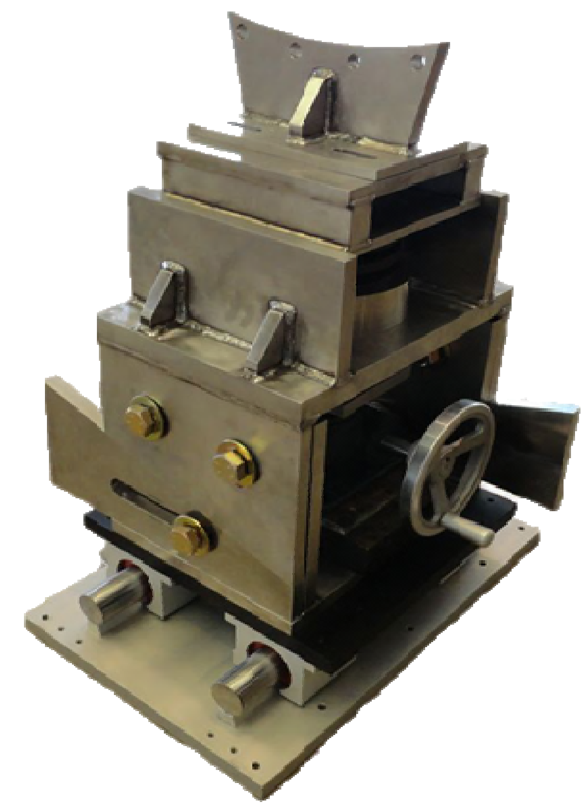
TurboTech

Titan 130 Engine Aft-Mount



Woody Newell

Michael Haworth

Dylan Perry

Prepared for:

Ismael DePaz – Solar Turbines

Joseph Mello – Cal Poly SLO

Solar Turbines Titan 130 Engine Aft-Mount Redesign

by

Woody F. Newell

Michael R. Haworth

Dylan R. Perry

Mechanical Engineering Department

California Polytechnic State University

San Luis Obispo

2013

Statement of Disclaimer

Since this project is a result of a class assignment, it has been graded and accepted as fulfillment of the course requirements. Acceptance does not imply technical accuracy or reliability. Any use of information in this report is done at the risk of the user. These risks may include catastrophic failure of the device or infringement of patent or copyright laws. California Polytechnic State University at San Luis Obispo and its staff cannot be held liable for any use or misuse of the project.

Table of Contents

EXECUTIVE SUMMARY

Solar Turbines' Titan 130 package has encountered a series of gearbox failures occurring on both Solar and Turbomach packages. It has been identified that one probable cause of the failure is due to engine – gearbox misalignment initiated by aft-mount failure. The goal of the engine aft-mount redesign is to find an improvement that can greatly benefit alignment repeatability and reduce set up time. The improved design will need to account for engine misalignment due to varying operating loads, turbine thermal expansion, and exhaust bellows forces.

These requirements were met by utilizing several different design aspects incorporated into the final aft-mount design. A removable mechanical jack and load cell were specified to accurately set initial turbine alignment instead of basing alignment on spring pack deflection alone. By placing the aft-mount on linear rail sliders allow for axial movement due to thermal expansion of the engine, the mount can move during operation and maintain a vertical load on the mount. Finally, wedges fastened to the mount were used to replace the current shimming technique to set and maintain the aft-mount height.

The mount was manufactured in Solar Turbines' Department 45 in order to satisfy ISO requirements. The main components were built using 316 Stainless Steel for its anticorrosion properties. The procurement cost for the stainless steel material came to a total of $3,091.00 with the cost to manufacture each of the details totaling $27,794.12. In addition, buyout items including the mechanical jack and the rail-slider system cost $2,132.80 and $2,012.58 respectively. The total cost of the prototype aft-mount came to be $33,064.59, with an additional $3,371.00 for tooling costs.

Verification of the aft-mount design requirements was performed in Cal Poly's engineering building 13, room 125. Static and cyclic compression testing was conducted on the MTS 322 Test Frame with a load capacity of 100,000 pounds of force. Using this machine, each engine type's loading characteristics were simulated during engine setup and normal running conditions. The repeatability of the mount was verified during the static compression testing portion of the test. This was accomplished through the strategic placement of linear variable differential transducers at points on the mount where deflection was expected to be greatest. As expected through beam bending calculations, the maximum deflection of the top plate for conventional and SoLoNOx engines were 0.032 inches and 0.035 inches respectively. Also, the deflection of the spring pack for conventional and SoLoNOx engines were about 0.62 inches and 0.73 inches respectively. By repeating these static load tests ten times, each of these numbers were found to be accurate to within ±0.005 inches, thus proving its repeatability. Cyclic load testing was also performed, however the data collection frequency was too low and the duration of testing was too short to make any substantial conclusions about the mount's abilities in actual operation. The tests do show that for the length of the tests, the mount is capable of withstanding

the weight and vibration of both engine types. Tests were performed on the rail-slider system to determine the static coefficient of the rails. These tests were conducted in order to determine the necessary force for the axial thermal expansion of the engine to overcome in order to move the mount and maintain the vertical force. For clean rails, the static coefficient of friction was found to be 0.18 and for dirty rails it was found to be 0.19. This corresponds to approximately 2400 pounds of force for the conventional engine to overcome and 2900 pounds of force for the SoLoNOx engine to overcome. Based on the opinion of a Solar Turbines contact, these forces should be sufficiently low for the axial growth to break the static friction and the mount should slide in operation. Through engine alignment simulations during testing, the mount installation time and the time it takes to set the turbine shaft alignment was drastically improved. During a simulation in the test frame, the alignment of the mount was achieved in less than ten minutes. After calibrating the Honeywell load cell, a conversion factor of 0.149 volts per kip was determined to convert the DAQ readout to thousand pounds of force. Using this number, a DAQ reading of 1.929V corresponds to the 12,927 pounds exerted by the conventional engine and 2.311V corresponds to the 15,484 pounds from the SoLoNOx engine.

There are several manufacturing recommendations that can be made for the aft-mount beginning with the top plate. Increasing the thickness to 2 inches should reduce the deflection of the plate to be less than 0.010 inches when the jack is removed from the cavity where it is used to lift the engine. As for testing, it is believed that further testing on an actual Titan 130 would be beneficial for more vibration analysis and real time rail slider proofing. It is fully expected that the mount will work under normal operation conditions for both engine types, and this test would provide conclusive data that proves that the mount would work during normal operation.

INTRODUCTION

Solar Turbines has presented our group, *TurboTech*, with the task of redesigning the engine aft-mount for the Titan 130 turbine package. The redesign has been identified as an important issue that leads to shaft misalignment and eventual gearbox failures on turbine packages installed in many locations across the world. The improved aft-mount design will have an easily repeatable method for aligning the turbine when installing the mount assembly and will account for axial and radial expansion of the engine in operation. There are several factors that affect the mount, including engine operating load, vibration, thermal growth, and the load from the exhaust bellows.

This project has a significant impact at many levels throughout the Solar Turbines corporation and customer base. There are over 350 Titan 130 packages operating around the world utilizing the current aft-mount design. Therefore, the results from this project can be used to retrofit current models as soon as their next overhaul, as well as be installed on new packages produced by Solar Turbines. With a properly functioning mount, the overall life of the turbine package can be increased, which will increase the performance of the machine and reduce downtime. If the mount can provide the proper alignment to the shaft, there will be less gearbox failures and more time between overhauls. This will not only help the owner of the turbine to reduce costs, but it will allow Solar Turbines to focus on other important turbine uprate projects.

By utilizing a set-based engineering design approach, each problem with the aft-mount is resolved by a separate design and then all the designs are consolidated into one viable option that incorporates the best characteristics of all the designs. Therefore, three separate concept designs are developed that make use of a unique way of solving the current aft-mount problems including an elastic, fluid, and mechanical approach. Solar Turbines recommended exploring one of their current concept designs for the final design which consisted of the currently used spring pack to lessen the stiffness of the mount. Shaft alignment is set by utilizing a hydraulic jack equipped with a load cell to measure the force on the aft-mount and is locked into place using wedges and fasteners.

The overall objectives for the aft-mount redesign consist of improving upon the methods used to initially set the turbine's alignment using the aft-mount, enable the aft-mount to manage any axial and radial loading due to thermal growth of the turbine, and to maintain the necessary stiffness to inhibit vibrational problems. In order to accomplish these objectives, several changes to the Solar design were made. Calculations were performed to determine whether the wedges were a viable option for maintaining shaft alignment, linear sliders mounted between the skid and aft-mount were added, and a high capacity mechanical jack was proposed to replace the hydraulic jack. All these additions were analyzed for compatibility with the system and changes to the overall dimensions of the aft-mount were made to accommodate them. Deflection of high

load bearing members of the aft-mount, fastener requirements, thermal expansion, and spring deflection characteristics were all calculated and used to specify the dimensions of all the necessary component parts of the aft-mount. These specifications are put into appropriately labeled detailed drawings with tolerances and geometric relationships specified to accommodate the weldments and tooling needed for installation. Due to Solar Turbines' International Organization for Standardization (ISO) certification, tight tolerances, and expertise needed to manufacture the aft-mount, Solar Turbines' Department 45 is required to manufacture the final design to ensure a quality prototype is built.

It was necessary to test the new design for the aft-mount in several different capacities to verify its ability to withstand the requirements aforementioned. All tests involving static and cyclic loading were performed on the MTS 322 Test Frame located in Cal Poly's building 13 room 125. In combination with the use of four linear variable differential transducers (LVDTs), deflection data was collected to verify the reliability, repeatability, and deflection calculations previously calculated for the mount. The static coefficient of friction associated with the rail-slider system was determined using and Omega Instruments load cell. This number was pertinent in the calculation of the force required from the axial thermal expansion of the engine. This minimum required force is what it takes to overcome the static friction of the sliders to retain a straight vertical load on the mount after the turbine is heat soaked. Based on the results of testing, several recommendations can be made in order to improve the overall design of the aft-mount which include manufacturing changes that would allow for improved assembly time and mount performance during operation. As for testing, it is believed that further testing on an actual Titan 130 would be beneficial for more vibration analysis and real time rail slider proofing. It is fully expected that the mount will work under normal operation conditions for both engine types, and this test would provide conclusive data that proves that the mount would work during normal operation.

BACKGROUND

Solar Turbines Titan 130 package, as seen in Figure 1, is the second largest set offered from the company. The cold end drive that we will be focusing on has a rated output power of 20,500 horsepower and weighs 85,000 pounds as a whole. Recently, this model has come to the attention of Solar after several unexpected gearbox failures in the field. This turbine has a few possible problems that the design teams are looking into now. The following will discuss the different problems ongoing in the turbine and what our team will be focusing on.

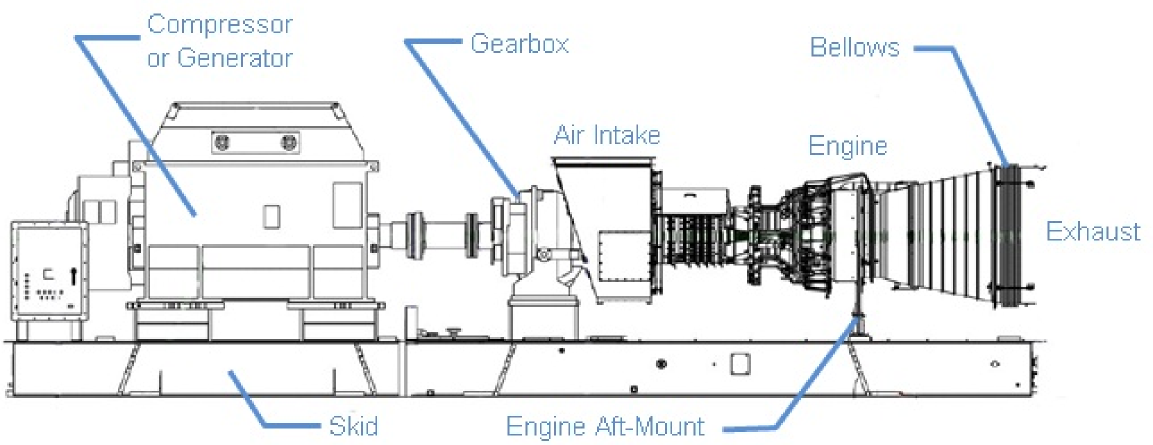


Figure 1. Side view drawing of the Titan 130 package showing relevant features of the platform.

First, the 940ºF exhaust from the turbine leaves the engine through the rear of the package. When Solar builds enclosed packages, they control the position of the exhaust outlet by building a case around the equipment and secure the bellows to the case. When the exhaust is left up to the customer to install, such as unenclosed package models, there are typically alignment issues when the engine package arrives. Figure 2 illustrates this problem. If the center point of the exhaust duct is not in alignment with the shaft, the forces from the misaligned bellows will affect the engine mount and gearbox on the other end of the shaft. Currently, design teams are looking into a flexible coupling between the package built by Solar and the customer's exhaust set up. Also, engineers are looking into being able to move the shaft by adjusting the aft-mount in order to align the exhaust outlet location, shown in Figure 3. The question is, what is the acceptable shaft misalignment tolerance to the gearbox without causing serious damage in the long run?

