

## UNDUE CONSERVATISM IN STRUCTURAL DESIGNS

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### ABSTRACT

Assuming that dynamic (transient/vibratory) loads are static in the design of structures is a long-standing engineering practice. The assumption is conservative for hardware that will have limited exposure to the specified dynamic environment and fatigue is not a concern. This is the case for most space flight hardware. Once out of the Earth's atmosphere, significant vibrations are non-existent for most hardware and vibration cycles are limited. Mass is, and has always been, a concern with respect to flight hardware design. How much unnecessary mass due to the subject assumption may be in structural designs was questioned and numerous tests have been performed to qualitatively address that. Tests have been performed on simple aluminum beams, beams with bolted joints and beams with welded joints. Results of those tests are reported in this paper. The objective of these efforts was to demonstrate the order of magnitude of conservatism associated with this engineering process via comparing static to dynamic tests and the longer term endgame is to develop a new or modified quick turnaround structural design process that circumvents assuming dynamic loads are static and results in lighter structural designs. Test results qualitatively demonstrate notable conservatism. Activities to devise the new method are underway.

Keywords: Lightweight, optimization, dynamic strength, frequency dependency of strain, undue conservatism, conservative loads

### 1. INTRODUCTION

In the context of this paper, “dynamic loads” refers to structural loads associated with space flight Components and Secondary Structures (CSS) which includes the majority of, possibly all, the many smaller items that attach to a Launch Vehicle (LV). The total dynamic load is a combination of the lower frequency (quasi static) dynamic response of the primary structure combined with the responses of the CSS. The latter is generally based on a specified random vibration environment, the CSS modes and modal effective mass, assuming a single degree of freedom system, and an assumed damping value. The long-term goal of these efforts is to realize relief from the effects

of the CSS design loads by means of leveraging the frequency dependency of strain and the material's capability to sustain dynamic loads as compared to static.

The design of many space flight structures is driven by dynamic loads. Dynamic analyses provide peak responses which are used to size hardware. It is common, possibly almost exclusive, to use dynamic loads and environments to develop a structural design load that is applied as if it were static within the design/analysis process.

Materials respond differently when subjected to dynamic loads as compared to static loads. So, while all indications are that doing so is conservative, assuming that a dynamic load is static, has no physics basis. This points to a need to develop a new set of material response capabilities and/or new design methodology to optimize designs.

No work to quantify or qualitatively assess the amount of conservatism this assumption introduces into the design has been identified prior to initiation of the efforts presented in [1], [2], [3] and in this paper. Therefore, one cannot readily determine how much this assumption costs a project or how much it limits LV or payload performance. NASA MSFC has exerted efforts to quantitatively or qualitatively characterize the order of magnitude of conservatism imbedded in this engineering paradigm and to then further efforts to devise a new or modified engineering design methodology that yields lighter structural designs.

Highlights from 2011 efforts were presented at the Spacecraft and Launch Vehicle Dynamic Environments Workshop (SCLV) in 2012 [1]. Findings from 2019 through 2023 efforts are presented in [2] and [3]. The objective of this paper is to summarize significant observations to date and discuss the path forward.

The ultimate endgame of these efforts is a new/modified quick turnaround design method that results in

lighter hardware. Hence, this project is labeled the Mass Mitigation Project (MMP). It should be noted that within the design phase of a new Launch Vehicle (LV), such as NASA's Space Launch System (SLS), or any flight hardware program, efforts are exerted to design hardware within defined structural mass specifications. However, it is common for the mass of a given piece of hardware to be greater than need be due to schedule. If an item is designed and it has positive margins of safety, it is likely more often than not signed off and engineering moves on to the next task. In many cases, mass could have been whittled away from that design but in that moment, meeting schedules is more important. Methods exist to design for dynamic environments that are closer to physics based as compared to applying a derived net Center of Gravity (CG) acceleration as a static inertial load in a stress analysis, and they may provide mass relief. Those methods are likely more time consuming than the current approach. They also, in cases, necessitate additional testing which takes budget and schedule. With Respect To (WRT) all the many small components and secondary structures that attach to a LV, very detailed analyses and tests are perceived as not feasible.

