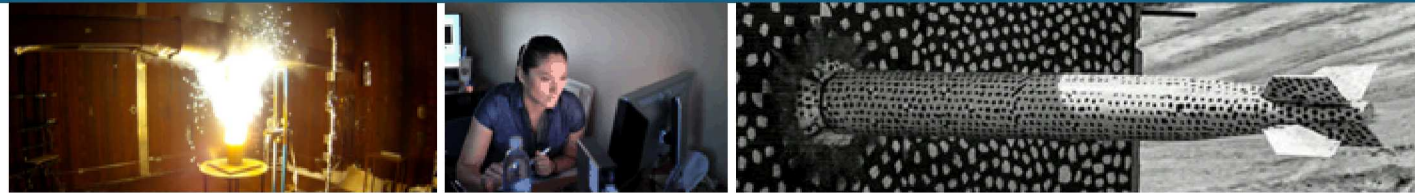


Comparison of Multi-Axis Testing of the BARC Structure with varying Boundary Conditions



PRESENTED BY

Dan Rohe

Ryan Schultz, Tyler Schoenherr, Richard Jones, Troy Skousen



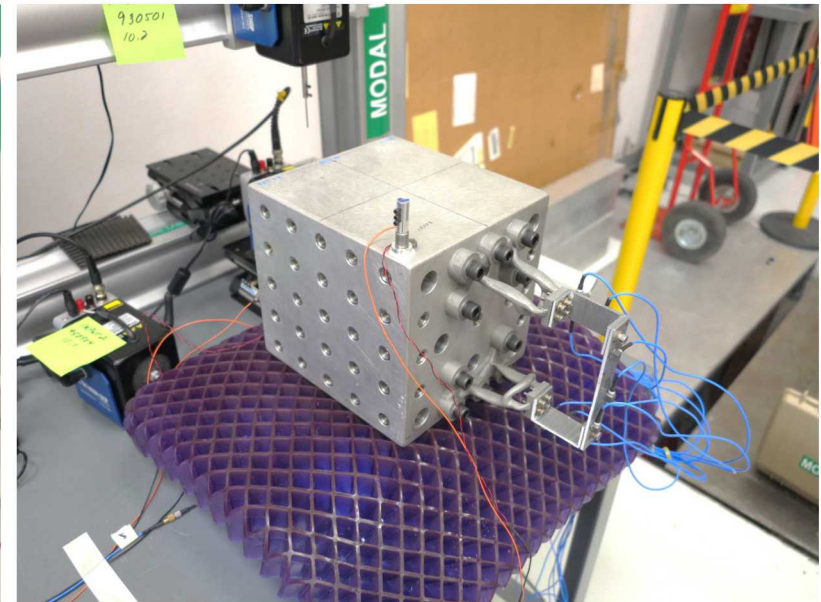
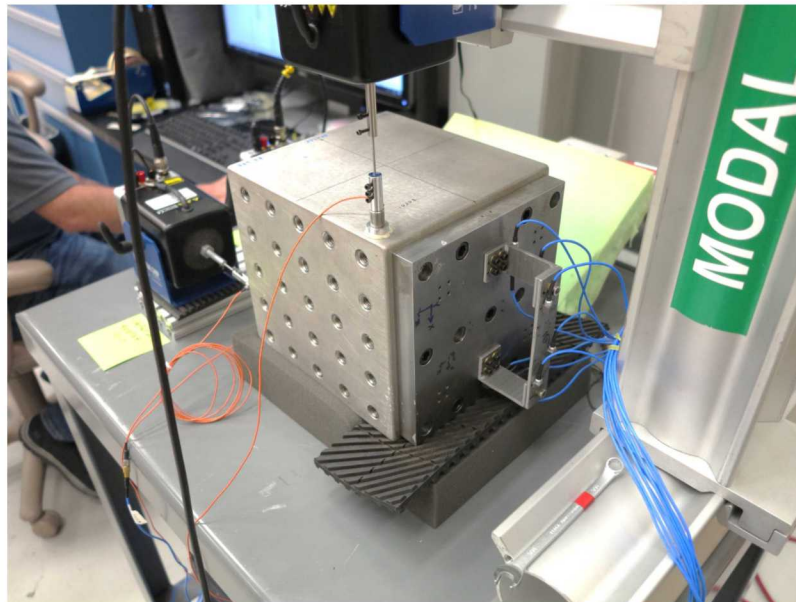
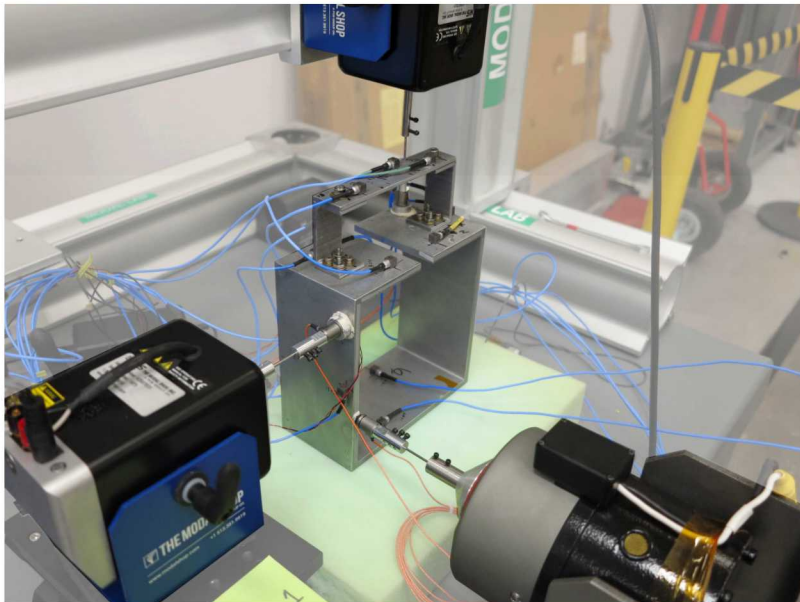
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The Box Assembly and Removable Component (BARC) structure was designed as a test bed to develop strategies for fixture design for component vibration testing.

Wanted to compare vibration responses on the removable component in a made-up “service” environment to those in a laboratory shaker test using a variety of fixtures.

- “Perfect” Impedance-Matched Multi-Axis Test (IMMAT)
- Traditional “rigid” plate fixture
- Topology-optimized, additively-manufactured fixtures

Investigate strains as well as acceleration response.



Truth Test

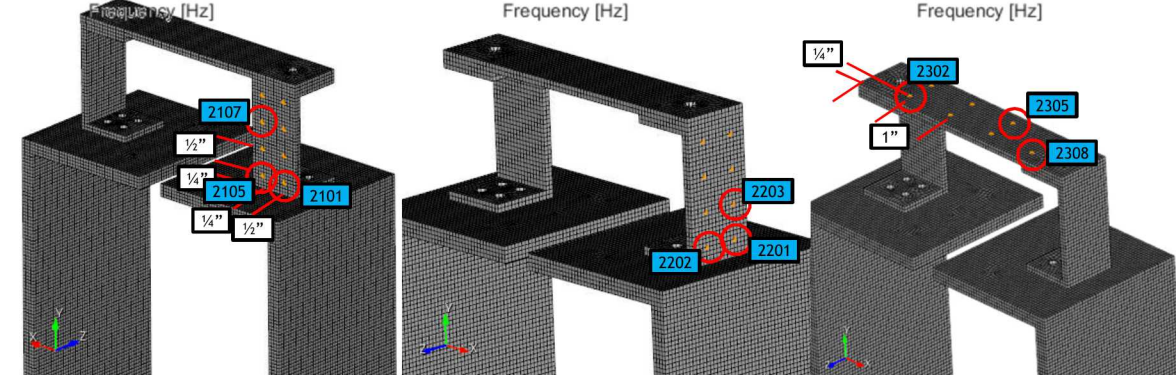
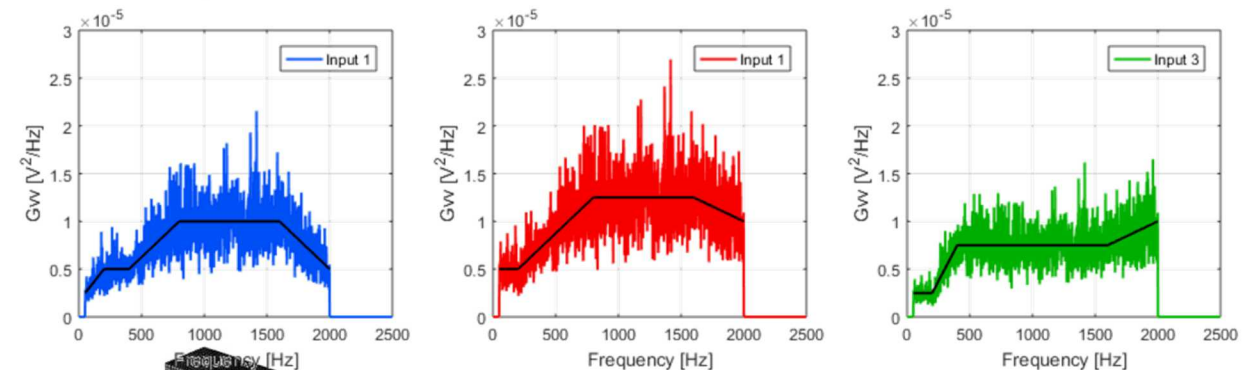
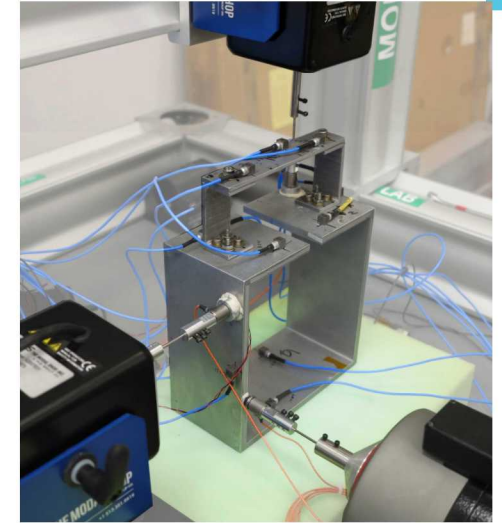
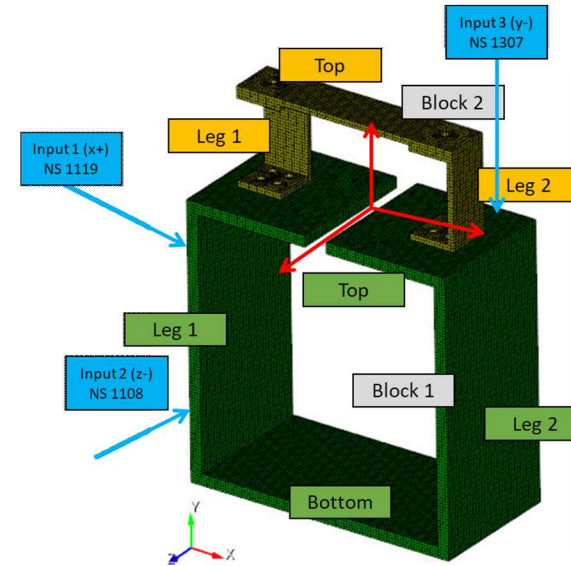
To investigate the fixture designs in some environment, that environment first needed to be developed.

- Vibration excitation was chosen over shock excitation due to controllability and existing capabilities.

A multi-axis input specification was developed that was representative of actual environments.

- 3 shakers were used to excite the structure.

Responses were measured at 9 triaxial locations on the removable component as well as a few points on the box itself (to allow identification of modes and shape expansion)



Control was performed in an open-loop sense using the arbitrary source capabilities of the B+K Lan-XI data acquisition systems.

An initial white-noise (buzz) test was performed in each configuration to develop transfer functions between the shaker amplifier input voltages and the response channels on the component.

The input cross-power spectral density (CPSD) matrix was derived using standard pseudo-inverse computations.

$$\begin{aligned}\mathbf{G}_{\ddot{x}\ddot{x}} &= \mathbf{H}_{xv} \mathbf{G}_{vv} \mathbf{H}_{xv}^* \\ \hat{\mathbf{G}}_{vv} &= \mathbf{H}_{xv}^+ \mathbf{G}_{\ddot{x}\ddot{x}} (\mathbf{H}_{xv}^*)^+\end{aligned}$$

This computation is performed frequency line by frequency line over all 1951 frequency lines between 50 and 2000 Hz, inclusive.

Time histories to play into the shaker amplifiers were created from the resulting voltage CPSD matrix using multiply-correlated signal synthesis [1].

A Constant Overlap and Add (COLA) method was used to synthesize longer time histories from multiple shorter frames while maintaining a smooth signal [1].

[1] D. O. Smallwood and T. L. Paez, "A frequency domain method for the generation of partially coherent normal stationary time domain signals," Shock and Vibration, vol. 1, no. 1, pp. 45–53, 1991.

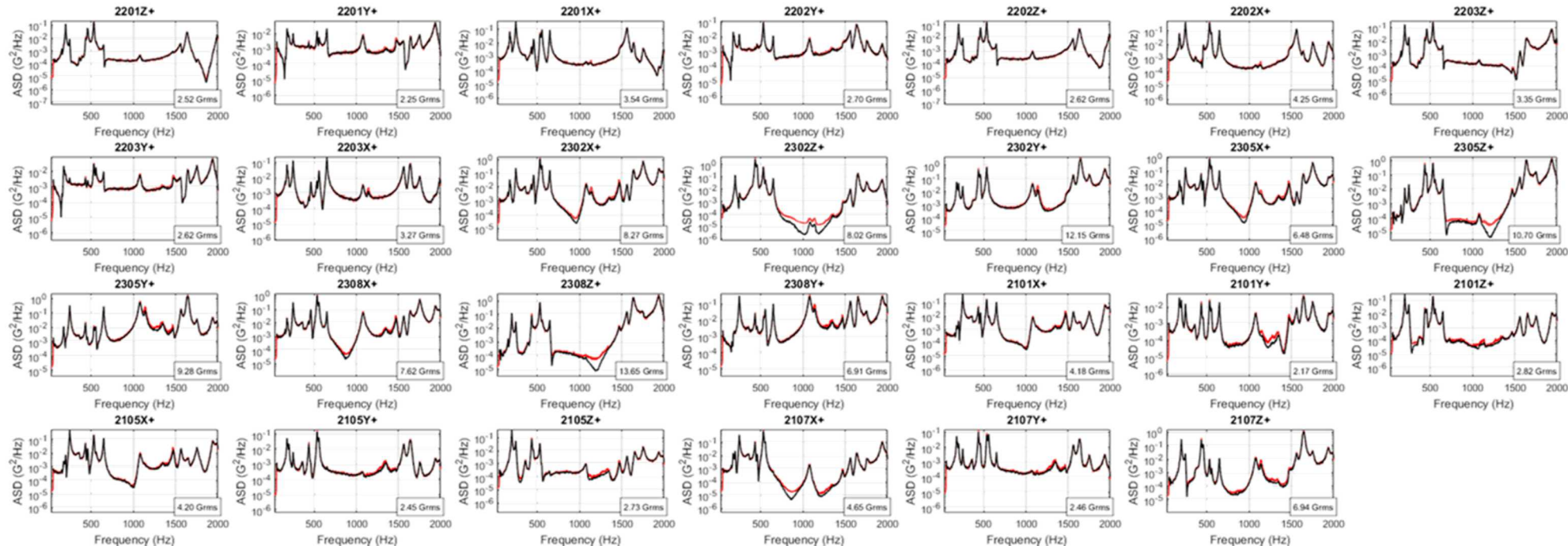
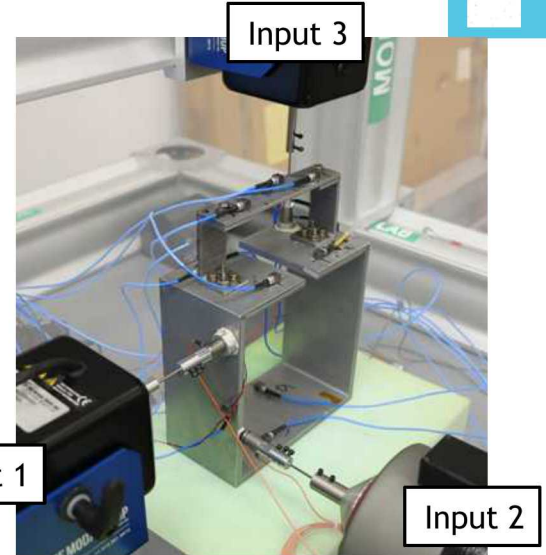
Replica Test

After the truth test was performed, a replica test was performed using the MIMO control.

This represents the “best” that an IMMAT approach could achieve.

- Exactly same boundary conditions
- Exactly same force input locations
- Forces are achievable

Matches to truth response data were very good, as expected.



Vibration Cube Test Planning

Design considerations in actual tests are:

- Controllability: Can the shakers force the test article to do what we want it to do?
- Force Required: Can we realistically achieve the force levels required to get the test article to do what we want it to do?

Use Finite Element Model to attempt to place shakers in some optimal sense.

5x5 candidate shaker locations on each face of the cube, brute force solution of control problem

Results from the control simulation were ranked from 1 (best) to 67,525 (worst) for both dB error on the response gauges and RMS force required.

The best set for dB error was not the same set for smallest force.

Summing the force and error ranking gave good compromise between control error and force required.

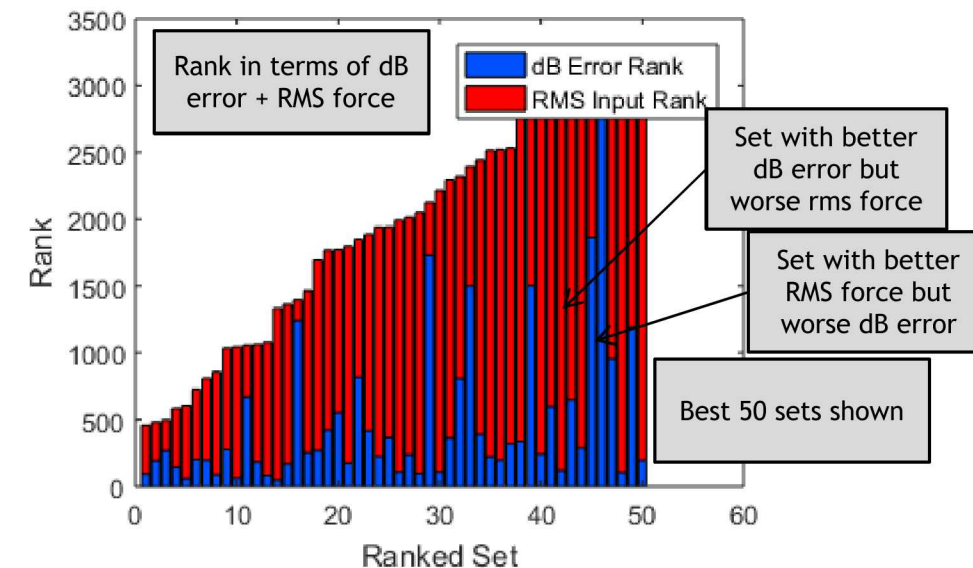
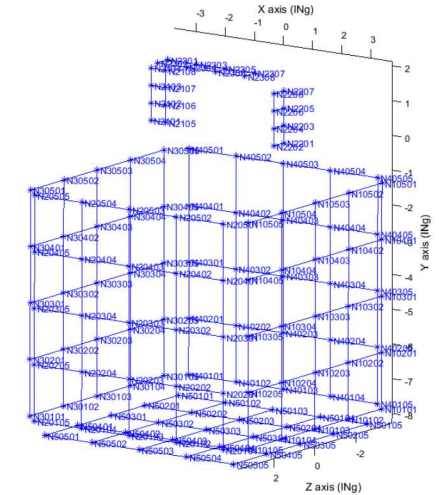


