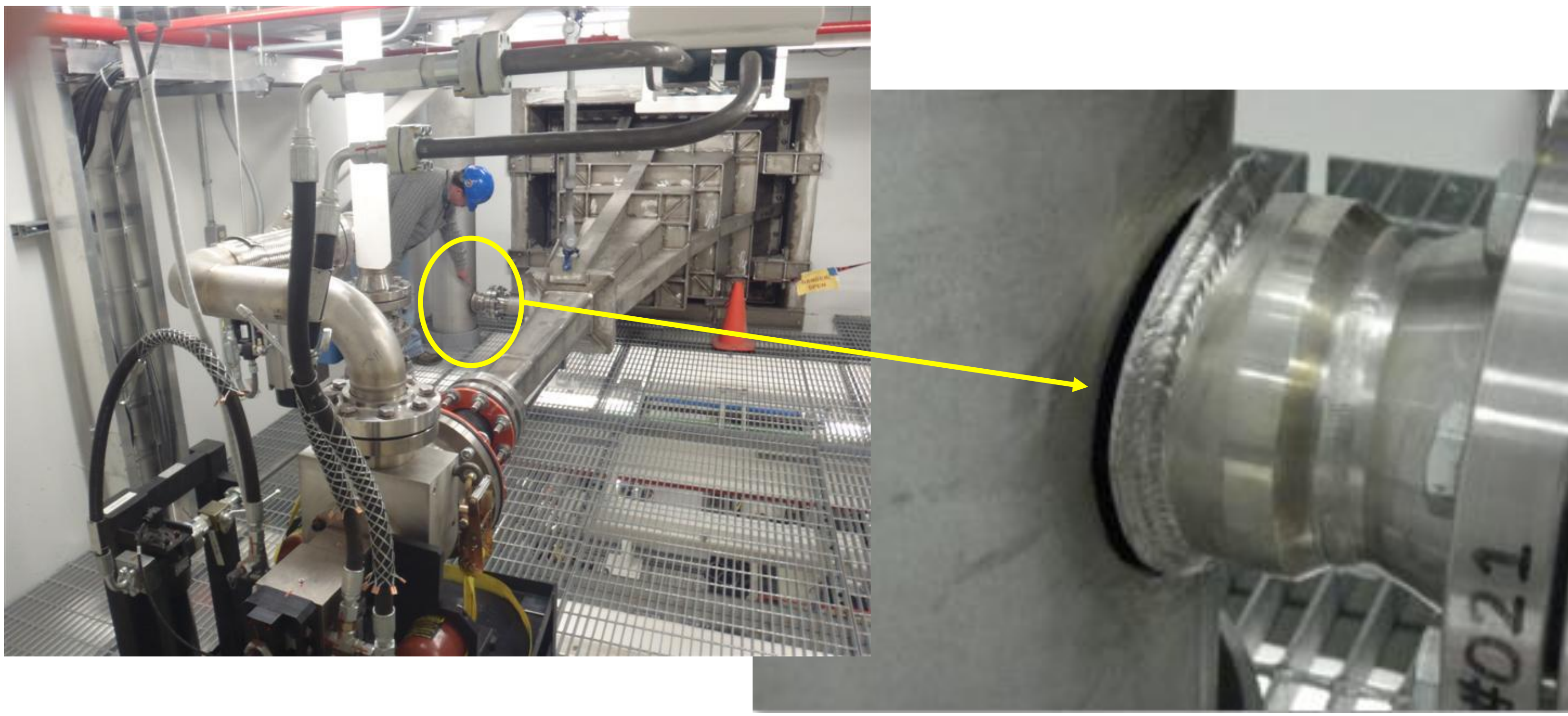


RATF Horn Room Piping System

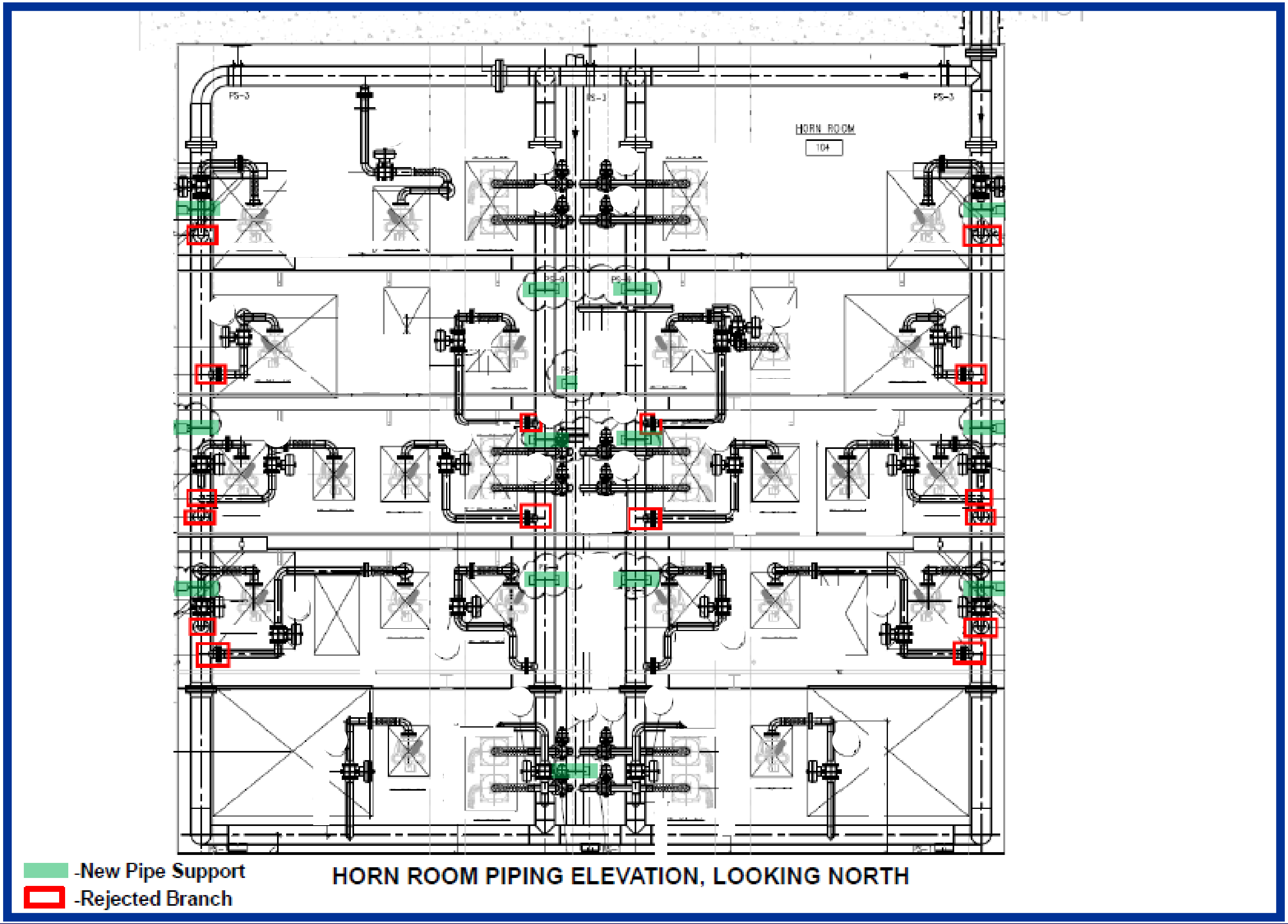


T-Junction Failures

- 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.