Minimal mass is very important in space flight hardware but in a fast and furious design project, "schedule is king." For this reason, proposed new engineering methods have to be quick turnaround methods comparable, WRT required labor, the current. The current vision for a first cut new or modified method is to:

1. Develop criteria to be used to deem accelerations above a frequency threshold for a given structural design and required dynamic environment negligible.
2. Develop frequency dependent strength properties for materials commonly used in space hardware.

These envisioned methods are to directly plug and play into the current engineering flow so little or no schedule impact occurs. [2]

## 2. MOTIVATION

The motivation for the work presented in this paper was to develop a business case for funding the development of a new or modified quick turnaround method to design hardware for dynamic environments that results in lighter hardware. And, doing so without necessitating development of new expensive "high performance" materials. Impacts or effects associated with assuming dynamic loads are static include:

1. Prevents optimizing structural designs WRT mass. There is significant undue conservatism in assuming a dynamic load is static so even with successful use of state-of-the-art optimization algorithms, hardware could still be made notably lighter.

2. CSS are sometimes notably heavier than need be due to design loads based on "higher" frequency accelerations. The unnecessary mass in a single small structure is often insignificant, but the sum of that conservatism in all the many CSS is likely very significant.

3. If the methodology used to assess the Primary Structure to CSS interface assumes dynamic loads are static and that includes unnecessary conservatism then that interface is more robust than need be. This can become an expensive engineering activity if the interface in question is a Primary Structure (PS) Orthogrid (OG) or Isogrid (IG) node, for example. It should be extremely rare, if ever, that PS, including the node of an OG or IG node where CSS may attach, is sized to accommodate CSS.

4. PS itself is designed, in part, to dynamic loads. Therefore, there is undue conservatism/mass in PS designs that could be removed.

5. A given LV has limited uplift capability. If a payload is designed per the current method, that may limit the LVs that could transport that payload to space, since the payload would be heavier than need be. This can be mission limiting or result in a project having to pay a higher rate for a larger LV.

Being able to leverage the subject facets of physics will enable a LV including all its subsystems being lighter which will allow the LV to lift heavier payloads. The same To Be Determined (TBD) method will facilitate the payloads themselves being lighter. Both effects add, resulting in a much more efficient industry.

The weight of hardware is dependent on the load that it was designed to sustain. Therefore, it is important that no unnecessary component of load be imbedded in the resultant design load. Focus on perceived overly conservative design loads and on developing methods to decrease high loads is longstanding. However, the manner in which loads are applied in the stress analysis process is equally important WRT the final design weight. Little, if any, work has been done to show how much the subject assumption is costing a project. With it demonstrated that there is significant gain to be realized, funding to develop a new or modified engineering method was and will be pursued. That method will include development of frequency dependent material strength properties and TBD criteria that will facilitate defining a frequency threshold, for a given hardware item and specified environment, above which oscillating loads can be omitted from the structural design load development process.

## 3. BACKGROUND

The technical origin of this project is based on the relationship between harmonic motion and strain. While loads

are often thought of as that which breaks hardware, strain is the real culprit. Static strain, depicted in Figure 1 and expressed in equation 1, is generally a small fraction of the displacement of a structural member under load.

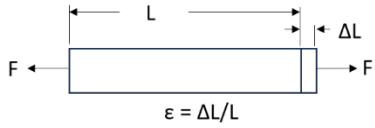


**FIGURE 1: STRAIN**

$$\varepsilon = \frac{\Delta L}{L} \quad (1)$$

Where,

$\varepsilon$  = Strain

$\Delta L$  = Change in length

$L$  = Undeformed length

The equation of harmonic motion is presented in Equation 2.

$$X = A \sin(\omega t) \quad (2)$$

Differentiating twice:

$$\dot{X} = A\omega \cos(\omega t) \quad (3)$$

$$\ddot{X} = -A\omega^2 \sin(\omega t) \quad (4)$$

Where,

$A$  = peak displacement

$\omega$  = frequency of oscillation

$t$  = time

$X$  = displacement at  $t$

$\dot{X}$  = velocity at  $t$

$\ddot{X}$  = acceleration at  $t$

Disregarding the sense of  $\ddot{X}$ , solving (4) for  $A$  and substituting that into (2) results in,

$$X = \frac{\ddot{X}}{\omega^2} \quad (5)$$

Equation 5 indicates that for a given acceleration, as one goes up in frequency, displacement goes down exponentially. Strain is directly dependent on displacement. Therefore, as one goes up in frequency, for a given acceleration (a 10 g design load for example) the structural significance of that acceleration goes down exponentially. In the current design paradigm, that acceleration is assumed to be at 0 Hz (static) and a global inertial load.

