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Development of a Compact, Low Cost Test Fixture to Evaluate Creep in High Strength Softgoods Materials under Constant Environmental Control

Vincenzo M. Le Boffe, Thomas C. Jones, and Winfred S. Kenner
NASA Langley Research Center, Hampton, Virginia

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Contents

Abstract

Space exploration is typically driven by two key factors: cost and payload mass. Therefore creating relatively low cost, low mass space hardware is critical. With the advancement of new high-strength flexible materials, a new chapter is opening in the evolution of pressurized space structures. Space-rated inflatable structures are becoming more attractive for their low mass, compact storage, and capability of performing as well, or better than, their rigid counterparts for applications such as habitats and airlocks. The lifetime properties of materials used for inflatable structures are not as well characterized as heritage composite and metallic materials; therefore, they require extensive testing to increase the level of confidence in their use. Creep testing is one method of evaluating the expected lifetime of materials that may be under constant load for many years in operational use. Conventional creep testing of high-strength softgoods webbings or cordage requires a large, temperature-controlled facility, with structural frames capable of supporting large dead weights, weighing thousands of pounds on each specimen. This is expensive and takes up a significant amount of volume in a facility for a long time. This paper describes the design and analysis of a low-cost, low-volume creep test stand that can be locally environmentally controlled, so that smaller, cheaper facilities can be used for characterizing the creep life of the softgoods materials used in inflatable space structures.

Introduction

The primary benefit of inflatable space structures is to provide compact packaging for launch, and a large volume in their intended configuration once in space. These structures can be used as pressurized crew quarters in space and on the surface of other planetary bodies. Inflatable structures are multi-layered, with an inner air retention bladder, a structural restraint layer (Fig. 1), and a series of outer layers for thermal and Micrometeoroid and Orbital Debris (MMOD) protection. The restraint layer is subject to tension loads due to the internal pressure. Restraint layers are typically constructed of webbing, cordage and/or fabric, arranged to optimize the load distribution. This approach creates a lightweight structure that can be packaged compactly. The materials used for the restraint layer require a high degree of reliability over the life of a mission due to their sustained loading and operational environment, therefore practical methods of evaluating the lifetime behavior of these materials is critical.

The materials used for the restraint layers of inflatable space structures are typically polymeric and viscoelastic, such as Kevlar or Vectran. The manufacturers of these products provide baseline data for the fibers but the end user typically performs testing at the webbing or cordage level. Strength and stiffness characterization can be attained using a tensile load frame of the appropriate capacity, but to obtain lifetime creep behavior a more specialized test fixture is required. Creep is the tendency of a solid material to slowly deform under the influence of a persistent stress. This can cause permanent plastic deformation in the material even if the material is subjected to loads below its yield point. Creep is well understood and characterized for metallic materials, however creep properties for softgoods components like Kevlar or Vectran webbing and cordage are not well defined. Creep testing equipment for metals and composites do not allow for the large deformations often exhibited by softgoods components. In addition, due to small production runs of the custom webbing or cordage used, and the length of time required to perform these tests, there is a shortage of creep data for the candidate materials proposed for use in inflatable space structures.

This publication addresses the development of a simple, low-cost, low-volume creep test stand that can produce creep data for high-strength softgoods materials. The design and development of this hardware has the following goals: operate in a relatively small space, have a low cost of fabrication, have the capability to control the thermal environment around the specimen, and provide accurate creep data.

Approach

To obtain creep test data, the conventional approach is to dead load a component, or designated material test coupon, at a tensile stress below the material yield point for an extended period. The applied load is a percentage of the average ultimate strength of the specimens. The primary advantage to the conventional creep test approach is that it guarantees a constant load (Fig. 2). The disadvantages are that this type of test requires a significant amount of space for high strength specimens, due to the size of the required dead load, thermal control of the entire facility, and the availability of the test area for the duration of the tests, which can be up to several years in length.

An alternative method uses a lever arm that acts as a force multiplier to introduce high loads while using significantly less weight and reducing the overall size of the test stand (Fig. 3). Given the project goals of reducing the footprint of the test stand and the required high loads (up to 12,000 lbs), a lever arm approach was selected for the test stand design described in this paper. The advantages of using this approach are the significant reduction of the weights required to conduct a test, reduction of the necessary test space, and the reduced cost of materials used for the weights. The disadvantage is that the design must account for the change in angle of the lever arm over time so that the load remains essentially constant.

For the creep test stand design, the maximum test weight required was 12,000 lbs, so a 20-to-1 load ratio was used for the lever arm, resulting in the need for only 600 lbs of weights. The lever arm length and pivot point location could be further customized if necessary to provide a different load ratio for a different specimen strength or type. As the specimen elongates under load, the lever-arm is constantly rotating which can add a small change to the load. This must be considered in the design. An inline load cell was added in the current test setup to monitor the change in applied load due to the change in the angle of the arm. This should be characterized for any reproduction or variant of this design during the initial setup, to quantify any load change present due to the angle of the lever arm beam.

An environmental chamber was also designed and implemented on five of the nine creep test stands that were fabricated to provide an insulated enclosure around those test specimens. Each chamber is equipped with temperature and humidity sensors that provide data and feedback to monitor and maintain a constant environment for each specimen. To minimize cost, a single heating/cooling system is ducted into the five environmentally controlled creep test stands (Fig. 4) that were built. The four stands without chambers were used to collect control data in a non-environmentally controlled facility.

Design

The design intent was to develop a creep test stand that met the following requirements:

1. Cost-effective construction.
2. Easily manufactured and duplicable so multiple stands could be fabricated.
3. Capable of testing nominally 48-inch long by 1-inch wide softgoods specimens.
4. Capable of applying creep loads up to 12,000 lbs.
5. Uses a lever arm system to significantly reduce the weight required to apply the maximum loading condition.
6. Allows for specimen displacements up to 1.69-inch (based on expected total strain).
7. Meets safety requirements for operations and tensioning set-up.
8. Compact to allow multiple stands to be accommodated in a small area.
9. Allows for an environmental chamber to be added separately to maintain a constant temperature around the specimen.
10. Provides a simple method for tensioning the specimen before the start of the test.

The NextSTEP-2 Habitat program, under which these stands were designed and fabricated, provided the funding to build nine creep test stands based on the above guidelines. Five with environmental control and four to provide a baseline in a non-environmentally controlled building. A secondary goal was to produce the entire set of nine stands at a lower cost than one commercially available creep test machine, which costs in the range of $50-100K. In addition, commercial creep test stands are typically designed for metals and composites, which are difficult to adapt for creep in high-strength softgoods materials, due to the order of magnitude larger deformations expected.

The engineering decision was made early in the design process to use standard stock materials. The selection of low-cost materials, reduced the need for expensive machining or ordering specialty items that yielded a cost-effective design. The use of custom fabricated components was minimized and applied only when the design and analysis required it. The design approach was to reduce manufacturing processes as much as possible to reduce the cost of each individual fixture. The average time to build one creep stand was 170 work-hours, which includes the manufacturing time, and incorporates the development of the initial prototype. The time to manufacture successive units was reduced as multiple parts could be manufactured at the same time.

The creep test stand was sized to fit a 48-inch long softgoods specimen that interfaced with the stand via simple pin grips, as shown in yellow prior to loading in Fig. 3. This length was based on the size of the seams and looped-end terminations used for the high-strength webbings NASA is testing. It was desired to maintain the mid-section of the specimen to be at least twice as long as the combined 8-inch stitched section and 6-inch loops at each end. The specimen dimensions drove the overall height of the test stand. The test stand dimensions are shown in Figure 5.

The lever arm beam was the most critical element of the creep stand design. Figure 6 illustrates the lever arm with the 20:1 load ratio at the maximum load case of 12,000 lbs. The beam needed to be optimized to provide minimum deflection yet remain practical for manufacturing and eventual assembly. The creep test stand uses a solid aluminum stock material, selected because of availability, to provide the required high stiffness desired in the arm. Deflection and bending calculations showed that the beam under load had negligible deflection (0.125-inches with 12,000 lbs load). The beam is a free-floating component that sits on top of the creep stand structure and is free to rotate in a low friction cradle using a knife-edge pivot point. The beam is captured in place by a simple retaining fender bracket that prevents the beam from jumping off the stand when the test specimen fails (Fig. 7).

The creep test stand is designed to be simple and easily duplicable for mass production. It could be modified to fit particular test requirements outside of the current testing as well if needed. Changes in requirements must be carefully considered in the engineering analysis when components or loads are scaled or modified.

The amount of stretching that the softgoods specimen is expected to exhibit during setup and while under creep loading is a critical feature of the creep stand. Softgoods components, due to their construction can stretch 2-3% of their length, prior to developing any nominal stresses as the yarns decrimp, followed by up to 0.5% of creep strain once loaded. The lever-arm beam was engineered to allow for specimen elongation of up to 1.69-inches to avoid the equipment bottoming out during testing. This was sufficient for the specimen length and materials used.

The design of the creep test stand must meet all pertinent safety requirements and design factors. Every critical component was analyzed and engineered to meet or exceed the maximum design load, including the required minimum factor of safety of 1.4 on the ultimate strength for test equipment. In addition, the setup procedure for loading and operating the stand was also evaluated, considering user risks and installation efficiencies. Appendix C documents all the necessary steps to install the test strap, generate the proper tensioning, and mount the necessary instrumentation to gather data.

The creep stand went through several design iterations, to optimize its structural features and to reduce its footprint. The final design of the test stand (Fig. 5) was sized to fit five units in a 12-foot by 12-foot room, which was available in the test facility, and represented a standard office floorplan and ceiling height. This was to demonstrate the ability of setting up long-term creep tests of high-strength softgoods components in a standard building rather than requiring a specialized facility.

To maintain a constant temperature around the specimens in the five environmentally controlled test stands, an insulating enclosure was designed and added to each creep stand (Fig. 8). The enclosure created an insulated, low-volume air chamber around the specimen that was connected to a system that pumped in heated or cooled air to all five chambers.

The final design consideration for the test stand was to provide a mechanism to properly tension the specimen prior to the start of the creep test. This was accomplished via a simple yoke at the base of the stand attached to a threaded rod that passes through the base and is secured with a nut and washer (Fig. 9). By rotating the hex nut, the threaded rod is lowered or raised to apply or remove tension in the strap as needed. This step is to take up the slack in the specimen until the lever arm is in a position above horizontal, after which load plates are added to take the specimen up to the test load condition. A secondary hex nut (jam nut) is used to lock the simple mechanism in place at the desired height.

Analysis

The softgoods creep stand was required to pass a rigorous engineering review prior to fabrication and use. All of the components for the stand were analyzed based on the type and magnitude of the maximum design load(s) expected, with a maximum specimen load of 12,000 lbs. All engineering calculations were presented at a preliminary and final design review. The factors of safety and margin for each component were presented. Components with critical local stress areas, or buckling requirements also included a Finite Element Analysis (FEA). These analyses are provided in Appendix A.

In addition to the analysis of individual components, a global analysis was performed on the entire assembly. The analysis showed that the components meet the safety requirements under the test loads, however additional analysis was performed to ensure that the structure was capable of handling buckling loads, and the stresses transferred from component to component without creating unexpected stress concentrations. Components in the creep test stand were analyzed for the following types of load: shear, double shear, tension, bending, column buckling, and compression. All analytical equations for the calculations are from “Mechanical Engineering Design 5th edition” by Shigley[1]. The minimum factor of safety of 1.0 yield and 1.4 ultimate required of each component was mandated by NASA-STD-5001B[2].

The desired temperature range, and volume of the chambers were provided to an outside vendor that sized the heating and cooling unit needed to meet the requirements of five connected chambers. The R-value for the thermal chamber was also taken into account for this evaluation, and R3 insulation panels were selected based on the requirements. The primary goals of the environmental chamber were to minimize the volume of air needed to be heated or cooled, minimize any leakage or heat transfer losses, and be capable of maintaining the desired set point temperature range of 72°F ± 2°F. The material for the insulation chamber is foam insulation sheeting used for building construction.

Fabrication

Nine test stands were fabricated, including five with environmental chambers. Prior to the start of the fabrication process, the following items were addressed:

- All stock material already available was gathered at a single location.
- All necessary hardware and all stock supplies not available on center were ordered in bulk purchases.

- All custom materials with specific heat treatment or hardness were identified early in the process and purchase requests with material qualifications were sent out to appropriate vendors for bids.
- The few custom machined parts were identified, and the tasks scheduled to speed up production when possible.
- Extensive usage of waterjet cutting was applied wherever possible to produce multiple parts in a single production run.
- Sub-assembly parts were fitted and mated before leaving the fabrication area to reduce re-work time.
- Quick design changes during fabrication were captured by engineer red line on production drawing as approved by NASA Langley Research Center (LaRC) engineering drawing revisions. The average fabrication time for each of the nine units was 170 man-hours with an additional 30 hours to manufacturer the weight plates. The plates could be purchased rather than fabricated if necessary, but the materials and fabrication shop hours were available for their production in this case.

Prototype

After the design and analysis phases were complete, a prototype of the creep test stand was built with the following goals:

1. **Design accuracy**. Ensure that the drawing package and design had no errors. The development of the prototype from working drawings verified that all of the drawings and components were accurate, and the mating parts had proper alignment and fit. (App. C).
2. **Material fabrication**. Explore manufacturing issues by reducing complex machining and implementing ways to simplify the fabrication process.
3. **Assembly**. Test out the complex and sometimes arduous assembly steps.
4. **Operation**. Demonstrate a test setup, and how the specimen and its required instrumentation are installed and operate within the constraints of the stand and environmental chamber.
5. **Preliminary load test**. Apply the maximum load of 600 lbs to the lever-arm beam and evaluate the beam deflection as well the performance of each component and the entire structure (Fig. 10).

The prototype met these goals, provided the design team with insight into improving the test stand fabrication, and was critical in developing the assembly and setup operations. No major issues in the design were identified during testing of the prototype.

Setup and Validation

The prototype stand (unit number 1) was used for the initial setup and validation. The first step was to carefully step load the lever-arm beam up to its maximum design load of 12,000 lbs to validate that there were no significant mechanical defects or anomalies. For this setup process, a sample Vectran webbing was used to load the lever-arm system in tension. The test stand and loaded specimen were left in the maximum load configuration over a period of three days until the test strap failed. A follow up inspection showed no observable damage or deformation to the test stand (Fig. 11).

The second step was to test the tensioning mechanism, using the same tensioning approach used for strap installation, to check that the tension in the strap is set prior to performing a creep test and observe if any of the installation steps created any technical or safety issues. This process proved that the current configuration was working as intended and no modifications were required (Fig. 12).

The third and final step was to setup the test stand with its environmental chamber and monitor the thermal performance. The chamber proved to be capable of maintaining the desired set point temperature range of 72°F ± 2°F.