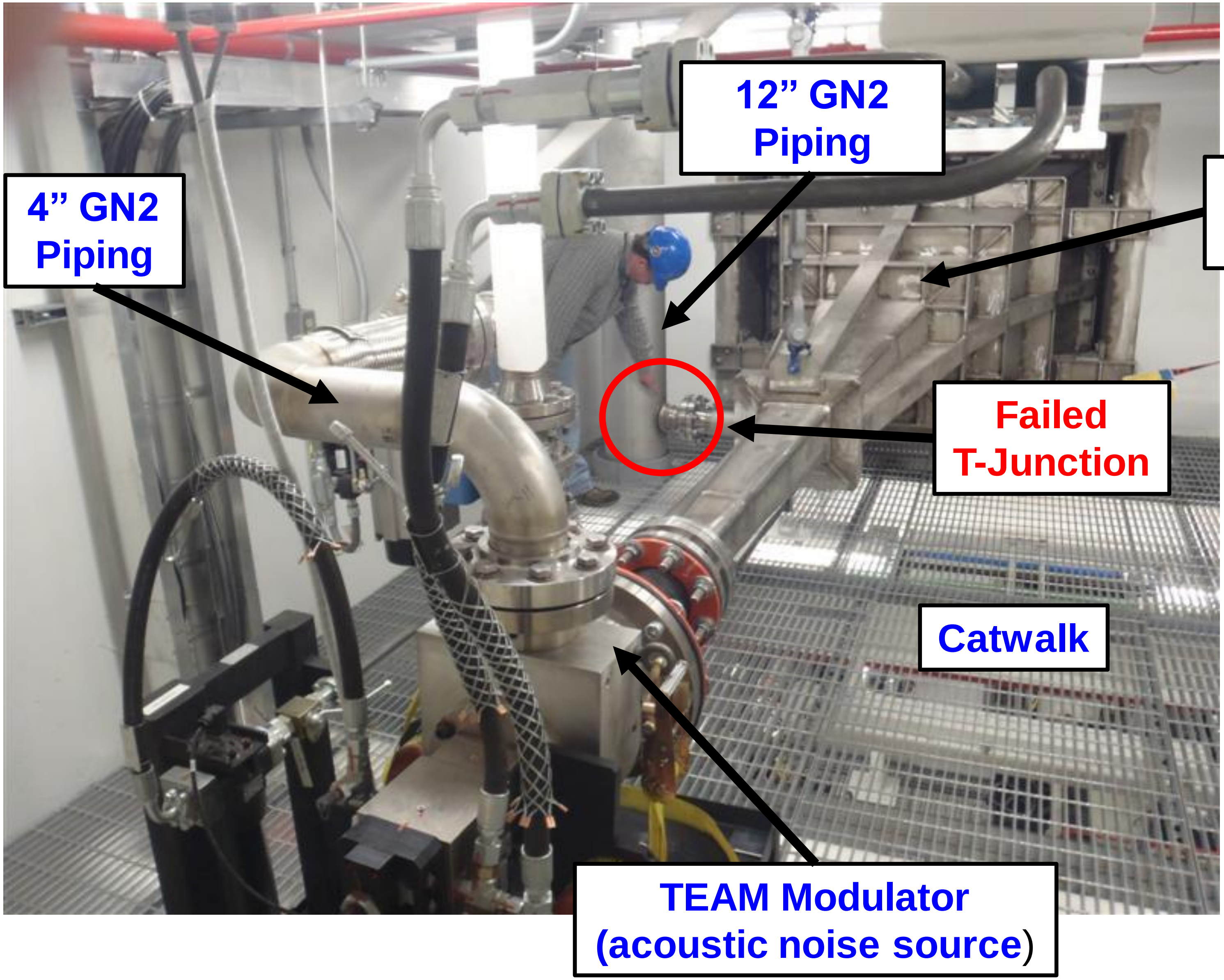


T-Junction Failures (in red) from initial Acoustic Checkout Testing



HORN ROOM PIPING ELEVATION, LOOKING NORTH

RATF Horn Room Piping System



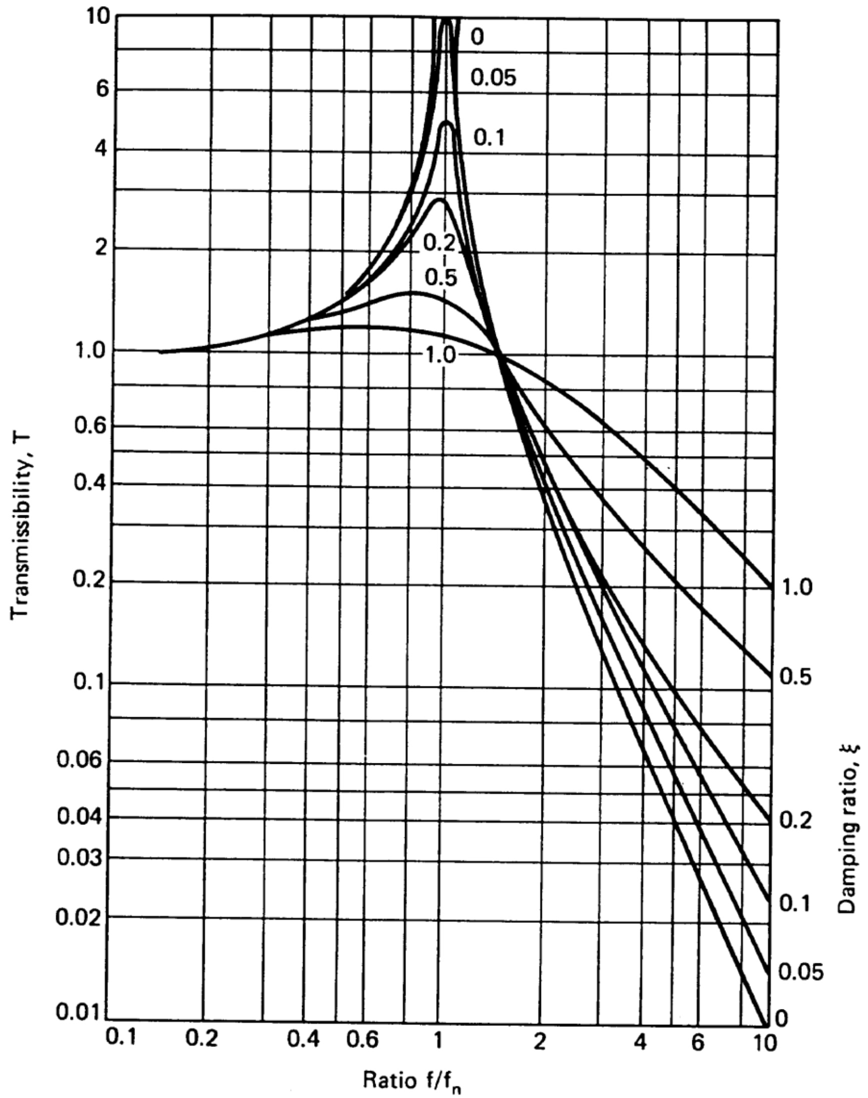
Analytically Assess Piping System

- 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 1:** "Low Frequency Prediction of RATF Response to Acoustic Excitation," by Bryce Gardner, ESI Report, Revision 4, October 28, 2008.
- **Reference 2:** "RATF Horn Room Catwalk Analysis," by J. H. Kincaid, Benham Report, Revision 2, March 18, 2009.

Design Goal: Eliminate Dynamic Coupling

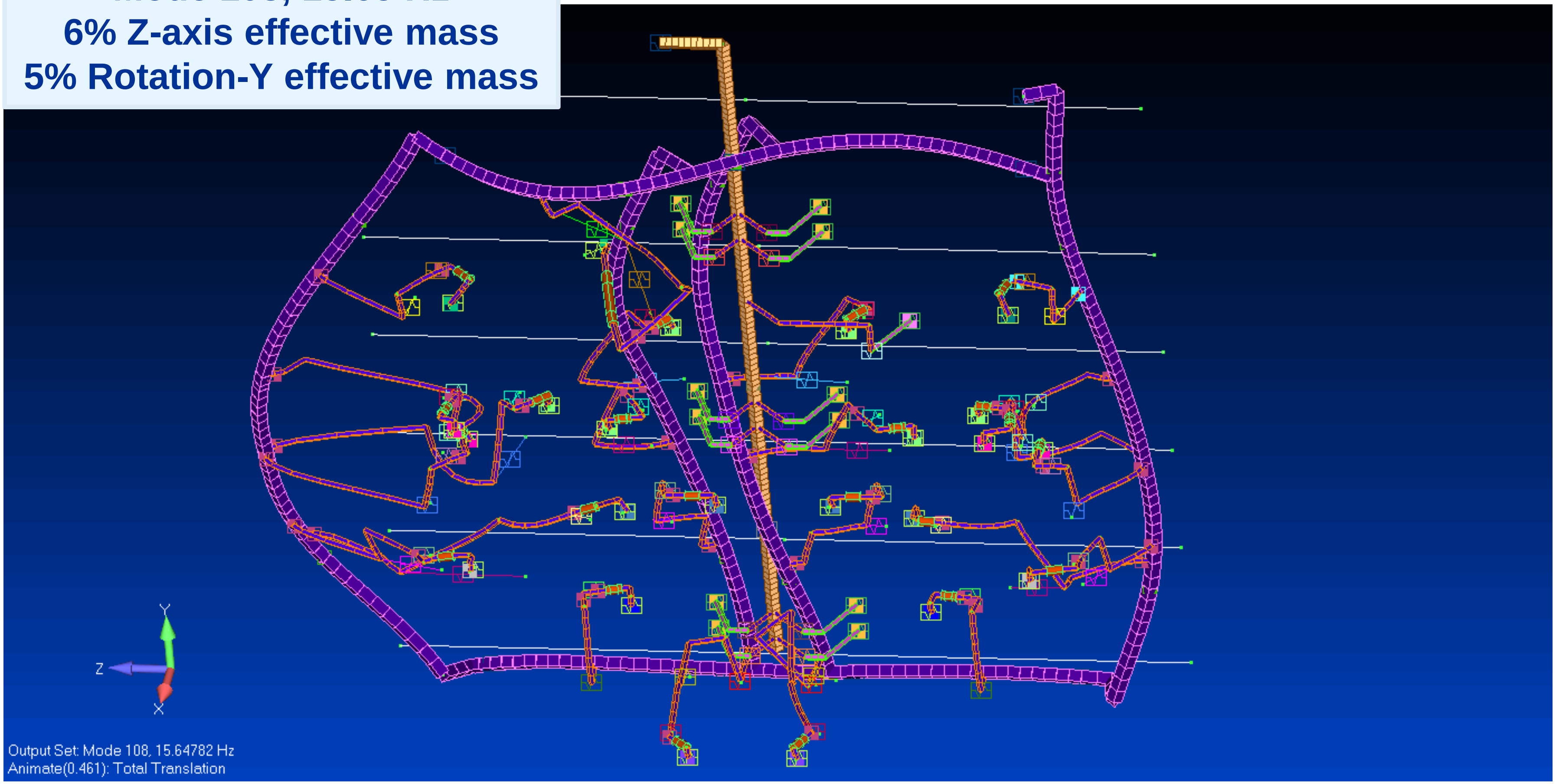
Reference: http://personal.cityu.edu.hk/~bsapplec/design2.htm