In 2011 [1] a simple sine dwell test of an Aluminum 6061 beam was performed. The intent was to design and perform two tests in which strain associated with a structure's first mode was dwelled at and compare the strains. The underlying premise of

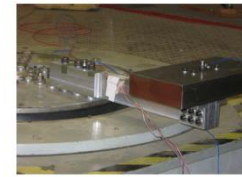
MMP is that accelerations at "higher" frequencies are less significant than those at "lower" frequencies WRT structural capability. Therefore, for this to be meaningful, one test had to be at a "low frequency" and the other at a "high frequency". Also, the effective load and the cross sectional area of max strain in both cases had to be the same. This was accomplished with a 14" cantilever Test Article (TA). The high frequency case was at 300 Hz and was the first mode of that TA. Adding mass without adding stiffness will decrease the frequency of a mode. To accommodate the low frequency case, a mass ( $\approx 40$  lb) was added to the end of the beam which dropped the fundamental mode to  $\approx 40$  Hz. The two configurations are shown in Figure 2.

In both cases, the moment that yielded the max strain was reacted at the fixed end of the beam. The amplitude of the input sinusoidal vibration was iteratively tweaked to get, per Newton's second law, approximately the same effective force. For the high frequency case that was  $\approx 98$  lb and for the low frequency case it was  $\approx 92$  lb.

Test observations supported the MMP principle of the structural effect of a dynamic load is dependent on frequency.



Configuration 1 – Beam without the mass  
 $f \approx 300$  Hz,  $M = 6.5$  Lb.,  $M_{CL} \approx 4$  Lb.



Configuration 2 – Beam with added mass  
 $f \approx 40$  Hz,  $M = 48.5$  Lb.,  $M_{CL} \approx 46$  Lb.

**FIGURE 2: 2011 TEST SETUP**

Within Figure 2,  $M$  is the total mass of the TA and  $M_{cl}$  is the cantilevered fraction of  $M$ .

The measured strain at the 40 Hz mode was approximately 3 times that at the 300 Hz Mode. The strain is shown in Figure 3.

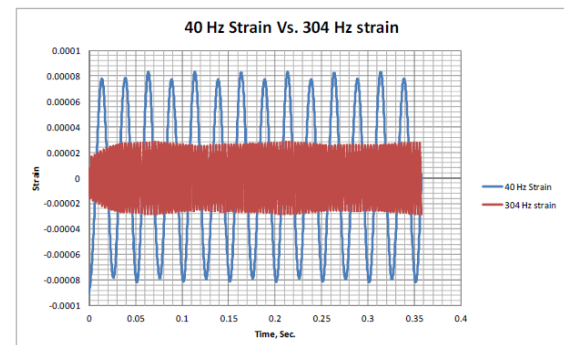


**FIGURE 3: 2011 TEST RESULTS**

In 2019, a TA was designed for continued testing. It was made of 6063-T5 Aluminum and its dimensions were

18"X2"X0.375" as shown in Figure 4. Two inches of the beam were clamped leaving a 16" cantilever.

Static tests were performed to demonstrate that loading the beam to the predicted ultimate load, approximately 80 lb, would result in Plastic Deformation (PD) of the beam. After unloading the beam, slight PD was visible.

Sine dwell tests were then performed with the same beam design. The intent was to get data or make observations that could be compared/contrasted with the static tests and to get strain data at numerous frequencies to simply look a little deeper than the 2011 efforts. The TA was cantilevered from the expander head of a vibration shaker table as shown in Figure 5.