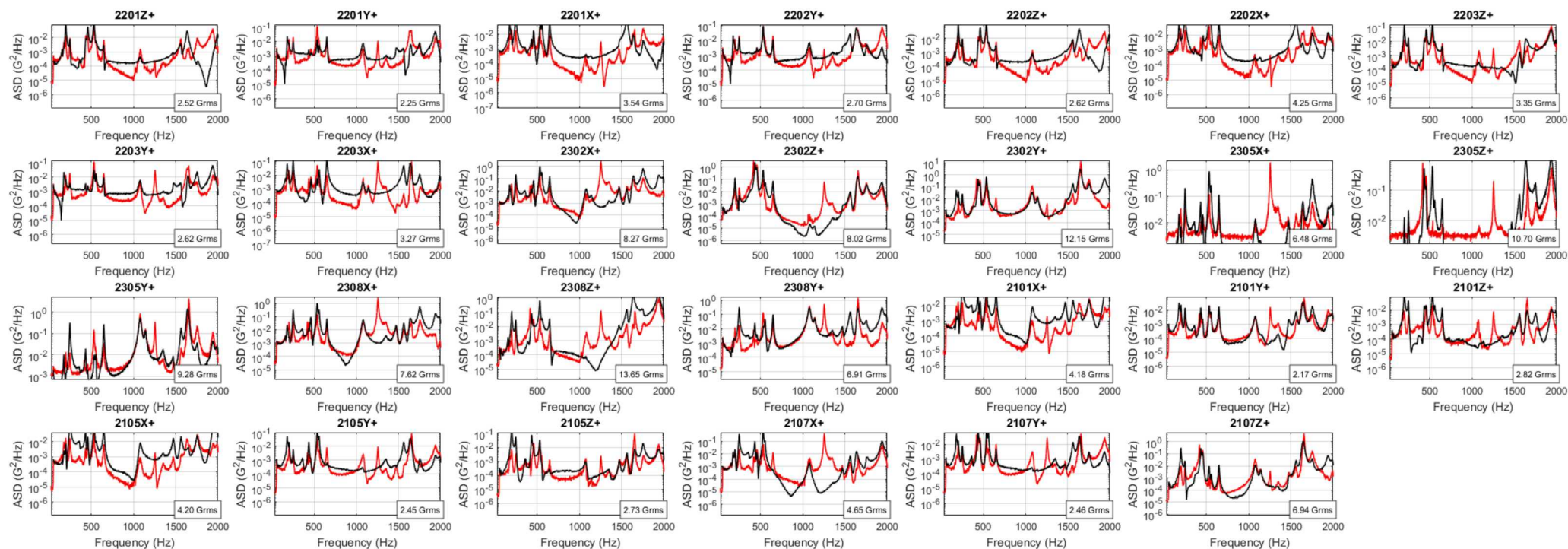
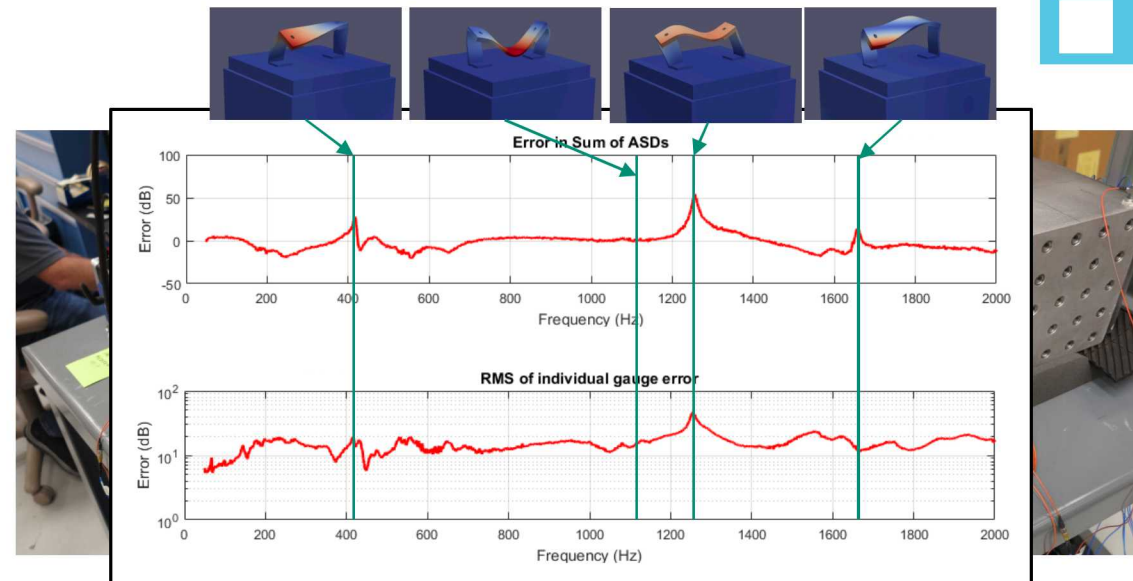
Plate Test

Plate fixture mounted to vibration cube with removable component attached to plate

Vibration cube set on foam for soft support.

Three shakers attached per best setup on previous slide.

Increase in mass required significantly more force to achieve similar levels, so linearity was assumed and forces were scaled down by -26 dB (approx. 1/20).



Additively Manufactured Fixture Tests

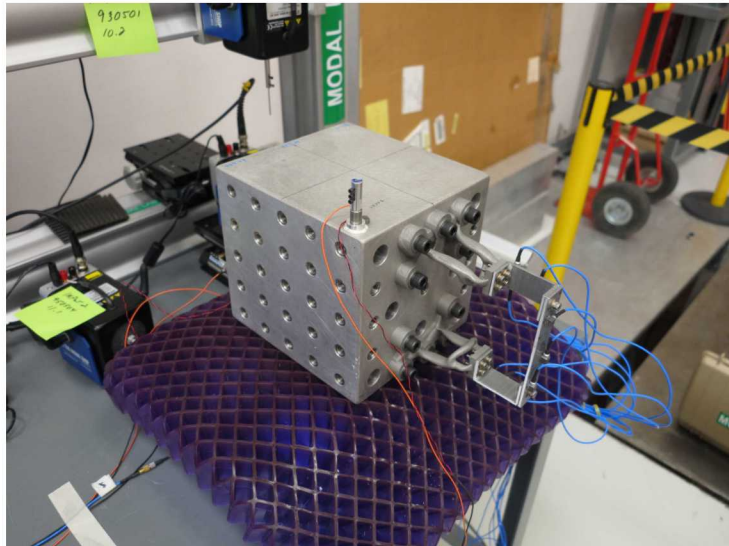
To improve upon the plate fixture, three fixtures had been designed using topology optimization to attempt to match the stiffness of the BARC structure

Optimization attempted to match static deflection due to gravity loads

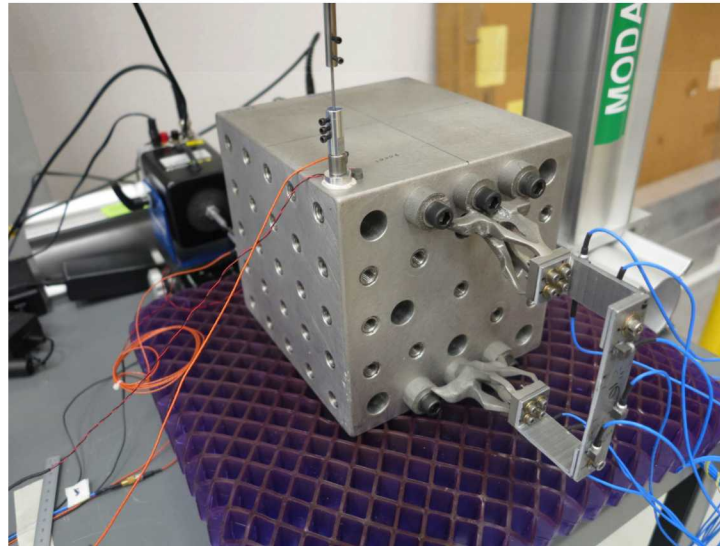
- Design 12: Objective to match six degrees of freedom (deflection and rotation) motion
- 3 Leg: Objective to match three degrees of freedom (only deflections) motion
- 2 Leg: Objective to match three degrees of freedom (only deflections) with the vertical direction weighted by a factor of 2

Identical shaker setup to the previous fixture

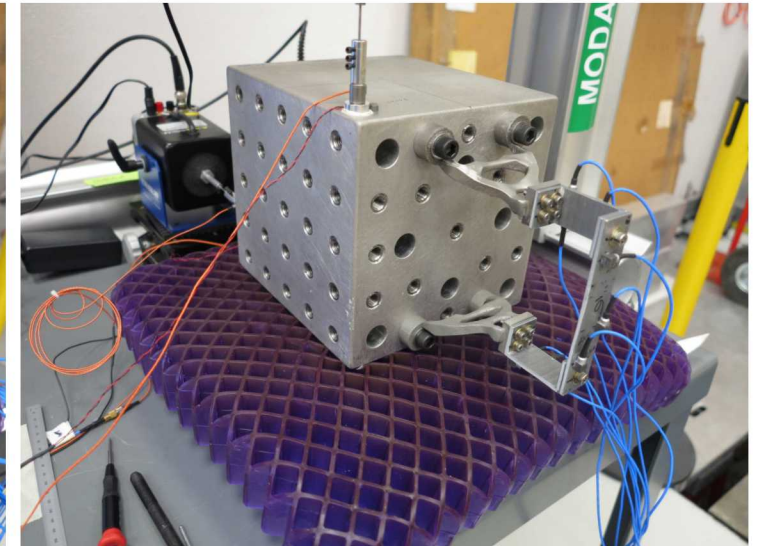
Increase in mass required significantly more force to achieve similar levels, so linearity was assumed and forces were scaled down by -20 dB (approx. 1/10) from the truth test.



Design 12

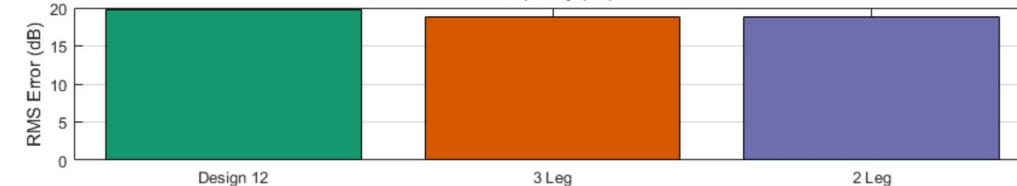
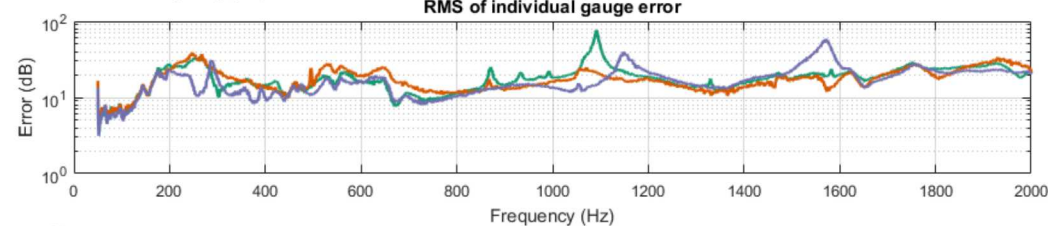
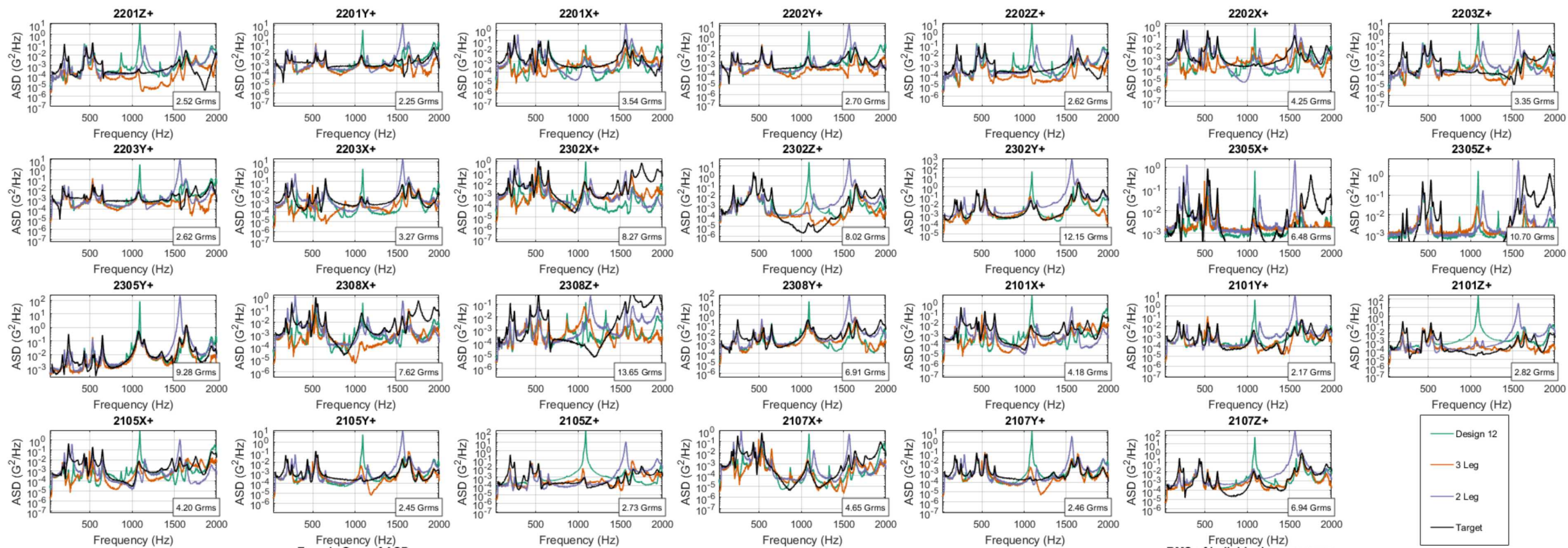


3 Leg

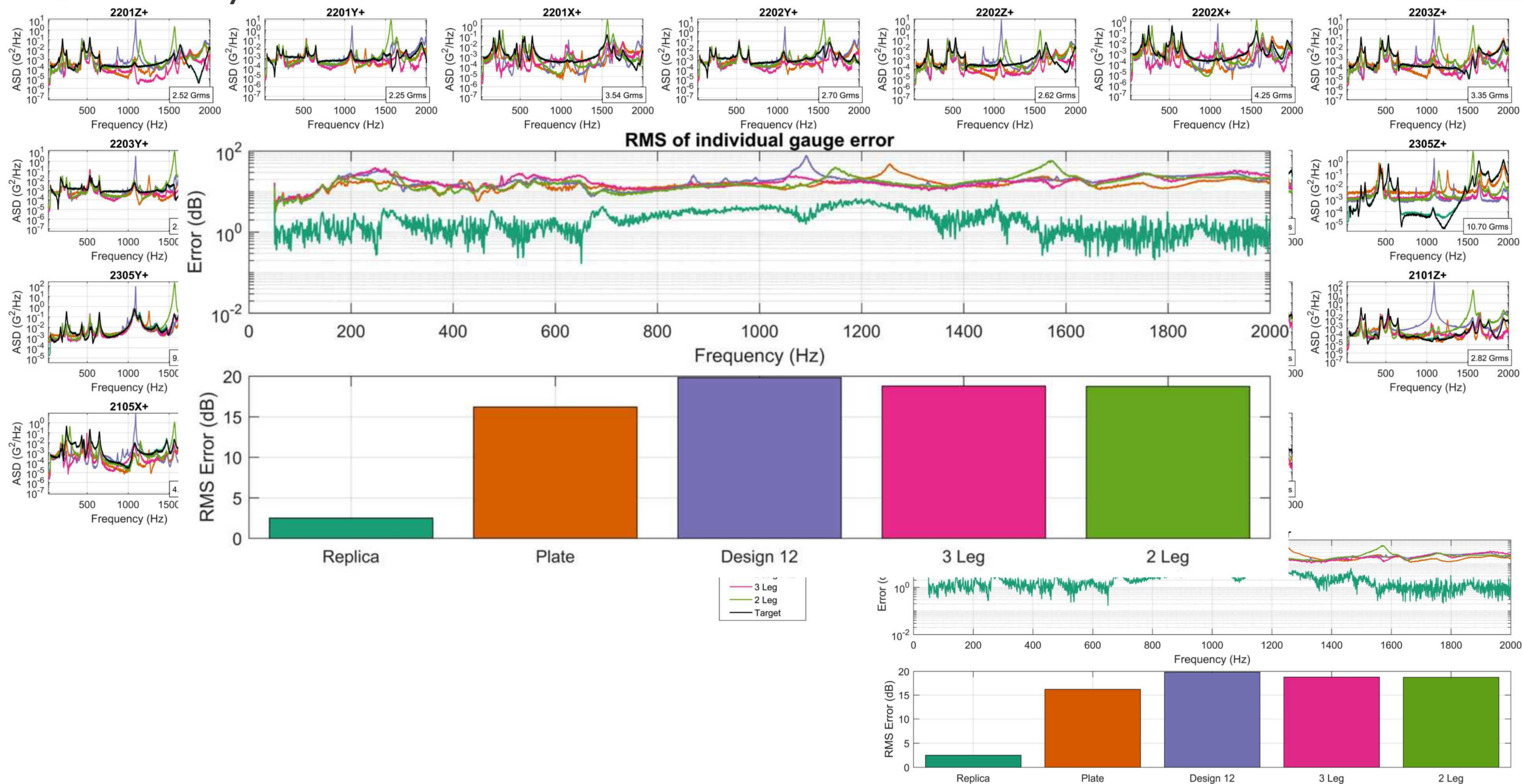


2 Leg

Additively Manufactured Fixture Tests



Summary of All Acceleration Results



Expansion to Strain

Typically, acceleration auto-spectral density plots are examined

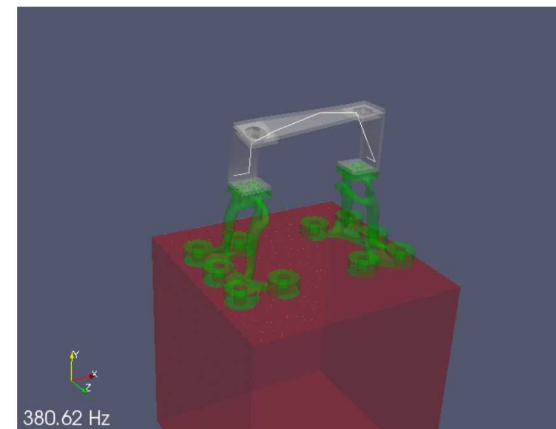
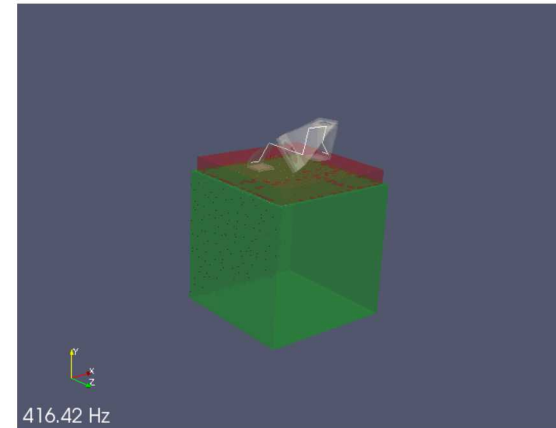
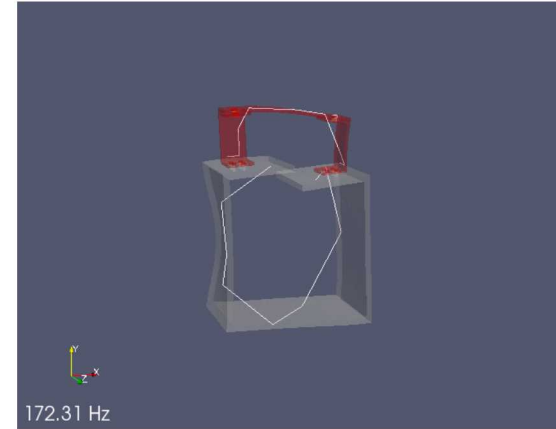
Stresses and strains are arguably more important to examine, though harder to measure.