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.

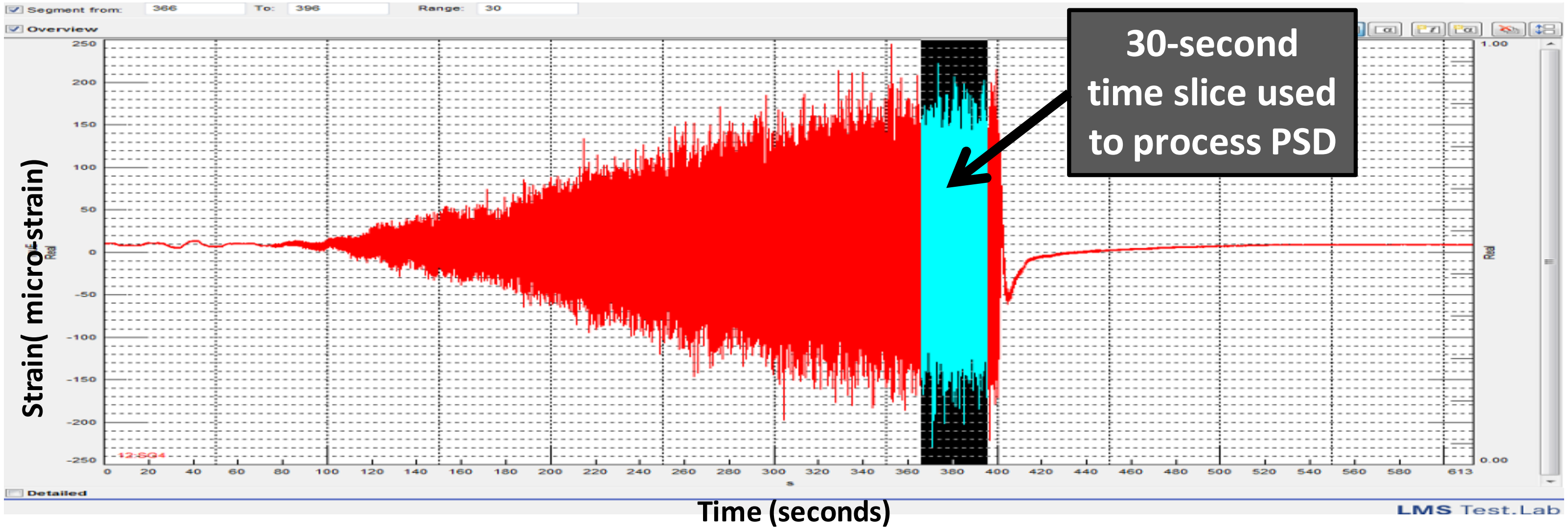
Configurations Analyzed Summary of Results

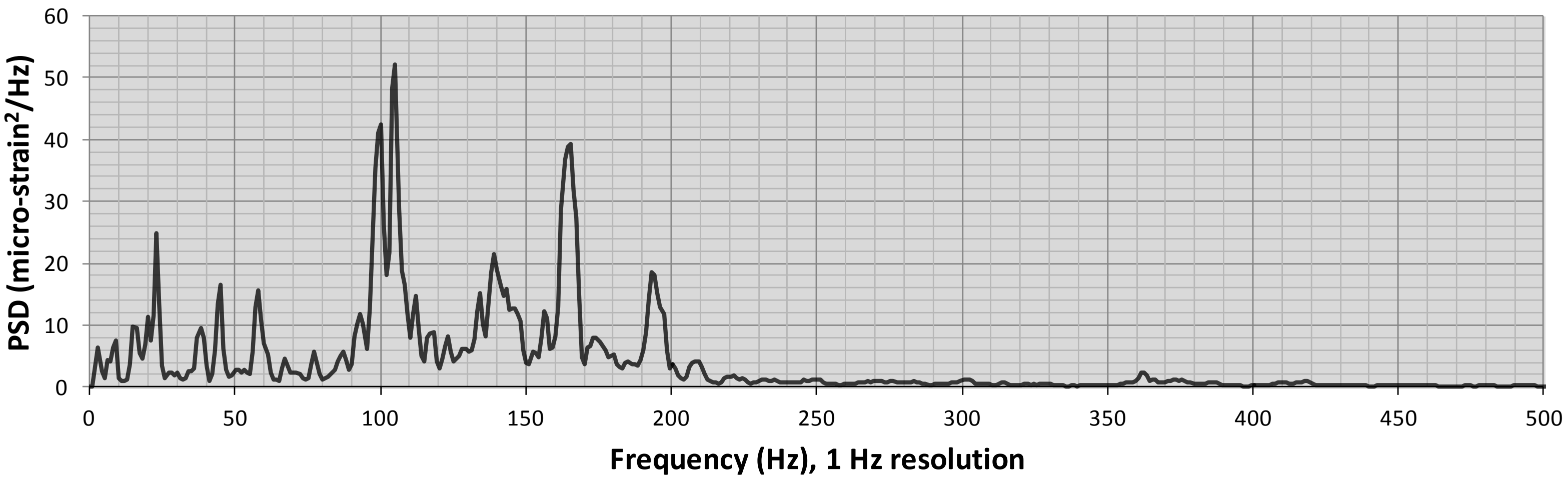
LEGEND:
[red box] = High effective mass piping modes

Configuration Analyzed	TEAM Modulator Piping Modes	WAS 5000 Modulator Piping Modes	Piping System High Effective Mass Modes
1. Baseline Configuration	3.00-50.26 Hz	3.91-49.37 Hz	**10.66 Hz**, 13.68 Hz, 15.65 Hz, **15.72 Hz**, 47.20 Hz
2. Adding lateral constraints to TEAM modulators	6.92-50.26 Hz	3.91-49.37 Hz	**10.44 Hz**, **15.75 Hz**, 13.29 Hz, 47.20 Hz
3. Removing all constraints from the TEAM modulators	2.52-50.27 Hz	3.91-49.37 Hz	**10.67 Hz**, 13.71 Hz, **15.72 Hz**, 34.92 Hz, 47.20 Hz
4. Add 500lb mass to the base of the TEAM modulators	2.35-50.19 Hz	3.91-49.37 Hz	**10.63 Hz**, 13.58Hz, 15.60 Hz, **15.70 Hz**, 47.17 Hz
5. Isolate the TEAM Modulators – **Gamma flex hose**	1.11-50.95 Hz	3.91-49.37 Hz	**10.17 Hz**, 12.03 Hz, 14.87 Hz, **15.40 Hz**, 47.33 Hz
6. Isolate the TEAM modulators – **Mason braided flex hose reoriented 90°**	2.90-50.80 Hz	3.91-49.37 Hz	**10.66 Hz**, 13.67 Hz, 15.63 Hz, **15.69 Hz**, 47.20 Hz
7. Add new SAIC-Benham recommended pipe supports	3.05-100.14 Hz	3.91-100.21 Hz	23.58 Hz, 33.51 Hz, **91.22 Hz**, **94.19 Hz**
8. Add new SAIC-Benham and NASA recommended pipe supports	3.05-100.19 Hz	3.91-100.21 Hz	30.96 Hz, 31.11 Hz, **91.32 Hz**
9. **Combine #6 and #8**: New SAIC-Benham and NASA recommended pipe supports **Mason braided flex hose reoriented 90°**	2.93-100.18 Hz	3.91-100.21 Hz	30.89 Hz, 31.23 Hz, **91.31 Hz**
10. **Combine #5 and #8**: New SAIC-Benham and NASA recommended pipe supports with soft connection to TEAM modulators using **Gamma flex hose**	1.10-100.09 Hz	3.91-100.21 Hz	1.13 Hz, 1.32 Hz, 8.85 Hz, 48.86 Hz, **90.43 Hz**

Adding piping supports increases high effective mass piping modes to 90 Hz or greater, decoupling from the RATF building and catwalk modes.