In the final test of the day, the test engineer was asked to increase the input amplitude as high as he was willing to go. The test dwelled at the TA's fundamental mode (about 43 Hz) with 15 g input. Two significant observations were made:

1. Measured strain was more than twice the yield strain for that alloy.
2. The tip oscillating displacement was estimated to be +/- 6 in. (Figure 6) while the tip displacement at the static ultimate level was estimated to be between 1 and 2 in.

The 2019 tests resulted in no visible plastic deformations.

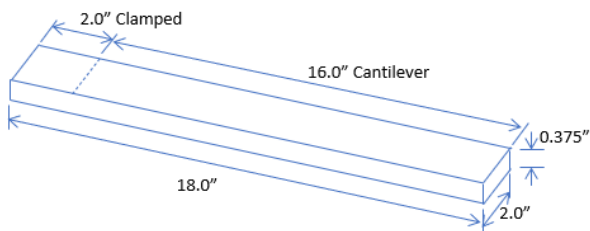


**FIGURE 4: TEST ARTICLE**

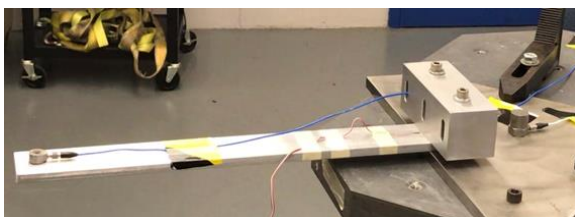


**FIGURE 5: 2019 Test Setup**

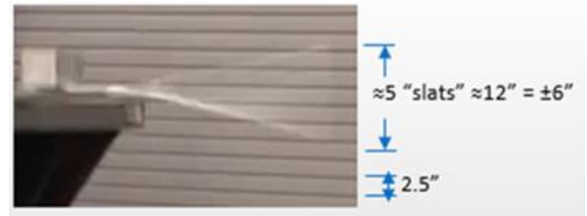


**FIGURE 6: 2019 SINE DWELL TEST**

This emphasized the point - this alloy sustained much higher deformation and strain while oscillating as compared to static tests.

More tests were performed in 2020 but in these tests one end of the beam was fixed, and the other end was forced to oscillate vertically. Test Support Structure (TSS) was designed to attach to a shaker and then kinematically interface with a point near the tip of the beam (Figure 7). The TSS included 3 load cells that measured forces applied to the TA.

Sine dwell tests were performed, and they were controlled to force. The test engineer was provided peak force and frequency as required test inputs. The 3 measured forces in the TSS were summed and the shaker controller controlled to that.

The most significant observation from these tests was that the end of the beam was oscillated with a 40 Hz  $\approx$  900 lb force without fracture or plastic deformation and  $\approx$  85 lb statically resulted in slight plastic deformation.



**FIGURE 7: 2020 TEST SETUP**

A next step was to make the same point for bolted and welded joints.

#### 4. BOLTED JOINT

A beam TA was designed via lap joining two steel members and fastening them with threaded fasteners (Figure 8). Analysis, indicated that the fasteners should plastically deform with a tip/point load of 377 lb. Static tests (Figure 9) plastically deformed the high strength fasteners at 390 lb.