Testing

The creep test stand is designed to test a 48-inch long, high-strength softgoods specimen with an allowed stretch of up to 1.69-inches. A suite of sensors are attached inline or in proximity to each specimen to record displacement, load, temperature and humidity. Displacement data is collected by using dual string potentiometer (string pot) sensors attached at the pin grips. Load is measured using a 25-kip inline load cell at the base. Temperature and humidity are measured using three thermocouples and a high-resolution temperature-humidity probe in the environmental chamber, and the same type of probe is used to measure the room temperature and humidity. A data logger collects and stores all sensor data at a rate of once every five minutes and can be accessed remotely via a network connection.

All the weight plates are cut from the same uniform steel and their weights individually recorded. Weights are added to the test stand until the inline load cell reads the desired test load. Smaller weights can be used for fine adjustment to a specified level if needed. The lever-arm beam is raised to its highest up-angle initially to allow for the most stretch in the sample as it is loaded. The specimens are preloaded five times to 25% of their ultimate load to remove most of the initial stretch, and to emulate inflation cycles on a habitat. The specimens are cut slightly shorter initially so that the final constructed length from loop to loop is 48-inches. After preconditioning, the sample is loaded on to the pin grips, and the lever-arm beam start angle is measured. An example plot of the displacement data collected from one of the initial tests is presented in Figure 13. The setup procedure for a creep test is provided in Appendix C.

Concluding Remarks

The low-cost, lever-arm softgoods creep test stand successfully demonstrates a small footprint, easy to fabricate creep test fixture that can be used to gather lifetime data on high-strength softgoods webbing and cordage. The design and plans for this test stand are intended to be adaptable, and provide a guide to building a robust creep fixture. Companies seeking to gather creep data for inflatable structures, or other softgoods applications, could adopt test stands similar in size, capability and cost.

References

[1]Budynas, Richard G, J K. Nisbett, and Joseph E. Shigley. *Shigley's Mechanical Engineering Design 5th Edition*. New York: McGraw-Hill, 2002. Print.

[2]National Aeronautics and Space Administration, *Structural Design and Test Factors of Safety for Spaceflight Hardware*, NASA-STD-5001B w/Change 2, October 5th, 2016.

Figure 1. Expandable habitat with inflatable structural restraint layer visible in center section.

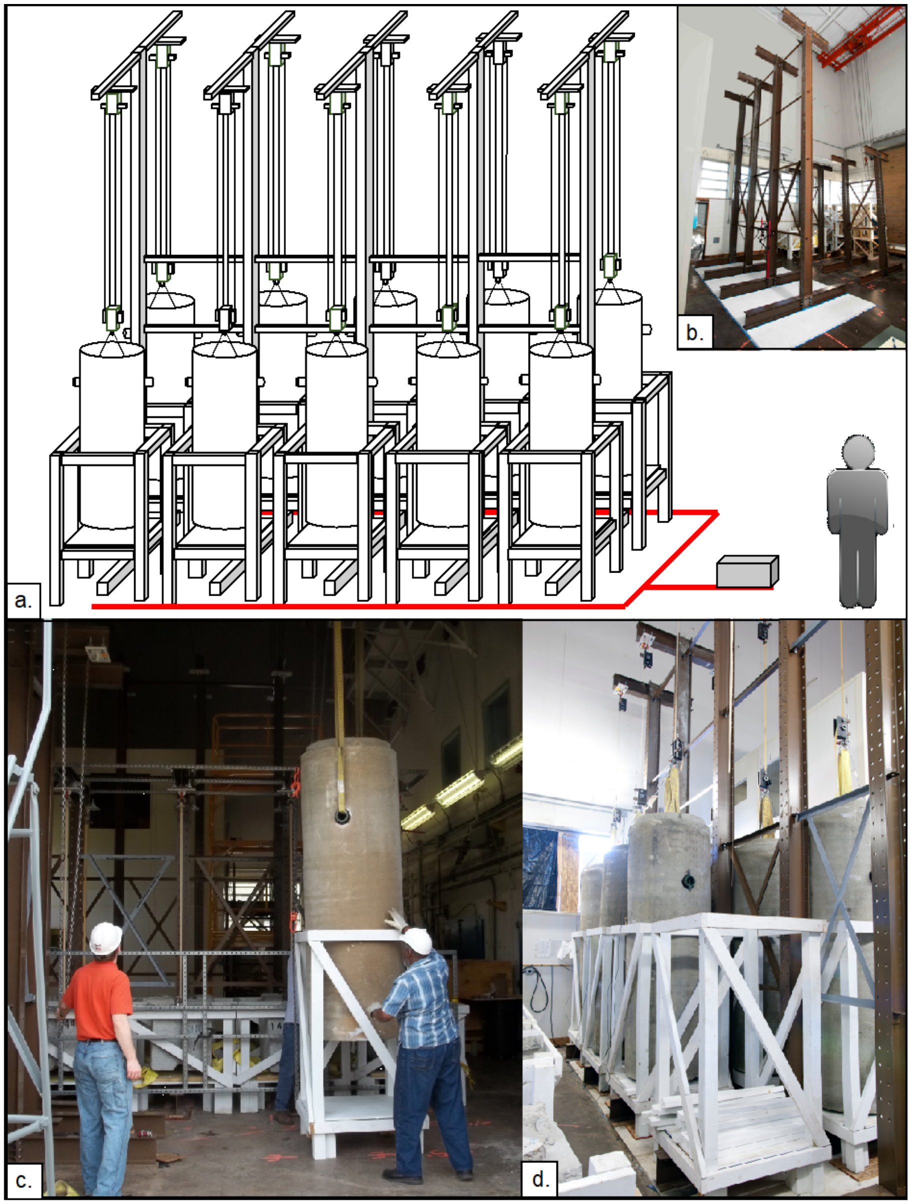
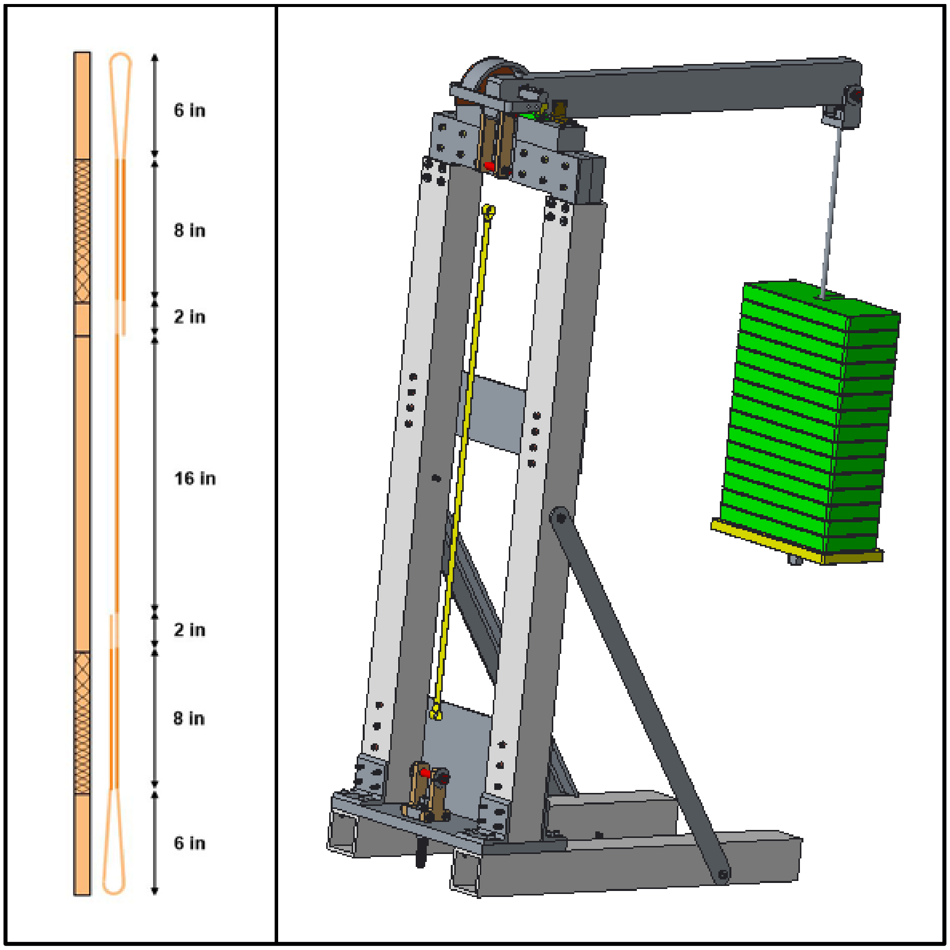


Figure 2. Typical high volume, dead-load creep test setup.



(a) **(b)**

Figure 3. a) 48-inch strap specimen, b) and low-cost, lever-arm creep test stand

Figure 4. Creep test environmental chamber.

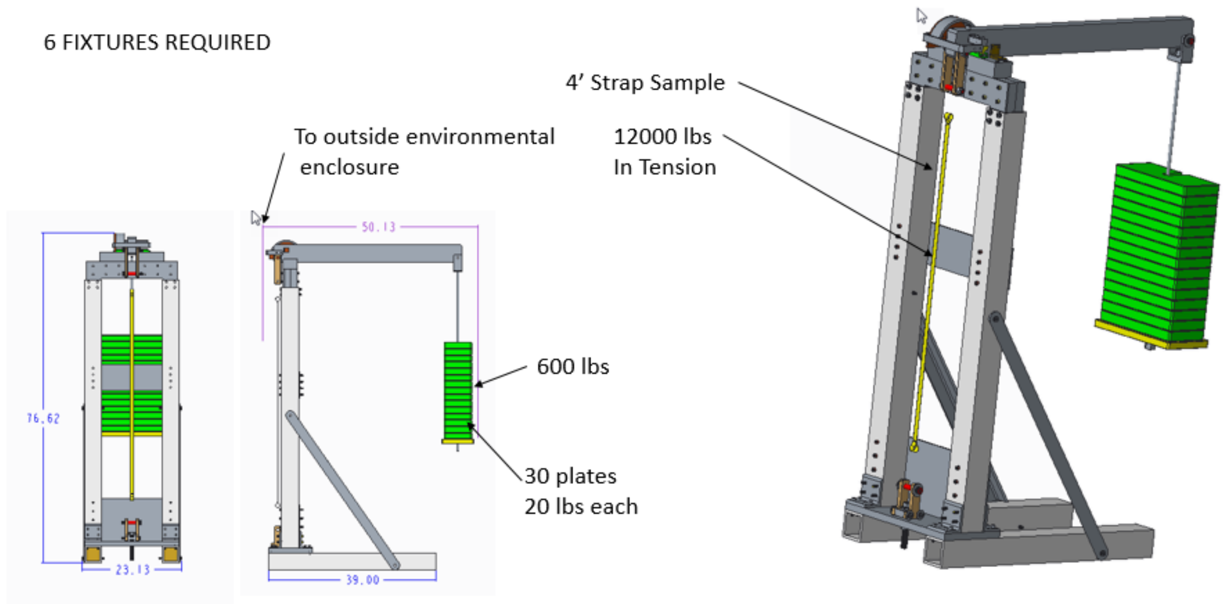


Figure 5. Creep stand overview.

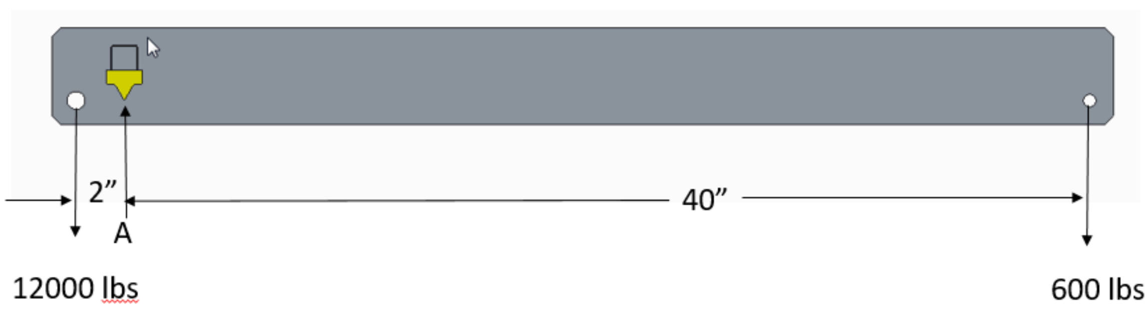


Figure 6. Creep test lever arm with 20:1 mechanical advantage.

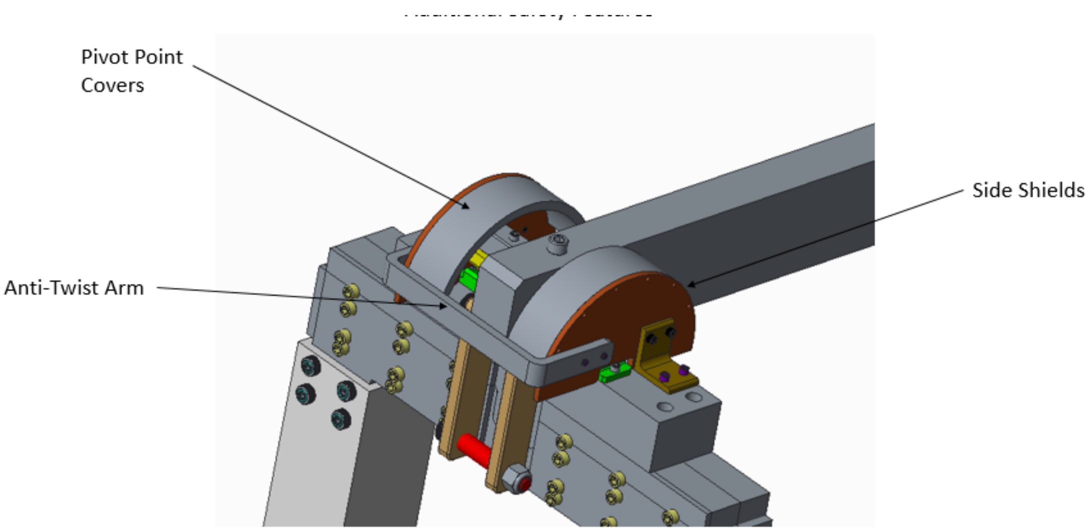
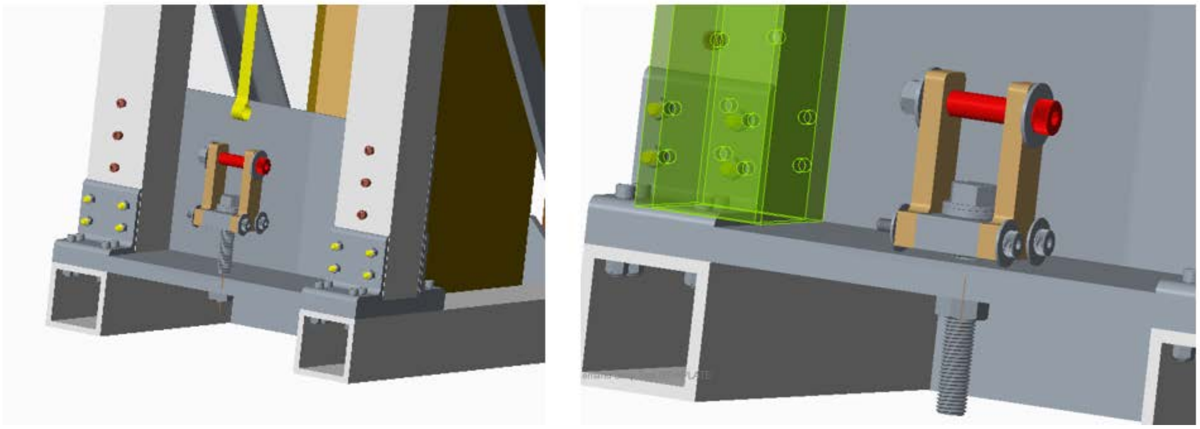


Figure 7. Creep stand cantilever beam fail-safe setup.