RATF Acoustic Verification Testing

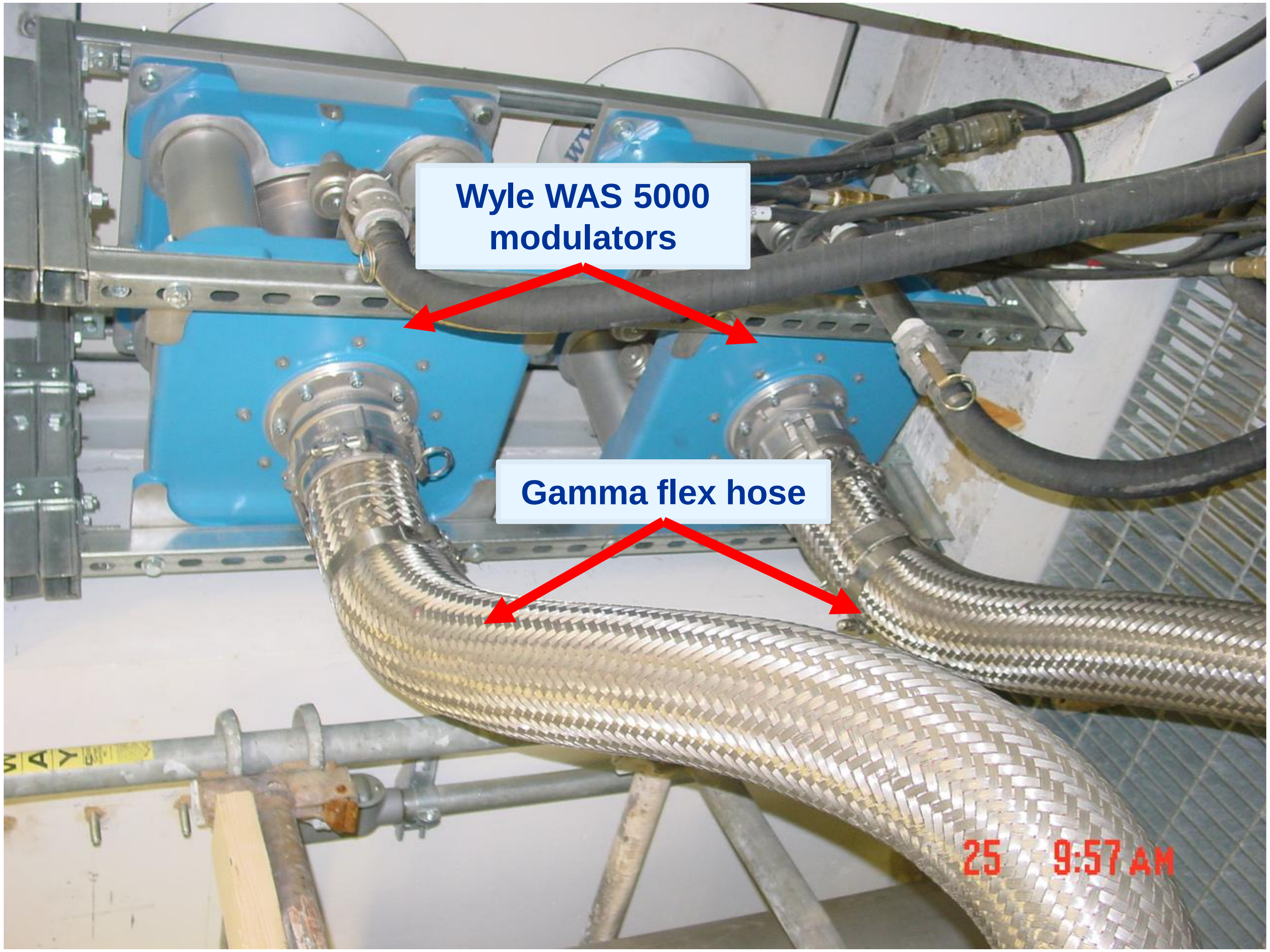




T-Junction strain measurements acquired during RATF acoustic verification testing indicates resonant modes at 99 Hz and 105 Hz, validating the finite element model and redesign goal of moving the major piping system modes to greater than 90 Hz.

Configuration Analyzed

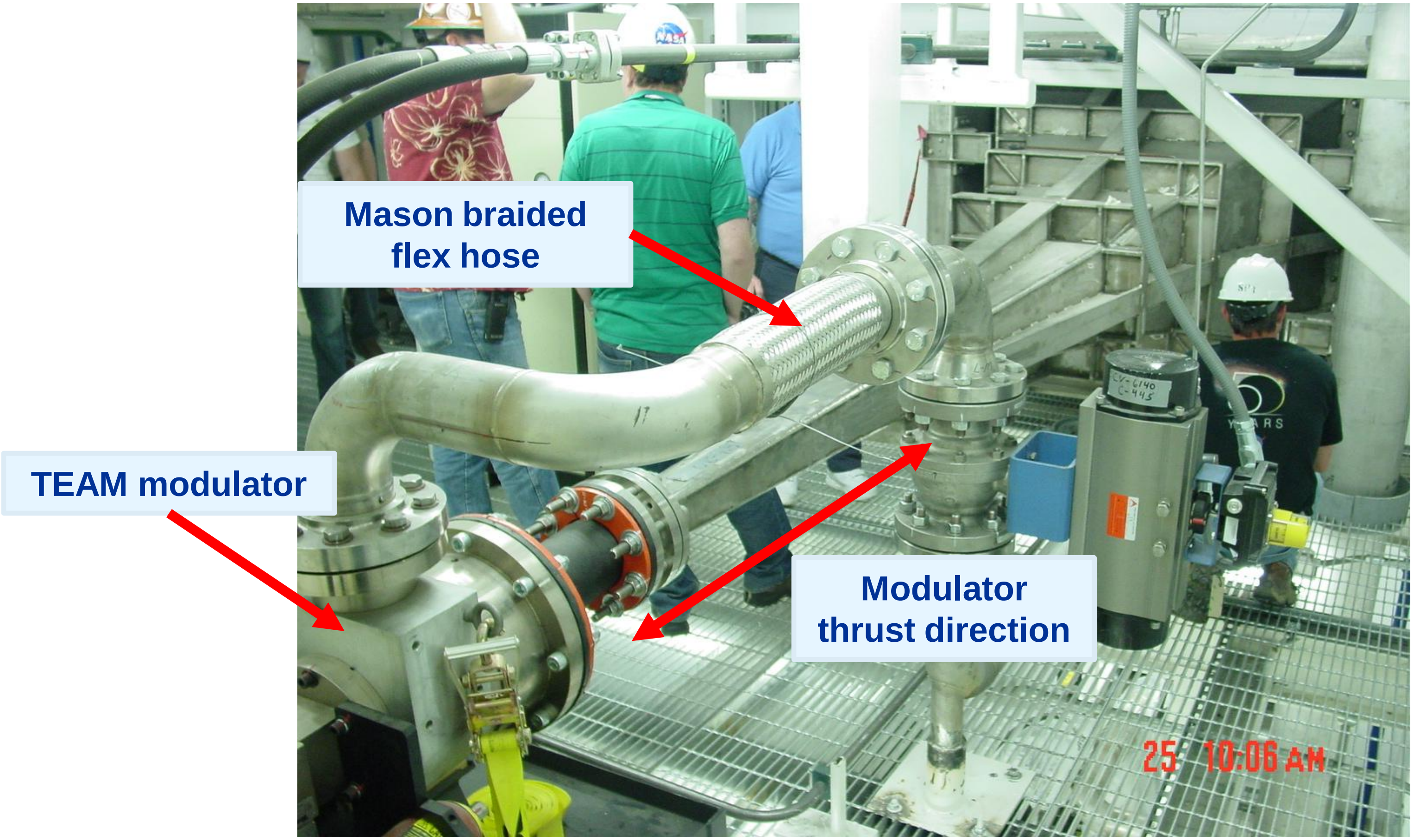
5. Isolate the TEAM modulators – Gamma flex hose



The Gamma flex hose provides a soft, flexible connection (4” bend radius) to the Wyle WAS 5000 modulators.