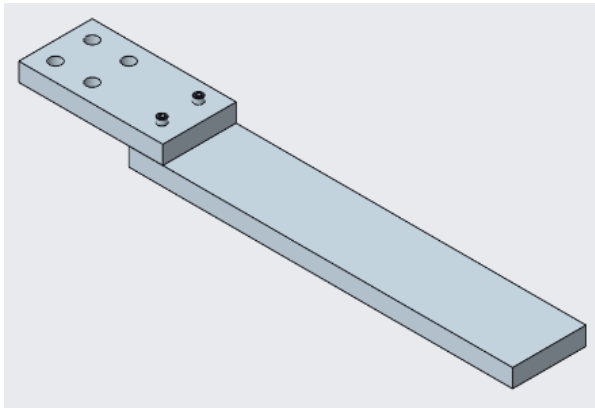


**FIGURE 8: BOLTED JOINT**

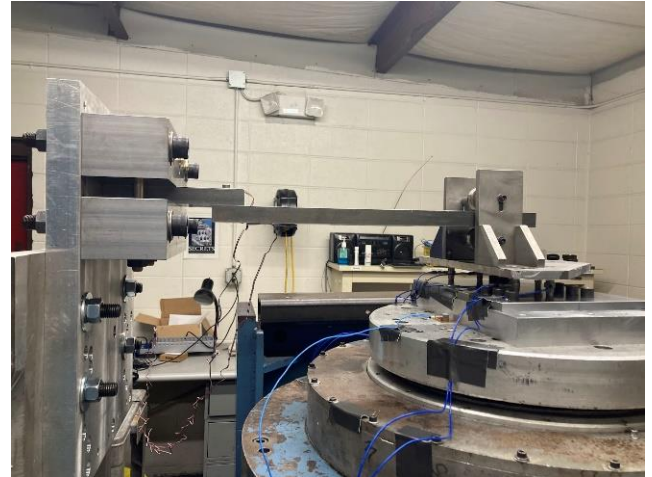


**FIGURE 10: BOLTED JOINT DYNAMIC TEST SETUP**



**FIGURE 9: BOLTED JOINT STATIC TEST SETUP**

The underlying premise in these efforts is that assuming dynamic/oscillating loads are static is overly conservative. So the same beam design was tested in the same manner as the 2020 Tests (Figure 10).

Tests were run from 20 – 300 Hz with peak forces ranging from 390 lb (the static failure load) to 2,000 lb (Table 1). Two fastener failures occurred at low peak loads at 20 Hz but the TA sustained much higher loads from 50 – 300 Hz. In part, perhaps significantly so, the two failures were due to the TA having 2 fasteners on an axis normal to the direction of motion thereby facilitating an axis about which ratcheting/prying occurred (Figure 11). Three or four fasteners in a plane (vs. on a line) would likely have had a different outcome.

| Load (lb) | Frequency (Hz) | Result  |
|-----------|----------------|---------|
| 390.0     | 20.0           | Failure |
| 780.0     | 20.0           | Failure |
| 780.0     | 50.0           | Pass    |
| 780.0     | 75.0           | Pass    |
| 780.0     | 100.0          | Pass    |
| 780.0     | 150.0          | Pass    |
| 780.0     | 200.0          | Pass    |
| 780.0     | 250.0          | Pass    |
| 780.0     | 300.0          | Pass    |
| 1170.0    | 50.0           | Pass    |
| 1170.0    | 75.0           | Pass    |
| 1170.0    | 100.0          | Pass    |
| 1170.0    | 150.0          | Pass    |
| 1170.0    | 200.0          | Pass    |
| 1170.0    | 250.0          | Pass    |
| 1170.0    | 300.0          | Pass    |
| 2,000.0   | 150.0          | Pass    |
| 2,000.0   | 200.0          | Pass    |
| 2,000.0   | 250.0          | Pass    |
| 2,000.0   | 300.0          | Pass    |

**TABLE 1: BOLTED JOINT DYNAMIC TEST RESULTS**

Independent of the suggested additional fastener, these observations highlight the MMP underlying principle. A structural design can sustain higher loads when they are higher frequency oscillating loads. The subject TA sustained 2,000 lb at 150 Hz but could not sustain 390 lb (the static failure load) at 20 Hz.

With the data presented, perhaps one could say that a design with those specific fasteners can sustain twice the static ultimate

strength if the load is a modal response or forced vibration at  $\geq 50$  Hz and four times the static strength if the load is oscillating at  $\geq 300$  Hz. In any case, it has been demonstrated that the fasteners and the fastened structural design's capabilities are a function of frequency and that at higher "frequencies" their capabilities are higher.