Figure 8. Thermal enclosure with ductwork.



Lower mount at maximum height Lower mount at minimum height

Figure 9. Lower mount tensioning fixture detail.

Figure 10. Maximum load test on prototype frame.

Figure 11. Creep stand under tension load evaluation.

Figure 12. Creep stands with specimens under tension generating data.

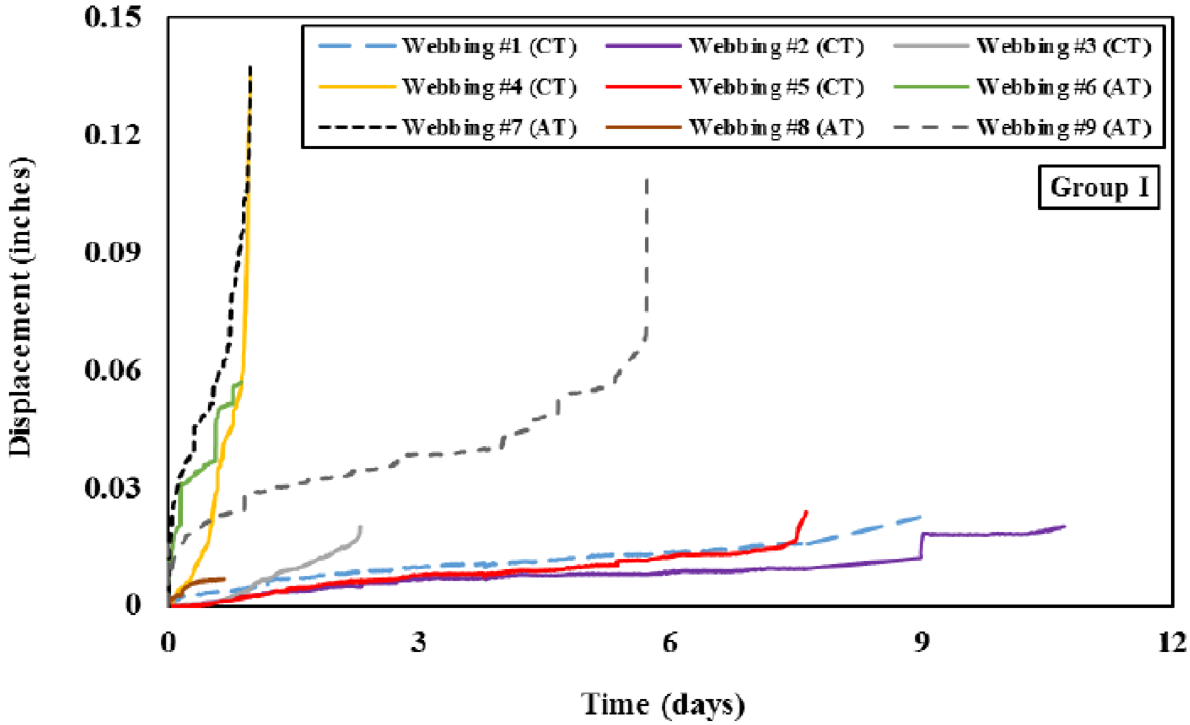


Figure 13. Example creep test displacement versus time plot.

Appendix-A

Structural Analysis

All items in this Appendix refer to the circled numbers in Figure 1-A and Table 1-A.

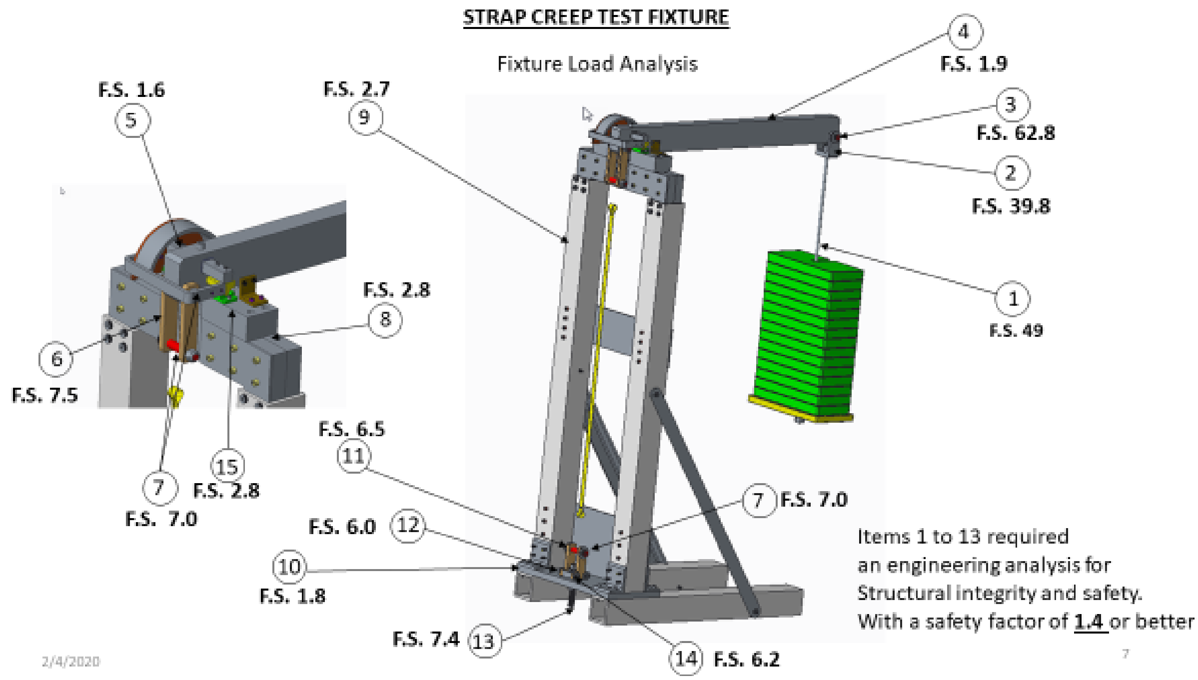


Figure 1-A. Safety factors on structural components determined via analysis.

Item	Part Name	Max Load (lbs)	Max Stress (psi)	Loading Type	F.S.
1	Threaded Rod	600	3055	Tension	42.6
2	Yoke	600	3638.6	Tension	39.9
3	Shoulder Bolt	600	~	Double Shear	141.5
4	Cantilever Beam	500	23375	Bending	1.9
5	Cantilever Mount	12649	89221	Bending	1.6
6	Strap Swing Plate	6000	19278.9	Tension	7.5
7	Shoulder Bolt	12000	~	Double Shear	7.1
8	Cross Beam (Single Case)	12649	15609	Bending	2.9
9	Main Frame	12049	16452.7	Compression/Buckling	2.7
10	Base Plate	12000	79280	Bending	1.8
11	Side Plates	6325	22142.6	Tension	6.5
12	Mount Block	12000	23818.7	Tension	6.1
13	Lower Threaded Rod	12049	20000	Tension	6.5
14	Shoulder Bolt	6000	~	Double Shear	6.3
15	Pivot Joint	6324	60233	Compression	2.8

Table 1-A. Analysis results for individual components in creep test stand.

Item 1 - Threaded Rod (P.N. 90322A158)
The threaded rod (Fig. 2-A) is analyzed for its ability to hold the weights to the mount at the end of the cantilever beam while in tension.

The equation for this analysis case is: $\sigma = F/A$
Where:

σ = Yield strength of material.
F = maximum load in lbs.
A = cross section area of rod in^2

Material: ½-13 inch, UNC high-strength threaded rod RC33 with 130 ksi yield strength.

Figure 2-A. Threaded rod detail.

Nominal area, A, of threaded rod = $\pi \times r^2 = 3.14 \times (0.25\text{ in})^2 = 0.196\text{ in}^2$
Load, F, applied to rod in tension = 600 lbs.
Tensile stress on the rod due to the load F/A = 600 lbs / 0.196 in^2 = 3055 psi.

<u>Factor of Safety (F.S.) = 130000 psi / 3055 psi = 42.6</u>

Item 2 - Weights Yoke (P.N. 1299321-1)
The yoke (Fig. 3-A and 4-A) is analyzed for its ability to react to the 600 lbs of weights at the end of the cantilever beam. This is done using an FEA model (Fig. 5-A and 6-A) with constraints at the location were the yoke is bolted to the beam, and with the load acting at the location were the yoke mounts to the threaded rod.

Material: 17-4 stainless steel, H-1025 ≤ 4.00-inch. 145 ksi yield strength.

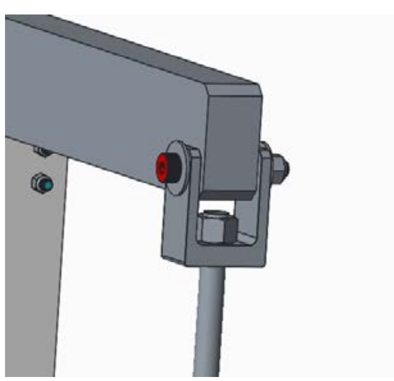


Figure 3-A. Yoke assembly details.

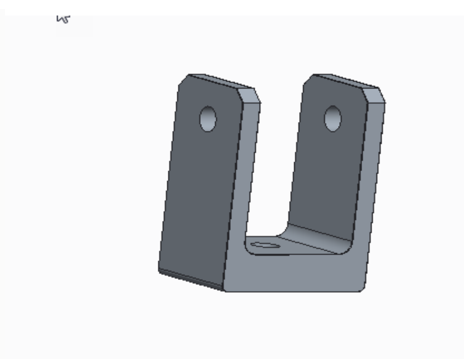


Figure 4-A. Weights yoke.

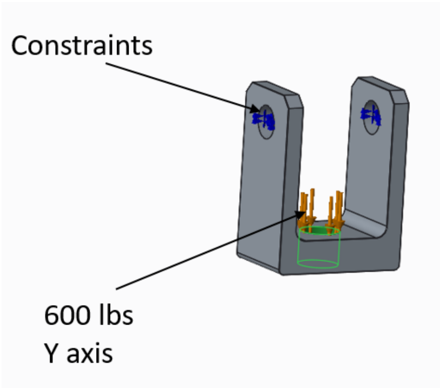


Figure 5-A. Loading and constraints.

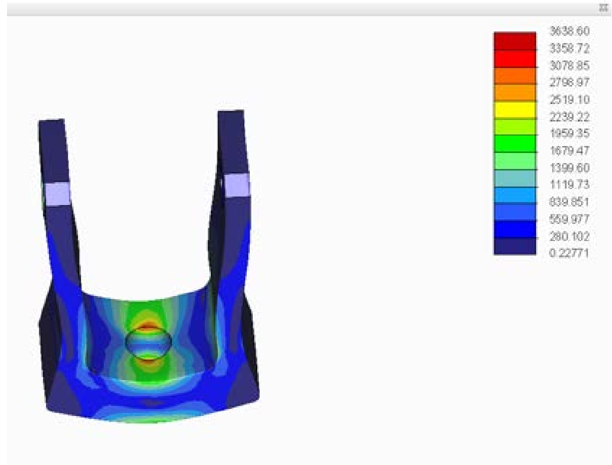


Figure 6-A. FEA stress resultants.

Component loaded with 600 lbs in the Y-axis, perpendicular to the ground.
Maximum Von Mises stress 3638.6 psi.

F.S. = 145 ksi / 3638.6 psi = 39.9

Item 3 - Shoulder Bolt ¾-inch **Diameter. (P.N.91259A848)**
The shoulder bolt (Fig. 7-A) retains the yoke to the cantilever beam. The bolt is loaded in shear across its body section.

Material: ¾-inch diameter steel bolt. 42440 lbs single shear strength, 84880 lbs double shear.

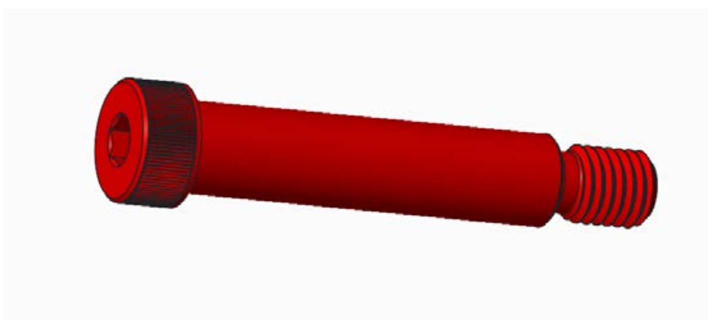


Figure 7-A. Shoulder bolt, ¾-inch **diameter.**

Maximum load is 600 lbs in double shear.

F.S. = 84880 lbs / 600 lbs = 141.47

Item 4 - Cantilever Beam (P.N.1299320-1)
The cantilever beam is designed to provide a load ratio of 20:1, with 600 lbs of weight on one end generating a maximum tension in the specimen of 12,000 lbs. The beam is analyzed for bending stresses and deflection using FEA.

Material: AL-2024-T3, 1.50-2.00-inch thick. 45 ksi yield strength.

Figures 8 to 10 below show how the FEA load case was set up with a 600 lbs load at the yoke mount position, the beam constrained at the mount socket, and the stress results.

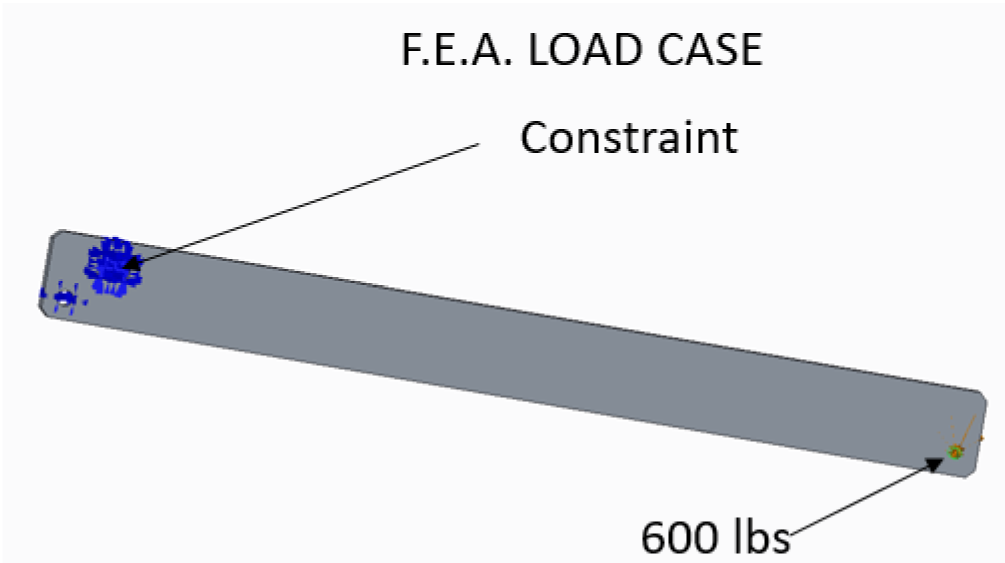


Figure 8-A. Cantilever beam loading and constraints.