Configuration Analyzed

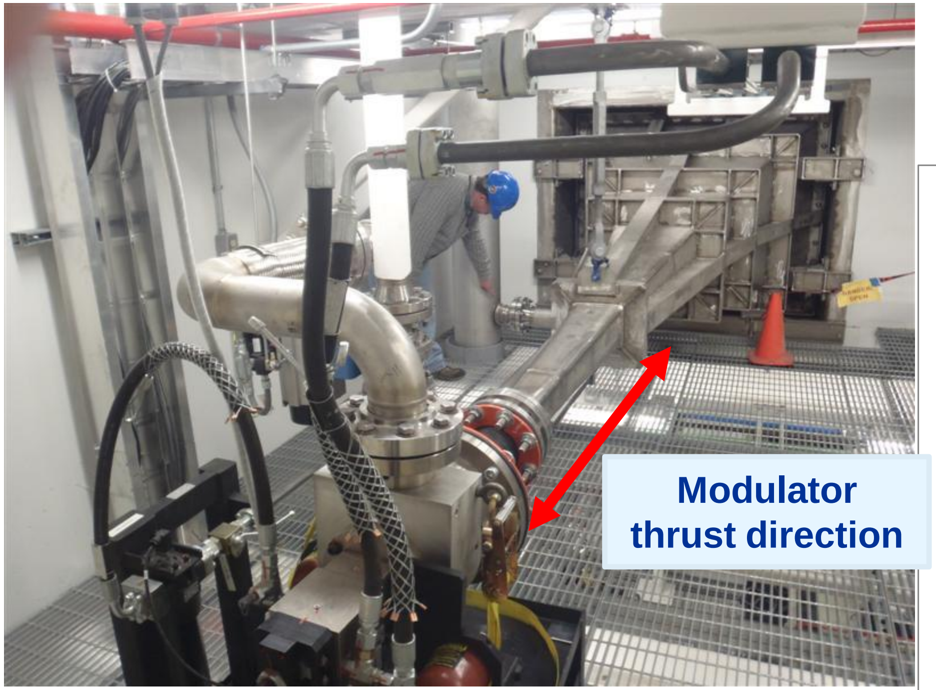
6. Isolate the TEAM modulators – Mason braided flex hose reoriented 90^{o}



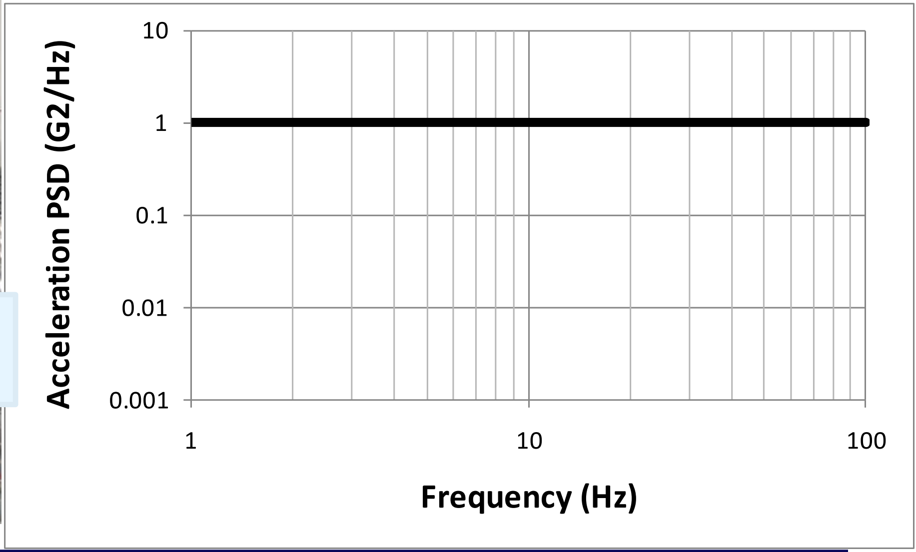
The as-built orientation of the Mason braided flex hose is **non-standard practice**. Need to reorient the flex hose 90^{o} so that it is perpendicular to the modulator thrust direction to limit piping vibration fatigue.

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 perform by applying a unit acceleration forcing function to the TEAM modulator thrust direction, and recover dynamic bending moments at the T-junction.

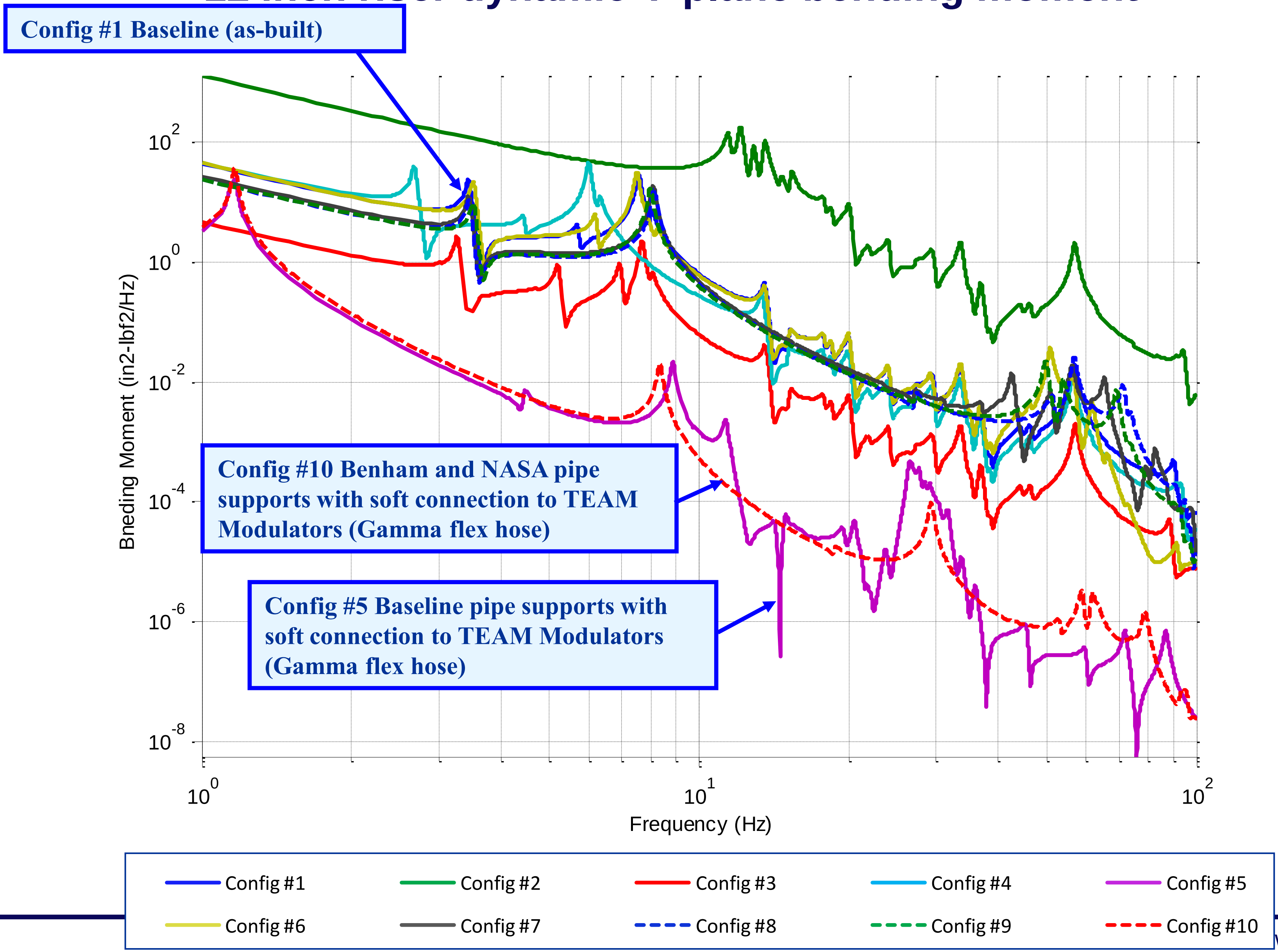


Unit acceleration forcing function applied in the modulator thrust direction from 1-100 Hz