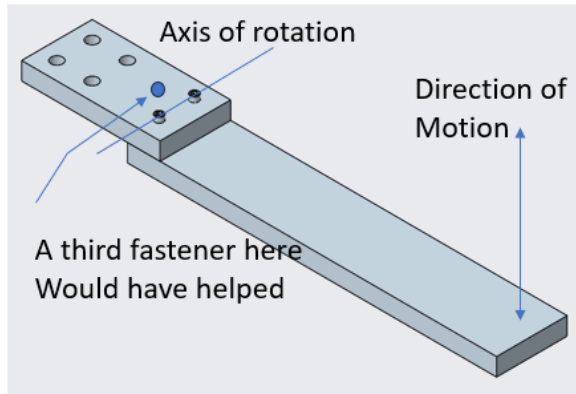


**FIGURE 11: BOLTED TA PRYING SCENARIO**

## 5. WELDED JOINT

A welded beam TA was designed. It incorporated a butt weld between two pieces of aluminum with 1 in. welds on both the top and bottom (Figure 12).



**FIGURE 12: WELDED TA**

Predicting failure of a weld is less certain than with a bolted joint. This is due to each weld being to a degree unique as compared to threaded fasteners that, with quality control in their manufacture, have very consistent properties. Nonetheless, predictions were made and resulted in a predicted static load of 77 lb to result in plastic deformation of the beam.

Knowing that there would be variance in the results from welded TAs, 7 TAs were fabricated (6 are shown in Figure 13).

Determining if plastic deformation had occurred was not as straight forward as it was in other tests. The intent was to do so via assessing a change in the angle between the top surface of the interface end and that of the cantilevered segment. In the first static run, grid paper was used and the initial configuration of the two surfaces was drawn on the paper. Then, after unloading the TA incrementally, whether that had changed was assessed

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visually. That run resulted in a failure load notably higher than predicted (121 vs. 77 lb). This significant difference caused suspicion in the method of assessing plastic deformation and run 1 results were disregarded. Following runs utilized a 9 in. long standard bubble type level and then, finally, a digital level was utilized.



**FIGURE 13: WELDED TAS**

Prior to static test of a TA, that angle was determined via the level. Then that was reassessed after incrementally unloading the TA. The results of the static tests (Figure 14) was the mean applied static load that achieved plastic deformation of the TA was 84.75 lb.

Dynamic tests were performed (Figure 15) as they were in the previous bolted TA test. Results are shown in Table 3. As was done in the bolted TA tests, the sine dwell tests were controlled to specified force.



**FIGURE 14: Welded TA Static Test**

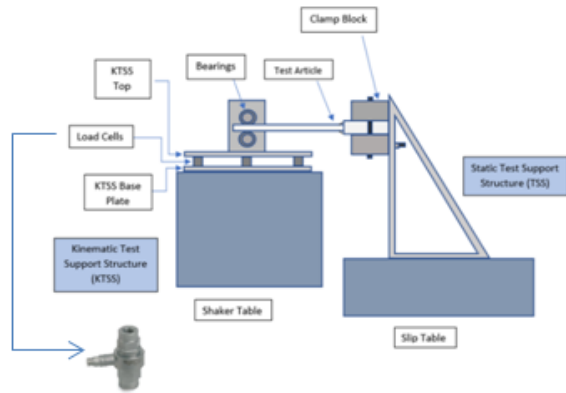


**FIGURE 15: WELDED DYNAMIC TEST SETUP**

| Load (lb) | Frequency (Hz) | Failure     |
|-----------|----------------|-------------|
| 160       | 300            | No          |
| 160       | 250            | No          |
| 160       | 200            | No          |
| 160       | 150            | No          |
| 160       | 100            | No          |
| 160       | 75             | No          |
| 160       | 50             | No          |
| 160       | 20             | No          |
| 240       | 300            | No          |
| 240       | 250            | No          |
| 240       | 200            | No          |
| 240       | 150            | No          |
| 240       | 100            | No          |
| 240       | 75             | No          |
| 240       | 50             | No          |
| 240       | 20             | No          |
| 400       | 300            | No          |
| 400       | 250            | No          |
| 400       | 200            | No          |
| 400       | 150            | No          |
| 400       | 100            | No          |
| 400       | 75             | No          |
| 400       | 50             | Yes, Note 1 |
| 400       | 20             | Note 2      |

**TABLE 3: WELDED TA DYNAMIC RESULTS**

Note 1: during this run, one of the fasteners in the TSS loosened and significant chatter occurred. Cracks were observed in the weld (Figures 16 and 17) after the completion of this run. It is likely that the impacts associated with the chatter caused the cracks.