Previous work showed SEREP could be used to expand test data to compute strains if they are present in the finite element model mode shapes

$$\begin{bmatrix} \mathbf{X}_{fem} \\ \boldsymbol{\varepsilon}_{fem} \end{bmatrix} = \begin{bmatrix} \boldsymbol{\Phi}_{fem} \\ \mathbf{E}_{fem} \end{bmatrix} \boldsymbol{\Phi}_{exp}^+ \mathbf{X}_{exp}$$

Using frequency domain integration and pre- and post-multiplying by the SEREP transformation, we can compute the strain CPSD matrix

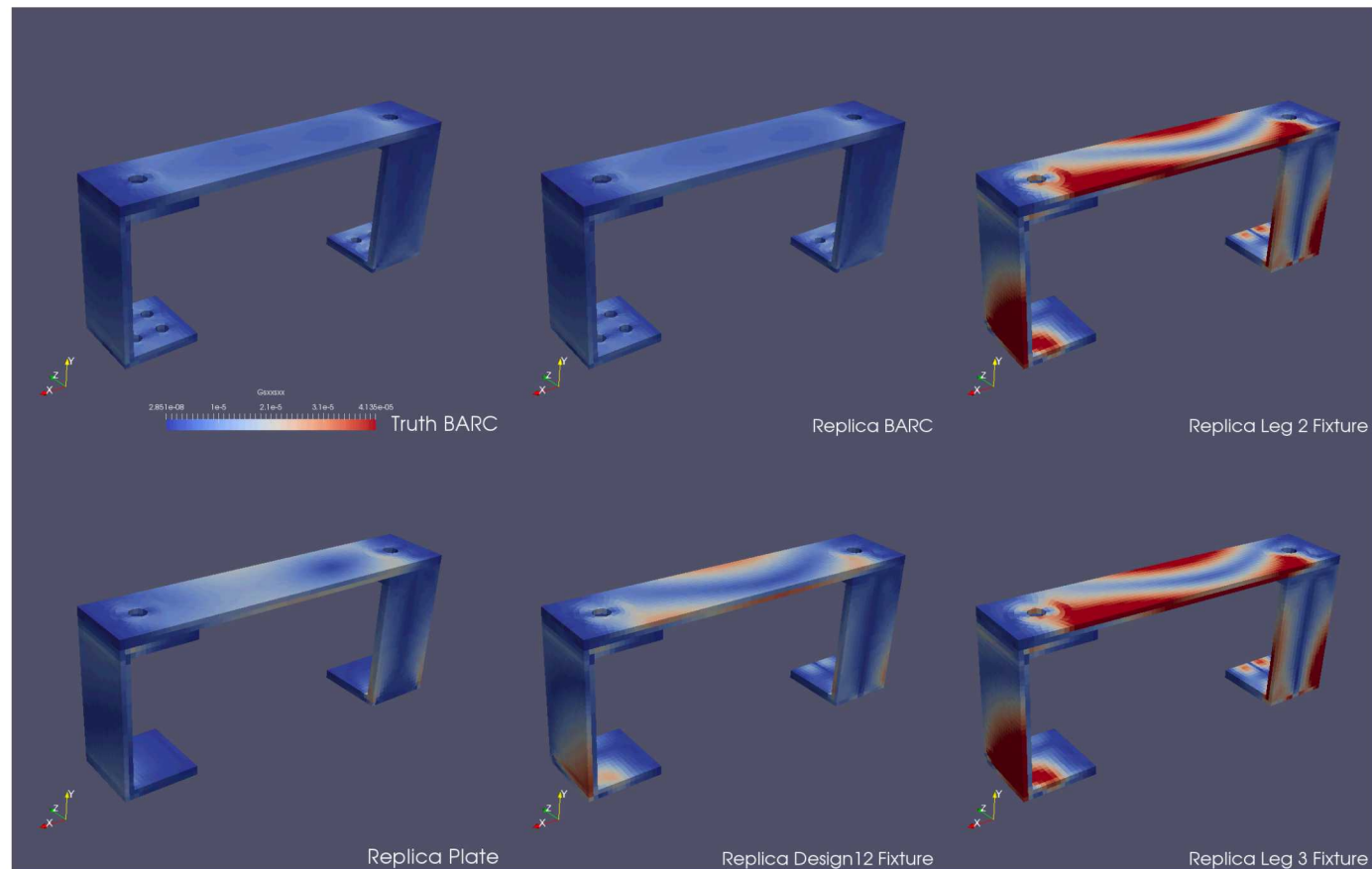
$$\mathbf{G}_{\varepsilon\varepsilon,fem} = \mathbf{E}_{fem} \boldsymbol{\Phi}_{exp}^+ \frac{\mathbf{G}_{\ddot{x}\ddot{x}}(\omega)}{\omega^4} (\boldsymbol{\Phi}_{exp}^+)^* \mathbf{E}_{fem}^*$$

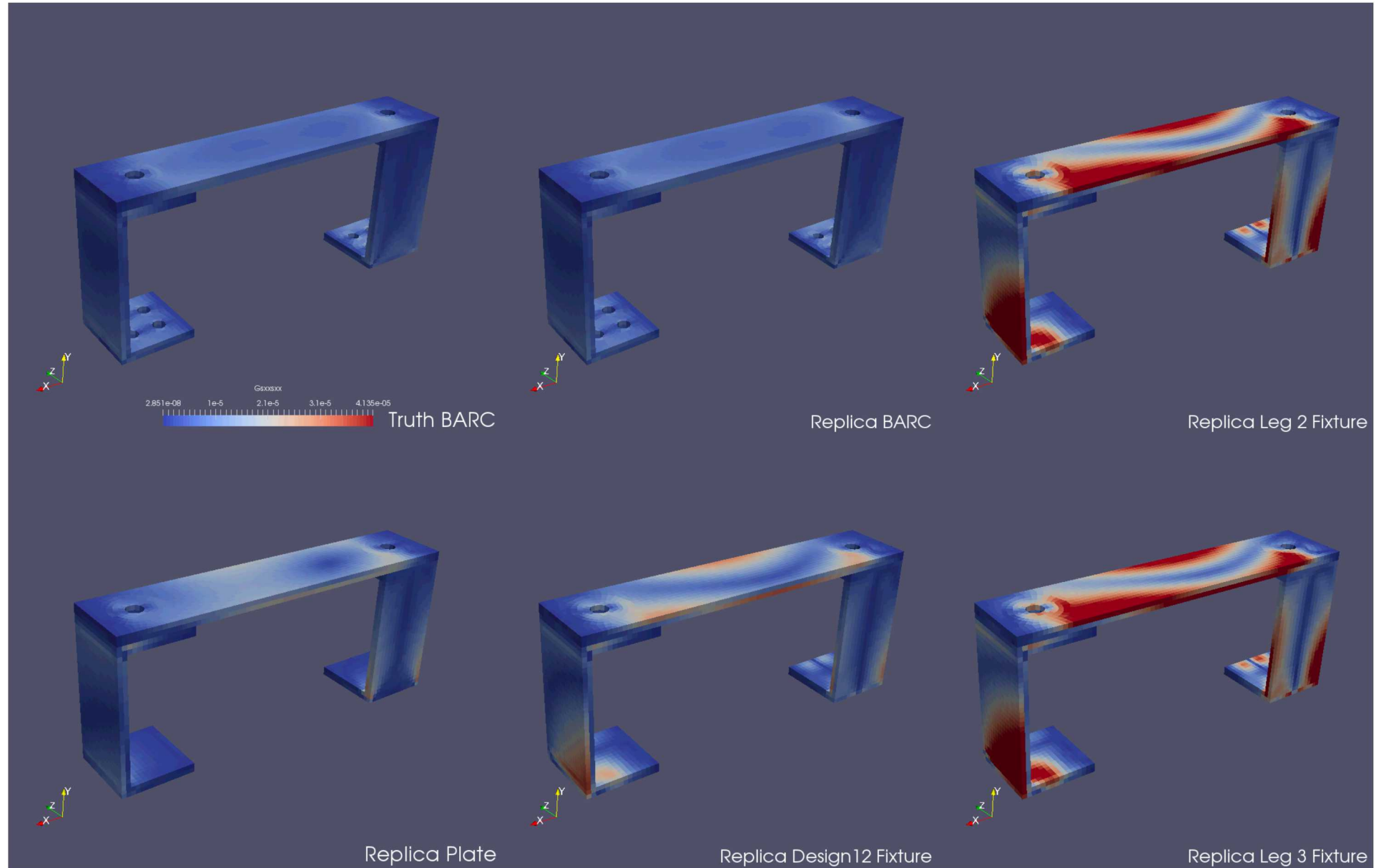


12 Strain CPSD Expansion

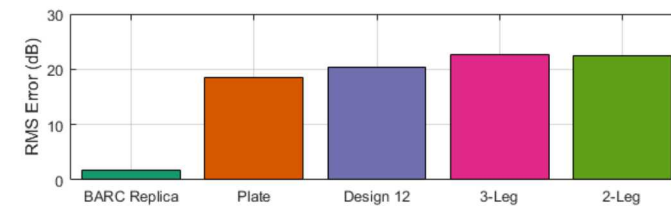
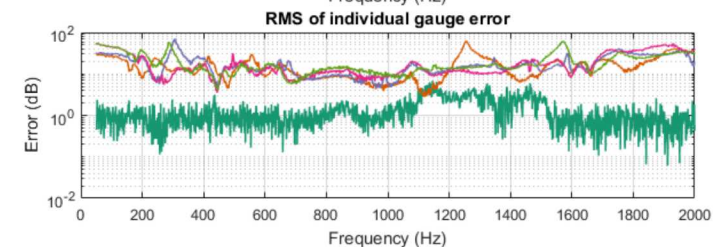
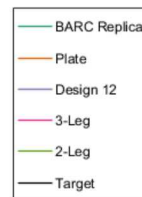
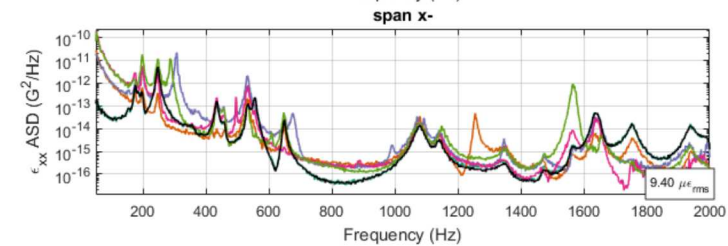
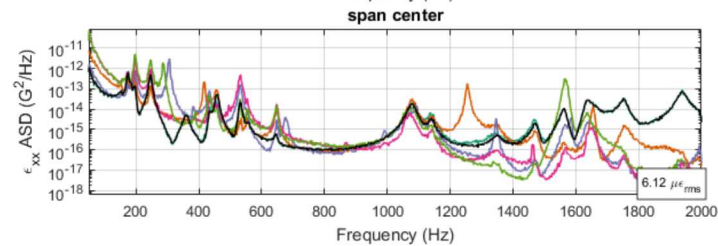
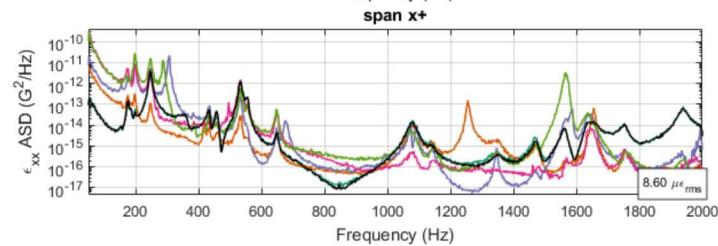
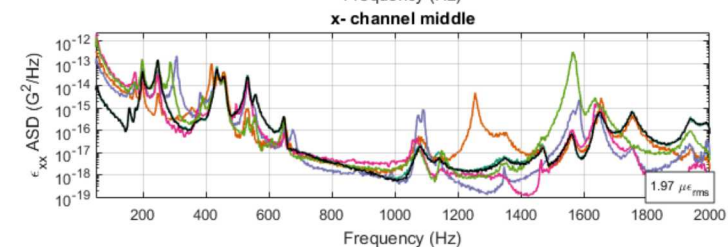
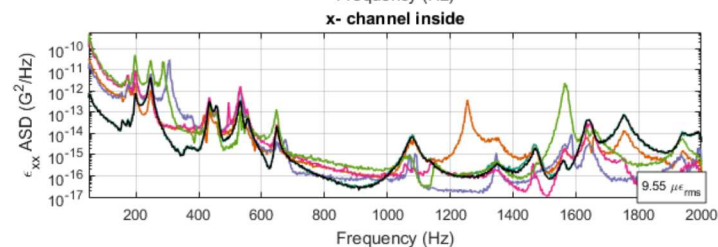
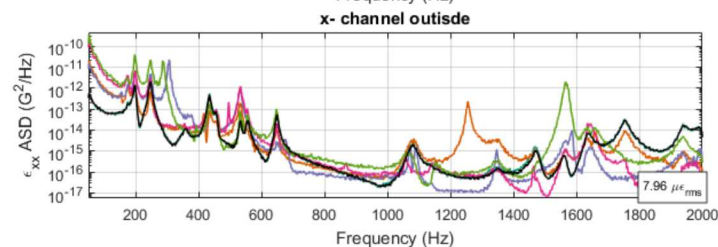
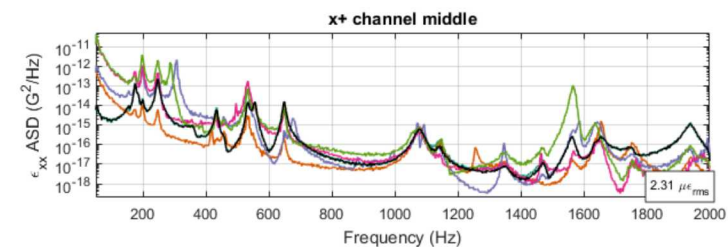
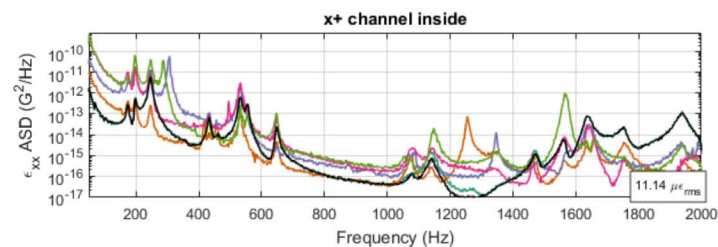
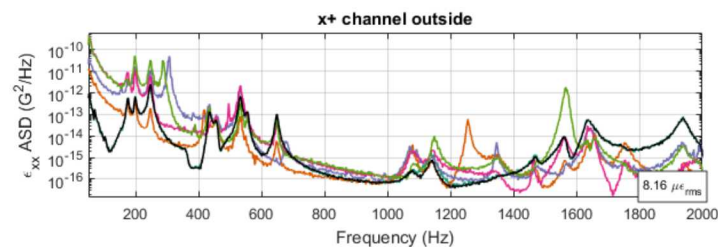
Autospectral densities (diagonals of the CPSD matrices) for the six strain components ϵ_{xx} , ϵ_{xy} , ϵ_{xz} , ϵ_{yy} , ϵ_{yz} , and ϵ_{zz} were computed for each element in the model for each frequency line of interest.

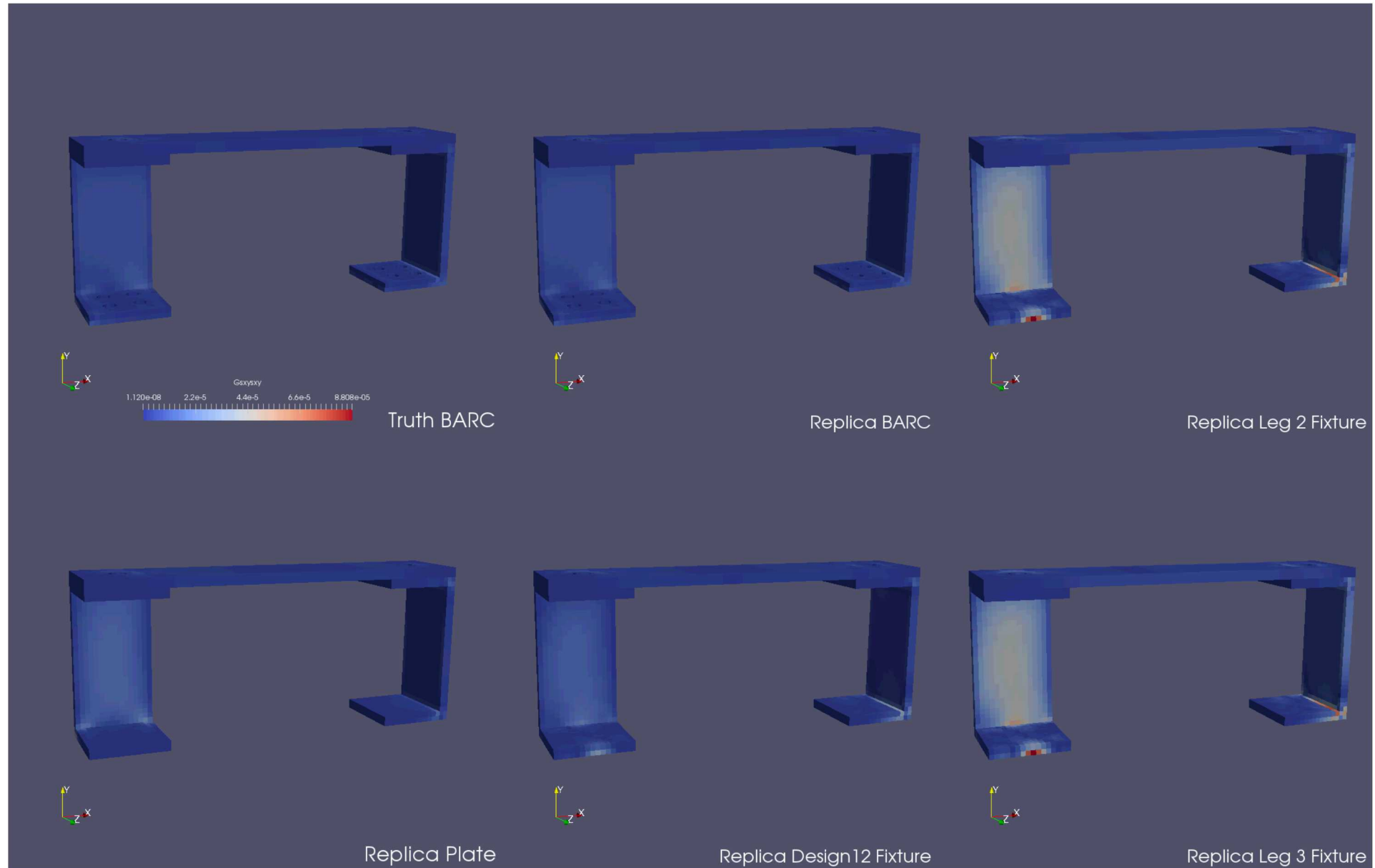
This was a lot of information to digest, so the RMS value of each strain component was computed as well to look for “hot spots” where we could investigate a handful of elements further.



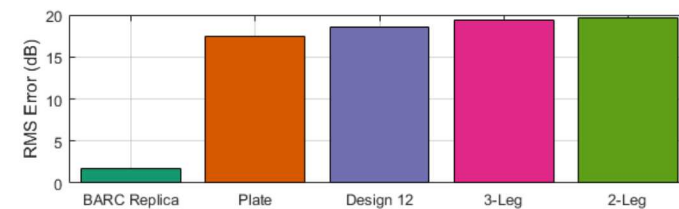
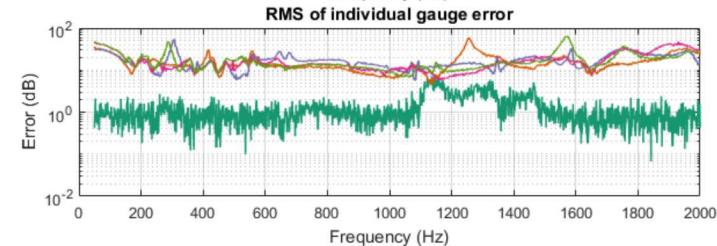
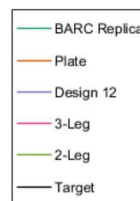
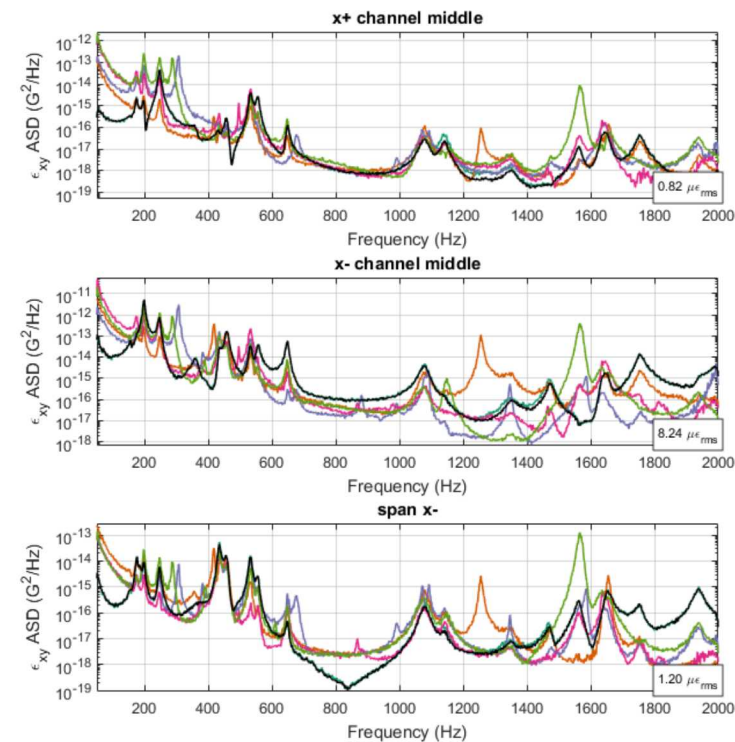
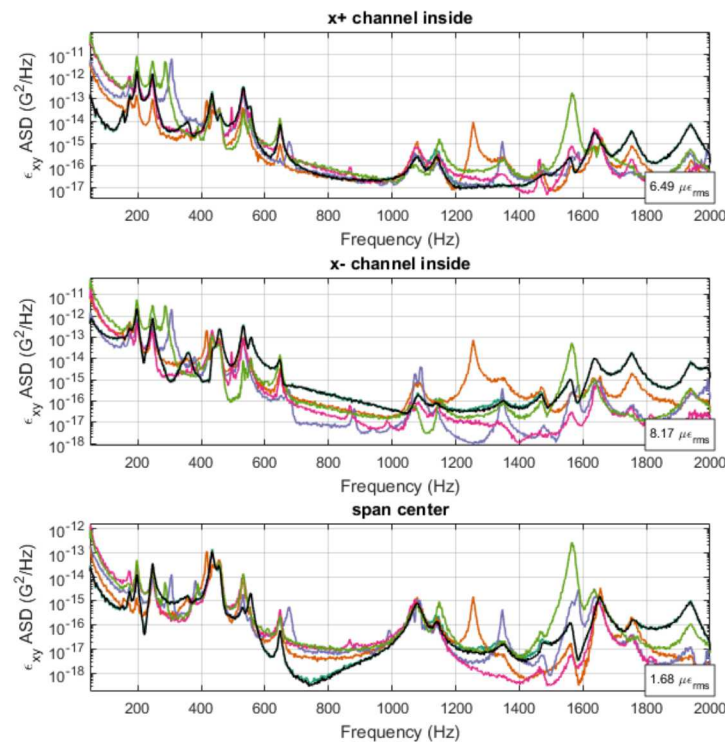
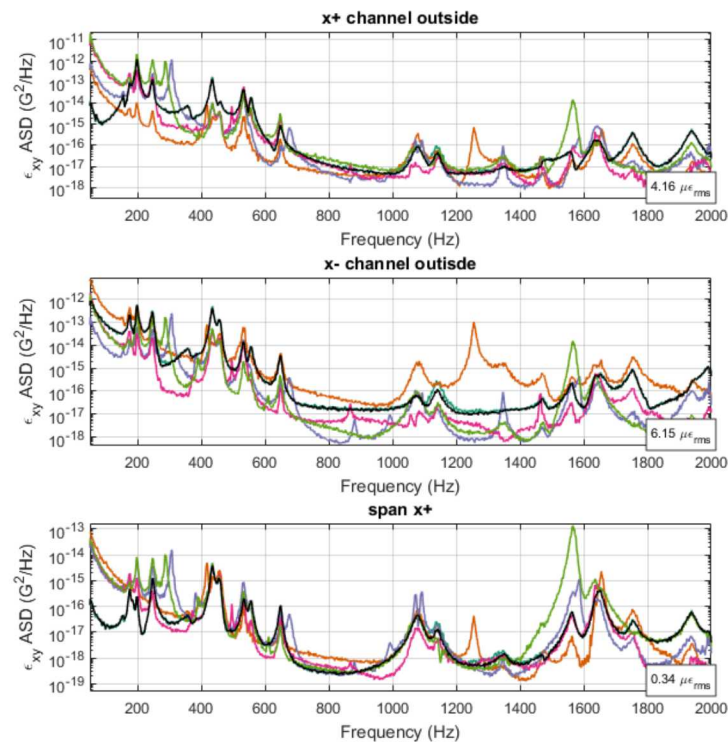