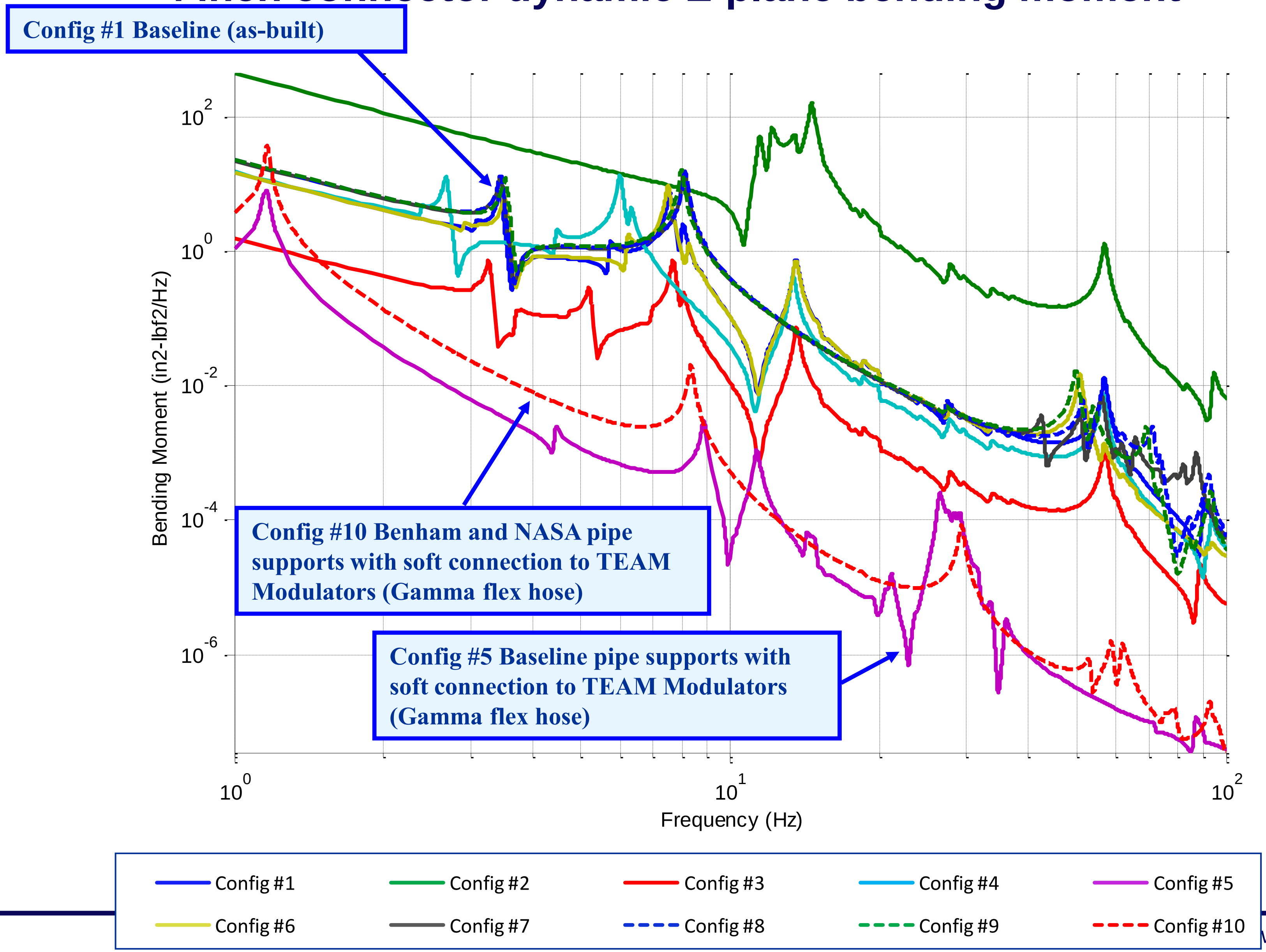
Forced Response Analysis

12 inch riser dynamic Y-plane bending moment



Forced Response Analysis

4 inch connector dynamic Z-plane bending moment



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).

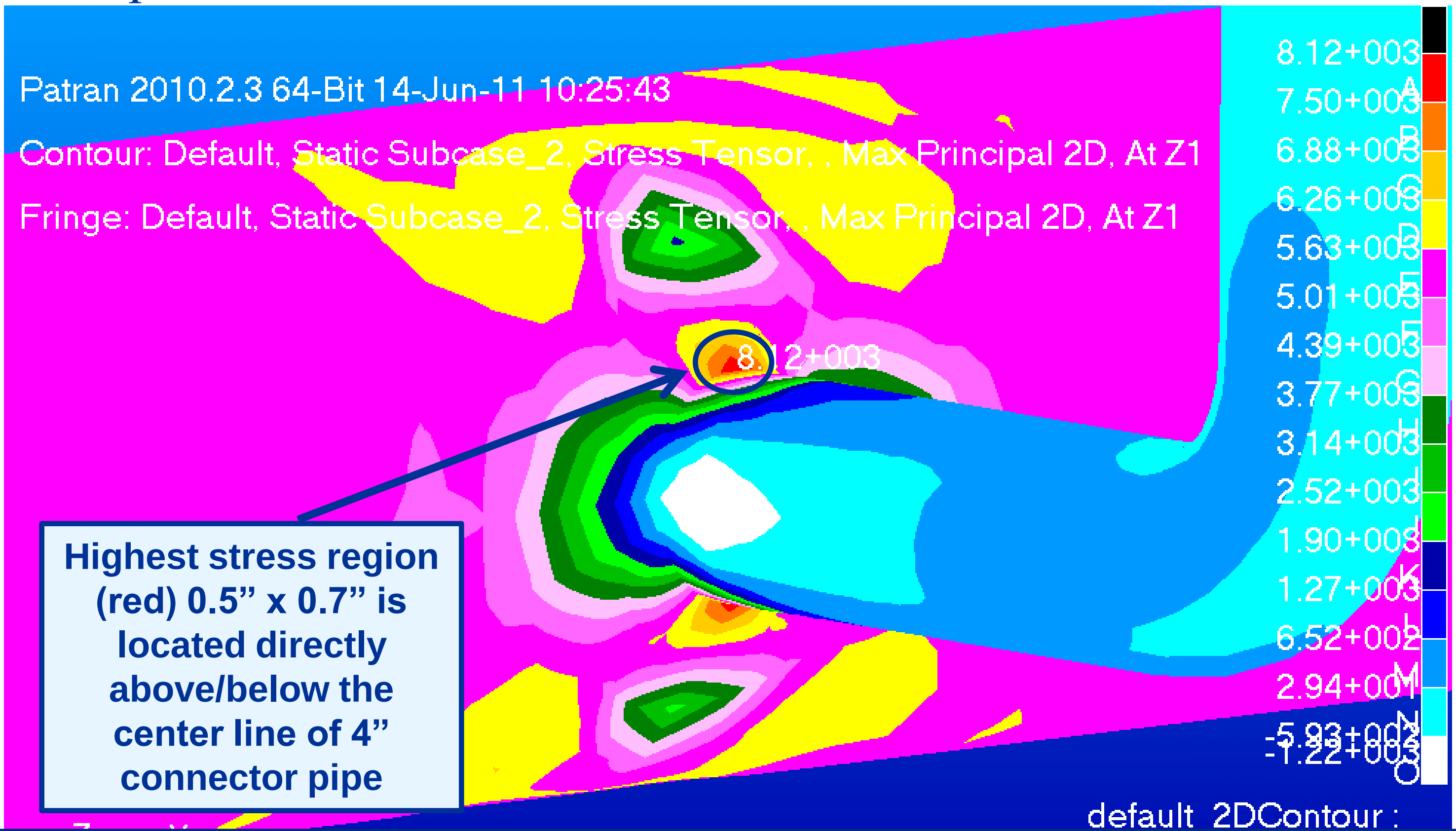
- Examining the bending moment results for the 4 inch and 12 inch riser indicates that Configurations #5 and #10 provide the largest reduction in bending moment compared to Configuration #1 (baseline).

To prevent long term **piping fatigue to 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.