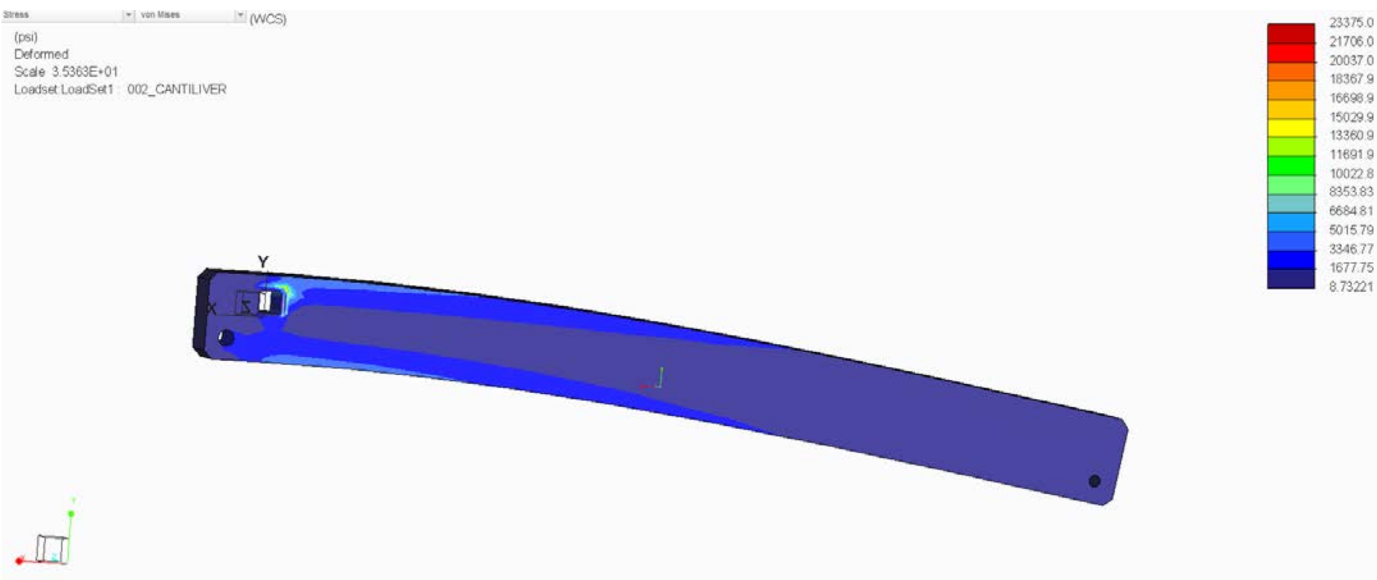


Figure 9-A. Cantilever beam FEA results.

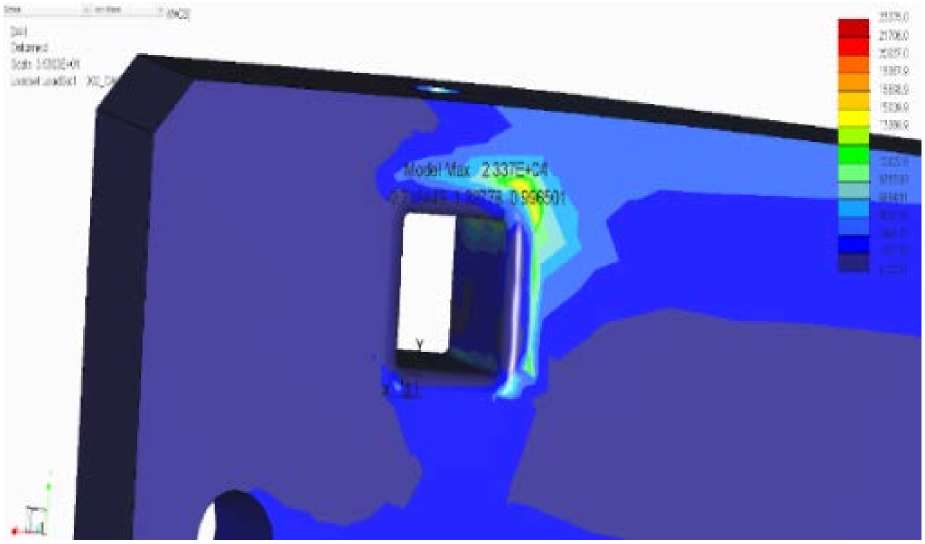
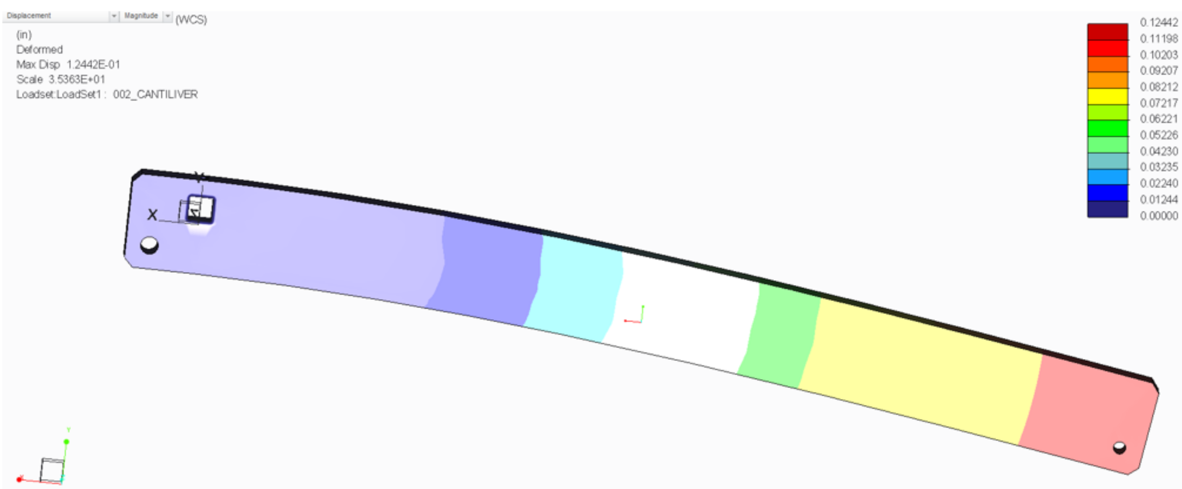


Figure 10-A. Cantilever beam FEA results detail.

The analysis model shows a maximum Von Mises stress of 23375 psi in the mount socket.

<u>F.S. = 45000 psi / 23375 psi = 1.93</u>

<u>STRAP CREEP TEST FIXTURE</u>
Item 4 Cantilever Beam



MAX DEFLECTION = **<u>0.125”</u>**

Figure 11-A. Cantilever beam bending deflection.

The cantilever beam under maximum load (Fig. 11-A) will have a maximum deflection of 0.125 inches. This was acceptable for the requirements of the creep stand.

Item 5 - Cantilever Mount Beam (P.N.1299318-1)
The cantilever mount (Fig. 12-A to 14-A) is designed to allow the cantilever beam to sit on two lateral pivoting points. The full load from the cantilever beam is transferred to the mount.

Material: Stainless steel 17-4 H1025 < 4.0-inch thick. With a 145 ksi yield strength.

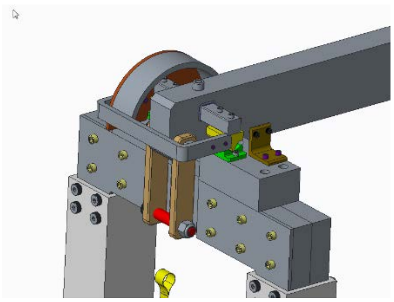


Figure 12-A. Cantilever mount, assembly detail.

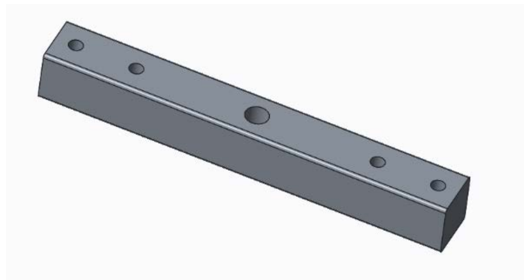


Figure 13-A. Cantilever mount.

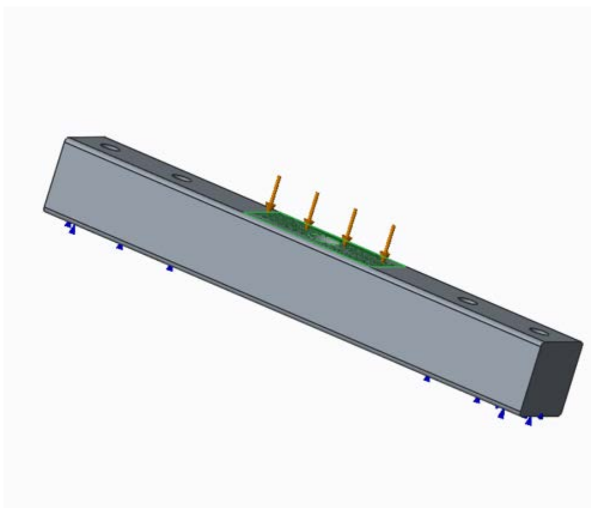


Figure 14-A. Loading and constraints.

Cantilever mount loading of 12649 lbs (total load = applied tension load + hardware weight + loads weights) in the negative Y-axis direction. Load is reacted at the mounting location of the male pivoting joint.

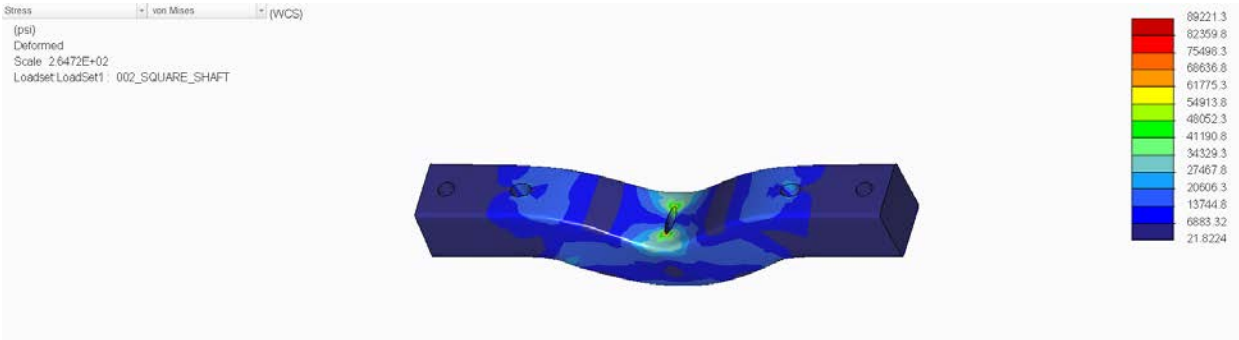


Figure 15-A. Cantilever mount FEA stress results.

The analysis model (Fig. 15-A) shows a maximum Von Mises stress of 89221 psi in the mount.

<u>F.S. = 145000 psi / 89221 psi = 1.63</u>

Item 6 - Specimen Swing Plate (P.N.1299322-1)

The swing plate is the connection point between the cantilever beam and the test specimen. Two plates are used in the assembly and each plate carries half of the total tensile load.

Material: Stainless steel 17-4 H1025 < 4.0-inch thick. With a 145 ksi yield strength.

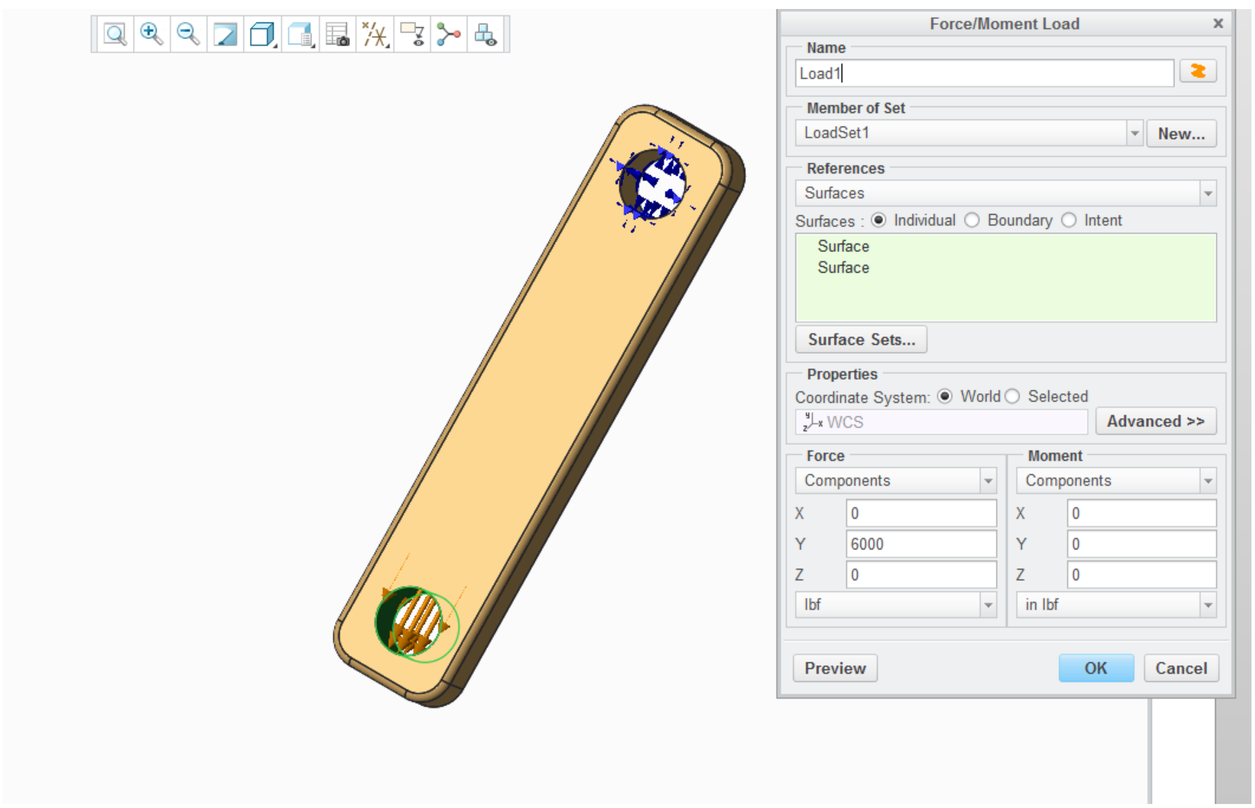


Figure 16-A. Specimen swing plate loading and constraints.