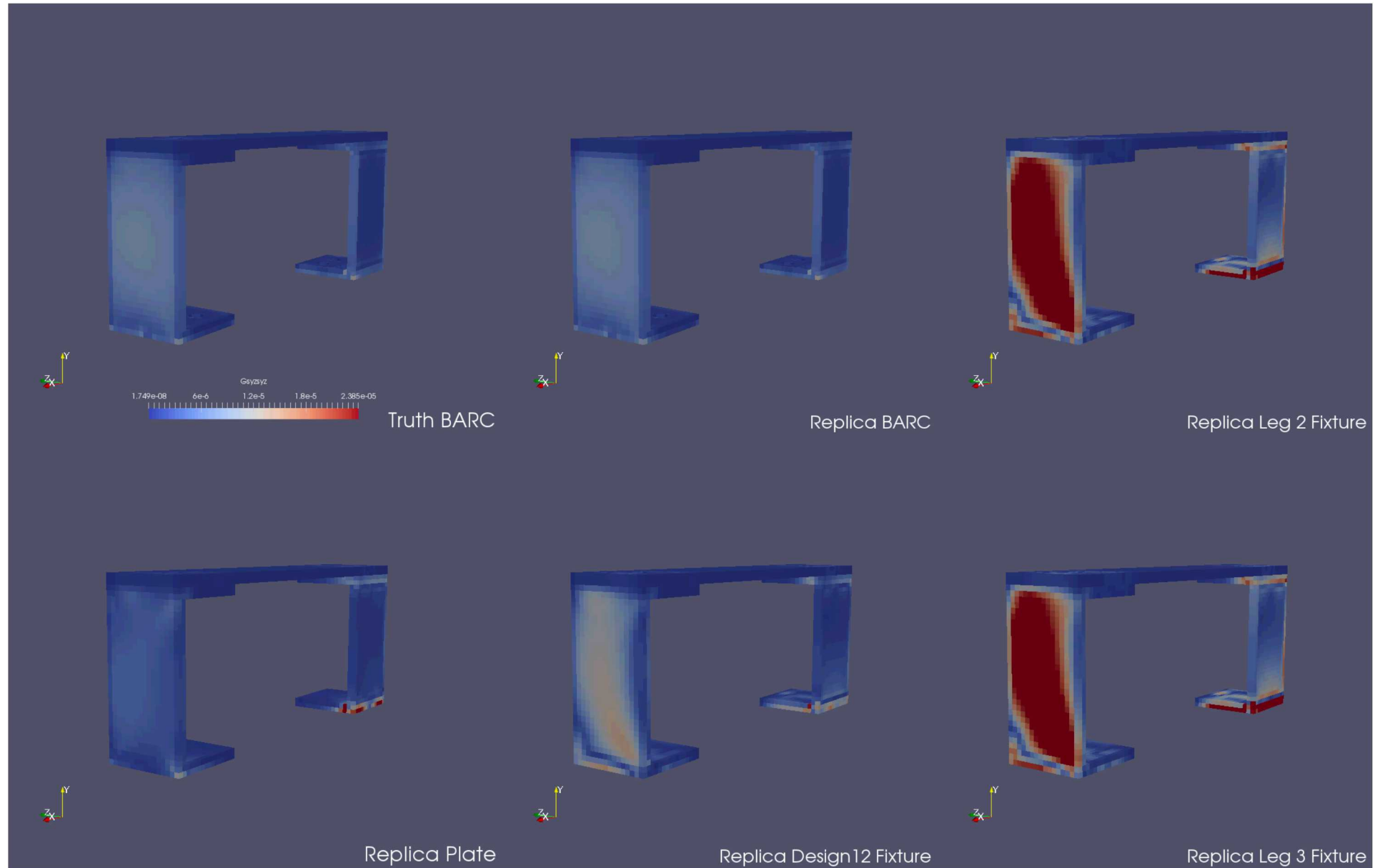
Strain Response at Local Elements: $G_{\epsilon_{xx}\epsilon_{xx}}$



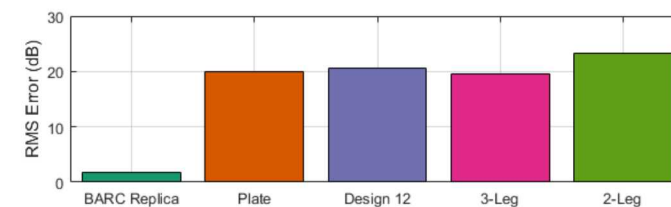
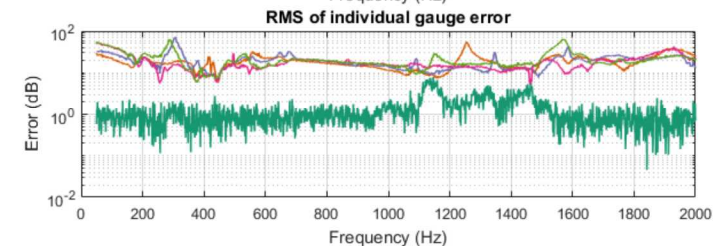
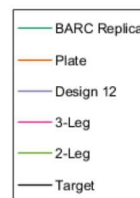
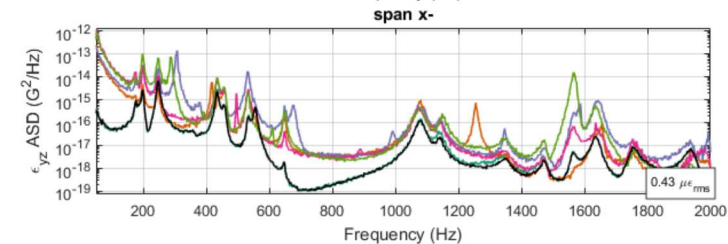
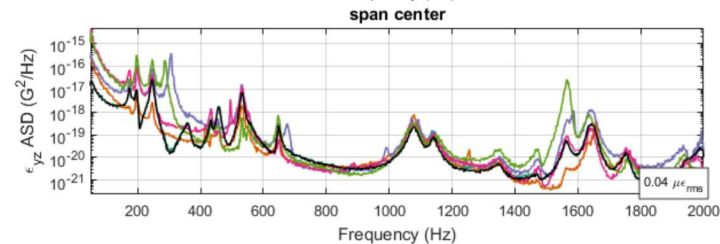
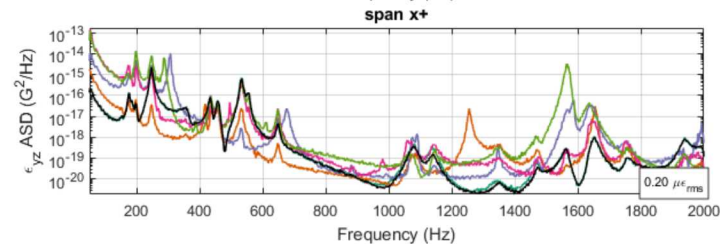
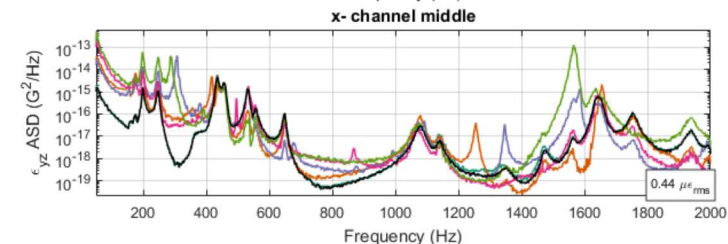
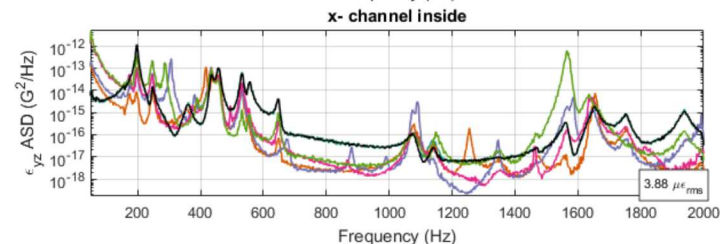
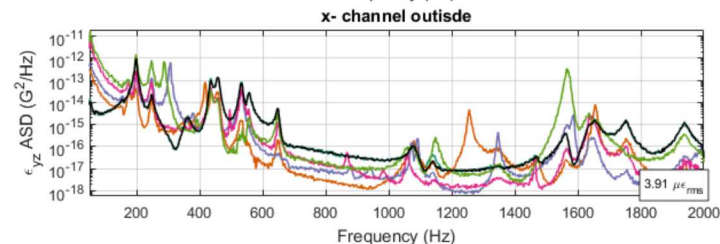
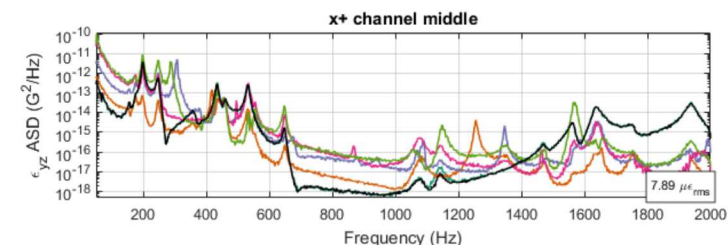
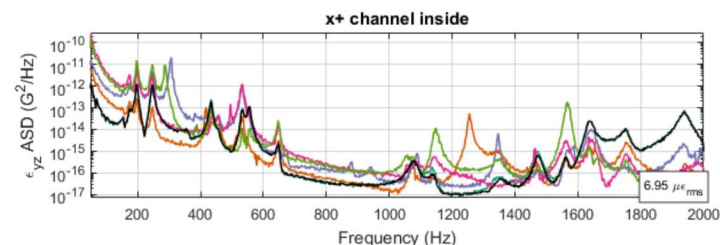
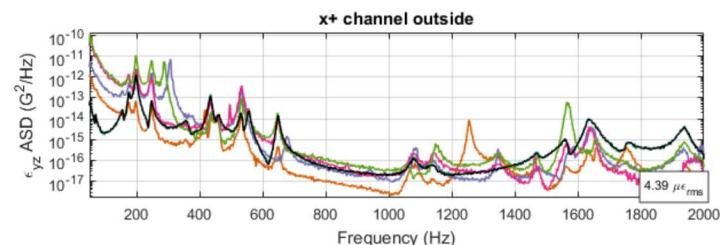


Strain Response at Local Elements: $G_{\epsilon_{xy}\epsilon_{xy}}$





Strain Response at Local Elements: $G_{\epsilon_{yz}\epsilon_{yz}}$



Finite element expansion seems to be a good way to compute strains, assuming sufficient instrumentation to perform the expansion.

- Can we control directly to strains on the component using a custom controller?

BARC Replica test matched truth test very well, strains also matched well

If your fixture has not been optimized for dynamics of the part, you are likely better off with a rigid fixture.

- Modes of the structure on the fixture that do not correspond to modes in the truth environment seem to be the largest source of error between the target and achieved responses
- Ideally, the modes of the fixture and component should match the modes in the service environment

Vibration Cube Test Planning

Use Finite Element Model to attempt to place shakers in some optimal sense.

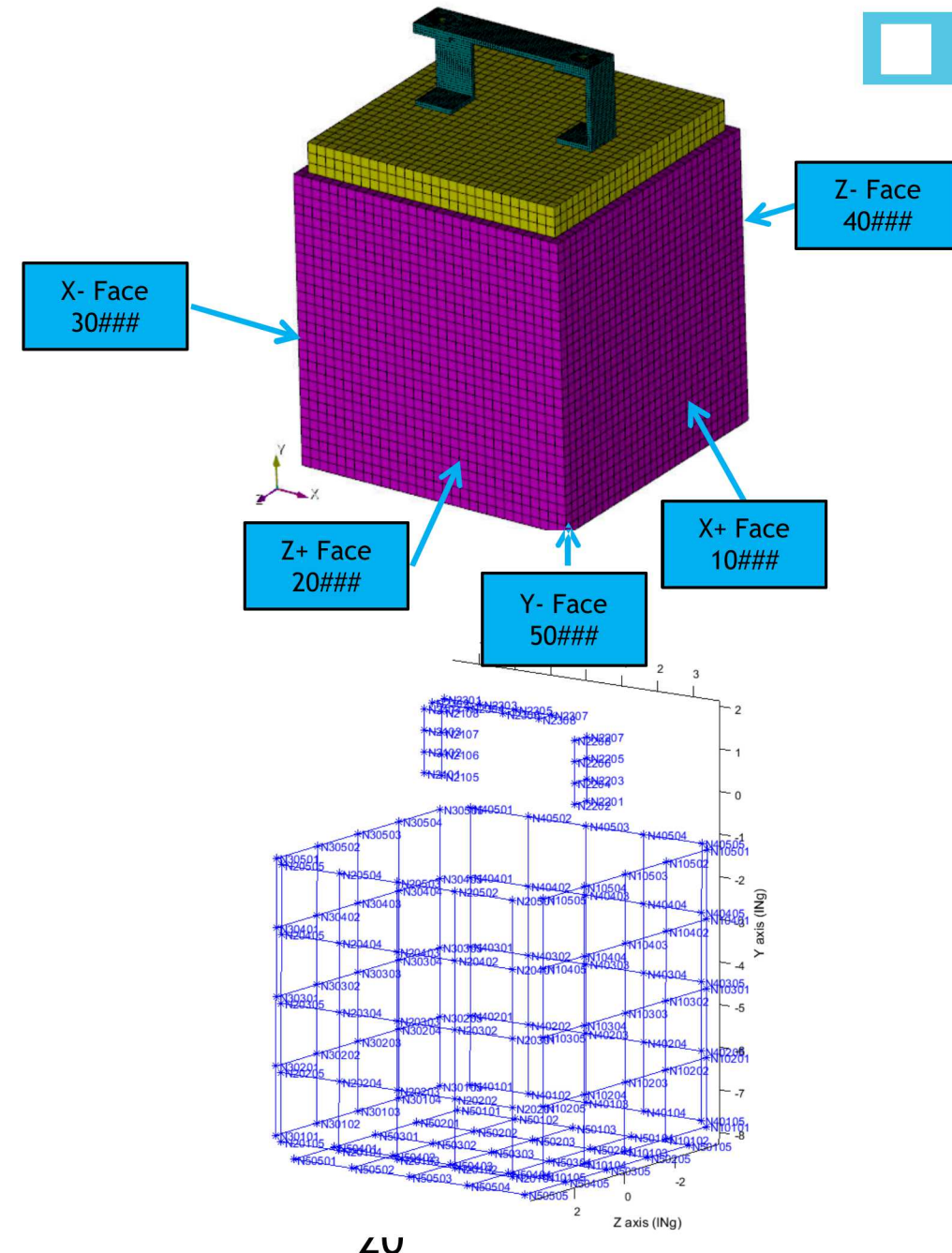
To investigate shaker locations, a coarsened grid was made with 5x5 candidate shaker locations on each face of the cube.

To make problem smaller, consider the cube rigid and only consider inputs on the 30000, 40000, and 50000 faces. (317,750 combinations to 67,525 combinations)

Solving the control problem at all frequency lines (1,951) would take approximately 2 days to run all 67k combinations, instead coarsened frequency lines to 88 frequencies in unique parts of the spectrum (peaks, valleys, etc.)

Design considerations in actual tests are:

- Controllability: Can the shakers force the test article to do what we want it to do?
- Force Required: Can we realistically achieve the force levels required to get the test article to do what we want it to do?



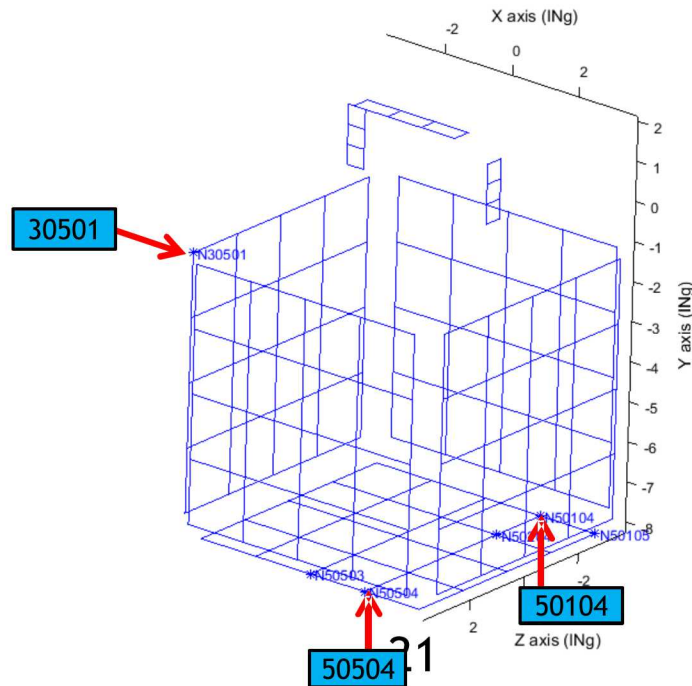
Vibration Cube Test Planning

Results from the control simulation were ranked from 1 (best) to 67,525 (worst) for both dB error on the response gauges and RMS force required.

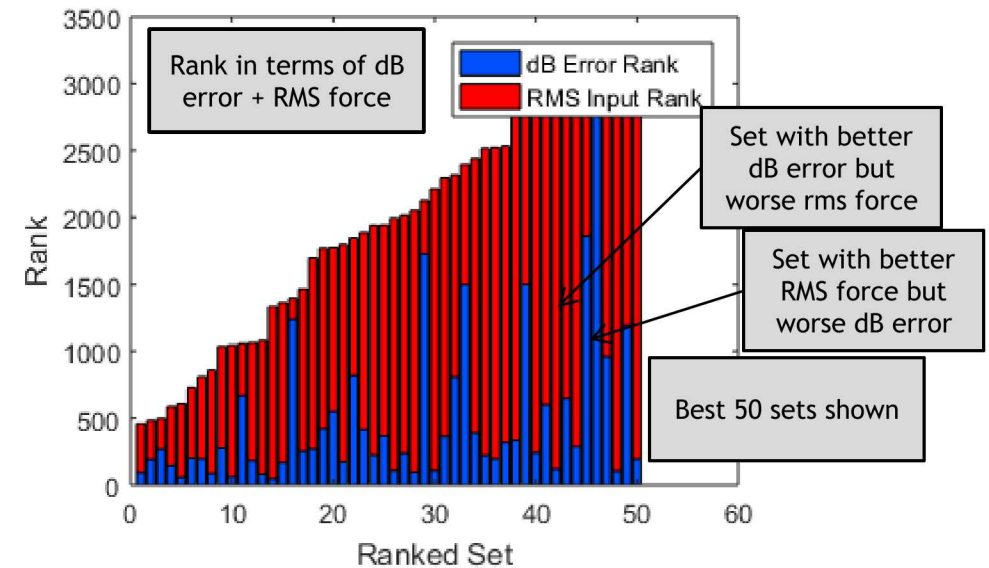
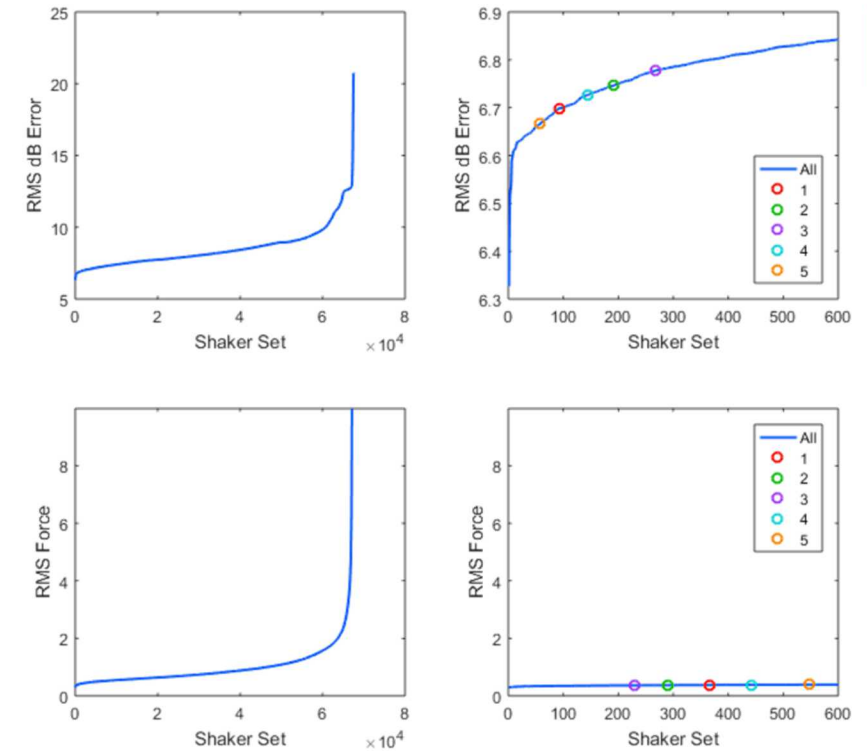
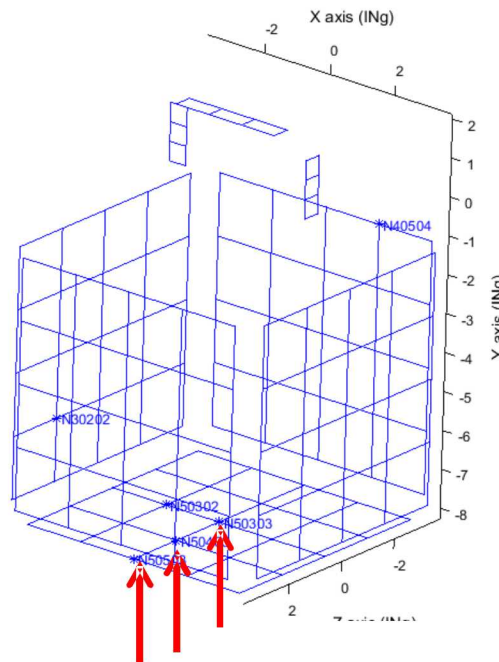
The best set for dB error was not the same set for smallest force.