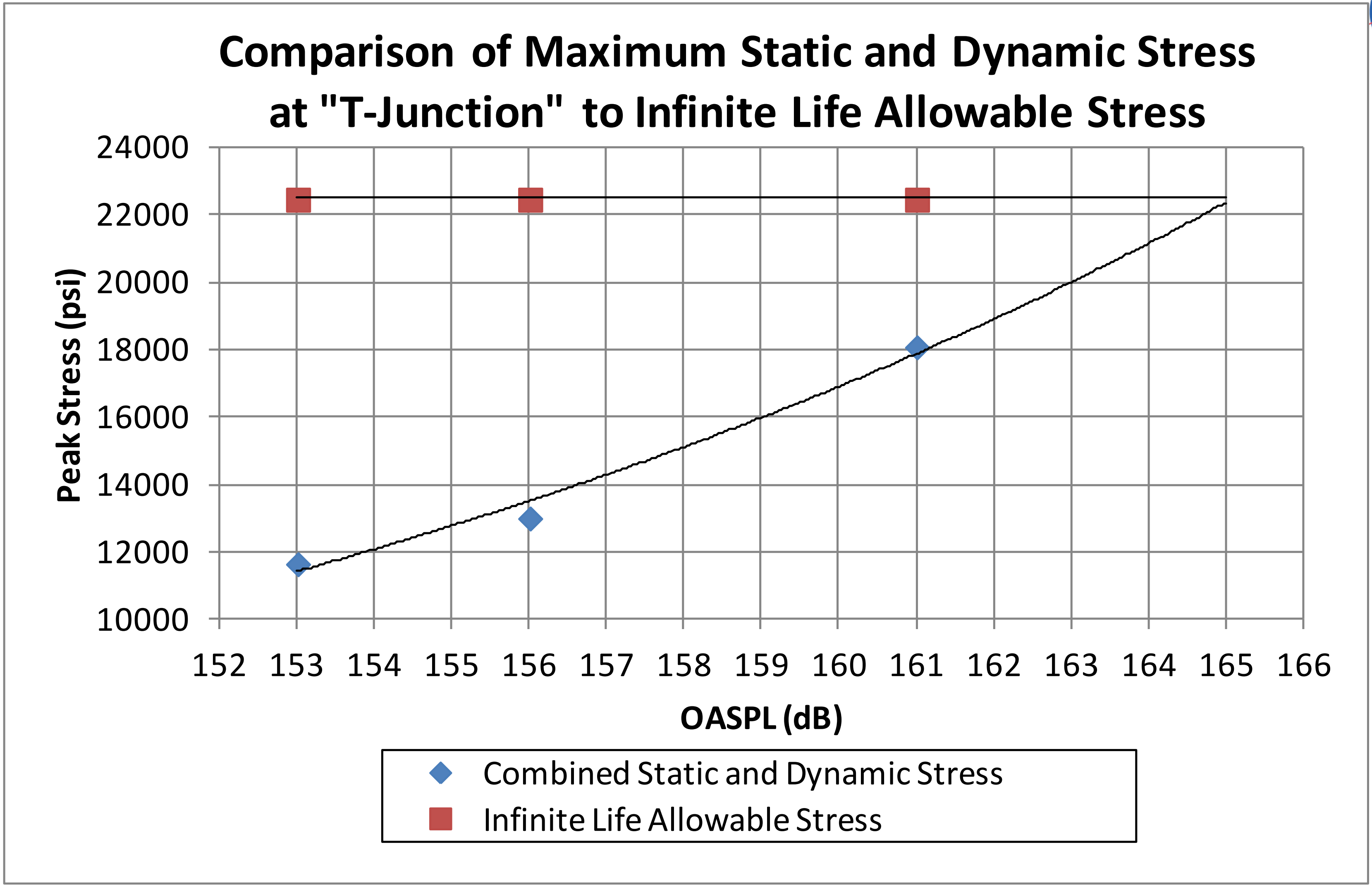
Stress Field Analysis of T-Junction

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 <u>maximum principal stress</u> (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

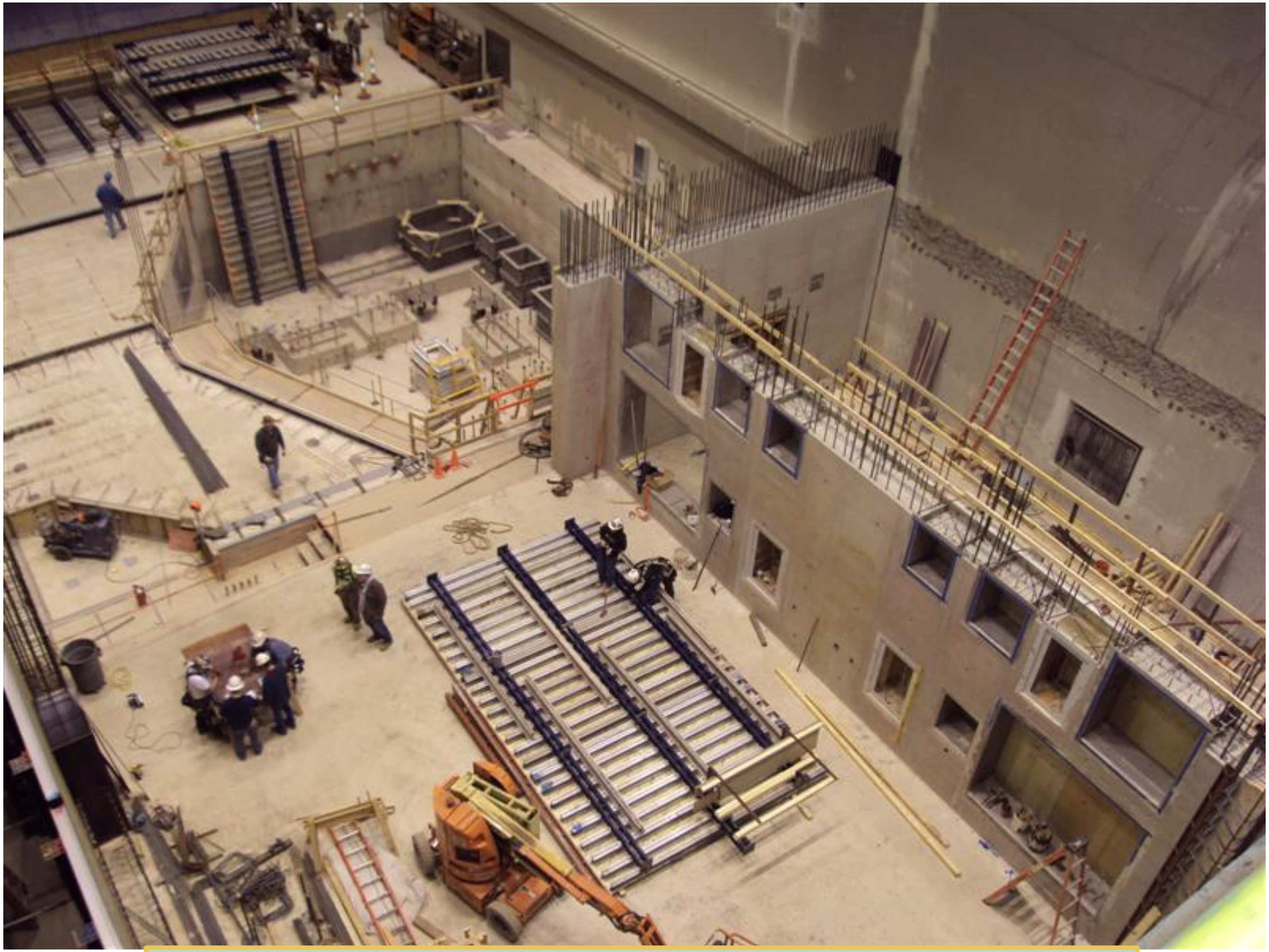
Overhead View – Preparation Horn Room Pour 1

Installation of horn frames and rebar

Overhead View – Horn Room Pour 1

Concrete pour #1 completed October 2009

Overhead View – Horn Room Pour 1



Concrete pour #1 completed with forms removed

Overhead View – Preparation Horn Room Pour 2

Horn wall level 2 horn frame and rebar installation

Overhead View – Horn Room Pour 2

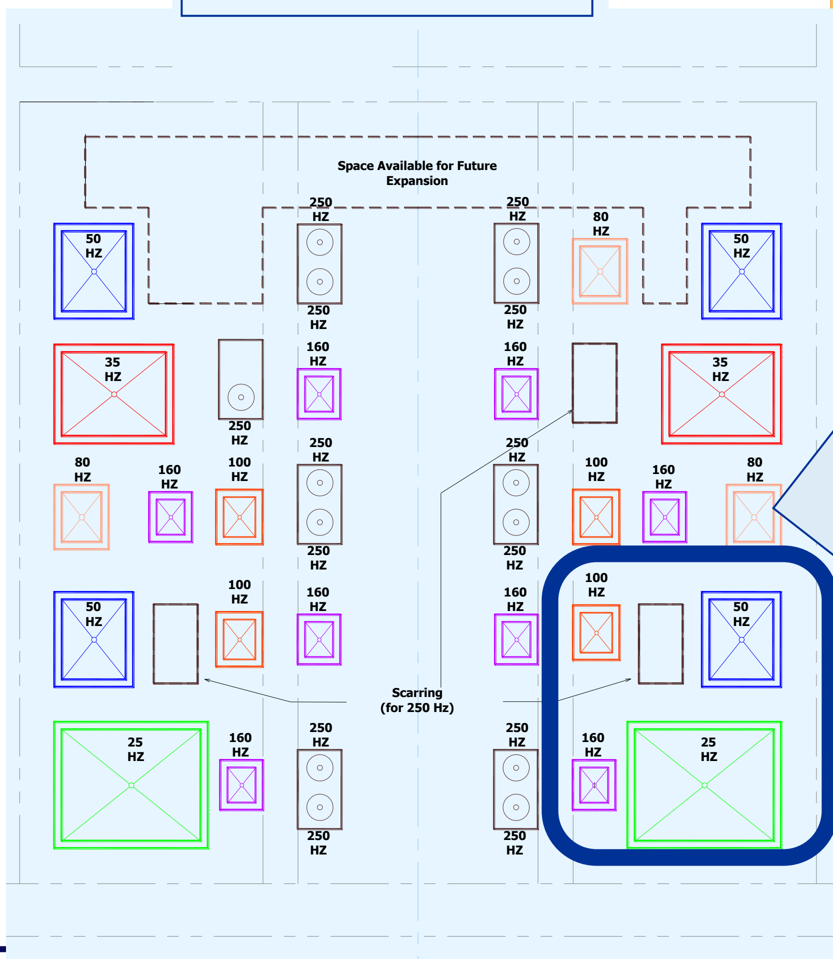
Concrete pour #2 completed with forms removed