The swing plates are modeled with surface constraints inside the hole (Fig. 16-A) that connects the cantilever beam to the plates via the shoulder bolt and are loaded at the opposite end with 6000 lbs at the location of the shoulder bolt that the specimen attaches to.

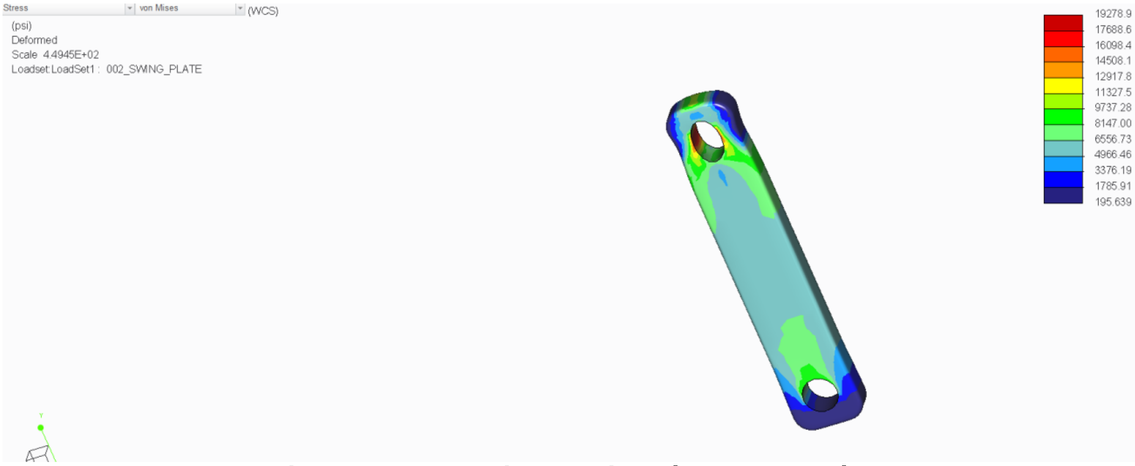


Figure 17-A. Specimen swing plate FEA results.

The FEA (Fig. 17-A) shows a maximum Von Mises stress of 19278 psi.

F.S. for each swing plate is: <u>F.S = 145000 psi / 19278.9 psi = 7.52</u>

Item 7 - Shoulder Bolt ¾-inch Diameter (P.N.91259A848)
The shoulder bolt retains the yoke to the cantilever beam. The bolt is loaded in shear across its body section (Fig. 18-A).

Material: ¾-inch diameter, 42410 lbs single shear strength, 84880 lbs double shear.

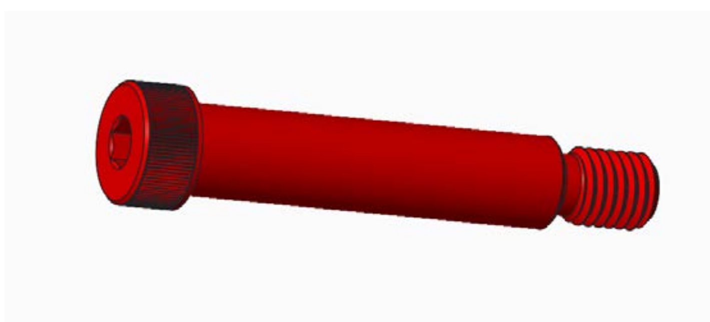


Figure 18-A. Shoulder bolt ¾-inch d**iameter.**

Maximum load = 12000 lbs in double shear.

<u>F.S. = 84880 lbs / 12000 lbs = 7.07</u>

Item 8 - Crossbeam with Mount / Rear Crossbeam (P.N.1299318-1/-2)
The main frame of the creep test stand is connected at the top by a crossbeam (Fig. 19-A to 21-A). This structural element is designed to hold the cantilever beam and provide a rigid surface for

mounting the pivot elements. The loading condition for this component is the same as item 5, 12649 lbs.

Material: AL-2024-T3 < 1.50-inch thick. With a 45 ksi yield strength.

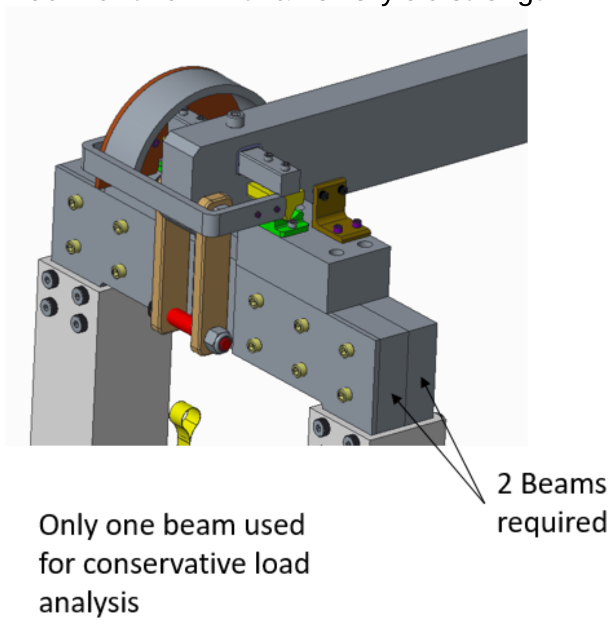


Figure 19-A. Crossbeam with mount assembly detail.

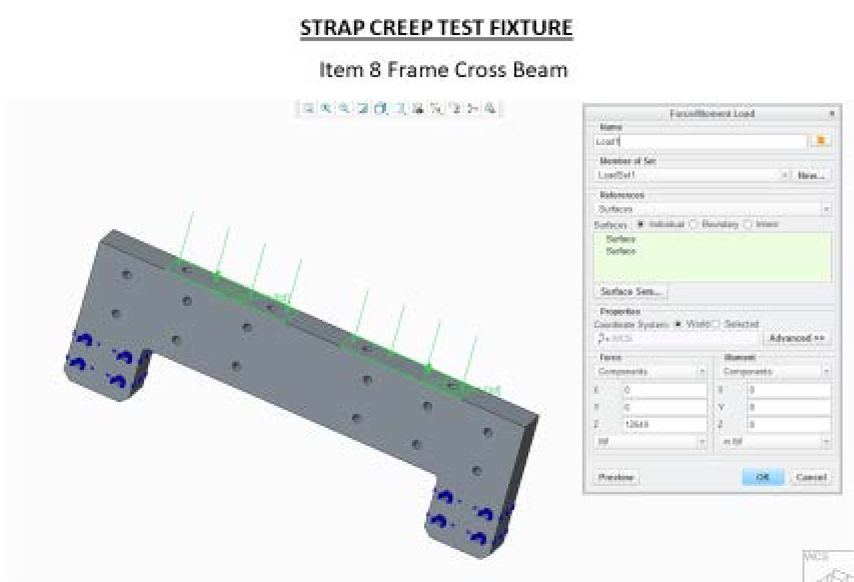


Figure 20-A. Single crossbeam loading and constraints (Single element for conservative analysis) two elements used.

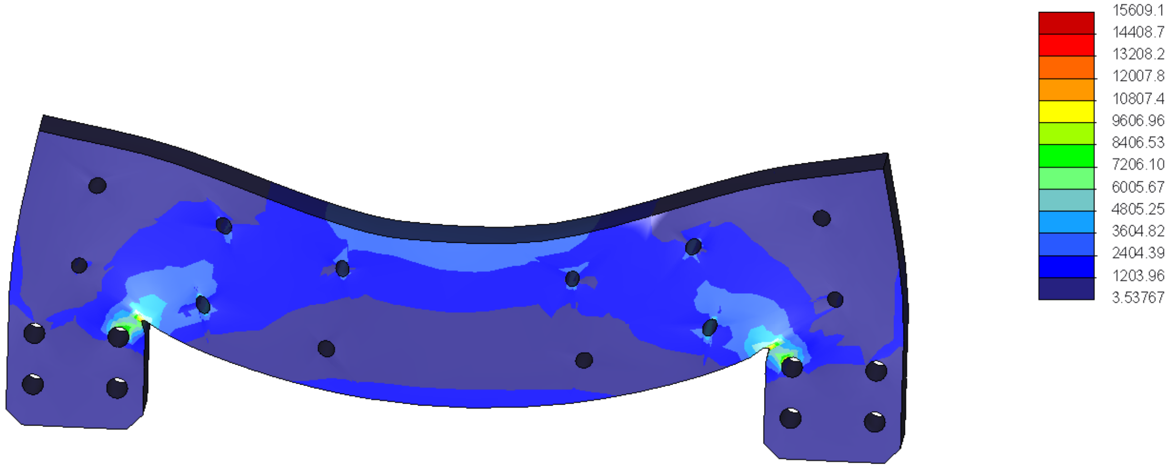


Figure 21-A. FEA stress and deflection results.

The FEA produces a maximum Von Mises stress of 15609 psi.

The F.S. for a crossbeam acting alone is: F.S. = 45000 psi / 15609 psi = 2.88

Item 9 - Main Frame (P.N. 1299307-1)

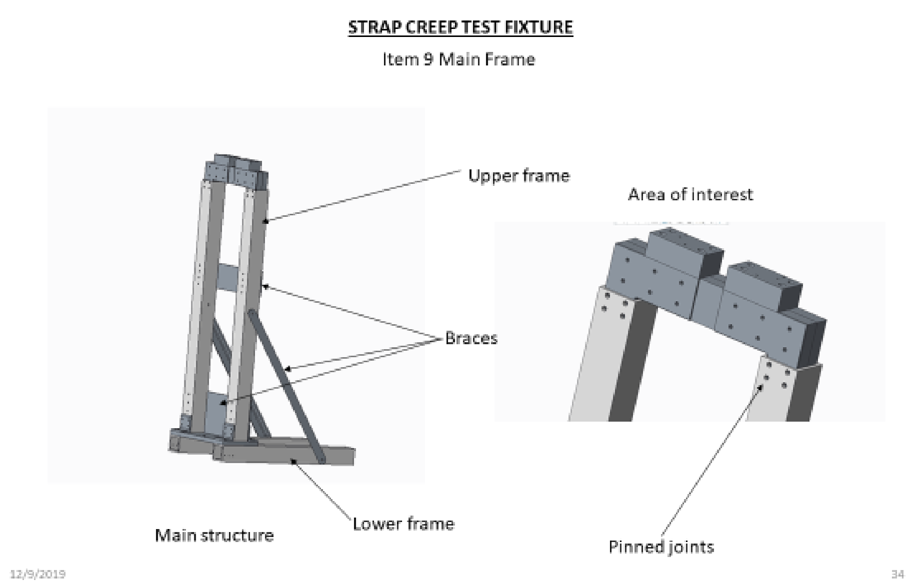


Figure 22-A. Main frame and crossbeam detail.

Materials: The structural assembly of the main frame (Fig. 22-A and 23-A) uses two materials. The side columns and crossbeam plates are AL-2024-T3 with a 45 ksi yield strength. The base plate that connects to the two side columns, and is also used as the mount for the lower yoke and threaded rod, is made from stainless steel 17-4 H 1025 < 4.0-inch thick, 145 ksi yield strength.

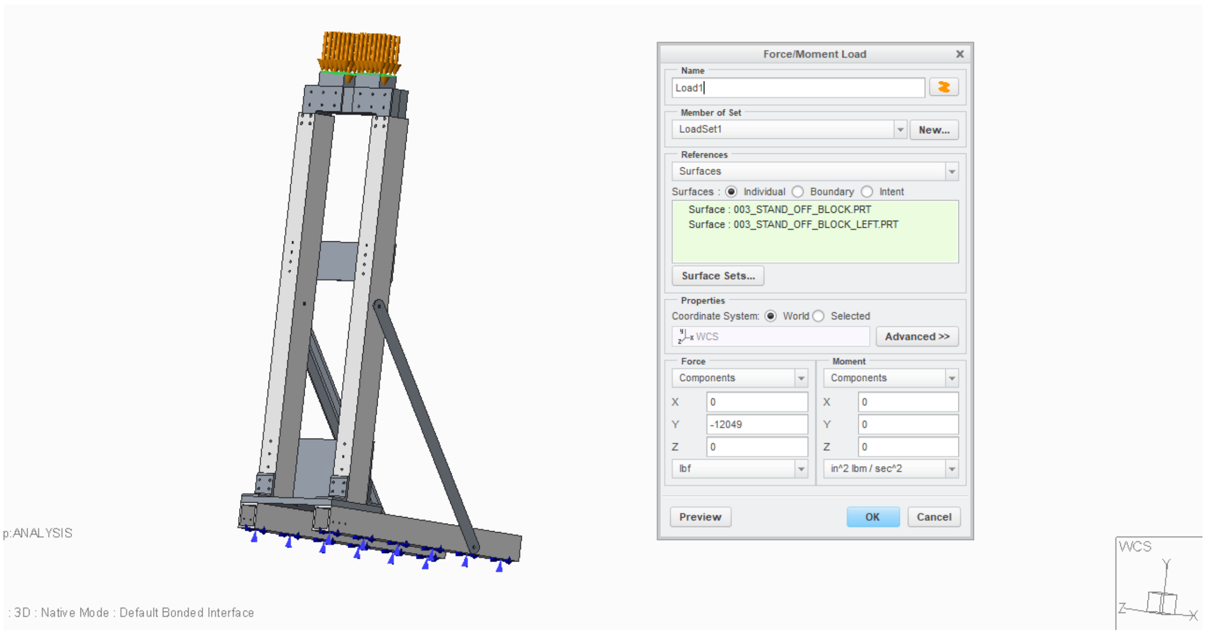


Figure 23-A. Main frame loads and constraints.

Figures 24-A and 25-A show how the frame reacts to the loads induced by the cantilever beam. The full load from the cantilever beam goes through the beam mount and the pivot points. The load is then transferred into the crossbeam. The crossbeam load in turn is reacted by the two side columns and transferred to the ground via the base frame.