Summing the ranks for each shaker configuration and then ranking by this new metric gave a good compromise between control error and force required.

Best Location



Worst Location

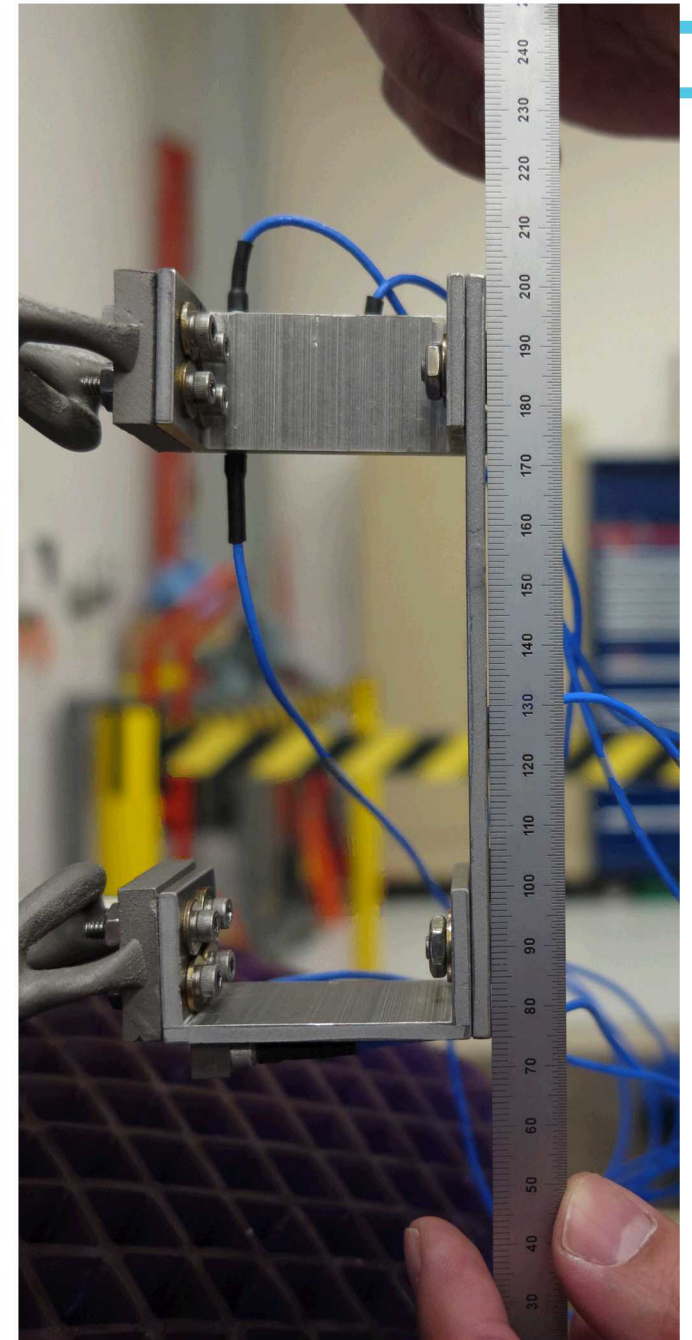
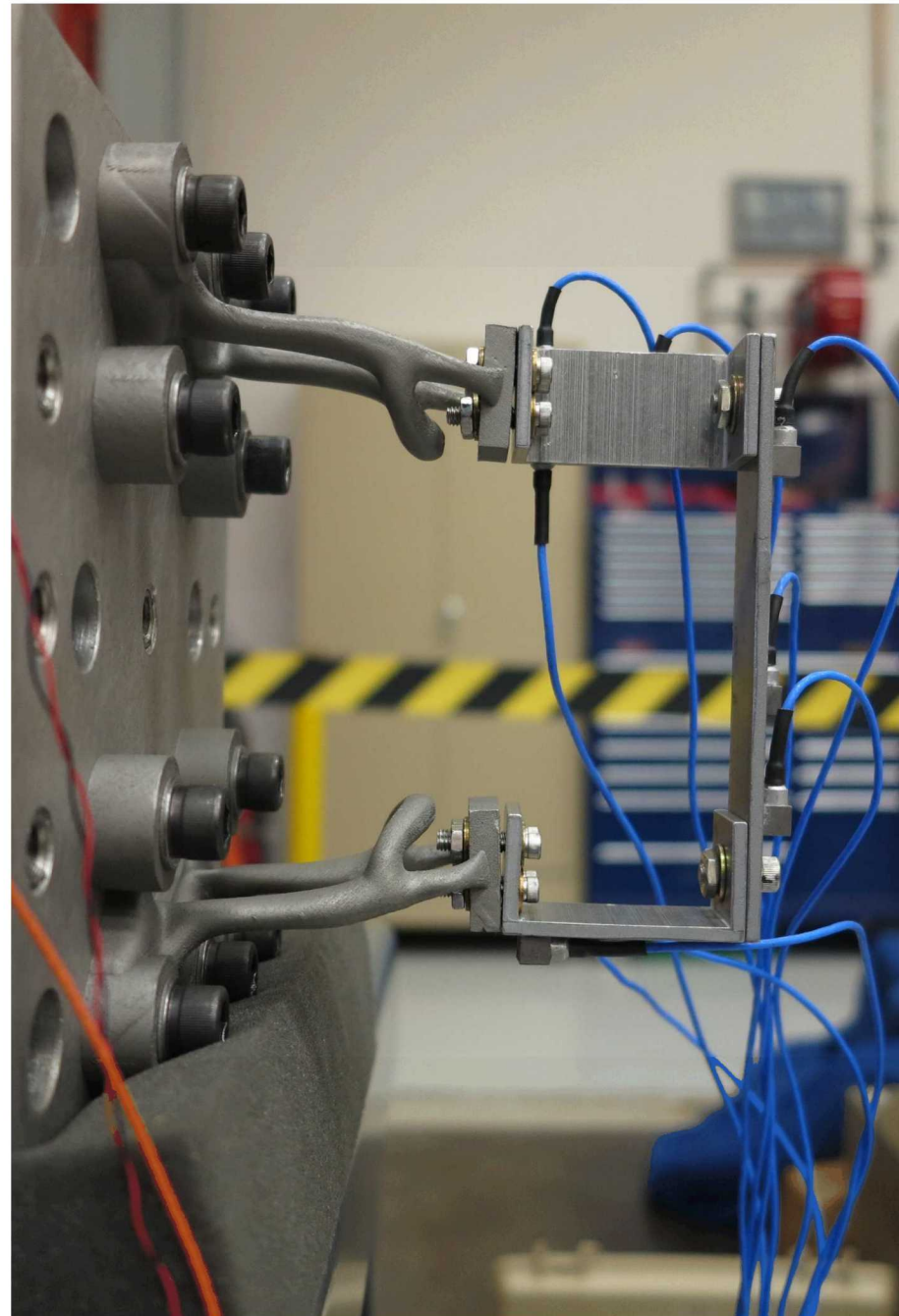


Aside: AM Tolerances

The tolerances on the AM fixtures were not great.

There were visible gaps when the structure was sitting on the fixture but not torqued.

Torquing the bolts resulted in deformation in the beam of the removable component.



Practical Considerations of Expanding to a FEM CPSD Matrix

For the SEREP expansion of a CPSD matrix, the resulting matrix is a complex, $n \times n$ matrix, where n is the number of degrees of freedom in the FEM. Even for relatively small models such as BARC, the $\sim 90,000 \times 90,000$ complex matrix will take up 140 GB of memory and therefore cannot be processed in-core.

Since only the diagonal values are of interest at the end, we can examine the linear algebra to reduce the math to a more manageable size.

Starting with the last equation, we make substitutions so the arithmetic is less cluttered.

$$\underbrace{\mathbf{G}_{xx,fem}(\omega)}_{\mathbf{B}} = \underbrace{\Phi_{fem} \Phi_{exp}^+}_{\mathbf{A}} \underbrace{\frac{\mathbf{G}_{\ddot{x}\ddot{x}}(\omega)}{\omega^4} (\Phi_{exp}^+)^*}_{\mathbf{Q}} \underbrace{\Phi_{fem}^*}_{\mathbf{A}^T}$$

Converting to Summation Notation, where m is the number of modes used in the expansion, we find that the diagonals B_{ii} can be expressed as:

$$B_{ii} = \sum_{l=1}^m \left(\sum_{k=1}^m A_{ik} Q_{kl} \right) A_{il}$$

This can be written in vectorized, efficient MATLAB code as `sum((A*Q).*A,2)` where no matrix bigger than $n \times m$ is ever needed. With $n \gg m$, this is a drastic savings in memory.

Developing the Proper Expansion Shape Sets

Previous experience in SEREP has revealed that the set of modes $\Phi_{fem} \Phi_{exp}^+$ used in the expansion can have a significant impact on the results. If not enough modes are used, real portions of the response can be filtered out. However, if too many modes are used, the expansion can also be erroneous and portions of the model away from the points being fit can behave unrealistically.

To investigate the proper basis for the SEREP expansion, we can perform the expansion on the modes within the bandwidth.

- Mode shapes tend to have intuitive, smooth shapes, so it is more obvious when overfitting is occurring.
- The modes in the bandwidth should form a relatively good basis for the responses in the MIMO control test, so a good set of FEM modes to expand the experimental modes in the bandwidth of interest should also form a good basis for expanding the results of the MIMO control test.

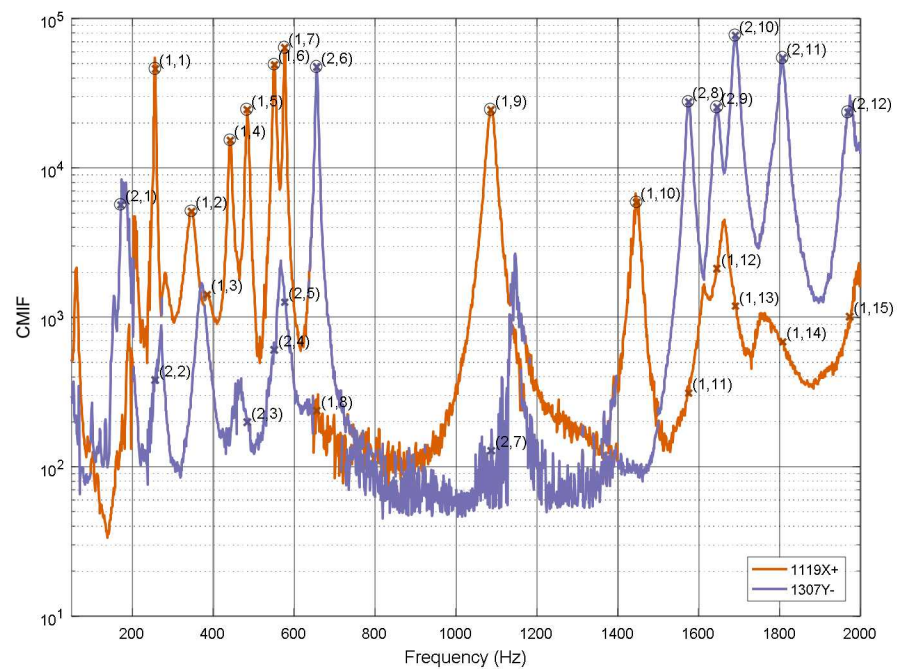
To fit modes, a force-acceleration FRF needed to first be computed, as opposed to the voltage-acceleration FRF matrices that were being used for the control transfer functions.

Modes were fit to these FRFs using the SMAC algorithm.

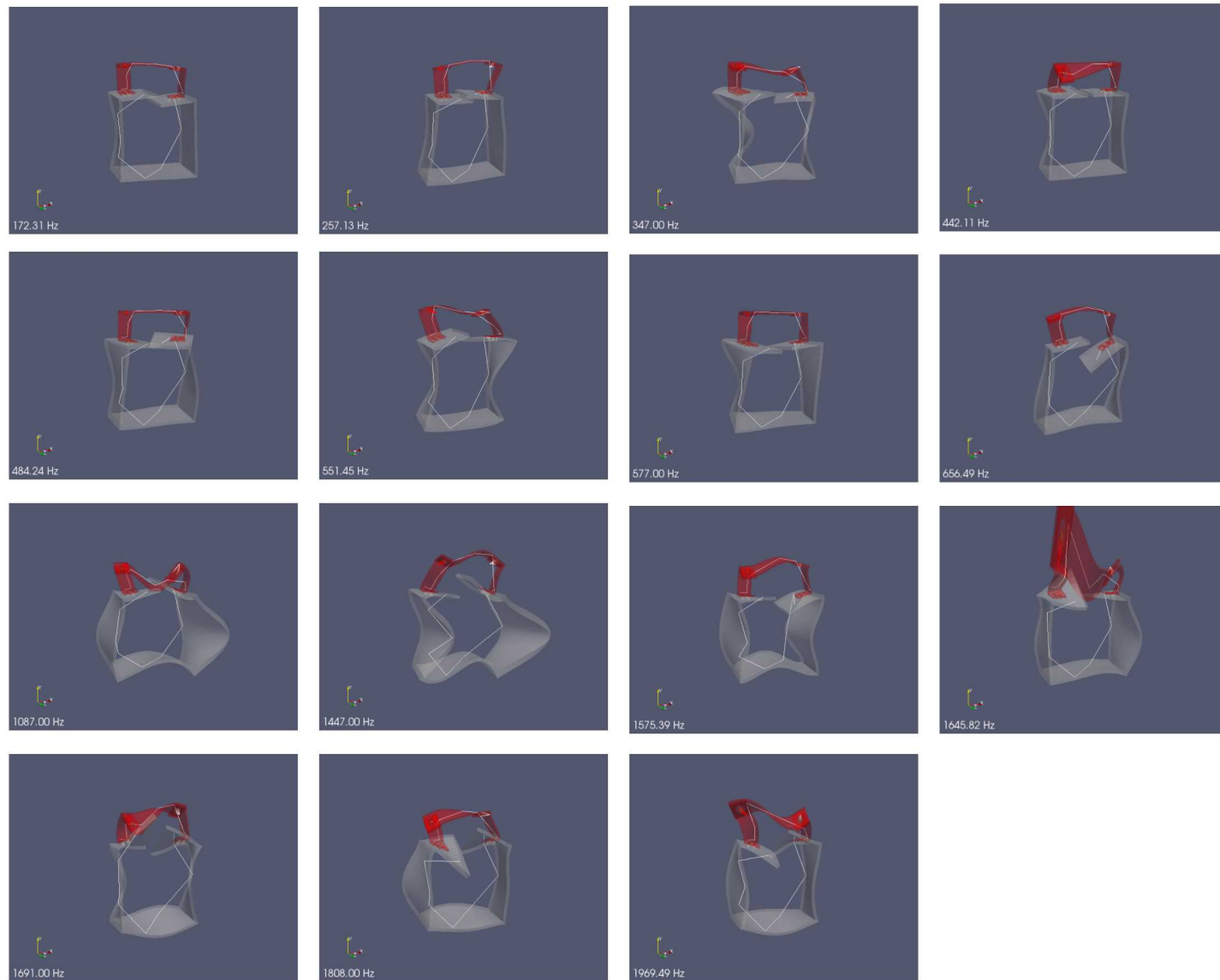
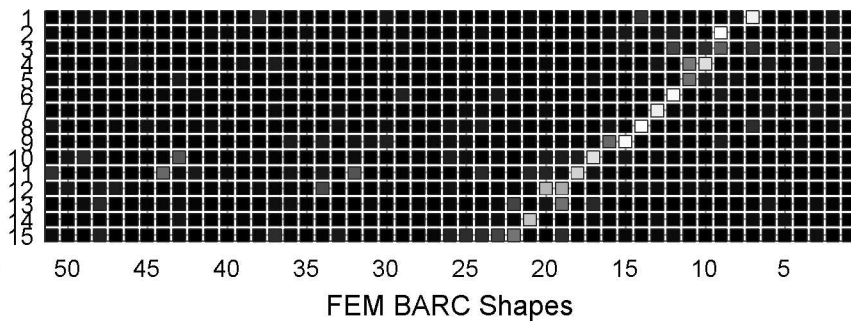
A modal assurance criterion matrix was formed between the experimental shapes and the FEM mode shapes reduced to the experimental degrees of freedom to determine which modes were important in the expansion.