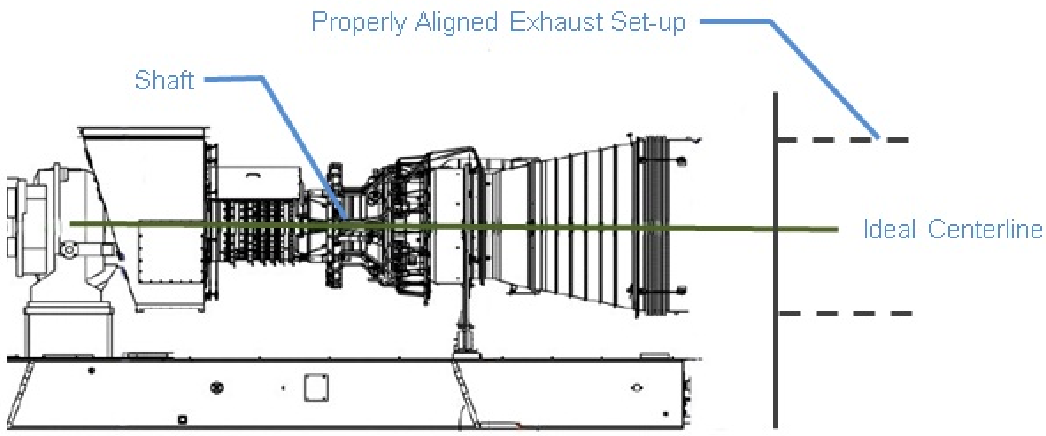


Figure 2. The shaft that runs through the engine should be centered with the exhaust outlet installed by the customer. When these are not in alignment, the forces caused by the misalignment will lead to wear on the gearbox at the other end of the shaft.

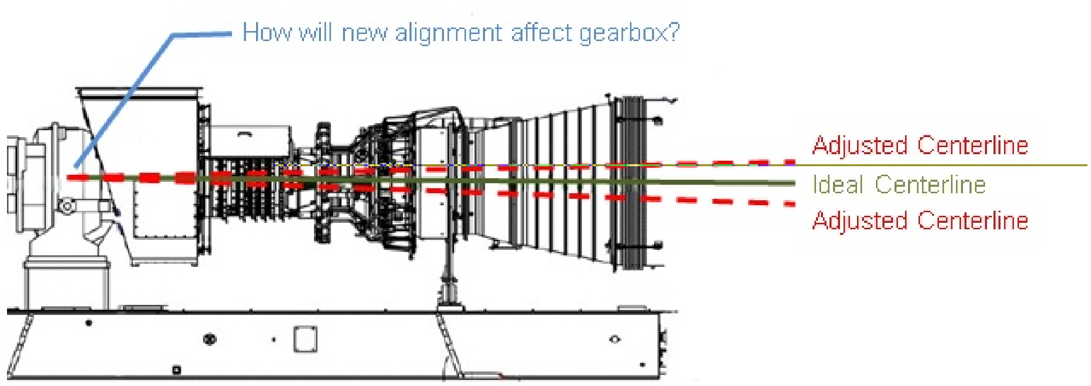


Figure 3. This illustrates how the shaft can be positioned in order to line up the exhaust with the customers pre-installed exhaust outlet. Notice how this alignment will affect the connection between the gearbox and engine.

Second, how does the connection between the gearbox and engine case allow for movement of the shaft? Figure 4 shows the two possible cases for this problem. Currently, the gearbox is rigidly connected to the skid and to the engine case. When the shaft is shifted to match the exhaust outlet, will the dog-bone coupling that connects the engine shaft to the gearbox follow the bend or remain rigid and cause problems in the gearbox?

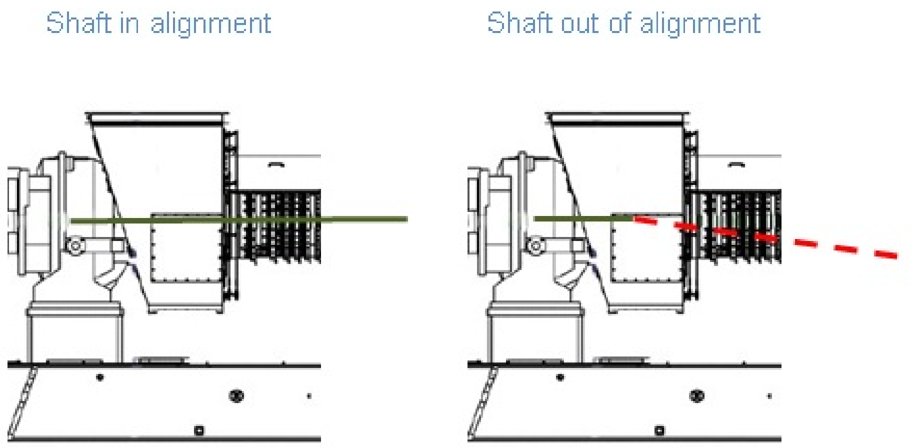


Figure 4. If the shaft is positioned to line up with the misaligned exhaust outlet (right), the dog-bone connection must be able to deal with the deflection. A team at Solar is currently looking into this problem.

While these two problems are being worked on elsewhere within Solar, our role in the project becomes clear. Our team will focus on how to better connect the turbine to the skid in order to increase repeatability of consistent alignment of the shaft to the exhaust outlet. The current design is shown in Figure 5. Solar would like to keep the current bolt pattern on the engine case, however they are open to changes to the skid itself. One important issue with the current design is repeatability. When the Belleville springs that Solar currently uses are new, they are easy to set because they are predictable and have not fatigued. As time goes on, it is harder and harder to get a precise, repeatable setting as specified by Solar on the springs because they have been exposed to vibration of the turbine, maintenance overhauls, and the environment. Secondly, the mount is designed to slide along the axis of the turbine (to account for thermal expansion and axial shifting) so that the weight of the engine will always be pushing vertically down on the spring pack. Due to the large engine bracket and significant weight of the turbine, the mount tends to bend and not push down vertically on the springs. The torque applied to the engine bracket does not allow the mount to slide back which puts uneven pressure on the spring pack which can lead to failure. Figure 6 depicts the ideal situation versus what is actually happening. The mount our team designs will need to solve these issues.

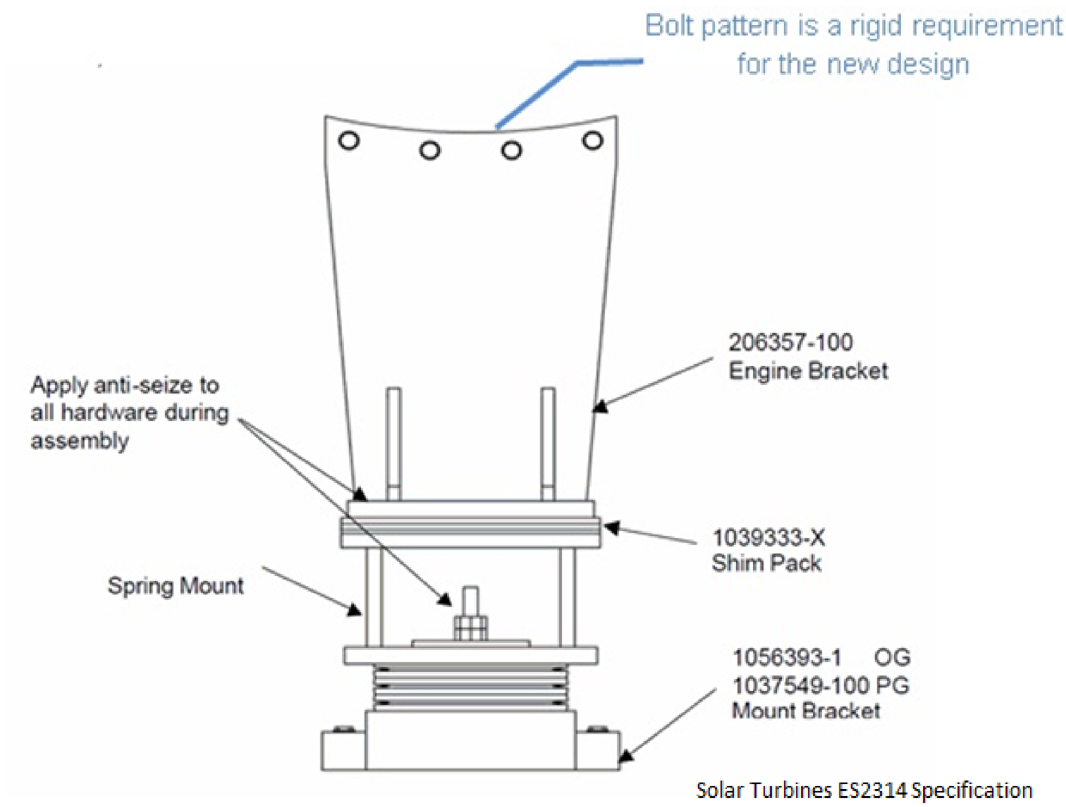


Figure 5. Front view of the current engine aft-mount used by Solar Turbines. One requirement is that the bolt pattern that connects the mount to the engine case must remain unchanged.

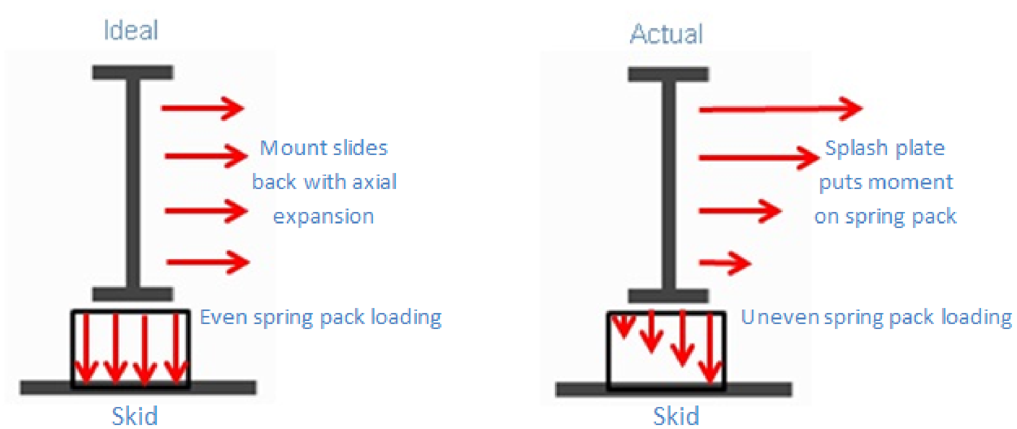


Figure 6. Simplified side view of the engine aft-mount. Ideally, the mount will slide back on rollers during thermal expansion and maintain even pressure on the spring mount. In actuality, the weight of the engine prevents the mount from sliding and puts an uneven amount of load on the springs.

A question that came up during our initial discussions with the sponsor was to clarify the differences between the engine aft-mount and the trunnions used to support the engine. The trunnions are external brackets that are installed after the alignment of the shaft has been set using the aft-mount. The placement of the trunnions is shown in Figure 7. It initially appeared that these trunnions took the place of the aft-mount after installation. After talking with Ismael DePaz, he clarified the use of trunnions. The Titan 130 is the only package that Solar produces that uses trunnions. They are only used in unstable environments, such as on the sea, where they will support the extra forces caused by the rocking of the package. On typical installations, trunnions are not used and the mount supports the full weight of the engine. From this point on, we will be ignoring the trunnions and only be focused on the aft-mount.

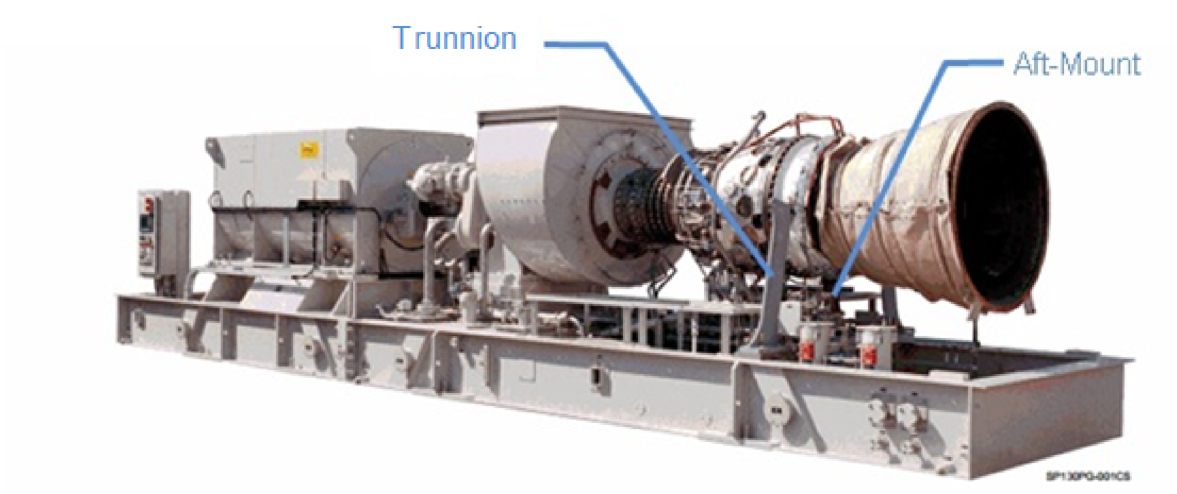


Figure 7. Solar Turbines Titan 130 package with trunnions. The trunnions are only used in conditions where the foundation is unstable, such as being used at sea.