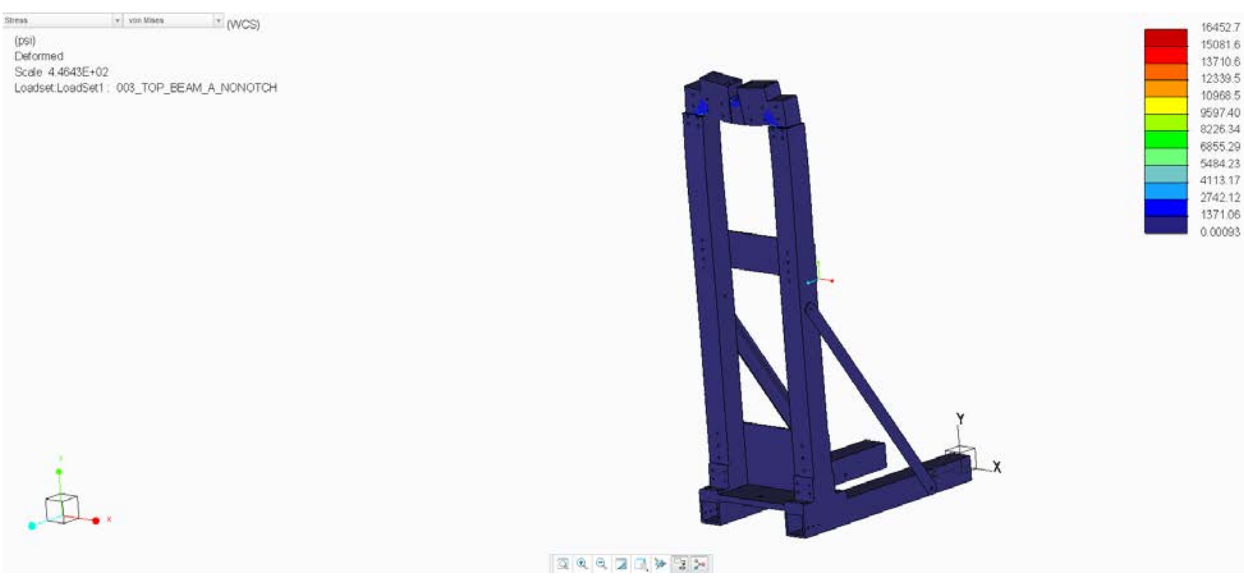


Figure 24-A. FEA results.

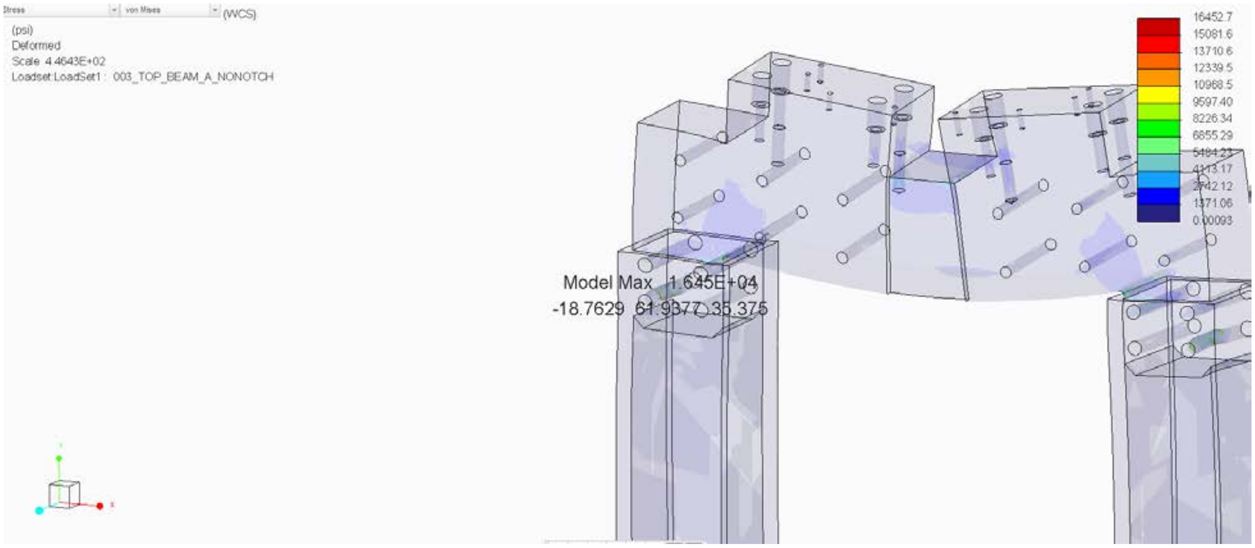


Figure 25-A. FEA results detailed view.

From the FEA, the maximum Von Mises stress is 16452 psi.

The F.S. for the main frame: F.S. = 45000 psi / 16452 psi = 2.74

The columns of the main frame structure are also analyzed for buckling loads. Initially a closed form Euler buckling calculation is presented that assumes a uniform distribution of the load on each column. This is followed by FEM buckling analyses of the uniform loading (Fig. 26-A) and a non-uniform edge loading (Fig. 27-A) to emulate the worst case loading condition.

Closed-form column buckling calculation.

Buckling Load, $P_{cr} = \pi^2 EI\ /\ (KL)^2$

Square tube $I = (B_o^4 - B_i^4)/12$ = [(4.00) ^4 – (3.25) ^4]/12 = 12.036 in^4
E = 10E6 Psi
L = 61 inches
K (Boundary condition) = 2.1

P_{cr} (K=2.1) = (3.14^2) (10E6) (12)/ ((2.1) (61)) ^2 = 72392 lbs.

Maximum applied load = 6,000 lbs.
F.S. = 72392 lbs / 6000 lbs = 12.1 (with uniform load)

FEA Results: Uniform Loading: F.S. = 80100 lbs / 6000 lbs = 13.3
Non-uniform Loading: F.S. = 77483 lbs / 6000 lbs = 12.9

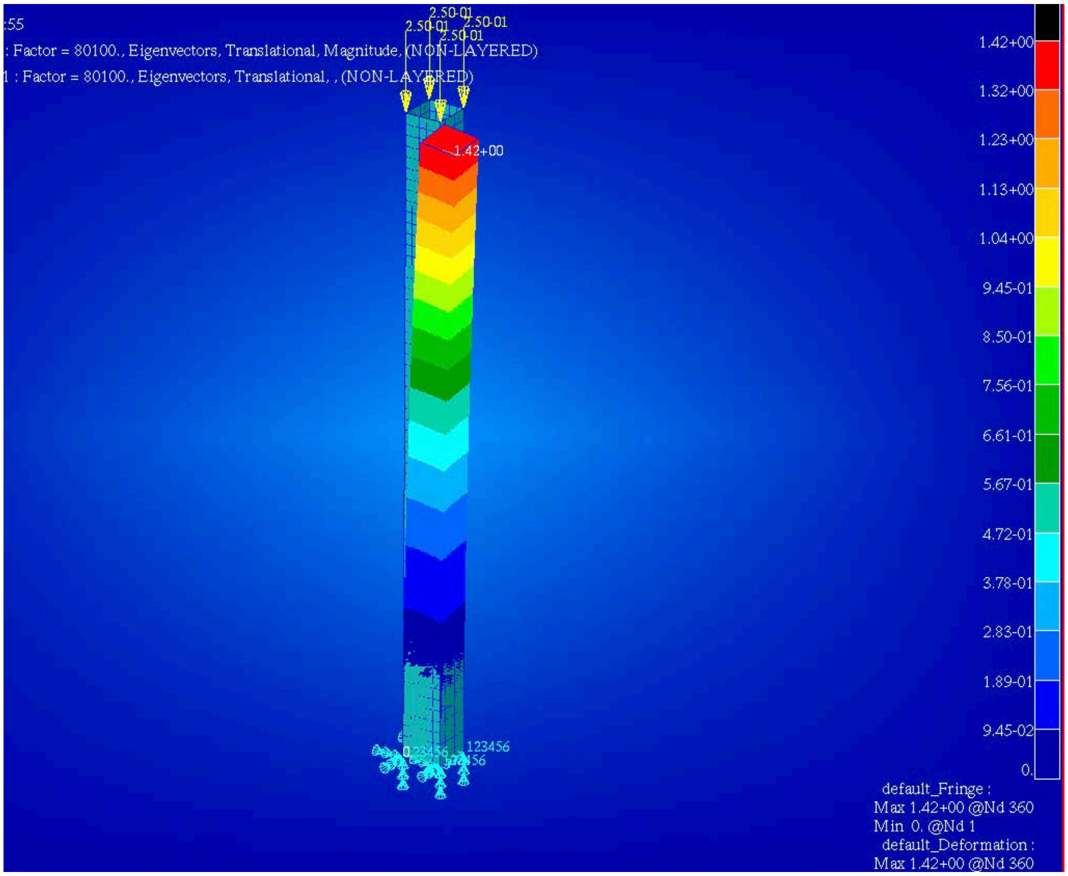


Figure 26-A. Column buckling analysis.

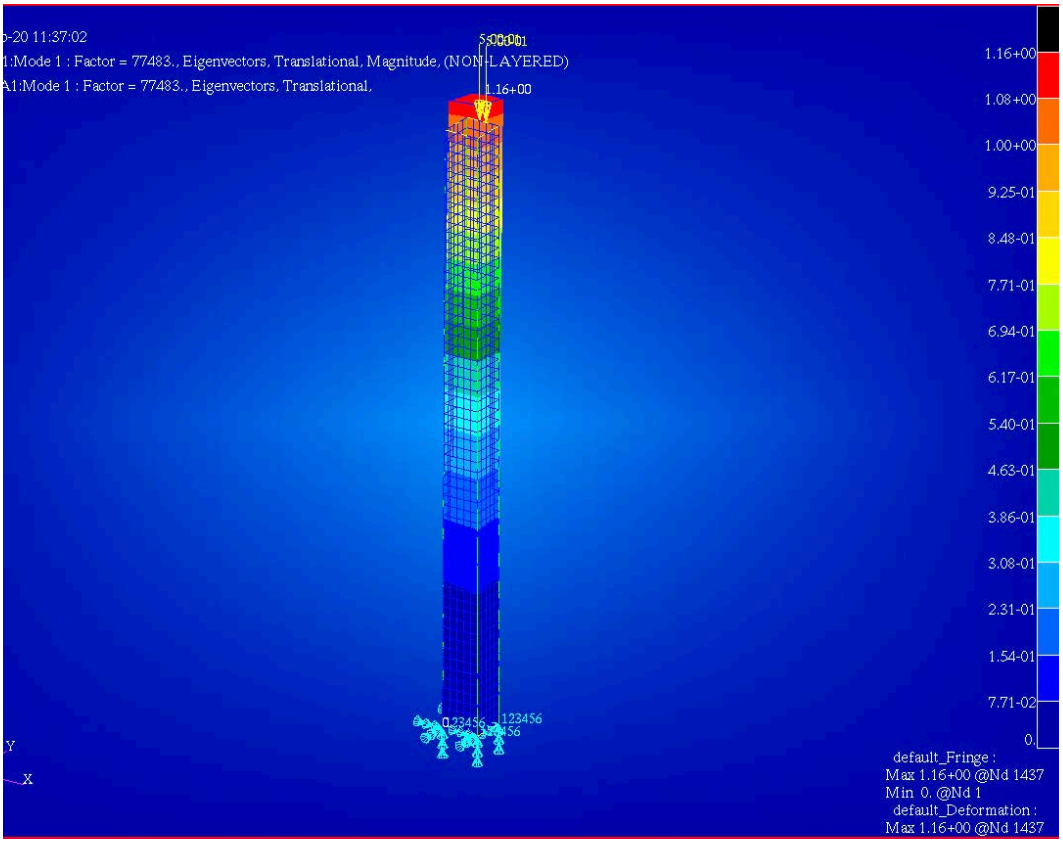


Figure 27-A. Non-uniform column buckling analysis model results.

Item 10 - Base Plate (P.N. 1299308-1)

The base plate (Fig. 28-A) is designed to provide a lower rigid connection to the frame structure, and serves as the attachment point for the lower pin grip. The base frame must react the maximum load with minimal deformation.

Material: Stainless steel 17-4 H1025 < 4.0-inch thick. With a 145 ksi yield strength.

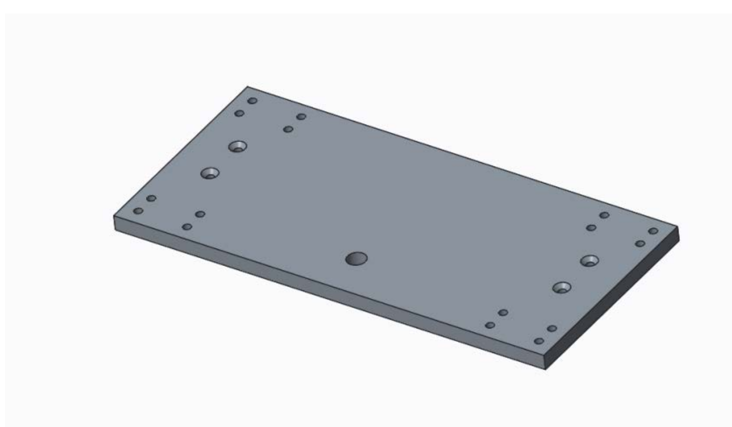


Figure 28-A. Base plate.

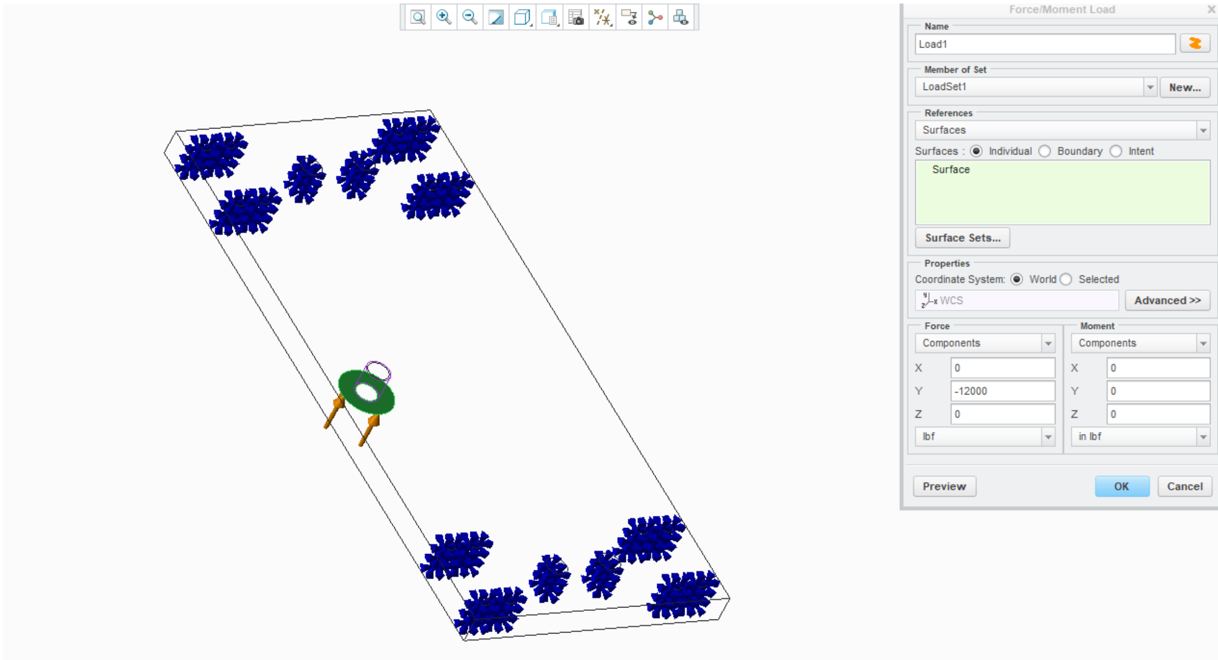


Figure 29-A. Base plate FEA constraints and load point.