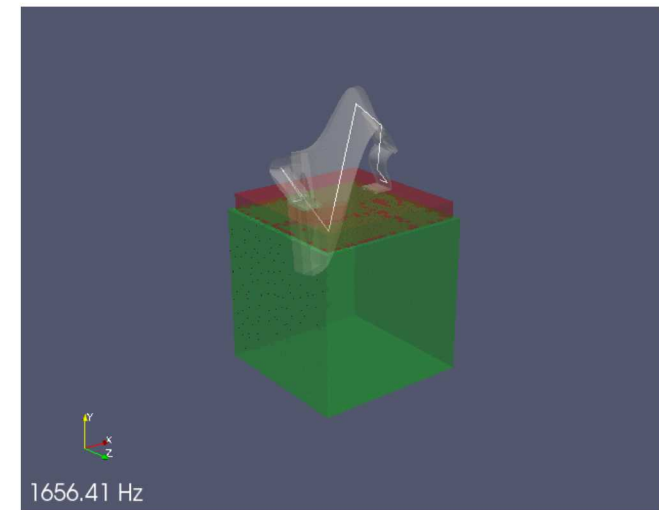
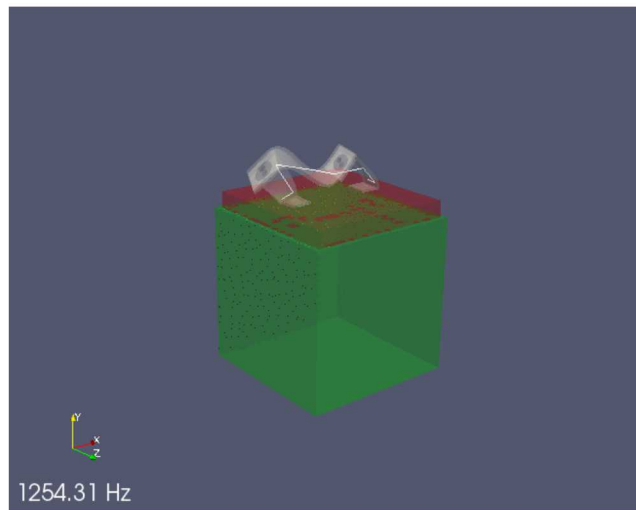
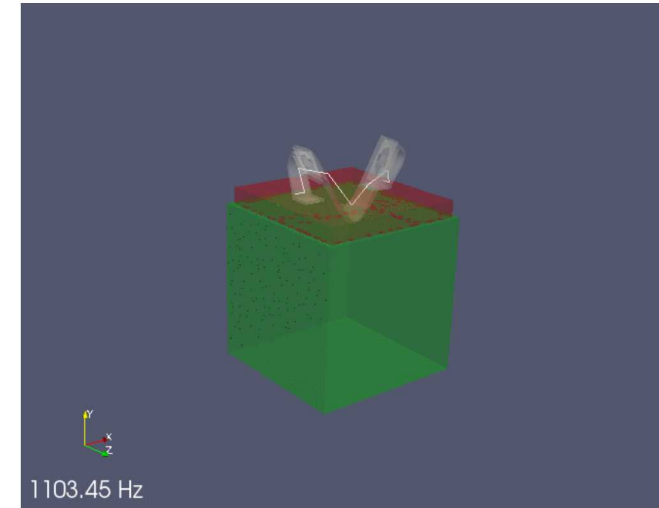
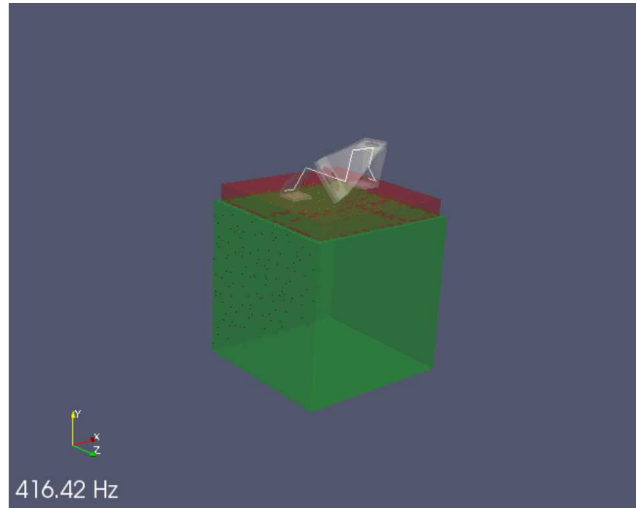
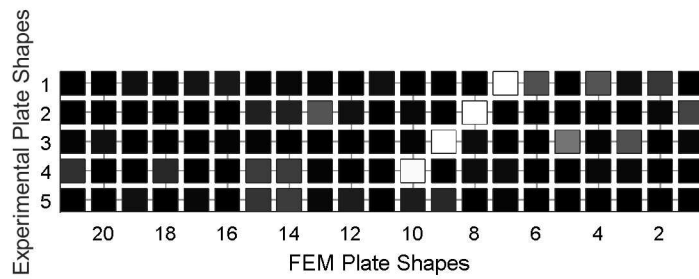
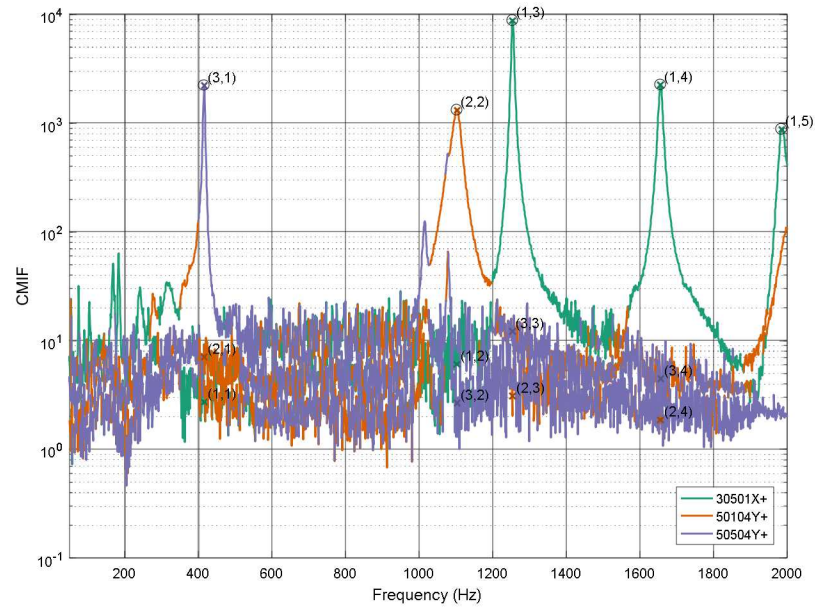
Experimental shapes were plotted on top of expanded shapes to ensure the expansion was not distorting the results.

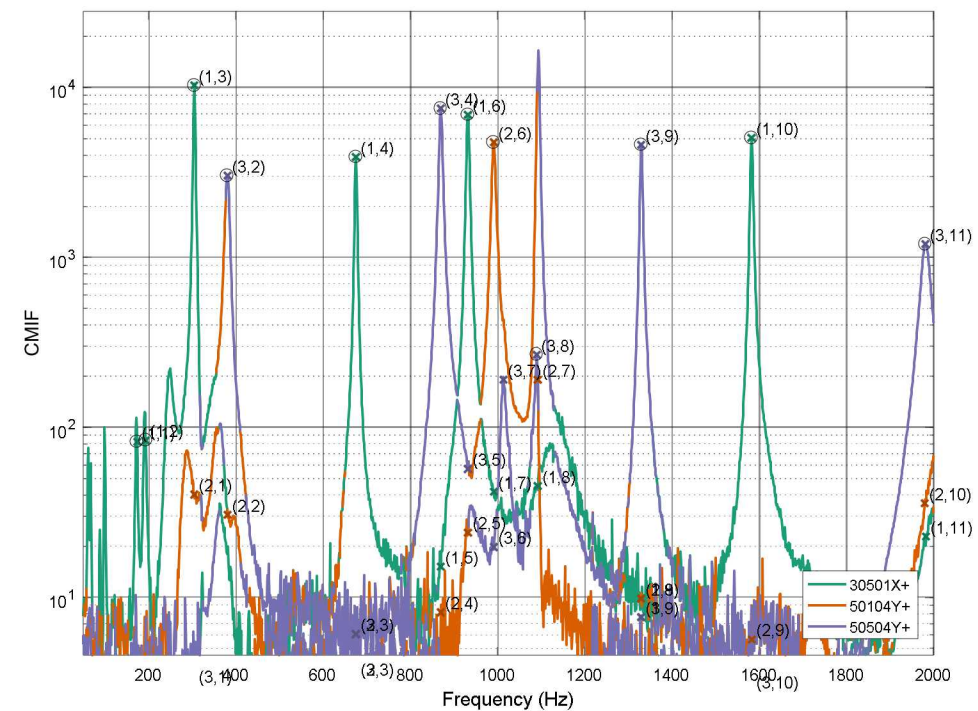
Truth/Replica Shape Expansion



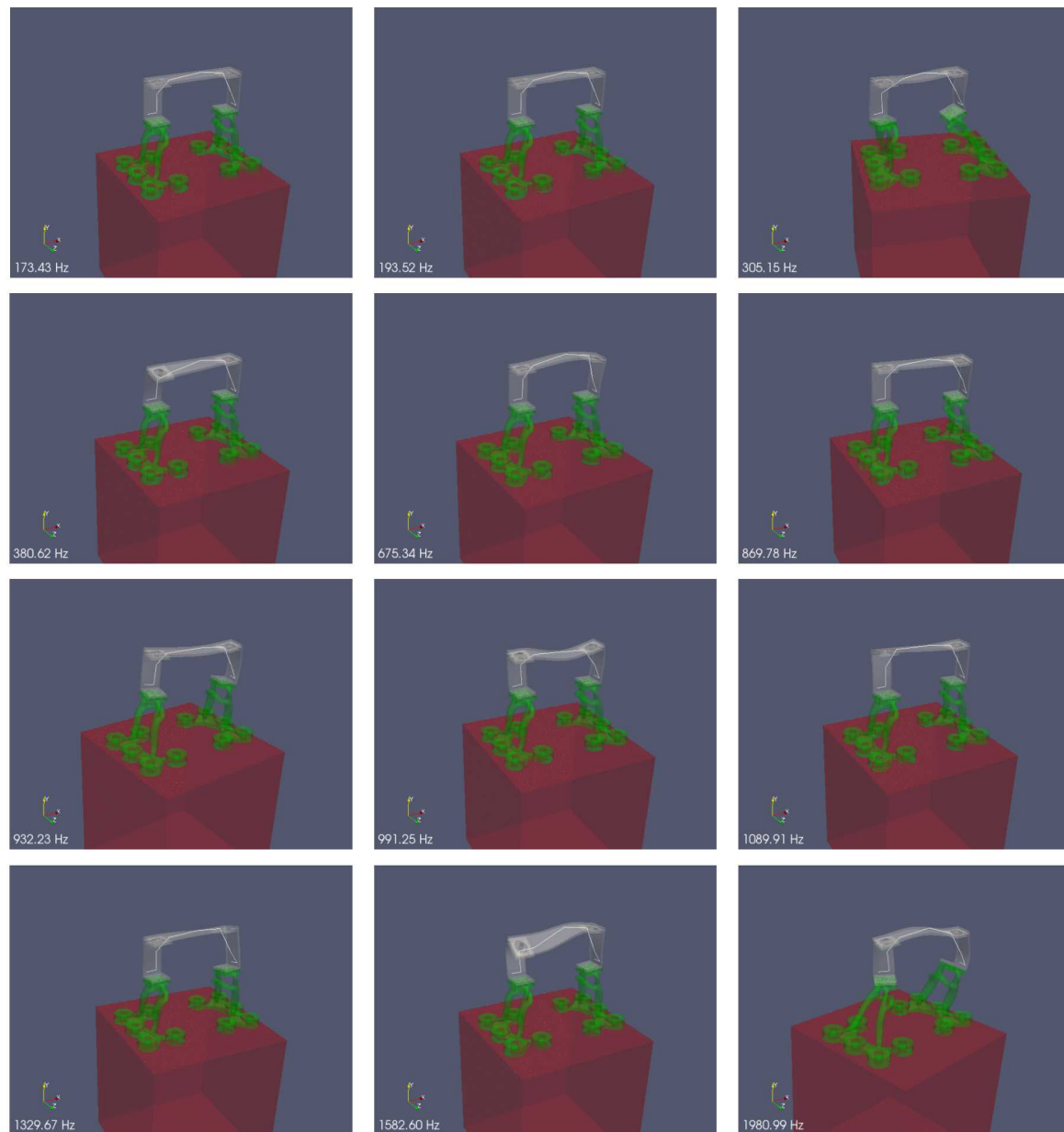
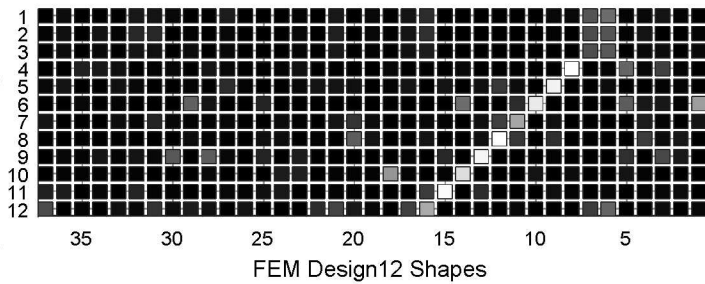
Experimental BARC Shapes

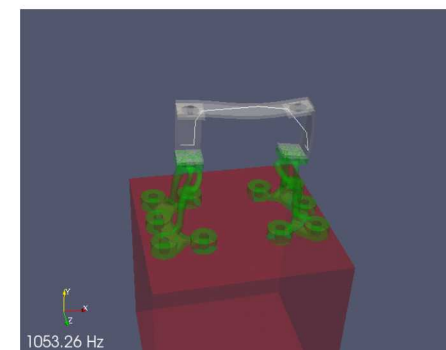
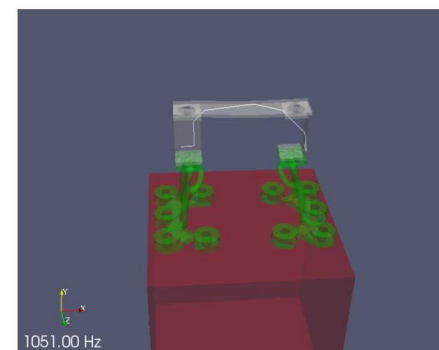
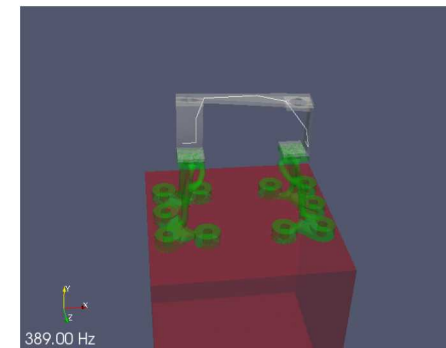
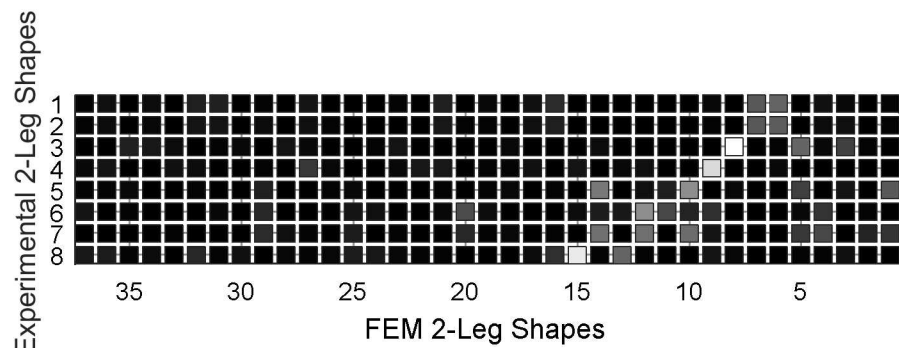
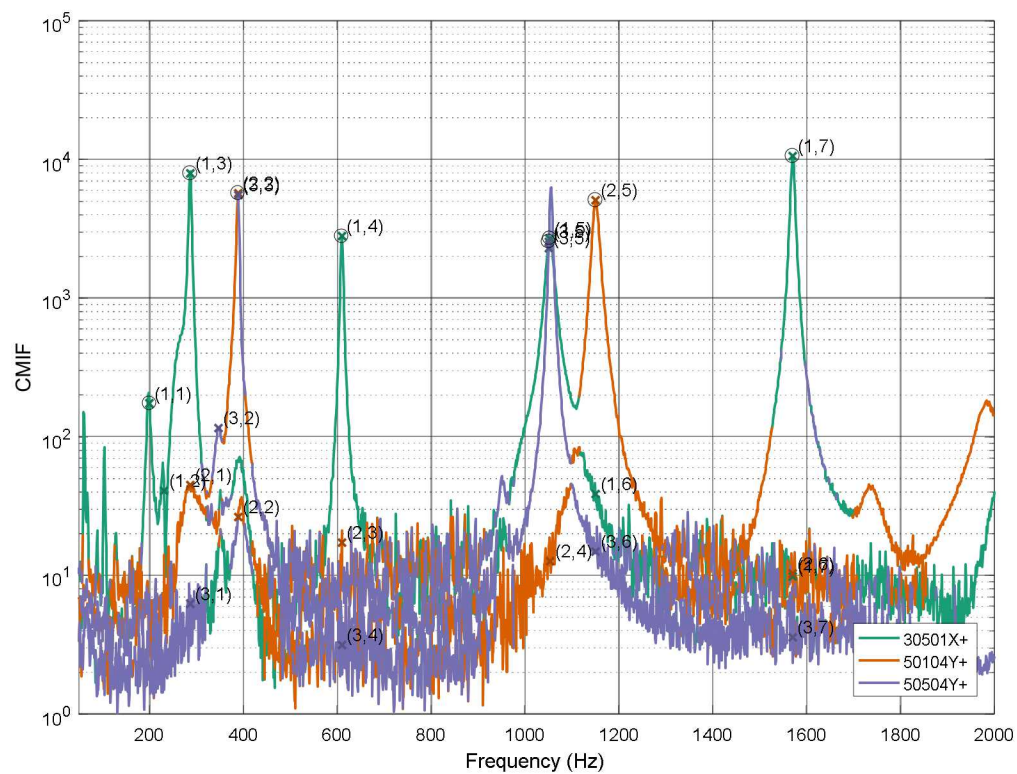


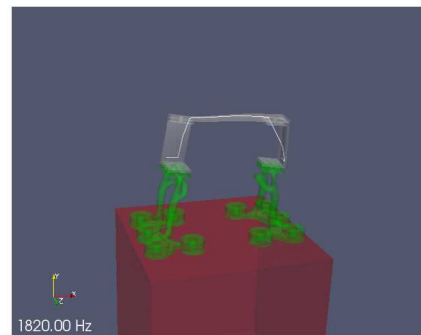
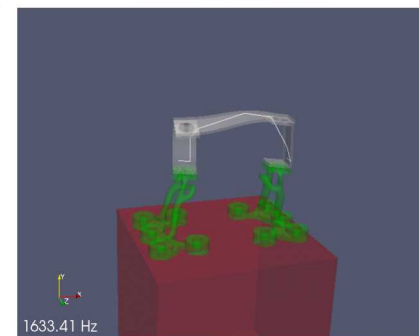
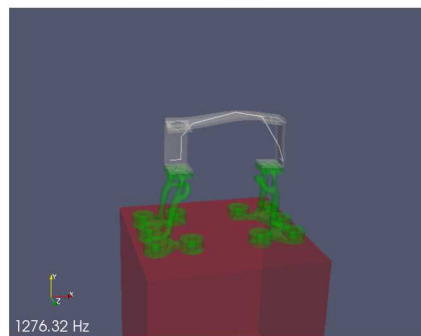
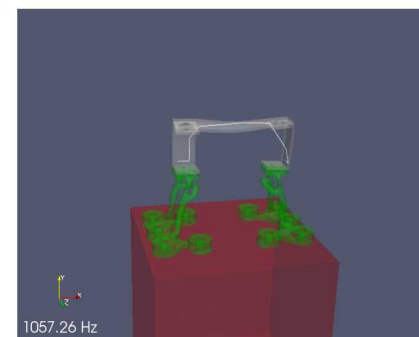
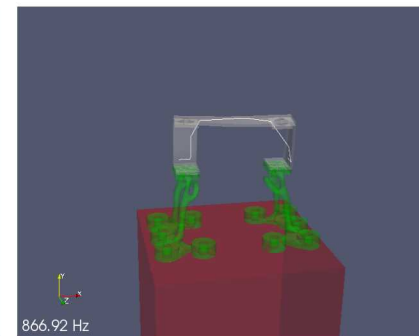
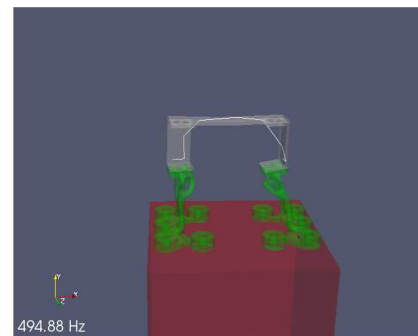
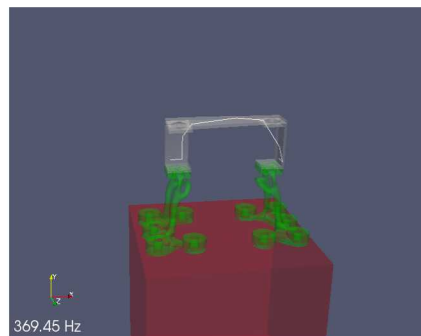
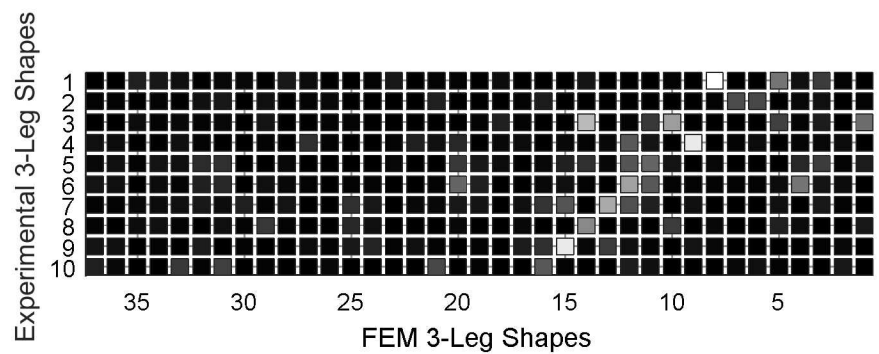
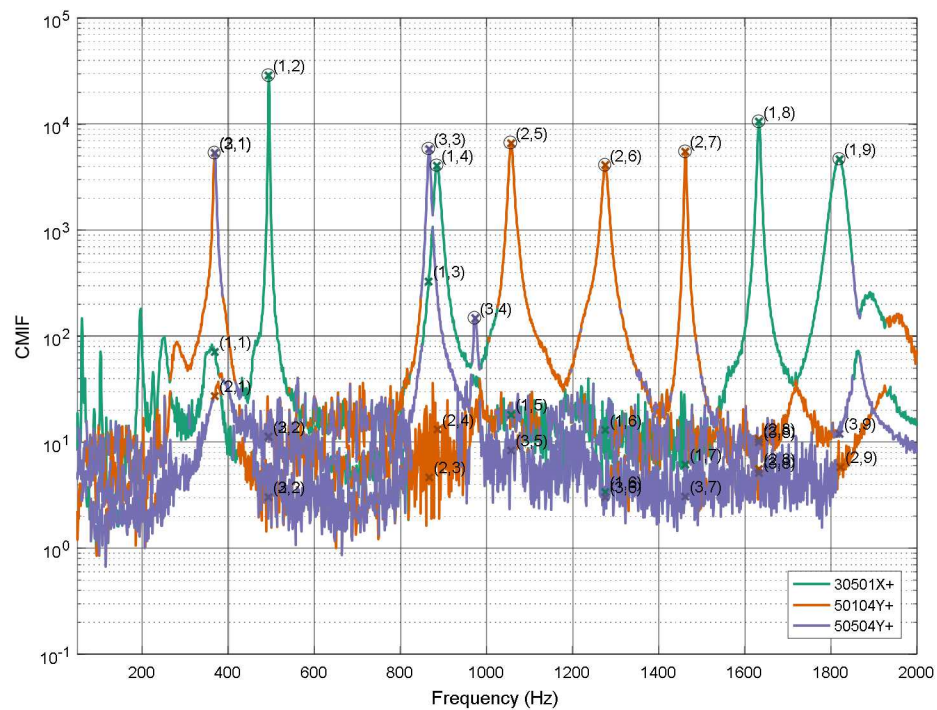