For the basis of our background research, our focus will be to understand the current mounting technology utilized by Solar Turbines. Solar provided an installation guide, a troubleshooting guide, and an analysis report performed by an outside company on the aft-mount in order for us to get a better understanding of the problem (Appendix C). Additionally, research time will be spent on non-traditional methods of maintaining shaft alignment and vibrational control. This is including, but not limited to, automobile engine mounts and damping systems, aircraft turbine mounts, and large scale commercial power generation. Finally, we will focus on looking into the repeatability of systems in order to get an understanding of what it takes to produce a system that can withstand the loading pattern of heavy turbomachinery. In order to judge our concept designs against one another, it will be important to clearly define the objectives in a way that will be easily quantifiable. We plan to meet with our team sponsor and

technical advisors every week to keep the project on schedule and to answer any outstanding questions we may have.

Our research began with the documents provided by Solar Turbines. We were given a positional change report authored by Bengal Resources and TERN Technologies. This provided information on a previously conducted experiment to better understand the effects of thermal expansion and rotor inertia on the levelness of the turbine as a whole, as well as the overall shaft alignment. A specification sheet (ES 2314) for the current mounting technology provided an overview of how the mount would be installed on the package. Finally, a technical letter to field service representatives was provided with information on how to troubleshoot a turbine to see if it was near failure. Specifically, it focused on the 2050 Hz vibrational frequency issues associated with the gear box failure. These documents collectively gave us an initial understanding of the problem at hand and how to begin our solution process. Further information on the Titan 130 package was found on Solar's website.

Next, we turned our focus to other designs currently being used by other companies such as General Electric or Siemens. Since specifics are proprietary and well-guarded trade secrets, our team came up empty handed in our searches. However, we were able to look into many other mounting systems such as automotive engine mounts and mounting systems for jet engines to aircraft. A great amount of time was spent looking through patents and journals in the hopes that an idea will trigger a possible design down the road. After realizing that information on the mount as a whole might be difficult to find, our team turned to looking at information for various components of the current design. The most obvious thing for our team was to look into the Belleville springs used in the spring pack of the current mount. Once we were familiar with this type of spring, we explored the strengths and weaknesses of other spring designs. Finally, our team was able to rapid prototype the current aft-mount in order to get a clearer picture of the overall layout as well as the dimensions of the part.

On November 1st, our team was able to take a tour of Solar Turbines' assembly facility in Kearny Mesa in order to see the mount up close. During the visit, we were able to gain more information on the problems facing the aft-mount as well as see the scale of the part we will be working with. The static forces acting on the mount can be seen in Figure 8 for a conventional engine and Figure 9 for a SoLoNOx engine. A SoLoNOx engine is heavier than the conventional engine because of the extra components needed to make the engine more environmentally friendly. During our visit in San Diego, we were also able to talk to Varad Sampathkumar who gave us information on tests that he has performed on the aft-mount assembly. From Figure 10, it is clear that the springs used in each mount do not always behave in a similar fashion. He also explained the different models of the Titan 130 package that are available to customers all over the world, as seen in Figures 12 to 14.

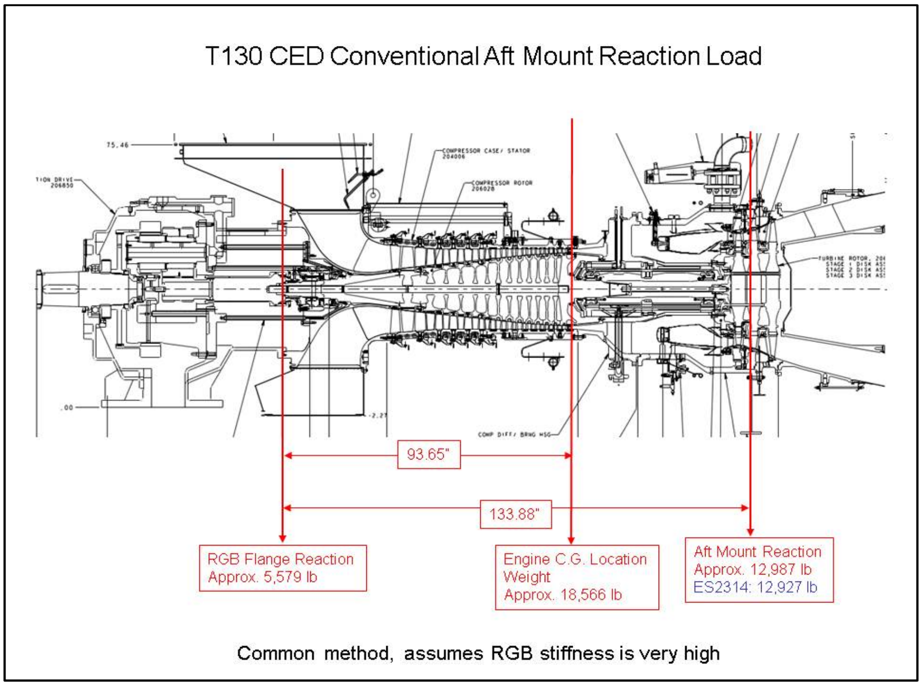


Figure 8. Diagram showing the reaction forces for a conventional Titan 130 engine at the gearbox-engine interface and the mount location.

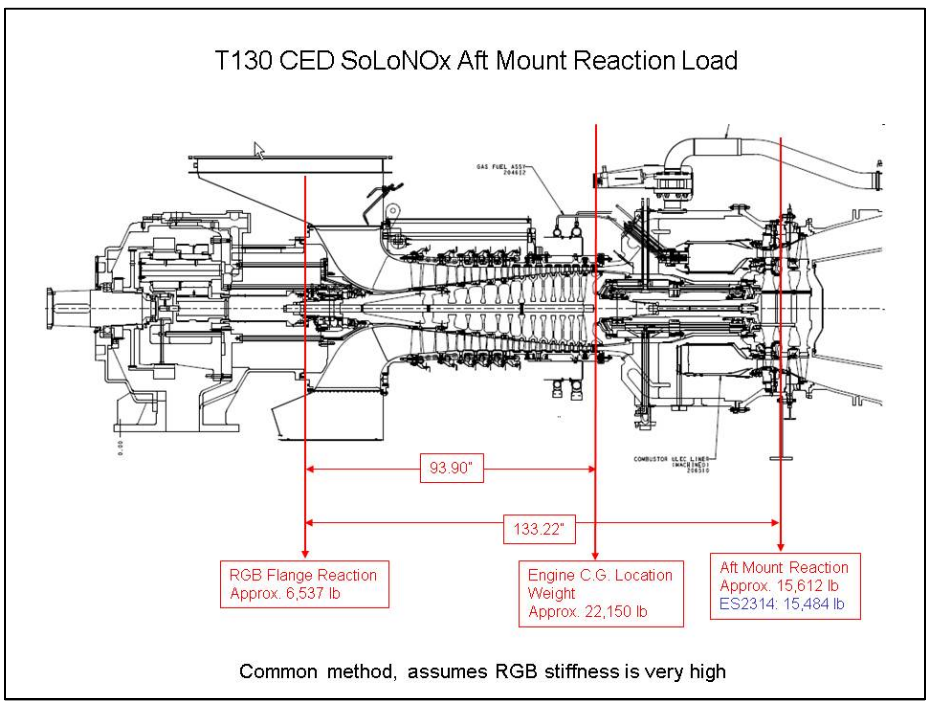


Figure 9. Diagram showing the reaction forces for a SoLoNOx Titan 130 engine at the gearbox-engine interface and the mount location.

T130 Aft Mount Spring Assembly Testing

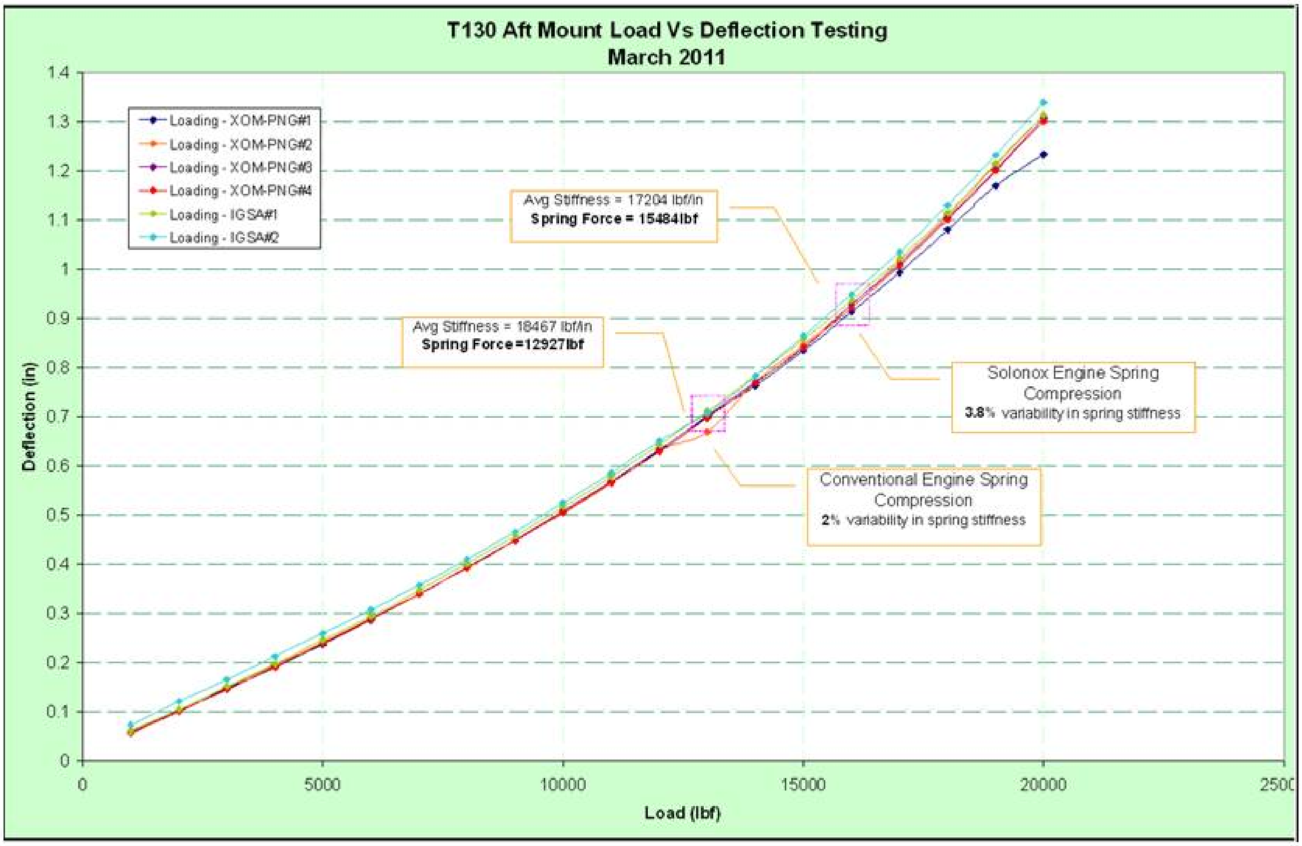


Figure 10. Load – Deflection curve for six different identical spring packs. Belleville springs do not have a linear spring constant as seen above. Notice the variability in the spring stiffness at the engine loading positions.

Figure 11 shows the results from a test that Solar Turbines performed to see if there was any correlation between engine loading and force exerted on the aft-mount. Fifteen total engines were tested, eight conventional and seven SoLoNOx versions, with different engine conditions were placed in a test cell with a load sensor under the mount. Measurements were taken at different stages of run up and compared to the specified tolerances. As seen in the diagram below, there was no pattern in how the loading on the mount was affected by the run up. In some cases, the load on the aft-mount increased and in some cases it decreased. The conclusion of the test was that there was no correlation between aft-mount load and engine run up.