For this analysis the base plate is constrained at the bolt hole locations (Fig. 29-A) where the plate is secured to the rest of the frame. The load is applied at the attachment location of the specimen lower mount that reacts a 12,000 lbs upward load (Fig. 30-A).

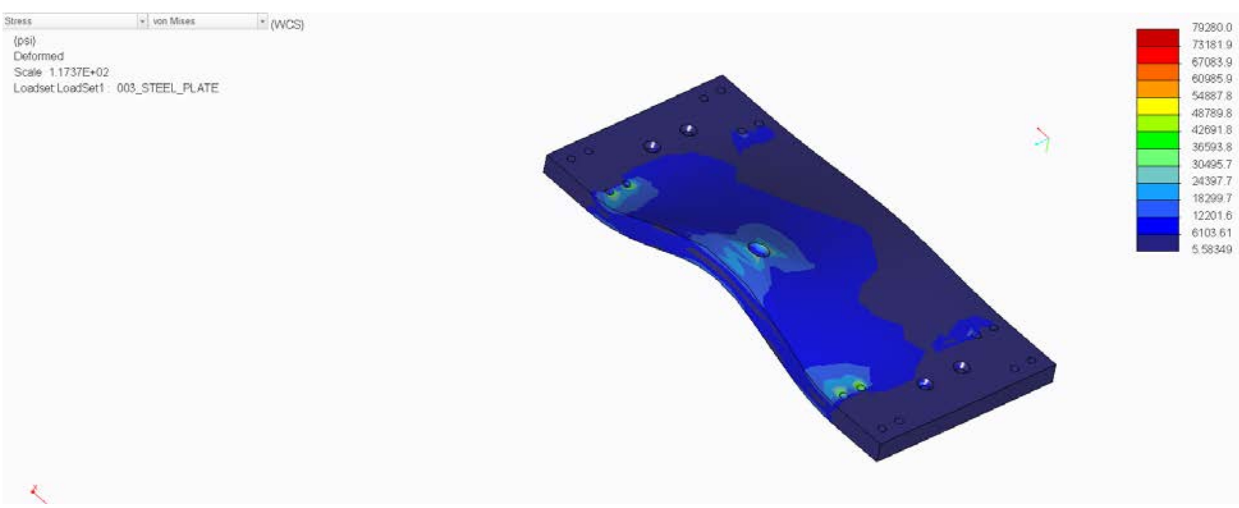


Figure 30-A. Base plate FEA results.

Maximum von Mises stress 79280 psi.

<u>F. S. = 145000 psi / 79280 psi = 1.83</u>

Item 11 - Lower Test Strap Mount Side Plate (P.N. 1299313-2)

The lower test strap mount (Fig. 31-A) is designed to mount the specimen to the base plate, while allowing for tensioning of the specimen, prior to creep testing. These plates allow the specimen to be secured at the lower end with a shoulder bolt. The center block, in conjunction with a threaded rod, allows the lower mount to be tensioned and to react the induced loads.

Material: Stainless steel 17-4 H1025 < 4.0-inch thick. With a 145 ksi yield strength.

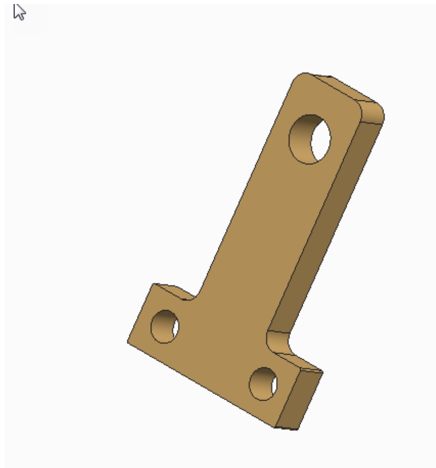


Figure 31-A. Lower test strap mount side plate (2 required).

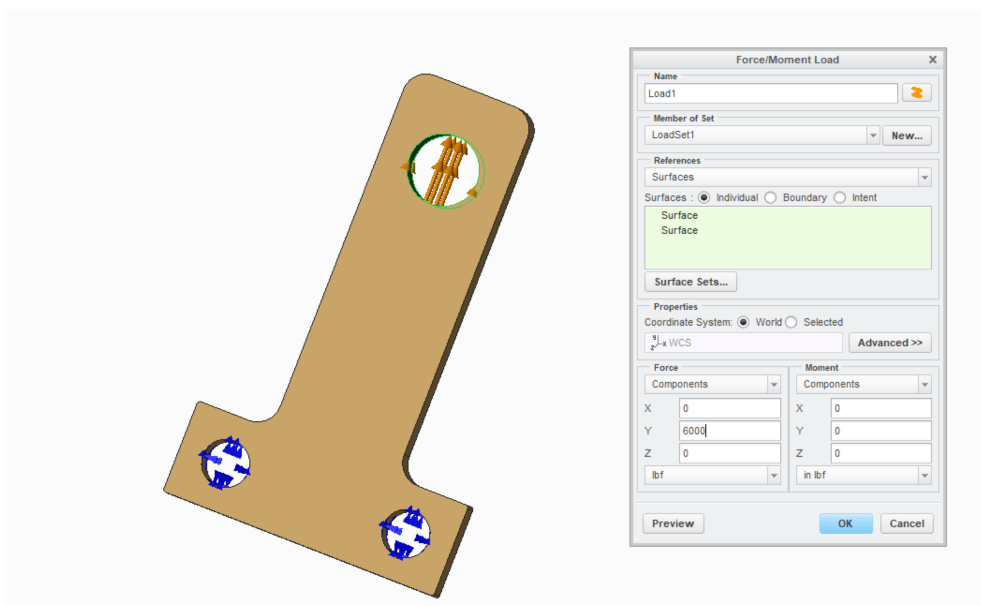


Figure 32-A. Side plate constraints and loads.

The side plate is constrained at the lower end where the plate bolts to the lower mount block. The side plate is loaded at the top hole where the shoulder bolts retain the specimen (Fig. 32-A). Since two side plates are required for the assembly each side plate is subjected to half of the maximum test load of 12,000 lbs.

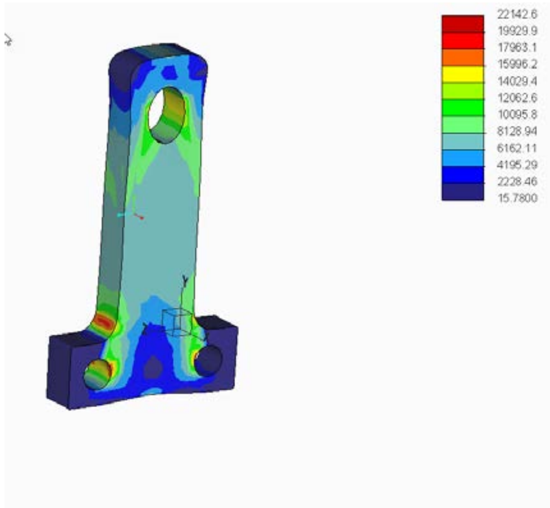


Figure 33-A. Side plate FEA results.

Maximum load condition = 6000 lbs. The analysis (Fig. 33-A) shows a maximum Von Mises stress = 22142.6 psi

<u>F. S. = 145000 psi / 22142.6 psi = 6.55</u>

Item 12 - Mount Block (P.N. 1299313-1)
The lower mount block (Fig. 34-A) connects the two mount side plates and reacts the 12,000 lbs load into the lower threaded rod and base plate.

Material: Stainless steel 17-4 H1025 < 4.0-inch thick. With a 145 ksi yield strength.

Figure 34-A. Mount block (lower mount).

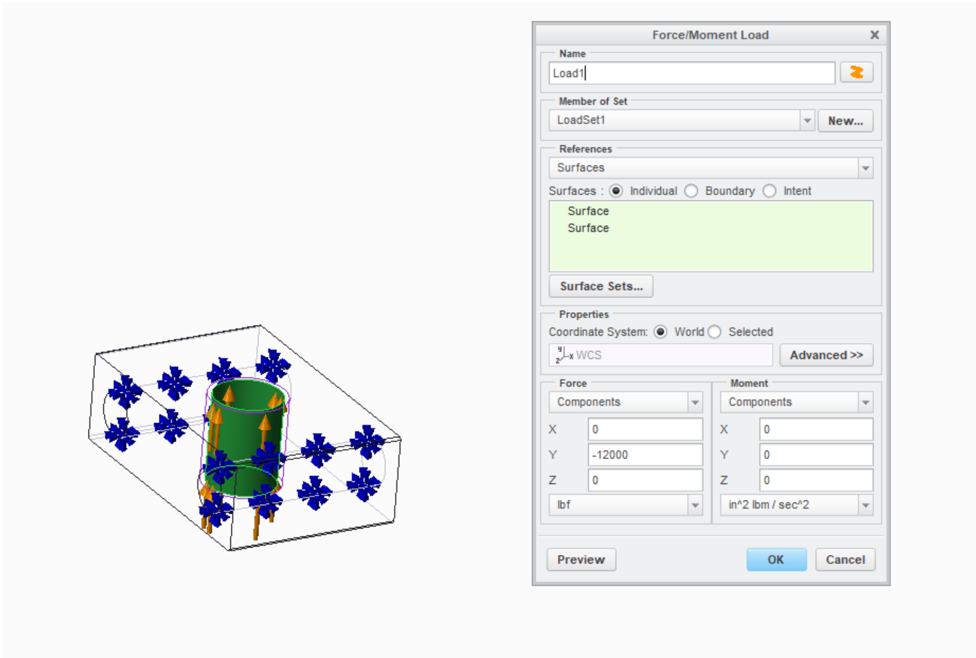


Figure 35-A. Loading configuration.

The mount block is constrained at the bolt location of the side plates. The mount block reacts the loads through the center where the threaded rod is connected (Fig. 35-A and 36-A).

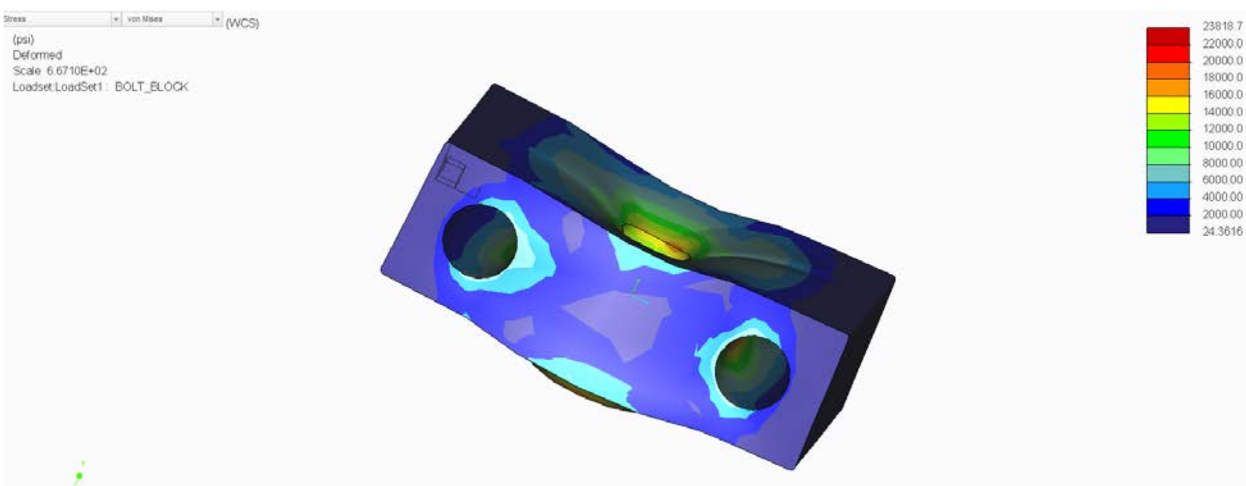


Figure 36-A. Mount block FEA results.

Maximum Von Mises stress is 23818.7 psi.

F. S. = 145000 psi / 23818.7 psi = 6.09

Item 13 - Lower Threaded Rod 7/8”-9 UNC (P.N. 90322A203)
The threaded rod (Fig. 37-A) is analyzed for its ability to sustain the load passing through the lower mount into the base plate. Equation 1 for critical stress is used.

Material: 7/8-9-inch diameter. UNC high-strength threaded rod RC33, 130 ksi yield strength

Figure 37-A. Threaded rod detail.

A (nominal area of threaded rod) = $\pi \times r^2 = 3.14 \times (0.875 / 2)^2 = 0.60$ in^2
Load applied to rod in tension: 12,000 lbs.
Tensile stress on the rod due to the load F/A = 12000 lbs / 0.60 in^2 = 20000 psi.

F.S. = 130000 psi / 20000 psi = 6.50

Item 14 - Shoulder Bolt ½-inch diameter. (P.N. 91259A848)
The shoulder bolts (Fig. 38-A) connect the side plates and lower mount block. The bolts are loaded in double shear.

Material: ½-inch diameter, 18850 lbs single shear, 37700 lbs double shear strength.
Maximum Load 12,000 lbs, distributed over two bolts. 6000 lbs per bolt in double shear.

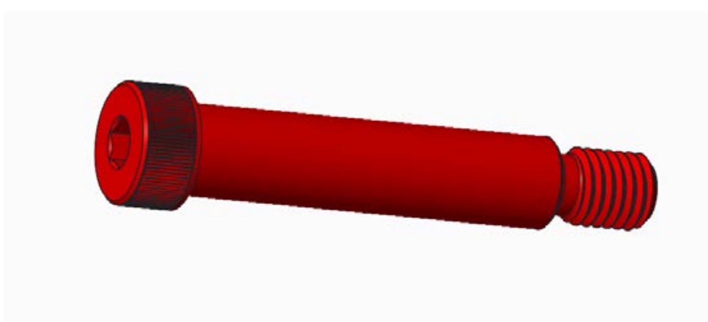


Figure 38-A. Shoulder bolt ½-inch diameter.

F.S. = 37700 lbs / 6000 lbs = 6.28

Item 15 - Male Pivot Joint (P.N. 1299319-2)
Two male pivot blocks are mounted to the cantilever beam (Fig. 39-A) and provide a very low friction, knife-edge for the beam to rotate about. The pivot knife-edge (Fig. 40-A) is located along the same axis that passes through the strap upper mount bolt hole and the weight yoke mount bolt hole. This maintains the distance ratio of 20:1 as the beam rotates. The male pivot is analyzed for a compression load. The analysis is intended to see if any permanent deformation is induced to the component Knife-edge under test loads.