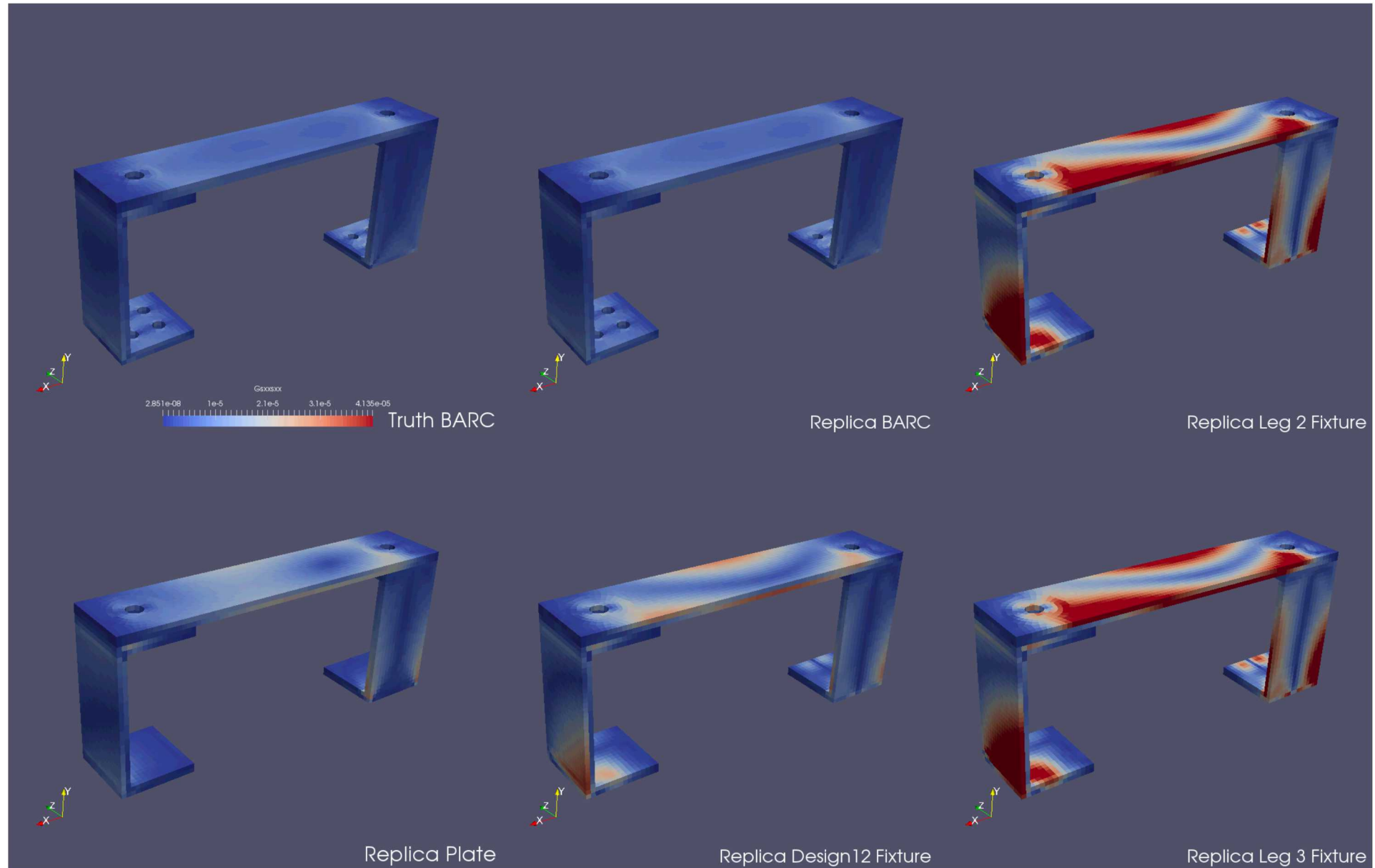


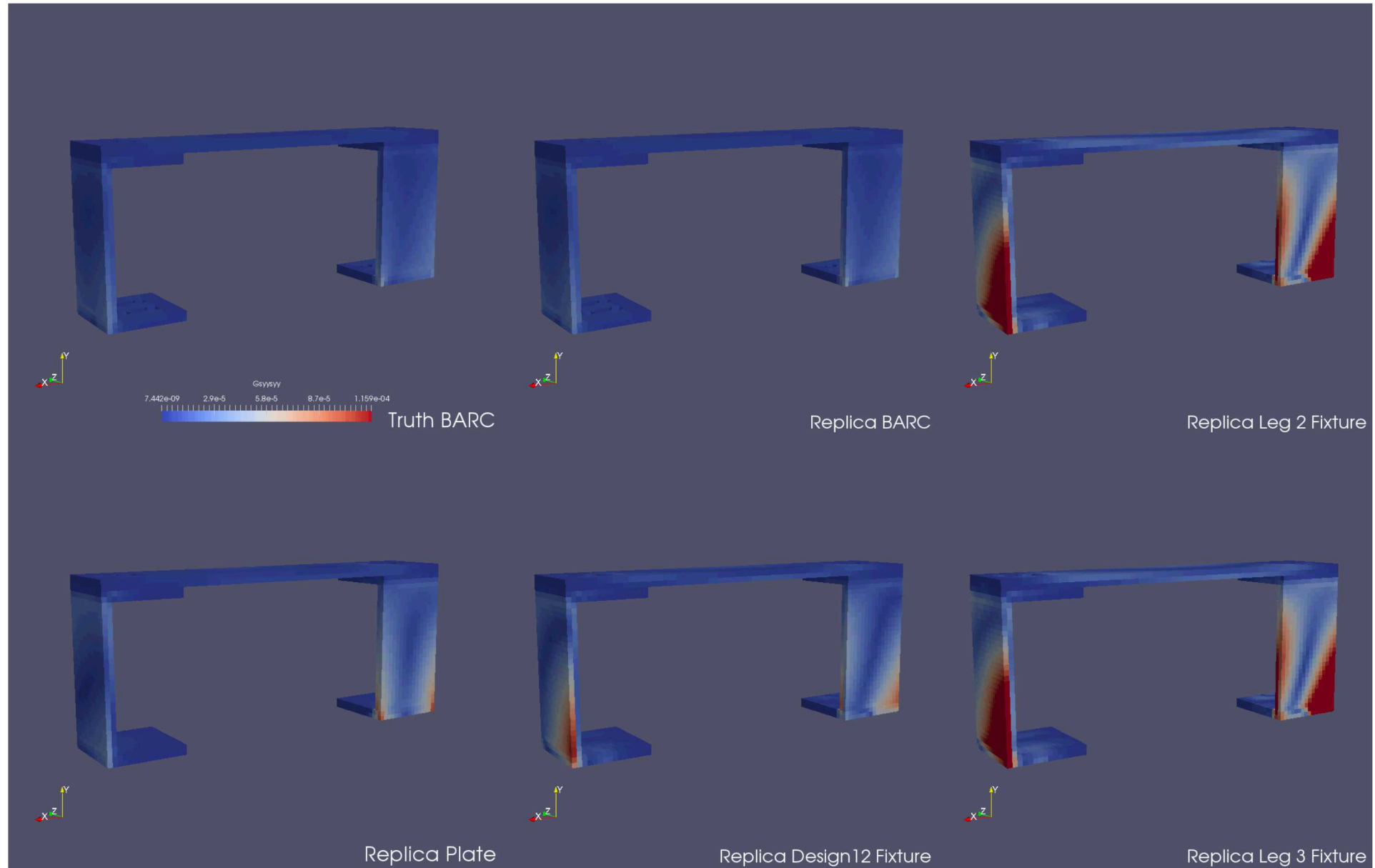
Experimental Design12 Shapes

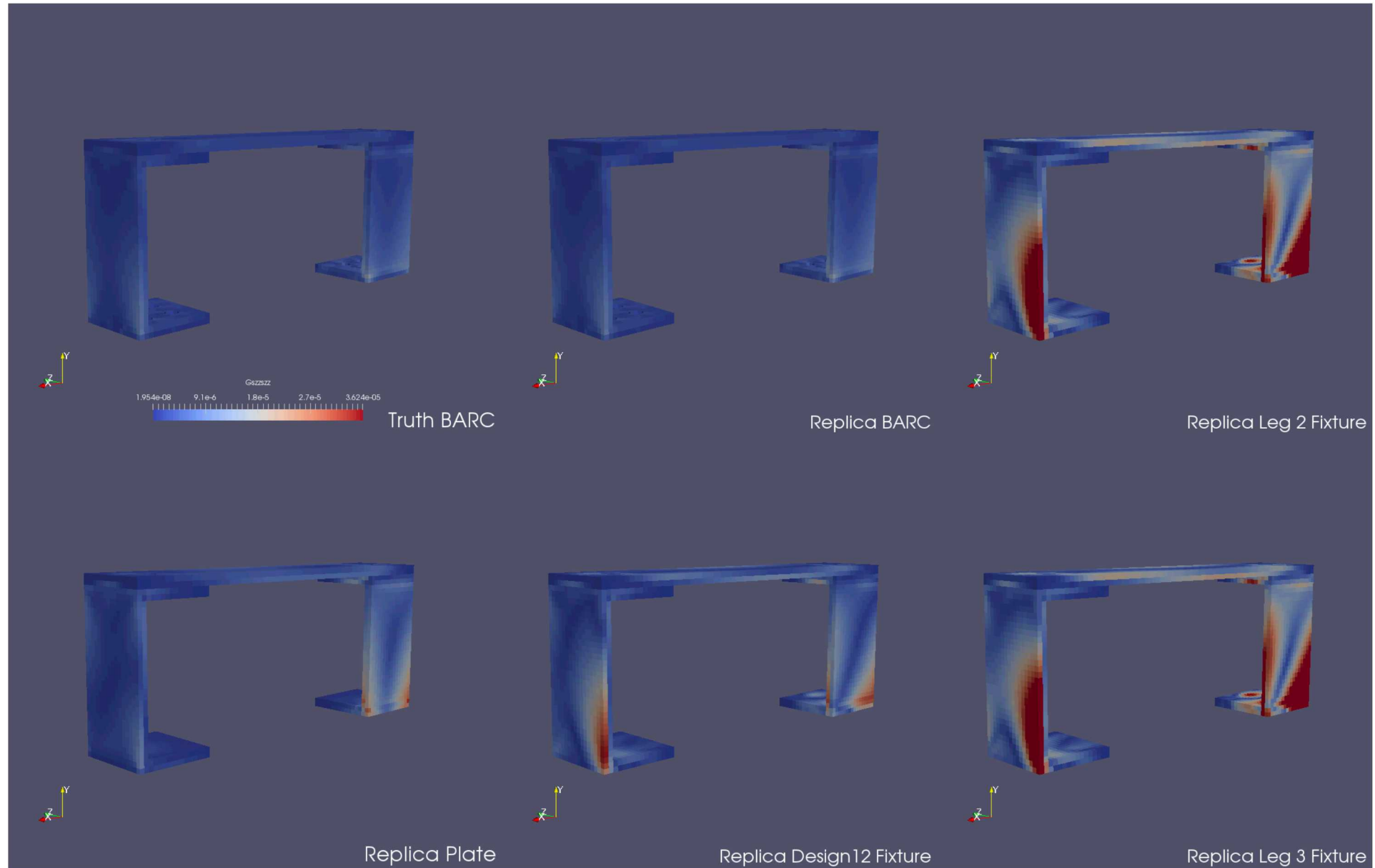


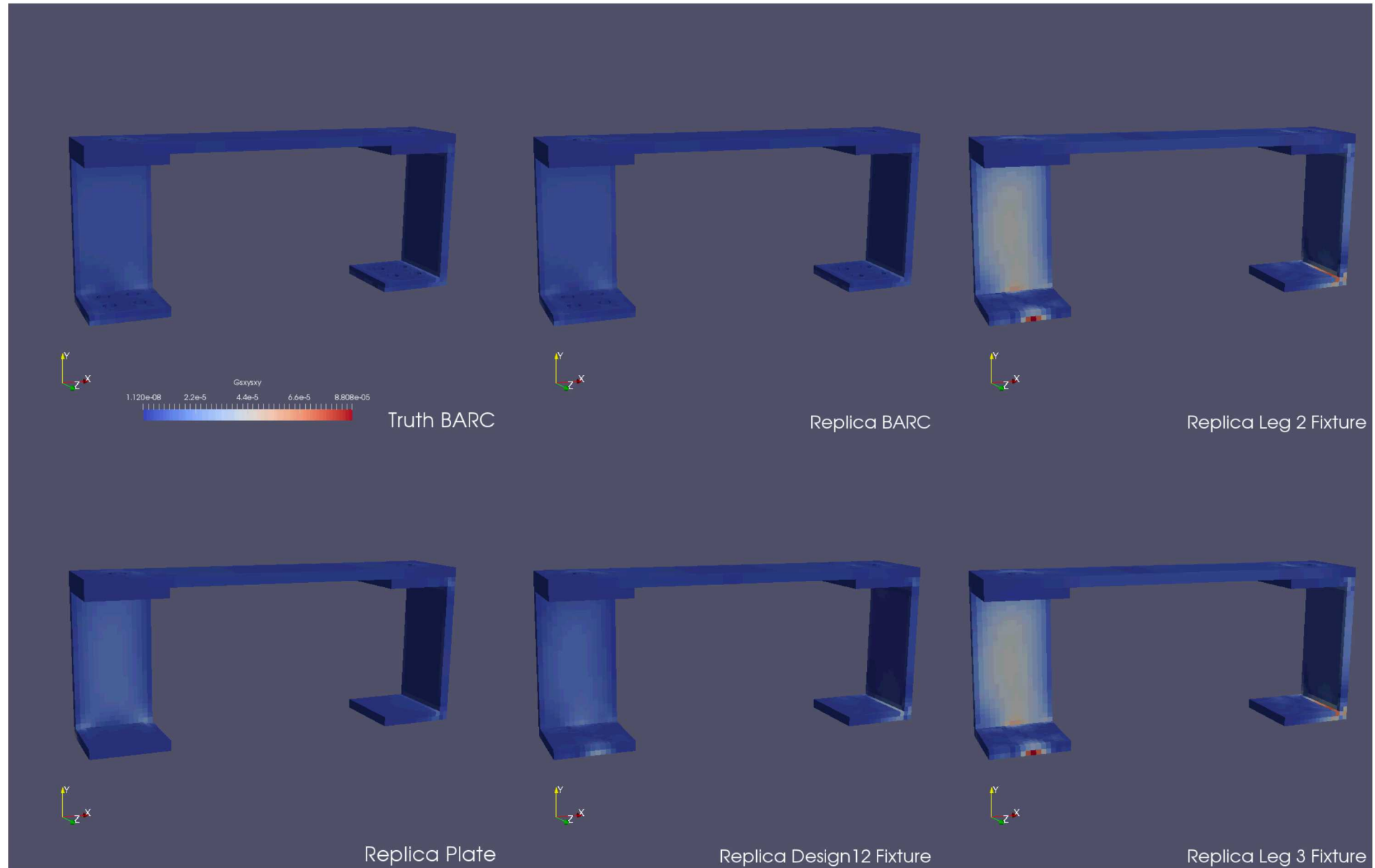


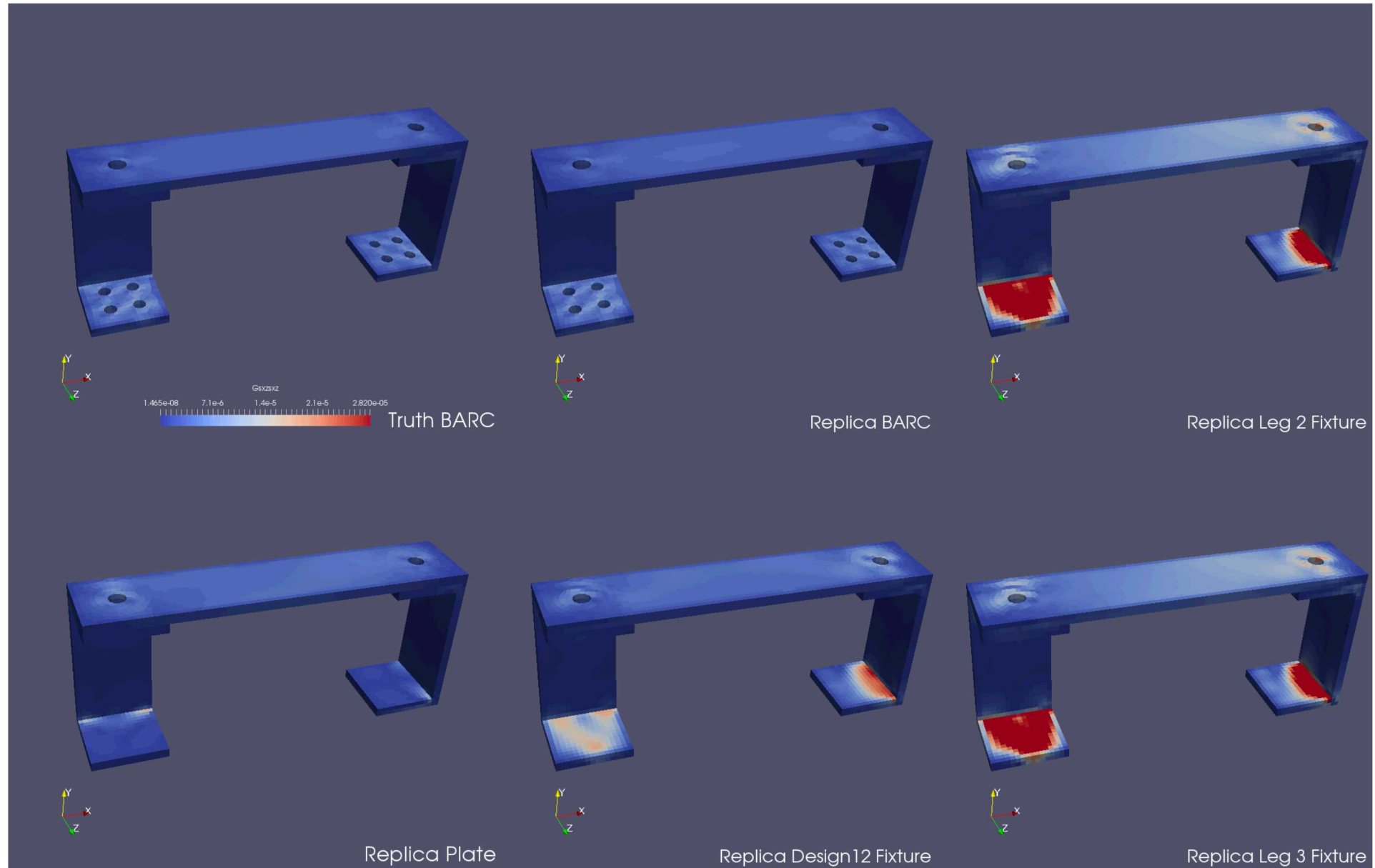


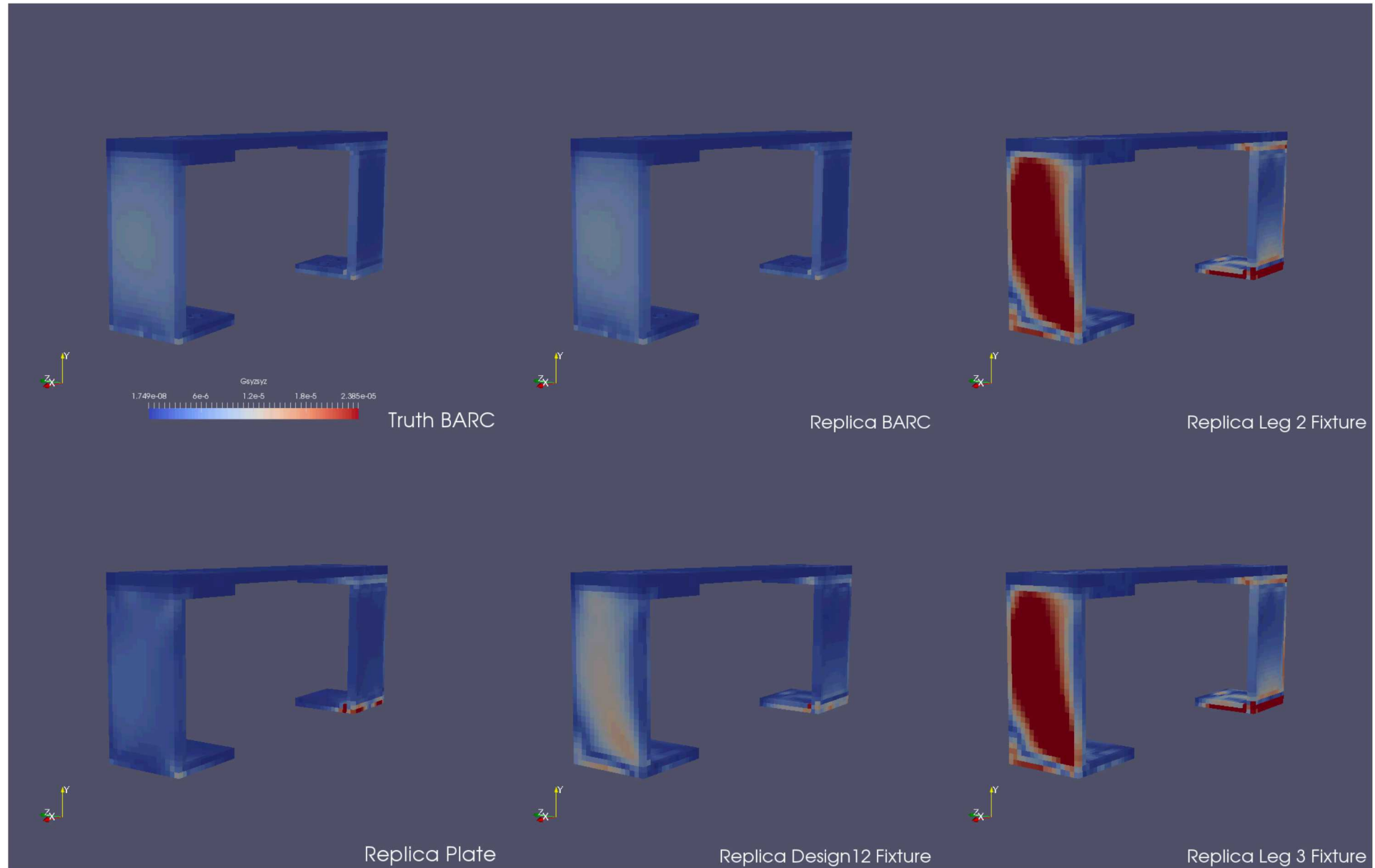




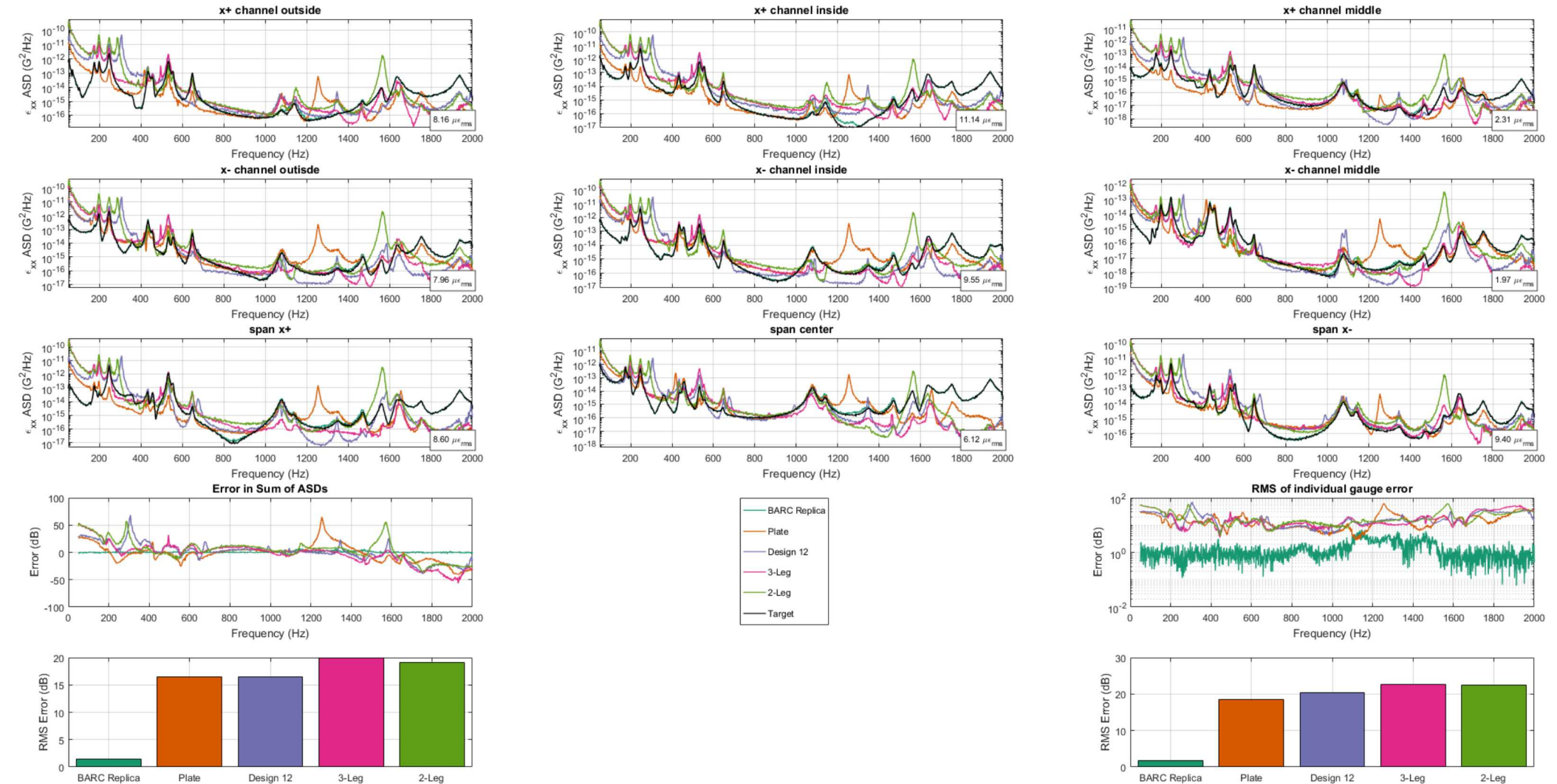




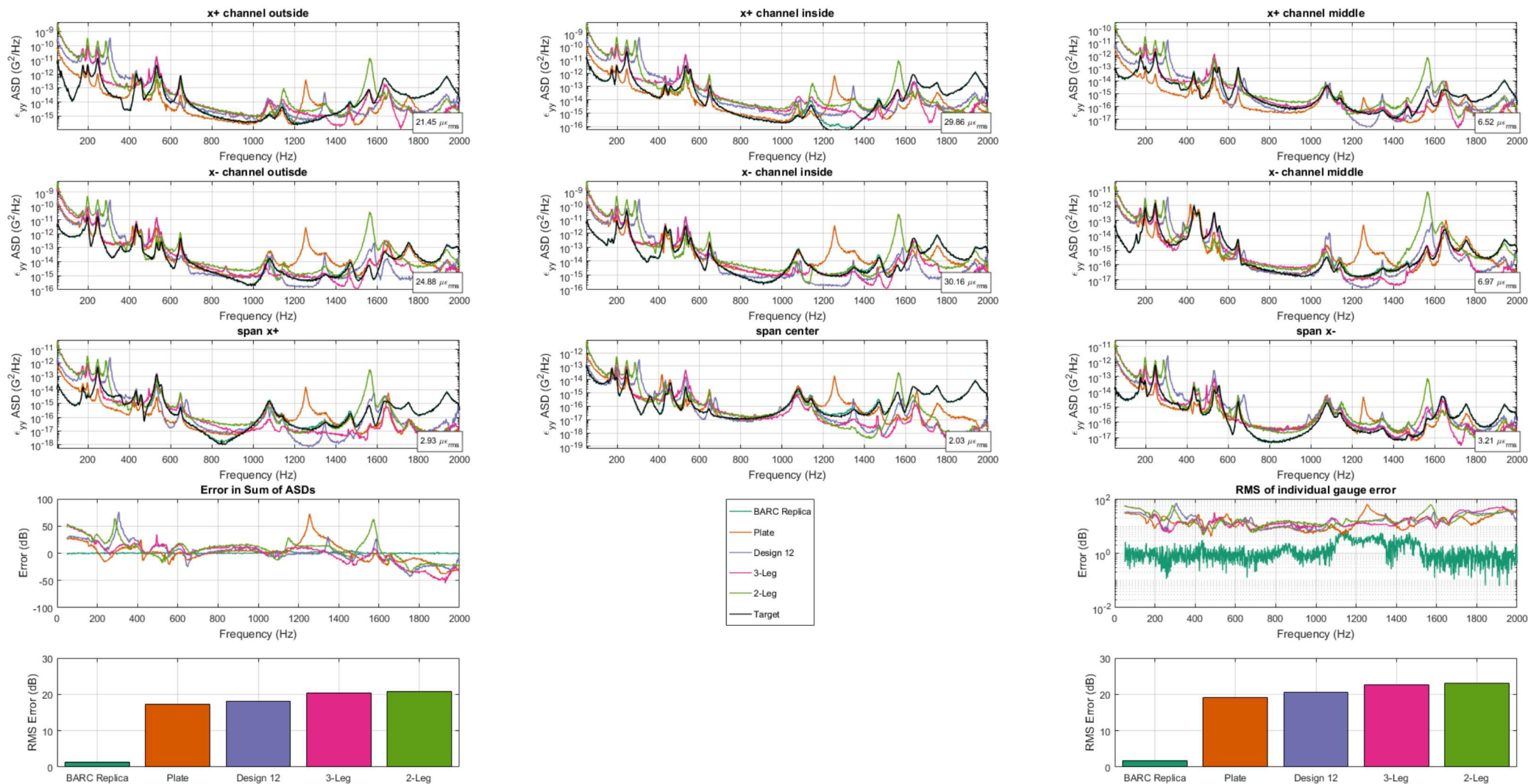




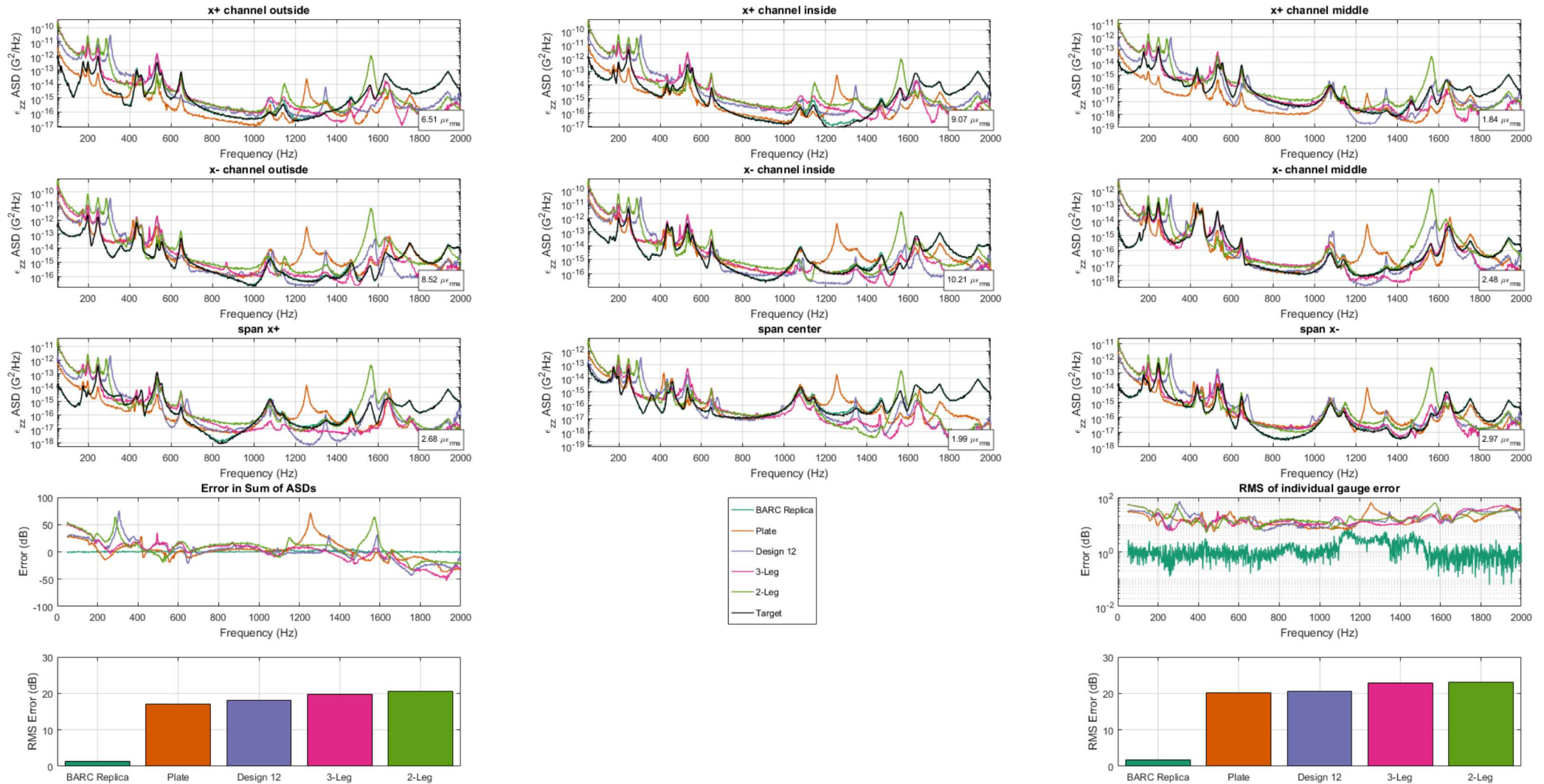
Strain Response at Local Elements: $G_{\epsilon_{xx}\epsilon_{xx}}$