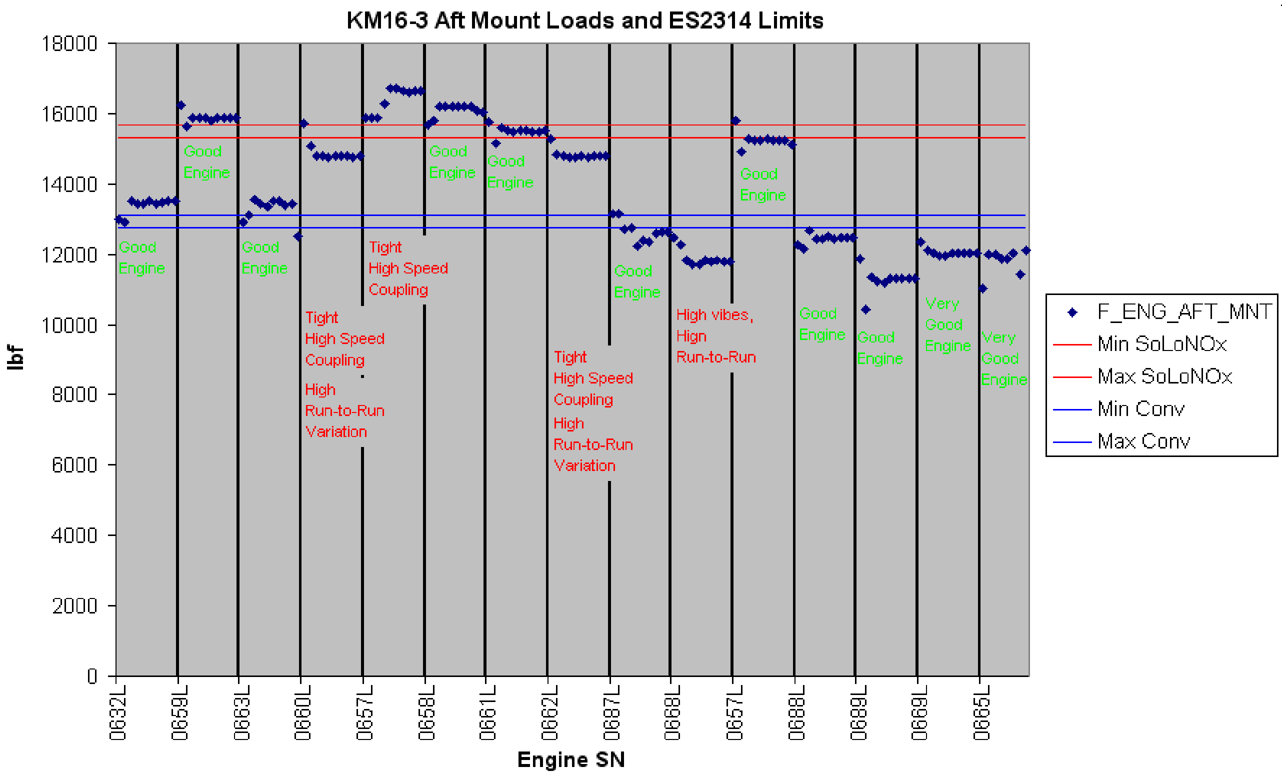


Figure 11. This diagram shows aft-mount loading for fifteen total engine packages (both conventional and SoLoNOx variations) through various stages of engine run up. The condition of the engine is shown in green for a good engine and red for a problem engine.

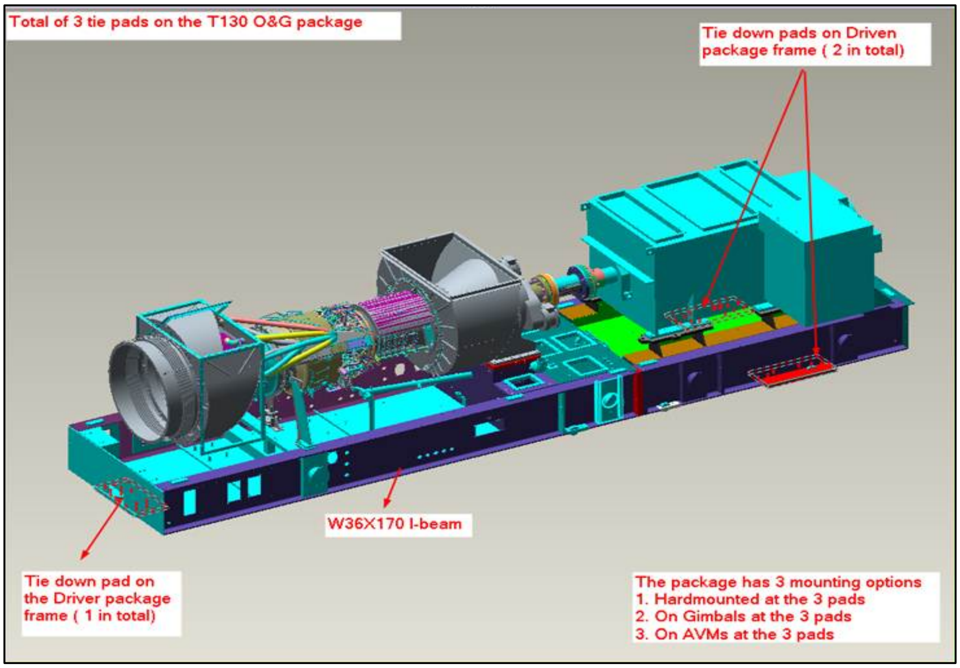


Figure 12. Solid model of the Titan 130 Oil and Gas package for offshore applications. This model features the stiffest frame of the three, and is mounted in three locations to the foundation.

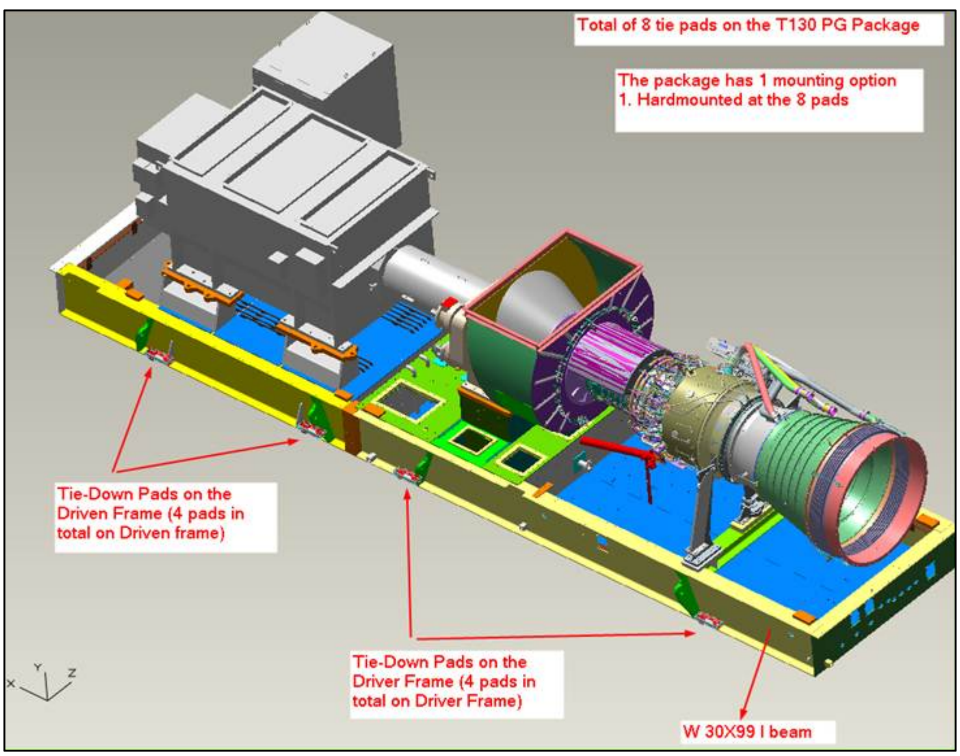


Figure 13. Solid model of the Titan 130 PG Onshore package. This model features eight tie down pads and a smaller frame than the offshore package.

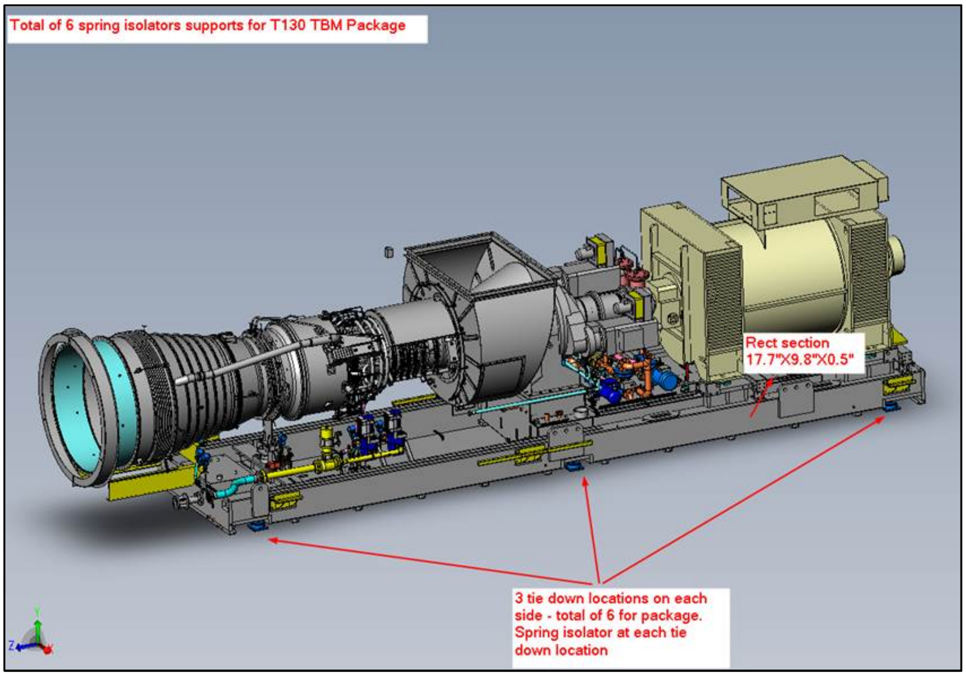


Figure 14. Solid model of the Titan 130 TBM package. This model is built by TurboMach in Europe and has a different frame design than the Solar Turbines counterparts. The six tie down pads feature spring isolators. Also, no trunnions are used on this model.

With an understanding of the current design, and some knowledge about different mounting systems, our team is ready to begin the concept design phase of the project. From the information that Solar Turbines provided, we feel that we have an adequate understanding of the problems with the aft-mount and the objective that Solar wants us to look into. As each proposed design is brought to the table, additional background research will need to be completed to solidify the design. A summary of our finding is in Table 1.

Table 1. Summary of key background into the engine aft-mount redesign.

Background	Brief Summary
Current Problem	Solar Turbines needs to be able to accurately set the engine on the mount every time regardless of engine, skid, and environment.
Focus on Aft-Mount	Even though there are other issues in that could affect the mount, such as exhaust position and the gearbox-to-engine interface, our scope is just on the mount assembly itself.
Driving Force	The mount is primarily load driven even though Solar Turbines currently uses deflection to set the mount.
Belleville Springs	Washer springs are great under high load and have low creep characteristics, however like all springs, the deflection becomes less predictable with time
Tall Splash Plate	The large engine bracket puts an uneven load on the spring pack during axial expansion of the engine, which puts excess wear on the washer springs .
Engine Types	There are two different engines that will sit on the aft-mount. First, the conventional engine applies a load of 13,000lbs to the mount. Second, the SoLoNOx engine applies a load of 15,500lbs to the mount due to the added weight of the environmentally friendly engine.
Skid Types	There are three different skids that the mount must be incorporated into. The Oil and Gas package has a very stiff frame with three feet that secure the skid to the deck. The PG onshore package has eight tie down pads and a smaller skid frame. Both of those models utilize trunnions to add support to the engine. The TBM package is built by TurboMach and has six tie down pads and no trunnions.

DESIGN REQUIREMENTS

Solar Turbines has given our team the task of designing an alternative engine aft-mount for the Titan 130 that will have higher repeatability and reliability characteristics than the current aft-mount design in order to increase the life of the package's gearbox and turbine. Repeatability is a difficult term to quantify. For instance, the engine mount must be able to be set into alignment both at the assembly floor and also at the customer's site. Also, after the turbine has reached its overhaul lifespan, at around 30,000 hours of operation, the turbine will be serviced and then reset on the mount. On the current design, after the turbine has been serviced, the spring pack has fatigued significantly and would compress more than tolerated once statically loaded. A summary of top design objectives and requirements can be seen in Table 2.

First, the static loading of the aft-mount will be analyzed so that after the initial installation the centerline will be aligned perfectly throughout the turbine and gearbox, providing a good starting position for any necessary future adjustments. Following the static loading of the turbine, we will analyze the thermal expansion and bellow loads that affect the mount. Our design will have to account for these forces and allow for small movements and deflections in the aft-mount in order to maintain proper centerline alignment tolerances throughout the turbine and gearbox interface. Also, our design will incorporate features that will avoid the creation of harmful system frequencies at approximately 2050 Hz that have been correlated with eminent gearbox failure. The time required to properly install the newly designed aft-mount will not be more than the current installation time required. Also, the aft-mount will not hinder any accessibility features that would affect the maintenance required to keep the aft-mount properly adjusted to maintain the package's centerline alignment. The current aft-mount design requires multiple iterations at setting the shaft alignment under static loading. We plan to look into designs that will allow the engine to be placed one time and allow for adjustment while the engine is secure. This limits the amount of time employees will have to spend under the suspended turbine, increasing the overall safety of the installation procedure.