Horn Room and Chamber Wall Pour

Concrete pour of walls completed with forms

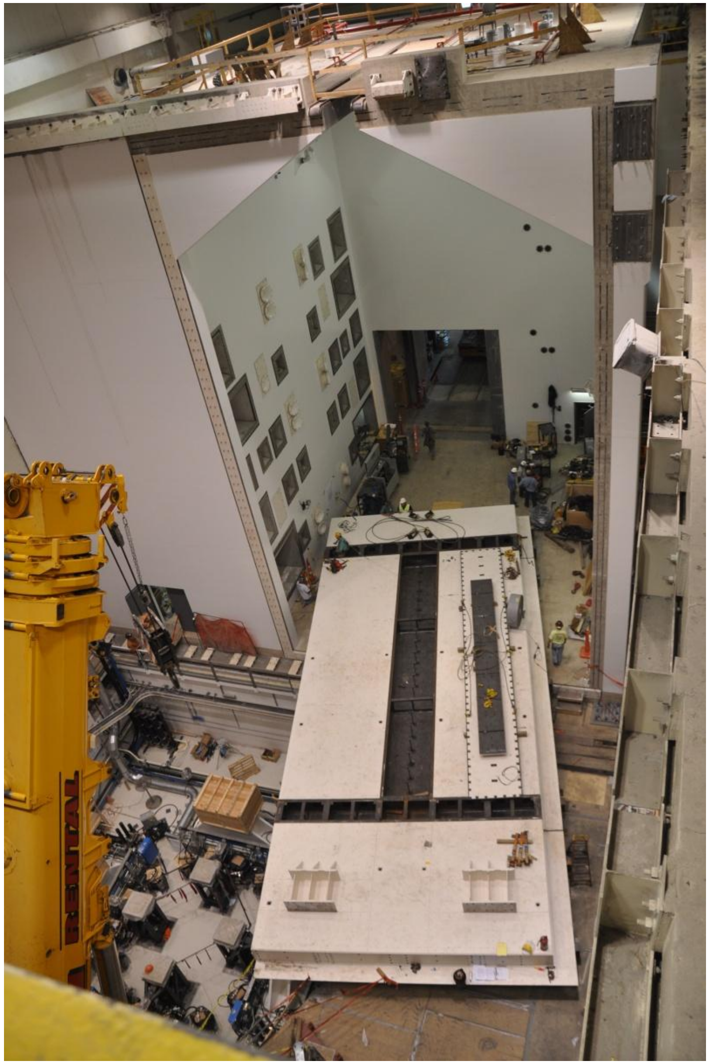
Horn Wall – Installation of Horn Frames

Legend	80 Hz
25 Hz	100 Hz
35 Hz	160 Hz
50 Hz	250 Hz



Construction photo showing the installation of the final RATF horn (25 Hz)

East Chamber Door
(September 2010)

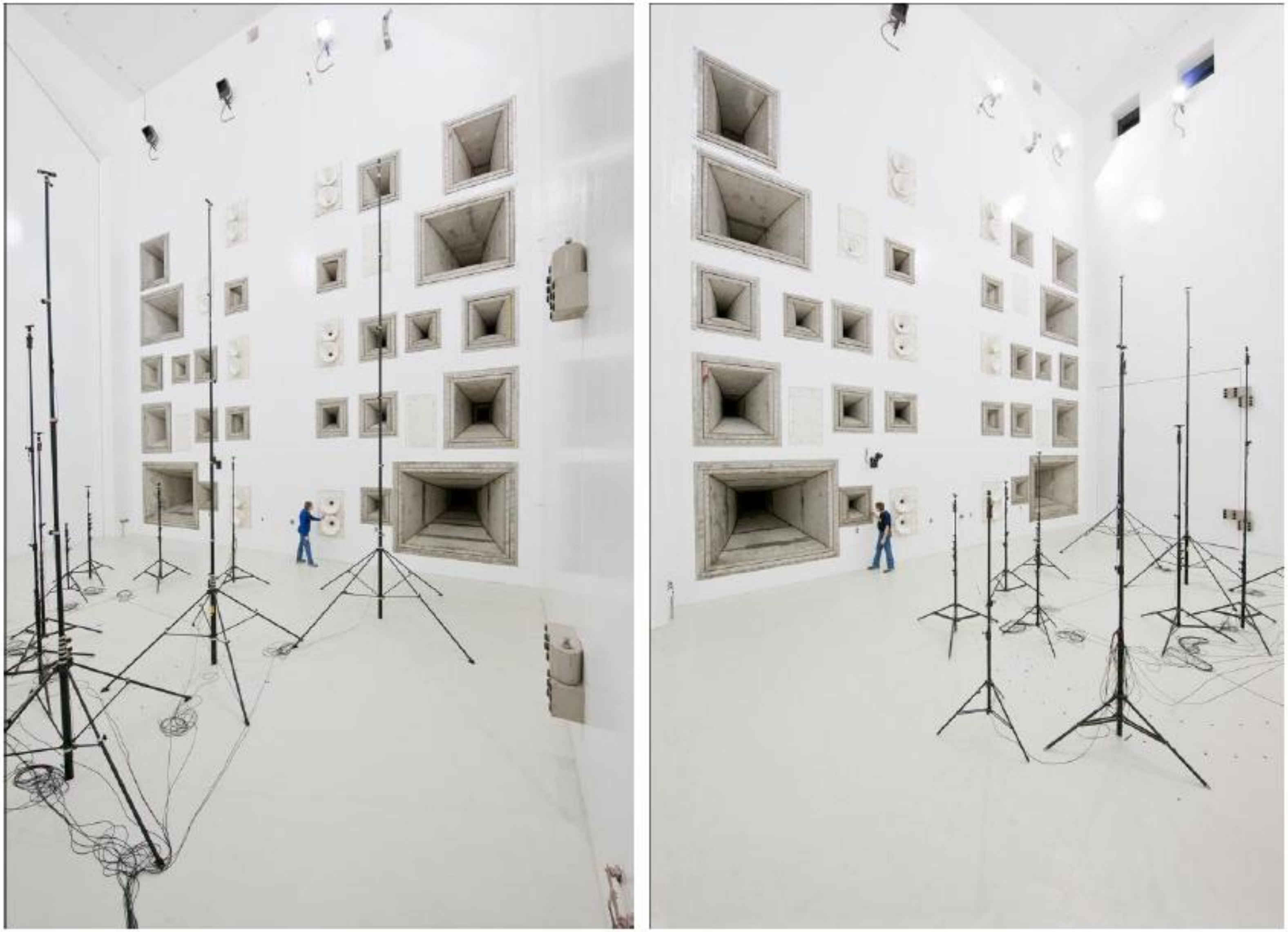


Installation of 675,000 lb door.

RATF Horn Wall
(September 2010)

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!

(Active) Reverberant Acoustic Test Facility	Location	Volume (ft^3)	Max. OASPL (dB) Empty Chamber	Year Commissioned
Lockheed Martin Missiles and Space, bldg.156, cell no.1, LVATF	Sunnyvale, CA	189,200	156.5	1973
NASA Plum Brook Station	**Sandusky, OH**	**101,200**	**163.0**	**2011**
Lockheed Martin Space Systems	Denver, CO	75,900	154.0	1985
Boeing Satellite Development Center (Boeing SDC)	El Segundo, CA	67,800	155.0	2004
Lockheed Martin Missiles and Space (LMMS), bldg.159	Sunnyvale, CA	64,000	157.3	1996
Mitsubishi Electronics	Kamakura, Japan	61,700	152.0	2002
Large European Acoustic Facility (LEAF) at ESTEC	Noordwijk, The Netherlands	59,000	154.5	1990
Northrop Grumman Space Technology (NGST), LATF	Redondo Beach, CA	51,600	154.0	1996

Reference:

“**The Development of the Acoustic Design of NASA Glenn Research Center’s New Reverberant Acoustic Test Facility**,” by William O. Hughes, Mark E. McNelis, Aron D. Hozman, and Anne M. McNelis, NASA Glenn Research Center, Cleveland, Ohio, NASA Technical Memorandum 2011-217000, July 2011.

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