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Note2: The 20 Hz run was not executed due to the cracks discussed in Note 1.



**FIGURE 16: TA 5B PRE-TEST 50 HZ RUN**

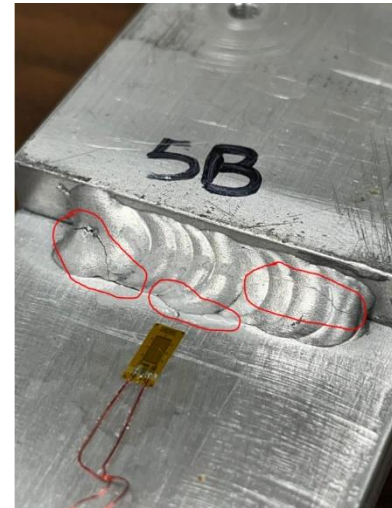


**FIGURE 17: TA5B POST 50 HZ RUN**

## 6. DISCUSSION

A long-standing engineering practice is to assume that dynamic loads (vibrations and low-level transients) are static within the structural design and analysis processes. This assumption is conservative but the magnitude of that conservatism, how much making that assumption is costing the project, is unknown. Initial efforts to assess the magnitude of that conservatism were exerted in the 2011-2012 time frame. Results confirm the notion that this conservatism can introduce significant unnecessary mass into a design. Further efforts resumed beginning in 2019 and have continued to date.

Test were performed with a beam cantilevered from a vibration shaker table, in which huge deflections and strain much higher than yield were observed without visible plastic deformation. Tests were also performed with one end fixed and the other loaded with a sinusoidal force. The later was performed using a beam made from a single piece of aluminum, a beam

constructed two members and connected with bolts and a beam constructed with a welded joint. In all cases, the premise of MMP was supported:

1. Applying dynamic loads or responses as static design loads is conservative. In other words, in each test scenario, the structure sustained much higher loads when they were harmonic as compared to static.
2. At higher frequencies tested, higher loads were sustained as compared to the lower frequencies tested. Minimal evidence of this was observed in the test results. The two failures that did occur did so, at least in part, due to design limitations (a third bolt would have helped) or unexpected test conditions (TSS fastener loosened) so it was not demonstrated that the failure was purely due to the low frequency. It is also true that they occurred at the lowest frequencies tested.

## **7. CONCLUSIONS**

It has been qualitatively demonstrated that there is notable conservatism embedded in assuming dynamic loads are static in the structural design process. Therefore, evolving a new or modified structural design methodology for structures that are design driven by dynamic loads would be a good investment of resources. With the yet to be developed new methodology in hand:

1. Mass savings will be realized.
2. Engineering development time and budget will be saved in the sense that engineering challenges associated with high design loads will be decreased.
3. LVs will be able to accommodate more payload mass thereby making space missions more economical.
4. Payloads can be lighter and therefore potentially ride on a less expensive LV.

## **8. CURRENT AND FUTURE WORK**

The envisioned finale of these efforts is to design and test a TA, possibly a simulated flight avionics box, by three methods and compare the weight of each. The three methods include:

1. Design via the current method of assuming dynamic loads are static and performing hand calculations and/or linear Finite Element Analyses (FEA).
2. Design via engineering of metallic fatigue methods.
3. Use of the TBD new/modified method.
4. Publish papers detailing the bolted joint and welded joint tests.

The fatigue methods seem to have notable uncertainty in them as well as potentially more conservatism than the static assumption. Therefore, current work includes fatigue testing of AA6061 beams to acquire data hoped to facilitate a TA design based on principles of fatigue.

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Future work will include design and analysis of a selected TA via the three methods discussed and continued efforts to further options for the TBD subject methodology.

## **ACKNOWLEDGEMENTS**

From 2019 to date, much of the MMP engineering labor has been performed by engineering students employed as interns. The bolted joint work was performed by Ms. Michaela Tarpley, a student (at that time) at the University of Alabama, Huntsville. And, the welded joint engineering was performed by Ms. Michelle Quinones, a student (at that time) at the New Jersey Institute of Technology.

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