The number one concern for the customer is repeatability. Solar Turbines would like to be able to control how the mount is set every time so that the alignment of the engine shaft can line up properly with the gearbox and exhaust outlet. The current design model utilizes Belleville springs that, while have a predictable deflection at first, begin to fatigue and lose their integrity over time. Moreover, under certain operating conditions, the springs have actually been found to crack after they have been loaded for long periods of time. In order to satisfy the customer's requirement, our team must make sure that the mount can be set up so that the engine is in alignment every time regardless of the mount's age or type of turbine it is supporting (i.e. conventional or SoLoNOx). In order to achieve proper alignment, the mount must be able to handle up and down, as well as left to right motion of the engine during set-up. Reliability of the new mount goes hand in hand with repeatability. In order to improve on the current aft-mount

design, the reliability of the part must be superior to the current model. If the turbine shaft's alignment is repeatable, but not reliable over the long term life of the system, it is of no use to Solar.

Along with design objectives, there are also a number of technical challenges that must be overcome in the new design. Once the aft-mount is able to be installed in a repeatable fashion both at the factory and at the customer's location, the next concern is how the mount will react to thermal expansion. During the run up cycle, the engine expands axially away from the gearbox. The current mount does not take this expansion into effect, which causes uneven loading on the spring pack which exaggerates wear on the springs. Also, the engine expands radially outward when in operation so this must also be taken into account when designing a new engine mount. Finally, the vibration that the engine produces must somehow be dissipated through the mount and frame and into the foundation. The resonant frequency of 2050Hz must be avoided at all costs in order to ensure that the package will operate safely.

Installation time is another big concern for Solar. The new design should be mounted in the same amount of time, or potentially less amount of time than the current mount, which takes one hour to assemble and an hour and a half to level the engine. A particular concern with installation time is safety. Since the engine will be suspended on an overhead crane, the time that the operator and assembler should be under the engine without rigid support must be kept to a minimum.

Maintenance of the package is another great concern to Solar. These machines go through an overhaul after a recommended operating time of 30,000 hours. It is important that the maintenance is completed in a timely manner, not only for Solar's crew, but also for the customer who is losing production time while the package is down. As far as the mount is concerned, accessibility is a big issue. Solar and our team will look into making certain that the overall mount configuration will not be a hazardous place for maintenance personnel to be around. Secondly, if something needs to be replaced on the mount, the availability of the parts is a concern while in the field. Ideally, the parts used in the new aft-mount design would be commonly available replacement parts and not one-of-a-kind pieces available upon special request.

An ever present objective in any design component is cost. Solar is a bit flexible with the cost of the new mount (because replacing the gearbox costs upwards of $600,000) but would rather keep the cost down in order to maximize profits for the company. If our design can prove to Solar that the new mount will solve the shaft alignment issues, they are willing to pay the price. Finally, our goal is to not significantly increase the total cost of the new design over the cost of the current method of manufacturing and installing the mount. Also, the new mount design must be able to be manufactured in a timely manner. If the new mount requires a great

deal of time to build, it will be very expensive to construct. Also, it is imperative that the new mount design must either be able to be built with conventional tooling that Solar has at its disposal, or purchased from an outside vendor.

Table 2. Summary of engineering requirements and objectives for the aft-mount redesign.

Objective	Brief Summary
Repeatability	Assembly staff or field installation team must know how to accurately set the alignment of the engine. Engine must be set into proper alignment regardless of the age, condition, or load on the mount.
Alignment	Alignment of the shaft relies on up and down positioning as well as left to right movement of the engine.
Axial Expansion	Mount must accommodate motion in the axial direction to account for thermal expansion.
Radial Expansion	Mount must accommodate motion in the radial direction to account for thermal expansion.
Vibration	Resonant frequency of 2050Hz must be avoided.
Reliability	Must not fail in operation and last for over 30,000 hours
Installation Time	Shorter than or equal to current design of one hour to assemble and an hour and a half to level the engine
Maintenance	Must be accessible and safe to work on
Manufacturability	Must be manufactured in a time effective manner or available to purchase from a vendor
Cost	Not a limiting factor considering failure of the mount leads to a $600,000 gearbox replacement, but the mount is support equipment to the engine so cheaper is always better

DESIGN DEVELOPMENT

In order to come up with several designs for the aft-mount redesign, our team decided to split up and brainstorm on their own. It was our goal to each have several designs in mind before travelling to Solar Turbines on November 1st. This allowed us to have some out of the box ideas before actually seeing the engine mount and the constraints that are around the aft-mount. After our trip down to San Diego, we came together and discussed the pros and cons of each of our individual designs. After throwing ideas back and forth for a while, we came up with several categories that our concepts would fit into. The first category of designs are grouped together as Elastic. Each of these deal with either retaining the current spring pack used by Solar Turbines or using a different type of spring than the current washer springs. The second category is Fluids. The designs lumped here deal with fluid supporting the engine using either pneumatics or hydraulics. Finally, the Mechanical category includes any design that uses mechanical components such as screws, gears, or cables. Each design is described in more detail below. After an initial concept review with Solar, we took these concepts and sorted out which qualities from each design satisfy key design requirements. From there, we were able to generate our top designs.

Elastic Concepts

One of the desires from the team at TurboTech is to allow for the final design to be able to be easily implemented into new and existing packages. We feel that this would be beneficial to Solar Turbines since it would allow them to easily replace existing mounts without needing a large retrofit. Another benefit to having a drop in replacement is that the installation procedure would be similar to the current mount.

Reposition of Current Design

The first concept design is to simply reposition the current spring pack. The current design utilizes a tall engine bracket (or splash plate) that connects the engine to the spring pack. During thermal expansion, the large moment generated by the engine bracket puts an uneven load distribution on the spring pack as seen in Figure 6. This can lead to wear on the washer springs and lead to fatigue and cracking. The first step of the new design calls for a shortened splash plate which will bring the spring pack closer to the engine. In order to raise the spring pack, the skid will need to be modified from the current design. From our tour of Solar Turbines on November 1st, we came to realize that there are two different skid designs that support the spring pack. One design is a cross bar that supports the spring pack while the other design uses a pedestal to support the mount. For each type of skid layout, a pedestal will be welded on to extend the height of the spring pack. A sketch of the proposed design for each skid layout is shown in Figure 15 through 17.

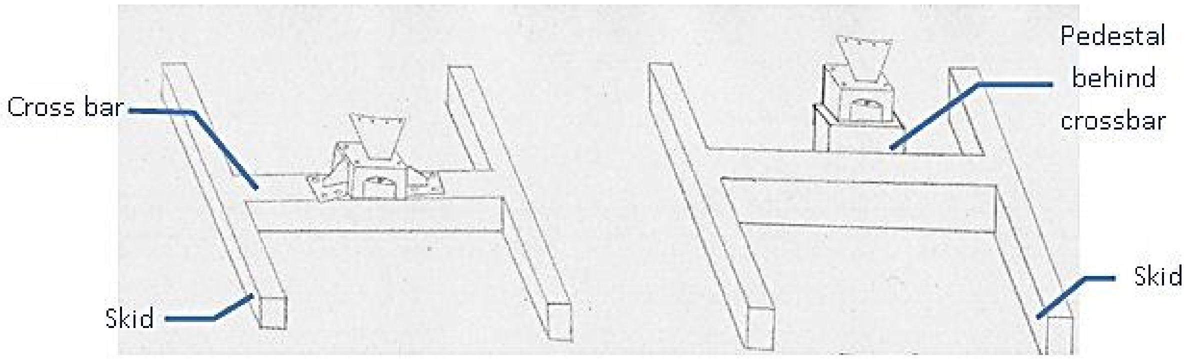


Figure 15. Sketches of the two types of current skid configurations used on the Titan 130. The mount is placed on a cross bar on the left while the mount on the right sits on a pedestal.

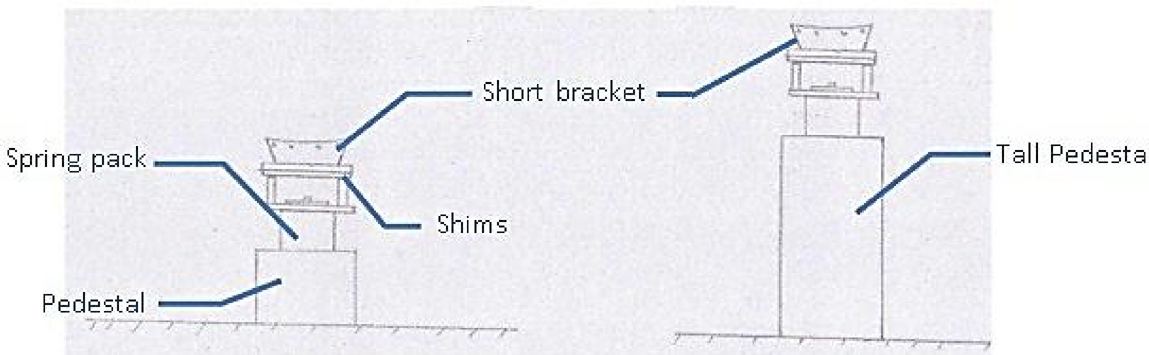


Figure 16. Front view of each skid. Both concepts suggest raising the height of the spring pack using a pedestal as on the left, or increasing the height of the current pedestal on the right.

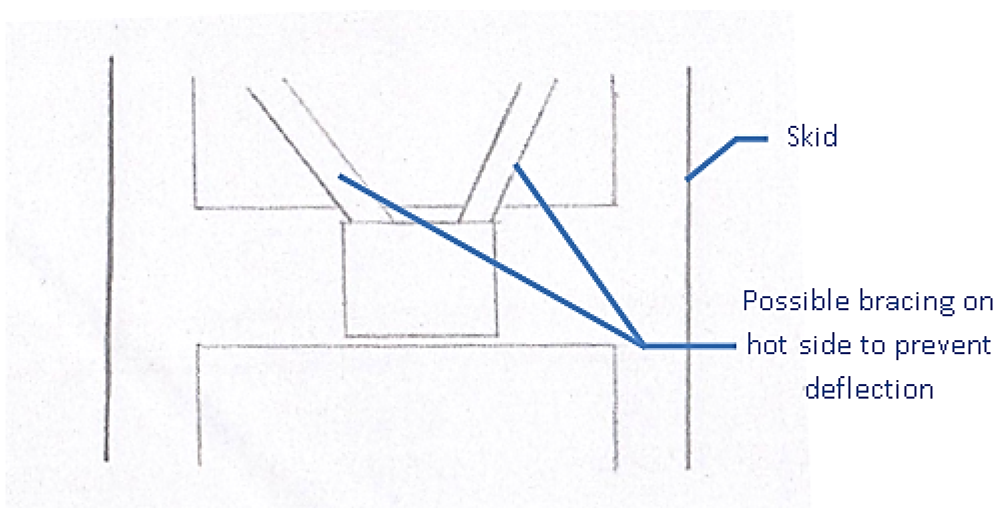


Figure 17. Top view of the extended pedestal skid. In this concept, support structure would need to be added to prevent excessive deflection of the pedestal.

The key features to this design is that it reduces the height of the splash plate and raises the height of the spring pack in order to avoid the uneven loading pressure on the spring pack which will reduce wear on the springs. Also, since the mount is now closer to the engine case, we believe that the spring packs will experience a more constant temperature range, albeit at a higher temperature. A benefit to this concept is that the current spring pack design is preserved which will reduce engineering costs and manufacturing will be unchanged. The only part that needs rework is the splash plate. Another advantage to this design is that the current assembly procedure is preserved saving the cost of having to train the assembly staff.

Even with the simplicity to this design, there are several drawbacks. First, this concept only fixes the problem of spring fatigue and does not do anything to account for the thermal expansion of the engine. Also, raising the spring pack will not help the variability between spring sets as seen in Figure 10. Another concern with this concept is that the space to place shims has decreased, which could lead to the possibility of a crew member being injured.

Sliding Mount

If springs are going to be preserved in the final design, we must be able to solve the problem of uneven spring loading caused by the thermal expansion of the engine as seen in Figure 18. A roller system will allow the spring pack to follow the direction of the expansion which will keep an even load distribution on the mount. The roller bearing concept shown in Figure 19 is designed such that the aft-mount can slide along the skid axially as the engine expands due to thermal forces. This allows the Belleville springs to have uniform loading while the engine is operating. The existing aft-mount does not allow for axial expansion and therefore has an uneven loading on the Belleville springs. These loads cause the vertical centerline of the springs to deflect and the upper portion of the spring to sit at an angle.

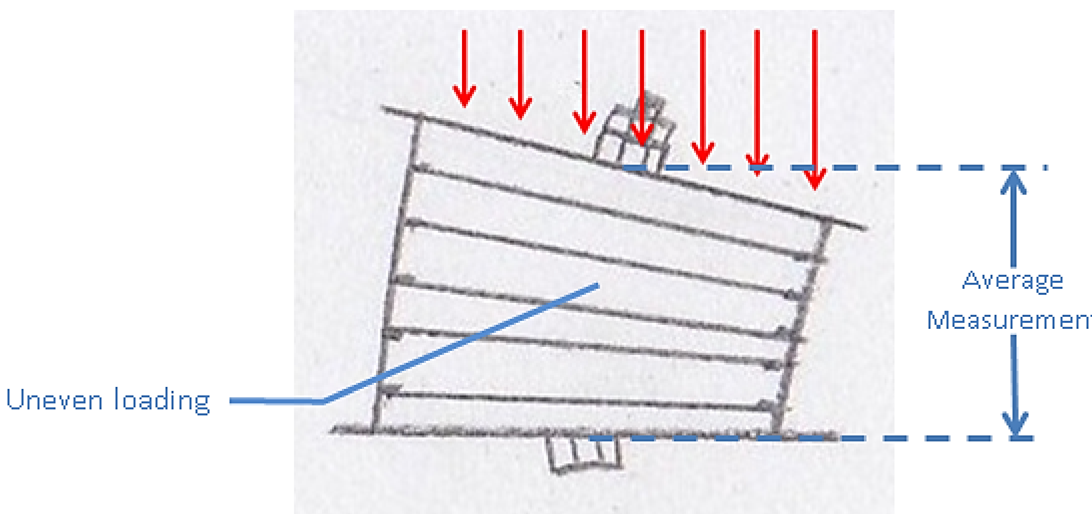


Figure 18. Side view of the spring pack showing uneven loading caused by the axial thermal expansion of the engine.