Figure 39-A. Male pivot joint attached to cantilever beam.

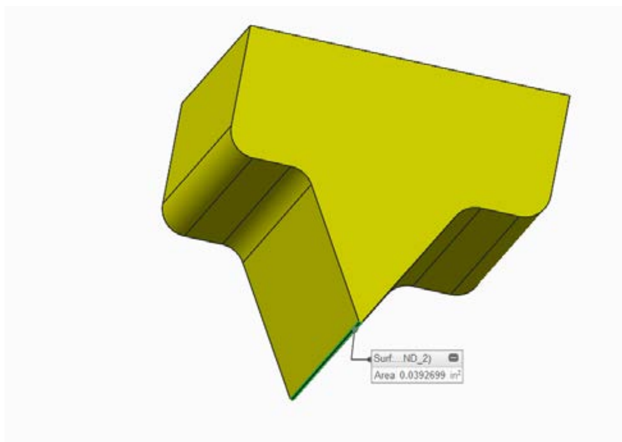


Figure 40-A Male pivot joint.

Two pivot joints are used to support the cantilever beam, and each sustain a total load of 6324.5 lbs. The load acts on the knife edge surface designed with a radius of 0.056 inches. The small radius produces a theoretical contact surface area of 0.056 x 1.875 = 0.105 square inches.

Material: Stainless steel 17-4 H 900 < 4.0-inch thick. With a 170 ksi yield strength.

F/A = 6324.5 lbs / 0.105 in^2 = 60233 psi in compression

<u>F.S. 170000 psi / 60233 psi = 2.82</u>

This calculation shows that the component will not deform under the load since the material compressive strength is greater than the pressure generated by the test load.

Appendix-B

List of official NASA drawings:

Main frame assembly	LA 1299307 A
Base plate	LA 1299308
Lower frame structure	LA 1299309 A
Upper frame structure	LA 1299310
Tie in plate	LA 1299311
Weight support plate	LA 1299312
Lower strap mount	LA 1299313
Angle mount	LA 1299314
Side brace	LA 1299315 A
Pivot hardware	LA 1299316
Lever-arm beam	LA 1299317 A
Crossbeam	LA 1299318
Pivot hardware	LA 1299319
Lever-arm mount	LA 1299320
Weight yoke	LA 1299321
Specimen swing plate	LA 1299322
Lever-arm beam support	LA 1299323
Thermal enclosure	LA 1299324 C items -2,-16,-17 not in use for this design
String pot mount plate	LA 1299325
Weights	LA 1299647

Appendix-C

Strap specimen installation and tensioning procedure

STEP 1 – Initial setup. With no weights on the rear weight plate, place the upper section of the wooden cantilever beam support in place and rest the lever arm on it. (Fig 1-C). Fit test the strap specimen to the upper mount by threading the shoulder bolt through the strap loop. Secure the shoulder bolt through the upper mount with a hex nut as illustrated in Fig. 2-C.

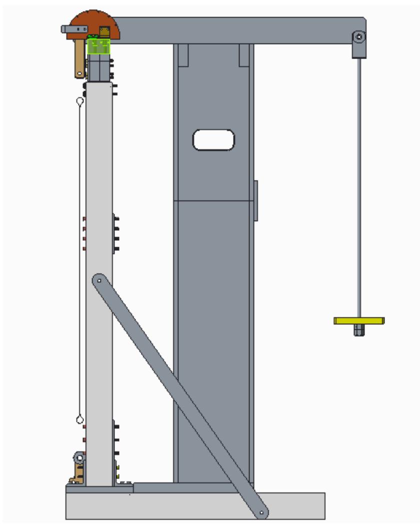


Figure 1-C. Initial test frame state.

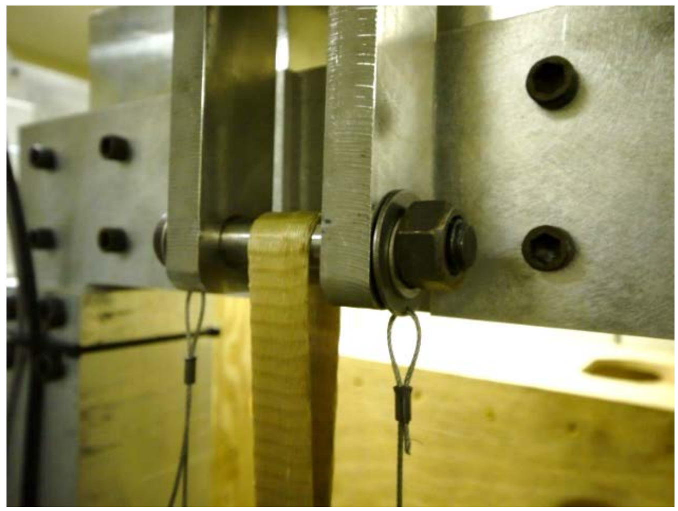


Figure 2-C. Upper mount strap connection.

STEP 2 - Lower mount assembly. If a load cell is being used in the setup, attach and secure the short threaded rod between the load cell and lower mount pin-grip using hex nuts as shown

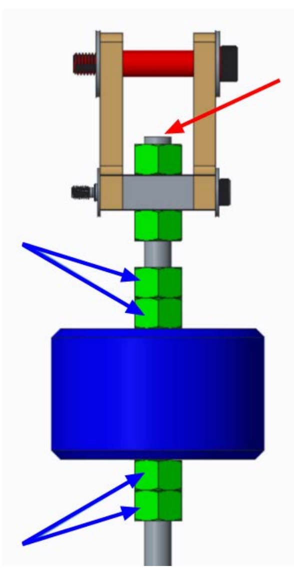


Figure 3-C. Hex nut locations.

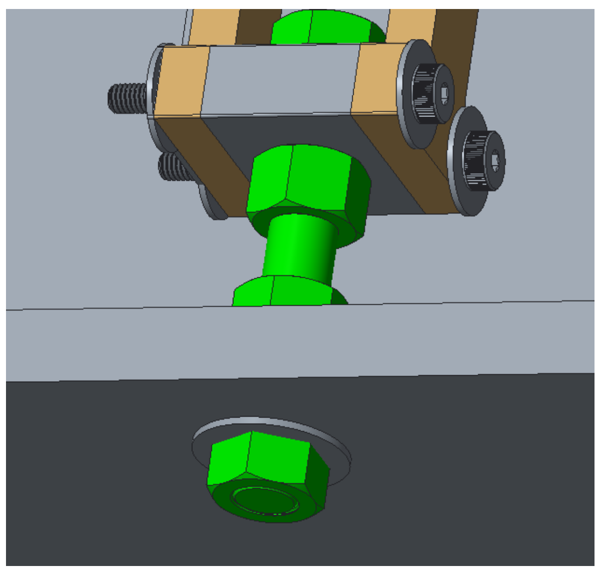


Figure 4-C. Test stand base connection (no load cell).

in Fig. 3-C. Ensure that the threaded portion of the rod has minimal exposure above the top nut as indicated by the red arrow in Fig. 3-C. This maximizes the available travel for tensioning the strap. Create a positive lock between the load cell and upper short threaded rod by using two hex nuts tightened against each other (blue arrows in Fig. 3-C). This operation is repeated for the threaded rod on the bottom side of the load cell that passes through the base of the test stand. Insert the lower threaded rod through the creep test stand base, and secure the lower mount assembly to the base with a washer and a hex nut below the base (Fig 4-C, shown without an inline load cell). Again, the threaded rod should be close to flush with the nut to allow for the maximum travel adjustment. When a load cell is used, tensioning can be performed by adjusting the lower mount's position on either the upper threaded rod or the lower threaded rod.

STEP 3 – Specimen installation. At this stage, the cantilever arm can be raised to a high angle, and the strap specimen can be attached to the lower mount (Figs. 5-C and 6-C). If the arm angle is small in this initial condition, then either the length of the test strap is too long, or some initial adjustment of the positions of the upper and lower mounts is needed.

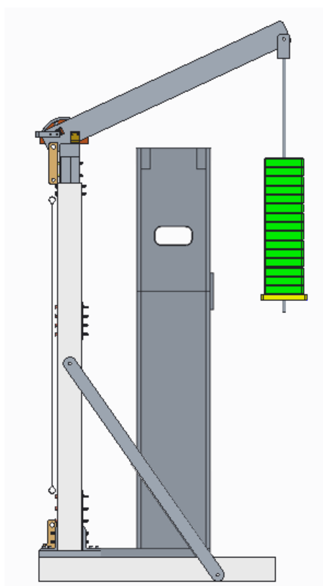


Figure 5-C. Side view of test stand.

Figure 6-C. Lower mount.

STEP 4 – Loading (A). Applying tension to the specimen is performed in several iterations of loading weights, offloading the tightening nuts and adjusting the height of the lower mount until the full test load is attained and there is room for the specimen to creep over time. If a load cell is not being used skip to step 7. Start introducing load to the strap by adding weights to the weight plate, until the cantilever beam comes to rest on the wooden support (Fig. 7-C). At this stage using a lift-jack under the weight base (Fig. 8-C) raise the jack until the load is reduced or removed from the specimen.

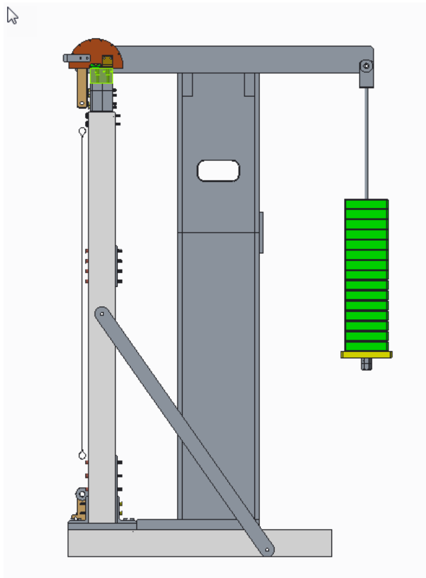


Figure 7-C. Arm resting on support.

Figure 8-C. Weights with jack stand.

STEP 5 - Tensioning above load cell. Loosen the hex nut indicated by the red arrow in Fig. 9-C, and move it down the threaded rod as much as is needed. Now, with the tension offloaded by the lifting-jack, adjust the hex nut indicated by the blue arrow to drive the lower mount towards the base. Stop when the threaded rod is a minimum of ¼-inch away from the strap (green arrow in Fig. 9-C). This process will relocate the cantilever arm again above the horizontal plane.

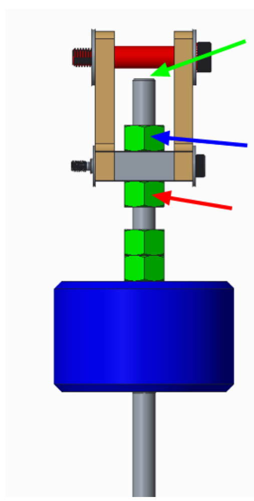


Figure 9-C. Adjustment of tension via upper threaded rod.

STEP 6 – Loading (B). Continue to add weight to the weight stack until the desired load is reached. If the cantilever beam settles again on the wooden support, reintroduce the jack to reduce or remove the load from the specimen (Fig. 8-C) and if sufficient travel remains repeat step 5. If not continue to step 7.

STEP 7 - Tensioning below load cell. Loosen the hex nut above the base plate and move it up as required. Then with the load again offloaded, tighten the hex nut under the base plate (arrow in Fig. 10-C) to drive the entire lower mount assembly towards the base, tensioning the strap. This will allow the cantilever arm to raise above the horizontal plane again. Continue these adjustments until the full load is applied and the arm is at a minimum of 35 degrees above the horizontal plane.

Figure 10-C. Test stand base plate and lower threaded rod.

STEP 8 – Stringpot installation. Once the full load is applied, introduce the lift-jack under the load to remove or reduce the load. Install the string pots as shown in Fig. 11-C and Fig. 12-C. Finally, remove the upper wooden support shown in Fig. 5-C prior to testing.

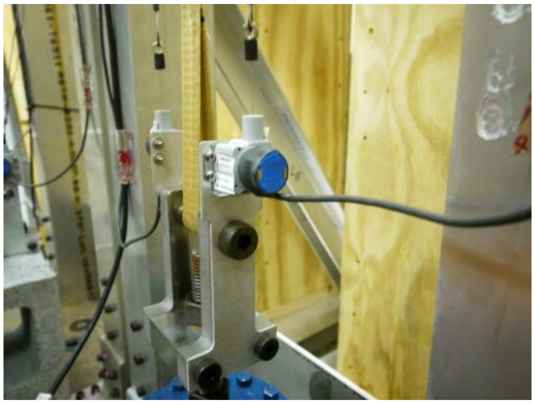


Figure 11-C. String pots attached to lower mount.

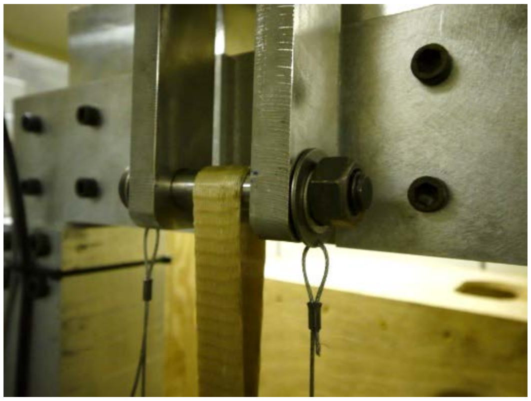


Figure 12-C. Upper attachment points to pin.

SAFETY NOTE: Ensure that only a few inches of space is left between the weight base plate and a rigid structure (stacked cement block) so that when the strap fails, the cantilever beam will only travel a short distance, thus reducing the dynamic loads on the test frame.

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14. ABSTRACT
Space exploration is typically driven by two key factors: cost and payload mass. Therefore creating relatively low cost, low mass space hardware is critical. With the advancement of new high-strength flexible materials, a new chapter is opening in the evolution of pressurized space structures. Space-rated inflatable structures are becoming more attractive for their low mass, compact storage, and capability of performing as well, or better than, their rigid counterparts for applications such as habitats and airlocks. The lifetime properties of materials used for inflatable structures are not as well characterized as heritage composite and metallic materials; therefore, they require extensive testing to increase the level of confidence in their use. Creep testing is one method of evaluating the expected lifetime of materials that may be under constant load for many years in operational use. Conventional creep testing of high-strength softgoods webbings or cordage requires a large, temperature-controlled facility, with structural frames capable of supporting large dead weights, weighing thousands of pounds on each specimen. This is expensive and takes up a significant amount of volume in a facility for a long time. This paper describes the design and analysis of a low-cost, low-volume creep test stand that can be locally environmentally controlled, so that smaller, cheaper facilities can be used for characterizing the creep life of the softgoods materials used in inflatable space structures.

15. SUBJECT TERMS
Inflatable Space Structures, Softgoods, Creep Testing, Vectran, Webbing, Cordage, Test Design.

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