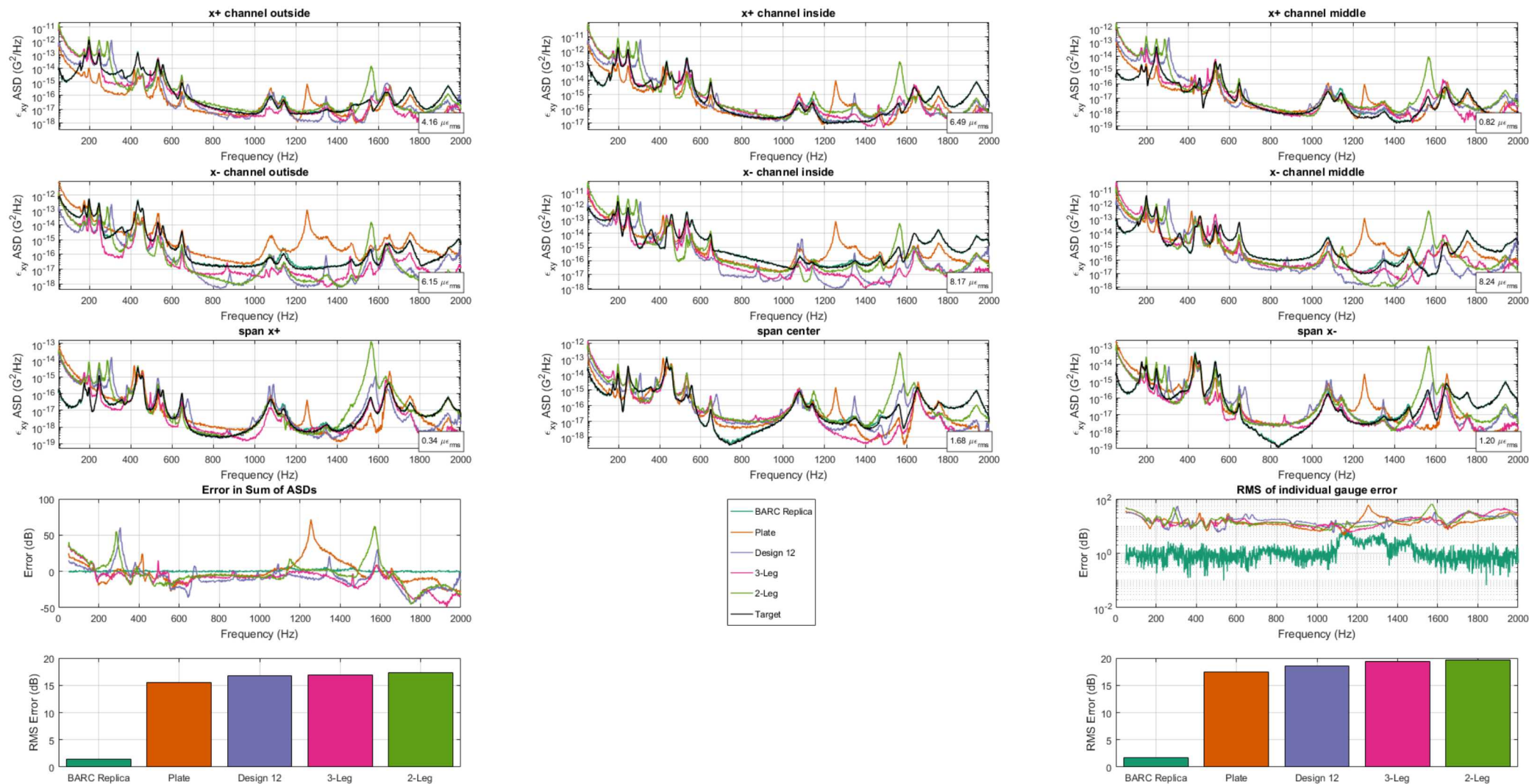
Strain Response at Local Elements: $G_{\varepsilon_{yy}\varepsilon_{yy}}$



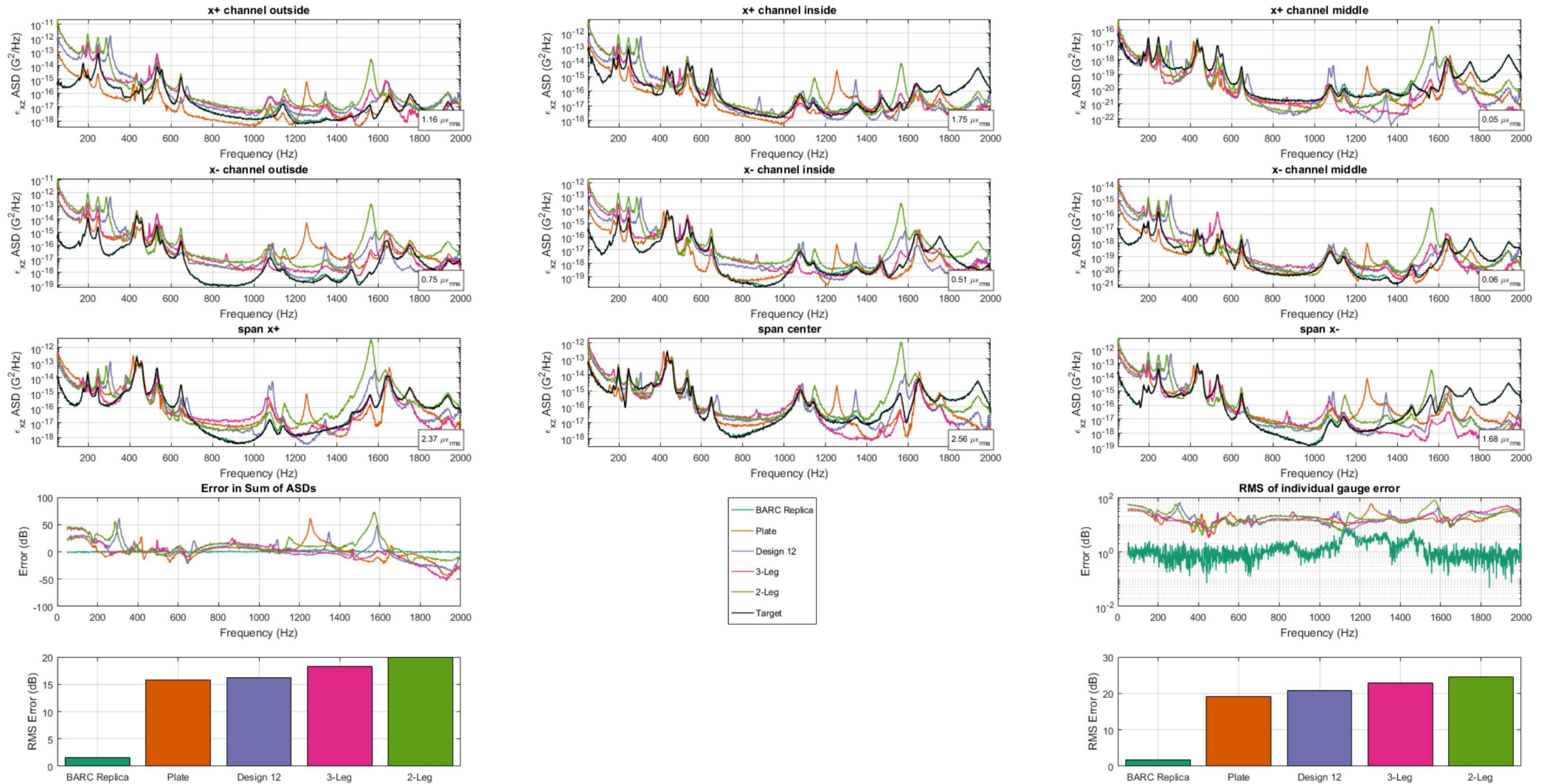
Strain Response at Local Elements: $G_{\epsilon_{ZZ}\epsilon_{ZZ}}$



Strain Response at Local Elements: $G_{\epsilon_{xy}\epsilon_{xy}}$



Strain Response at Local Elements: $G_{\epsilon_{xz}\epsilon_{xz}}$



Strain Response at Local Elements: $G_{\varepsilon_{yz}\varepsilon_{yz}}$