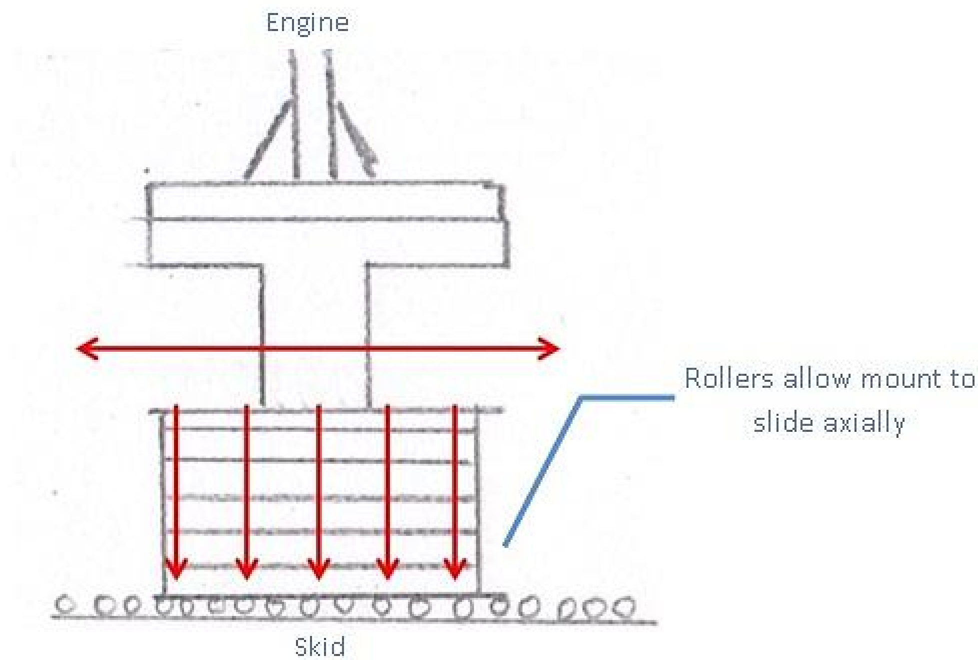


Figure 19. Side view of the current aft-mount. Any expansion or contraction of the engine would be taken into account by the rollers, allowing for a constant load distribution across the spring pack.

The benefit of utilizing roller bearings at the interface between the spring and the skid is that the uneven spring forces will force the aft-mount to shift axially until the spring centerline is aligned vertically again. This will keep a uniform load on the Belleville spring, decrease the effects of fatigue loading on the spring, and increase the life and reliability of the spring. Also, this design calls for very little change to the existing aft-mount allowing for an inexpensive, yet effective solution. Our team will need to look into how much load a roller or sliding bearing can support. Also, this component will be exposed to prolonged heat and vibration which must be considered in the final design.

Machined Springs

The second concept in the elastic design section is to replace the Belleville washer springs with a machined spring. A machined spring could replace the springs in the current design or be incorporated into the concept laid out previously. There are several advantages to machined springs such as spring rate accuracy and linearity. Using finite element analysis, the spring rate can be predicted to +/- 1%. Since the coils of the spring never rest on each other during deflection, the spring constant is linear. Other advantages include one piece design, long life, no cocking or twisting, operation in compression, and can be designed to avoid resonant frequencies. The biggest problem with this solution is that machined springs can cost between 10 to 100 times more than a conventional coil springs. However, a machined spring is ideal for applications that require precise alignment and consistent spring rate.

The first company we looked into is Helical Products Company located in Santa Maria California. An example of what one of their machined springs looks like can be found in Figure 20. This company requests several design requirements such as dimensions, operating conditions, and force-deflection data. From there, they will provide a possible solution if there is one. Initially, they responded to our request saying that the load for the given dimensions was too great for one of their springs to handle. Helical uses only a coil spring design, but there are many more possibilities that can be used in the final concept from other manufacturers . More research will be done with this company and others to see if a machined screw or a combination of machined screws will work for our application. Also, after talks with Solar Turbines, they have given us some liberty as to the envelope that these springs are required to fit into.

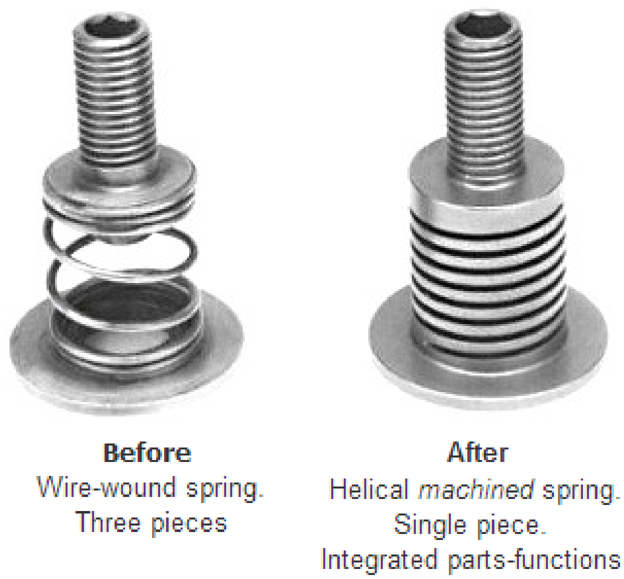


Figure 20. Helical Products Company produces machined springs that can deliver precise alignment and a constant spring rate in one package.

Fluid Concepts

The utilization of the hydraulic mounting method is essentially to provide an option that eliminates the use of a spring pack in the mount. Currently, there are several issues associated with the repeatability due to fatigue and varying environmental conditions that could potentially be alleviated by converting the system to use hydraulics. The working fluid for each design could either be hydraulic fluid or air.

Hydraulic Lift

Hydraulic lift designs are used widely in industry as devices to lift large loads that require precise positioning. For example, hydraulics can be used to lift cars in a mechanics garage, raise bridges, place jet engines, lift stages in a theatre, and many other applications. The Titan 130 has a similar need in that it requires a precise placement of an engine that is applying a load varying between 12 and 15 thousand pounds of force on the aft-mount. A hydraulic lift could alleviate the issues associated with fatigue in the spring pack and shorten the time necessary for aligning the shaft of the turbine.

There are several types of hydraulics that could be utilized in the design for an effective lift. In our case, a single-acting cylinder would likely be most effective as there is only one direction of force that would be necessary in the process of mounting the lift. A double-acting cylinder would be used if there were some need to pull the turbine down, which there is no need present. If the turbine needs to be moved down, pressure would simply be released, applying less pressure and the ram would descend. Analysis for necessary force is linear based on the equation: $F = PA$ In this equation, P is the pressure applied by a pump in the system, and A is the area within the bore of the cylinder. If the area is held at a constant, then varying the pressure will provide different amounts of force on the engine. This can be very useful in that a single hydraulic lift could be used for both the conventional and SoLoNOx engines by using the predictable force curve to set the engine height.

There are several different methods for mounting the cylinder to the skid. Four options for mounting consist of a tie-rod configuration, welding, threaded, and flanged. The flanged option would likely be the best option in the case of the engine aft-mount for Solar Turbines because it offers an easy installation option as well as fast removal for repair or replacement.

Figure 21 shows two different concepts that could be used for the aft-mount. The benefit of hydraulic cylinders is that they can support some vibration similar to the spring pack currently in use. They also provide a tremendous amount of force for very little input. High stiffness in the system would eliminate the vibrations associated with normal operations of the engine. There are a few downsides to this design however. The first issue being that, overtime, the likelihood that the system will lose pressure is inevitable. This could pose a serious danger to the life of the

engine if the shaft loses alignment, and could result in a catastrophic failure. We would have to find a method to lock the ram in place during operation so that the engine would not drop. Also, the operating temperature could be an issue. Seals used to prevent leaking do not operate properly at very high temperatures and could fail at temperatures as low as 150 ^{0}F. If the seals rupture, then high pressure hydraulic fluid could escape and cause damage to other components, including personnel. Finally, another important issue is space. The pump required to maintain the pressure in the system could pose a problem as far as placing the components on the skid.

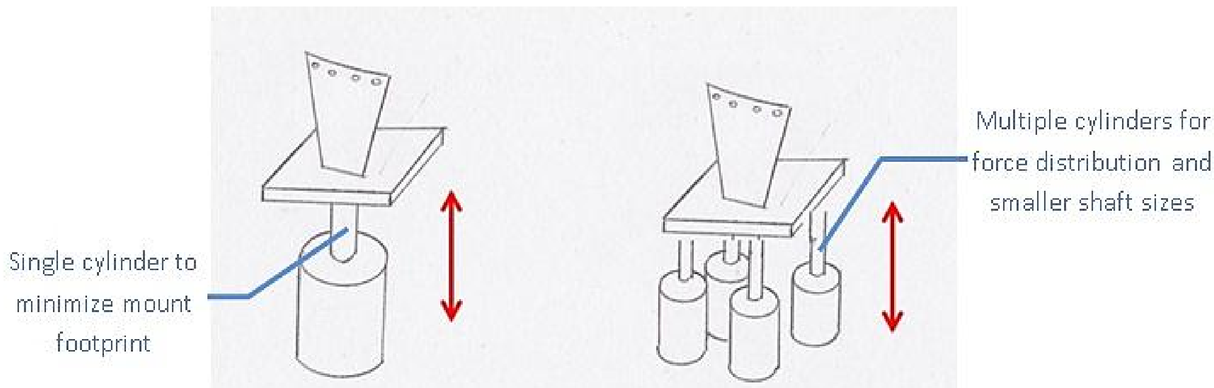


Figure 21. The sketch on the left shows one hydraulic lift connected to a shortened splash plate while the drawing on the right utilizes four lifts to distribute the load.

Hydraulic Wedge

One method for fine-tuned adjustment, as well as decreased leveling time of the aft end of the turbine, is by using hydraulic powered wedges to move the shaft up and down as seen in Figure 22. Instead of the current process of lifting the engine with a crane and adding a shim for a spring pack that is too tall and removing shims for a pack that is too short, we would utilize two hydraulic cylinders to move wedges in and out to compress a spring more or less. In this design, a double-acting cylinder would be ideal. At times, it would be necessary to apply a push force to move the wedges together to effectively compress a spring more. This is analogous to adding shims in the current design. Conversely, a pull force would also be required to move the wedges back out in the event that the spring is too short. This would be equivalent to removing shims in the current system.

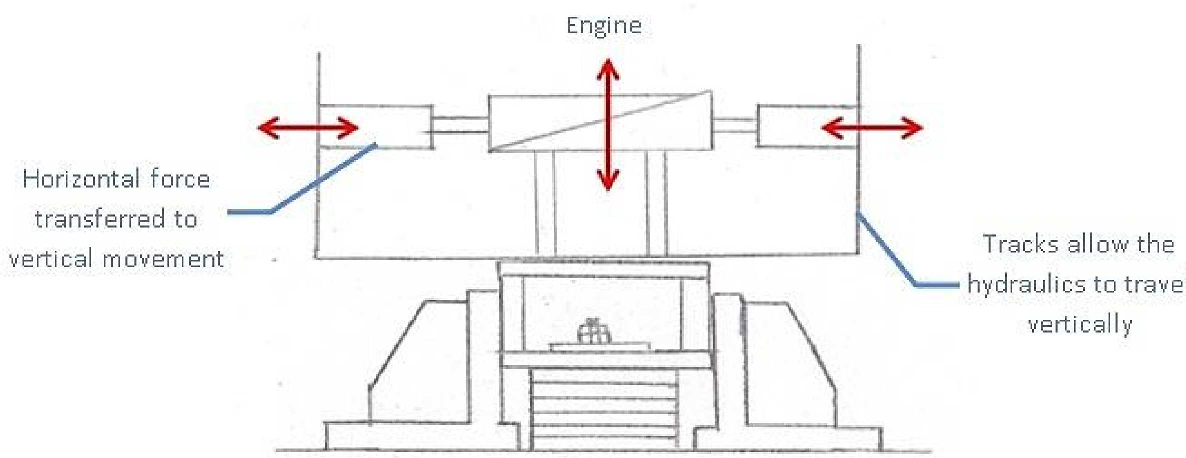


Figure 22. Hydraulic rams drive in wedges to raise and lower the engine.

The hydraulic cylinders would need to be mounted horizontally. In order to accomplish this, there would need to be some sort of track system to allow the cylinders to move up and down as the wedges are pushed together or pulled apart. As the wedges are pushed in, the cylinders would be sliding away from one another, creating a larger vertical distance between the horizontal forces. Also, when the wedges are being pulled out, the vertical distance would be decreased.

Similar to the hydraulic lift concept, this design would need auxiliary equipment that could take up necessary skid space for other components. There is still a concern about the cylinders losing pressure over time, as well as leaks, and the operating temperature of the seals being above the recommended value. One aspect that is different from the lift concept is that there will be bending forces acting on the rods of the hydraulics that are generated from the wedges reacting to one another. This would necessitate some sort of support system possibly for the hydraulics, too. This may result in a fairly complex and expensive structure. Finally, there would be a need for lubricant as the wedges slide against one another to level the turbine and would require upkeep if the turbine ever needs to be leveled in the future.

Mechanical Concepts

After the initial background research, one of our main questions was what type of component could replace a spring. Some options that we looked into were the use of screws, rollers, and gears to solve some of the design requirements. A common theme among the designs is eliminating the shimming process used to initially obtain the required spring deflection in the spring, or to eliminate the springs entirely.

Power Screws

When looking into alternatives to springs, we came across the idea of using a screw. Power screws are capable of handling large loads in a compact, simple to design and easy to manufacture component. Power screws are typically used to generate large forces, yet still obtain a precise axial movement. Since screws offer a large mechanical advantage while still providing precise positioning, they are ideal for this particular application. Another advantage to screws is that they are self-locking so they will maintain position over time.

The first concept using power screws would be to completely replace the current mount with a single power screw as seen in Figure 23. The benefit to this design is that it replaces the springs used in the current design and provides a way to precisely move the position of the engine. For a certain amount of turn in the screw, the engine height can be finely adjusted. A power screw would be simple to manufacture and easily replaceable if needed. Since there are no shims in this concept, the installation time would decrease. Also, since the screw can be made to be self-locking, the alignment will be maintained during operation. One option to help ease the adjustment process would to be to attach a motor to the power screw as seen in Figure 24. This allows the installation crew to be safely away from the engine during the installation process. However, with the additional component, there will be additional cost and maintenance.

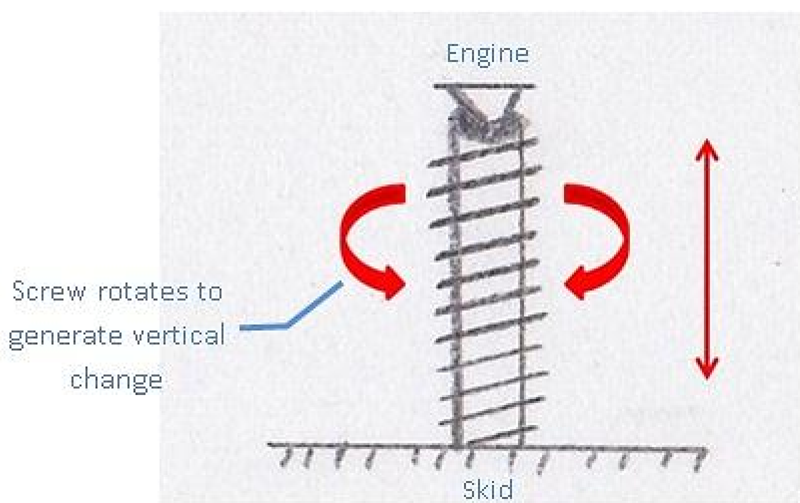


Figure 23. A single screw can be rotated in order to finely adjust the height of the engine.

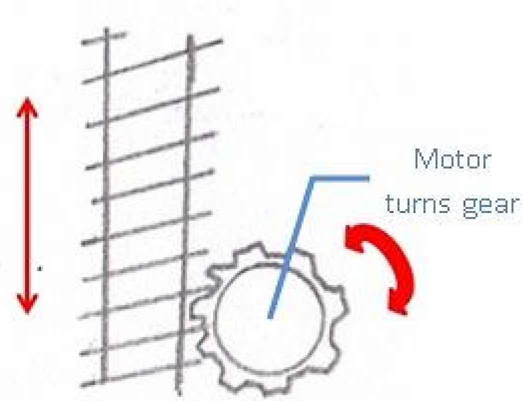


Figure 24. A motor would drive the gear which would in turn move the engine either up or down.

There are several concerns with this concept however. First, is it even feasible to support 12,000lb to 16,000lb with a single screw without buckling? After some initial calculations, the screw diameter would be between four to six inches which is reasonable. Next, since the screw is self-locking, it would probably be very difficult to turn manually. Also, this plan calls for a complete redesign of the skid and engine bracket. Finally, since the springs have been replaced by a mechanical component, the vibrations would go directly from the engine down to the skid without any means of dampening.

Since vibration is such a big concern, another concept using power screws would be to replace the shims in the current aft-mount with a power screw as seen in Figure 25. Since the spring is still in the design, it will reduce vibrations going into the skid from the engine as well as preserving the current assembly procedure. The only difference is that instead of having to lift the engine and replace the shim over and over, the power screw can be adjusted to provide the proper setting. A drawback to this concept is that the additional components to the mount adds cost yet still doesn't account for thermal expansion and uneven spring loading on the mount.

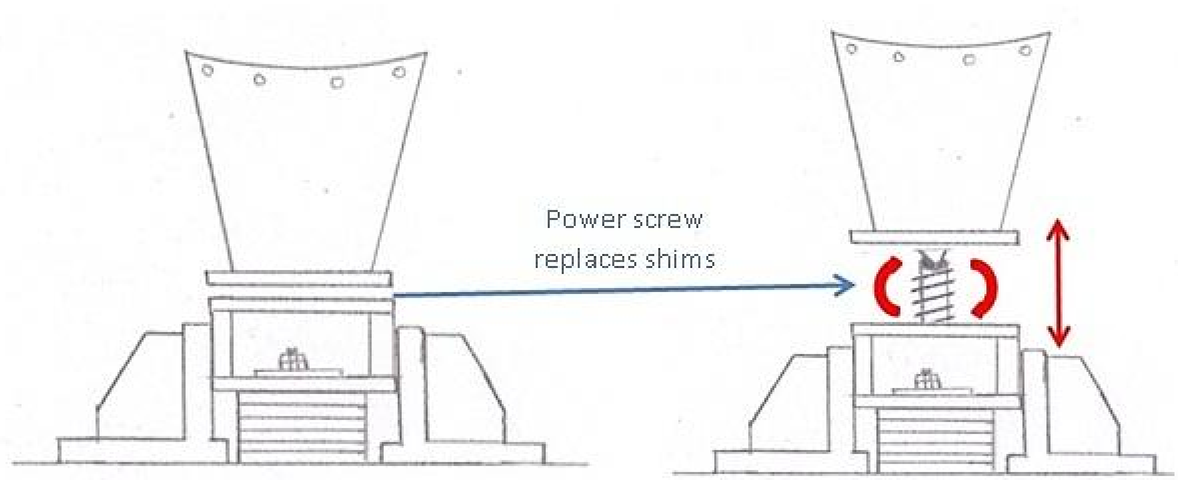


Figure 25. This design replaces the current shim pack with an adjustable power screw.

Another concept design fixes the screw to the skid instead of allowing the screw itself to rotate. Instead, a nut can be turned on the thread to adjust the height of the engine as seen in Figure 26. This design would be a bit simpler to construct since the screw is stationary. Again, vibration, buckling, and effort to adjust is a concern.

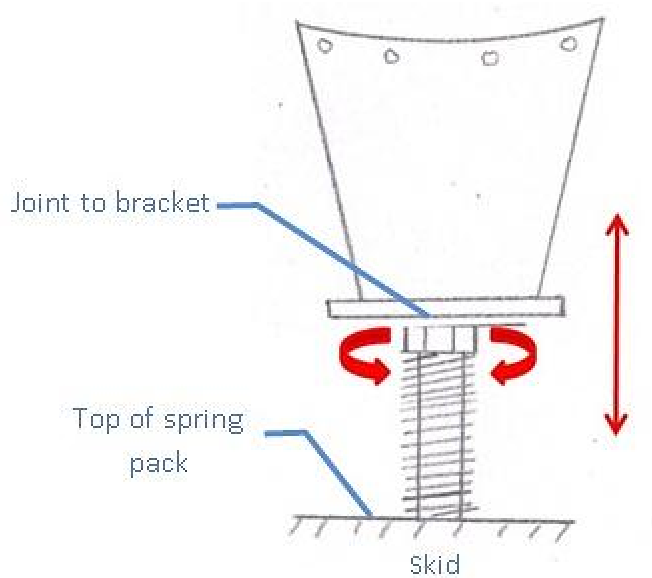


Figure 26. In this design, the screw is stationary while the rotation of the nut causes vertical motion.

Multiple Smaller Power Screws

As an alternative to using a single large power screw to adjust the height of the turbine shaft, four power screws could be used instead with smaller diameters as seen in Figure 27. This would allow the load to be more evenly distributed between the smaller screws instead of all on one screw at the center of the plate. In this design, each screw would be between one and two inches in diameter instead of four to six inches using the single screw. In order to make the height adjustable, a gear train could be designed to transmit torque from a motor out to each of the power screws. By creating favorable gear ratios, this design could yield a method for leveling the turbine very precisely that would eliminate the need for shimming. These screws could be laid out in any square or rectangular pattern as long as the screws would be in contact with the driving gear.

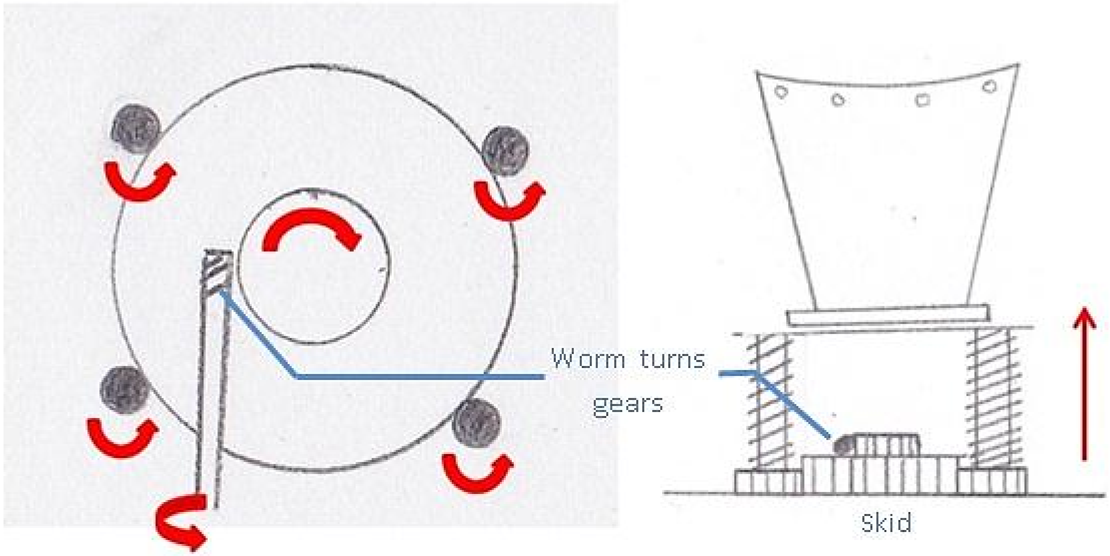


Figure 27. In this concept, a motor can control the vertical change in height through a series of gears and power screws.

There are of course some possible disadvantages to this concept. One of which is that it would require a complete redesign of the current system. This could be expensive to design and implement on current turbines in the field. Also, it would ideally eliminate the need for the current spring pack, which brings up the concern of handling the engine vibrations during normal operation. Finally, the motor necessary for the turning of the gears could be potentially taking up space on the skid for more important components to the system. Also, since there are moving parts in this concept, maintenance will be a concern in the long run.

Jack System

A major concern for the engine aft-mount is the ability to set the level of the shaft of the turbine quickly, efficiently, and in a repeatable fashion. One possible method for accomplishing this goal is to use a scissor jack style mount inspired from the automotive industry as seen in Figure 28. In this design, the rotation of a screw can be translated into vertical motion. This mount type can be found commercially to provide precise adjustment of height by the turning of the main lead screw. By using a gear ratio, we can make it so that the jack is raised or lowered at a very low speed and the overall height of the shaft can be changed precisely. This would eliminate the need for shims and, in fact, can provide a continuous spectrum of adjustment heights and is not limited to 0.005” increments.

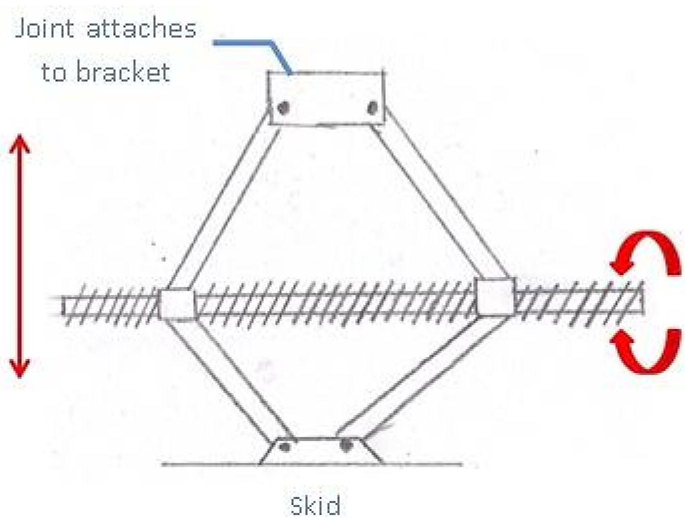


Figure 28. A common jack takes rotational motion and converts it into a linear movement. For this design, the turn of the screw can raise and lower the engine into alignment.

This design is simple and reliable as it is commonly used throughout industry in several applications. For that reason, many different types of scissor lifts are commercially available to buy and would not necessarily need to be further designed for this particular mounting application. This would be a viable option for retrofitting the current turbines in the field, as well as those currently being built.

There are some possible drawbacks that could be associated with this particular design. First and foremost, there is a concern about whether or not there are any current models that are capable of handling the weight of the conventional and SoLoNOx engines. On that note, the large loads may cause the jack to buckle which would result in a catastrophic failure of the gearbox. Also, this design would essentially eliminate the spring pack, so its ability to handle the vibrations generated from the normal operations of the engine is in question. Finally, this would require a motor to provide the torque necessary to turn the lead screw, which utilizes possibly crucial skid space for other components.

PowerJacks.com

A useful site we found while doing research for the scissor jack concept was www.powerjacks.com which is a large manufacturer of high performance machine screw jacks. They offer a wide range of power jacks that come in configurations such as rotating, translating (Figure 29), anti-backlash, and anti-rotation. This manufacturer also provides the supporting equipment such as the motor that would drive the jack. A benefit to using this design is that it is a commercially available, drop in place solution to adjusting the height of the engine shaft. A big concern is that this is a vendor that Solar Turbines would have to deal with. Since they are located outside of the United States, the supply chain would have to be analyzed to make sure that everything arrives for assembly on time. Another drawback to this option is that each spring

jack comes with supporting equipment, such as the motor, that would have to be incorporated into the skid. More research into this manufacturer will be required, but other manufacturers of similar products will also be looked at if this design is utilized.

Figure 29. Translating screw jack produced by Power Jacks

The M-series screw jack delivers precise actuation for 0.25 ton up to 250 ton capabilities. In addition to the features provided in Table 3, these jacks come with many add on features. One feature that could be very useful in our application is a position indicator which would allow the operator to know precisely where the engine centerline is at all times. Another option is to attach a motor directly to the free shaft which will allow the installation team to make adjustments without being in harm's way. Most M-series imperial machine screw actuator models with higher ratios are self-locking and will hold heavy loads in position indefinitely without creep, in ideal conditions. However if self-locking is critical, a brake motor or other restraining device should be considered. They can be used to push, pull, apply pressure and as linear actuators. They are furnished with standard raises in increments of 1 inch. Depending upon size and type of load, models are available with raises up to 25 feet.

Table 3. Summary of M-series features provided by Power Jacks.

Feature	Description
Precise Positioning	Can be controlled accurately for positioning within thousandths of a millimeter
Self-Locking	Will normally hold loads in position without creeping when using the higher ratio units, as long as the actuator unit is not subject to vibration. If self-locking is critical a brake motor or other restraining device should be considered
Sure Operation	Positive acting, for accurate response to motive power
Anti-Backlash Option	Reduces vertical backlash between the screw and the worm gear nut to a practical minimum for smooth, precise operation and minimum wear. Typically used when changing from tension to compression regularly
Corrosion Resistant	Stainless steel models available to prevent corrosion

In order to get some product specifications, we decided to choose the M-series model which is the imperial equivalent to the companies E-series model. A transitional jack was chosen since we do not need anti-backlash or a rotational jack. There are several models within this category that range in capacity form 0.25 tons to 250 tons. We chose model M1815 which has a 15 ton capacity. The SoLoNOx engine has the largest load on the mount at just under 8 tons so we decided to be on the conservative side by using a 15 ton jack. The specifications for this model are summarized in Table 4.

Table 4. Summary of M-series 15 ton jack specifications for model M1815.

Specifications	Quantity
Capacity (US short Ton)	15
Lifting Screw Diameter (in)	2.25
Lifting Screw Pitch (in)	0.5
Lifting Screw Form	Square
Weight with Base travel 6” (lb.)	66
Weight for each additional 1” (lb.)	1.5
Worm Gear Ration	24:1
Turn of Worm Gear for 1” Rotation	48
Maximum HP per Jack (HP)	1.5
Start Up Torque at Full Load (in. lb.)	820
Efficiency (%)	12.9

Design Requirement Components

After going through the above concepts with Ismael DePaz, he directed us to look into the aspects of each design that met a specific design requirement and to begin to group strong concepts together. Repeatability is the main objective at Solar Turbines, and without solving this problem, there is no reason to continue on in the design. In a design that doesn't use the spring pack, we must be able to determine how to properly set up the mount. This problem is a huge issue that none of the previous concept designs accounted for. After repeatability is nailed down, the focus will shift to axial and radial displacement, followed by vibrational concerns. Cost is secondary to Solar in this project because to replace a gearbox costs the company about $600,000. Our goal is to look at the main design requirements and mix and match designs to achieve the optimum solution.

Repeatability

The main problem that Solar has with the current aft-mount is that the engine cannot be set in a repeatable fashion both at the factory and in the field. Due to various loading conditions, the reaction at the aft mount is observed to be different and is suspected to cause problems with normal engine operation. In addition, after the specified overhaul period, a refurbished engine is replaced on the mount and it is again observed to have different deflection characteristics. This is likely due to fatigue of the spring material. In order to be able to successfully implement any new design concept, there must be a proven method of installing and adjusting the new mount to ensure proper turbine centerline alignment. When the aft-mount is initially installed the static loads on the mount are usually determined and adjusted using calipers to measure spring deflection. Then shims above the spring mount are placed to increase or decrease that deflection. If we decide to use a concept design that does not include springs, we will have to determine how installation teams can determine the proper alignment of the engine. Solar was also interested in the idea of having some type of device that the shop crews could use with the current design to check alignment based on load forces.

The first idea our team had to solve the problem of setting the engine precisely was to use a load cell. If an installer was able to read the load on the mount, he or she could adjust the mount in order to get the desired loading. This would eliminate the initial variability of the spring packs since the adjustment would be load driven instead of deflection driven. Initial research into load cells shows that there are many available models on the market today that can meet the loading requirements. Initially we were looking for a load cell that would permanently be installed on the package, however after speaking to Solar, they suggested a possible solution where the load cell would not be permanently integrated into the mount. A tool that the installers and maintenance crews could use to check the alignment may be more favorable considering that it would not have to have as rugged characteristics to endure variable operating conditions.

Many of the previous designs convert rotational motion into vertical motion of the engine. Another way to help the installation crews set the engine with a design like this would be to install a turn counter on the motor shaft. This would give the operators a number that could be recorded at the factory and then duplicated out in the field. This turn counter could be combined with a load cell to give the crews two measurements that can lead to accurate engine alignments.

These concept designs are another addition to the collection of designs that we hope to combine in order to achieve one viable solution to the problem. Since repeatability is Solar Turbines number one objective, these concepts are critical to the final design.

Thermal Expansion

Earlier in this report we discussed how thermal expansion in the axial direction affects the load distribution on the spring pack. The next few concepts expand on the idea of eliminating the effects of thermal expansion on the aft-mount. The first concept simply goes into more detail from the idea of rollers discussed earlier. Hilman and several other companies produce sets of machine rollers that are built to handle large loads. These rollers could be placed under the mount so that when the engine expands axially, the mount will move in the direction of the expansion. Another concept is to use a rail and slide system. Similar to the rollers, this allows the mount to move so that it will always have a uniform load applied to it. Rails may have a longer lifespan than the rollers but this will be looked into further if the final design calls for them.

Consolidated Concepts

From the design development, our team came up with several solutions for the engine aft-mount. While no one design solves all of the design requirements, there are components of each that do solve the challenges faced in the current design. From this toolbox of concepts, we began to combine ideas in order to bring forward three leading designs. Each of those designs is discussed in detail below.

Design I – Elastic Concept

As a replacement concept for the current mount, a machined spring with a slider system at the base could be implemented. A layout of the proposed design can be seen in Figure 30. Maintaining a design utilizing a machined spring would allow for normal engine vibration created by the Titan 130 during its operating time to be isolated from the rest of the skid. In addition, the spring's deflection can be used to account for the radial expansion of the engine during heat soaking and initial shaft centerline positioning. However, the Belleville spring design currently used is suspect to some serious faults. The axial thermal expansion of the turbine causes uneven loading of the spring pack due to its inability to slide with the expansion and harsh environmental conditions cause the current spring packs to crack or yield in such a way that a proper turbine centerline cannot be consistently achieved. A revised spring design could eliminate the negative characteristics of the traditional spring pack as well as enhance the beneficial features of the design. This too, would not require a major redesign of the package's skid and would not necessitate extra bulky hardware.

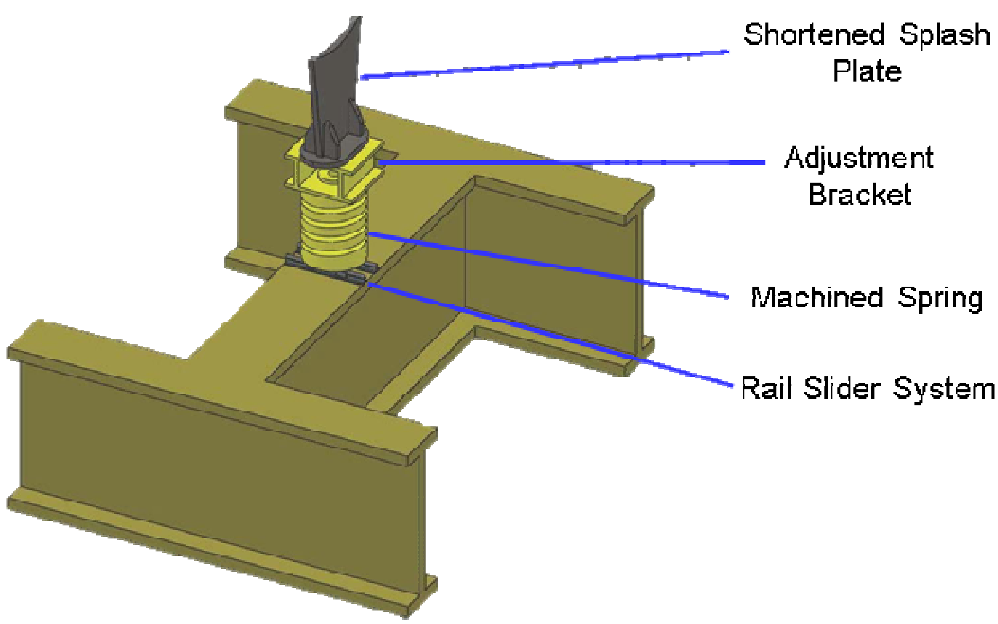


Figure 30. Elastic concept design layout showing the machine spring and slider system.

With a plethora of spring types and designs available, the most appropriate spring types are difficult to narrow down for most applications. However, because there is a significant amount of test data generated from the current springs, it is possible to select the best spring type for this particular application. First, the spring constant is a very important feature and a major factor in choosing a new type of spring. The current spring pack delivers a spring constant of 18,467 lbf/in for a conventional engine spring compression and 17,204 lbf/in for a SoLoNOx engine spring compression (the spring compression necessary to provide enough radial force to align the turbine and gearbox.) There is approximately 3% variability in spring stiffness when multiple spring packs were tested in operation as seen in Figure 10. Thus improvements can be made to attain a more consistent spring force. The large loads placed on the aft-mount eliminate the possibility of using many common spring types that would not be able to produce the needed spring force or deflection demands. A common approach to heavy loading conditions calls for using commonly available Belleville springs (or washer springs) as the current design utilizes. It has become apparent though, that these springs are not appropriate for the operating conditions the aft-mount experiences in the field all over the world.

Through research performed on the different spring designs available for heavy loading, harsh environmental conditions, and linear spring constants, it was decided that a machined spring would be the most effective in satisfying these requirements. One of the most useful features of a machined spring is that it can be built to nearly any specification the user requires. For instance, a multitude of spring constants, end loading conditions, and internal damping can be achieved. One design for a machined spring that is proposed for the aft-mount has a spring constant of 17,836 lb./in and a 1.12 inch deflection under a 20,000 lb. compression load applied. Moreover, it would have a minimum life of 10 million cycles when loaded with at most a 13,000 pound load alternating plus or minus 1000 lbs. However, simply replacing the current spring pack with a new machined spring would not satisfy all the conditions necessary for repeatable loading and unloading of the aft-mount. Therefore, more redesign is necessary to deal with the uneven loading and thermal expansion issues that arise during turbine operation. When the turbine is operating, it gets very hot (exhaust gasses at approximately 900°F). Even with a substantial amount of thermal insulation the aft-mount is heavily affected by the thermal expansion of the turbine. The axial thermal expansion pushes the mount aft towards the exhaust bellows and the radial thermal expansion increases the vertical force on the aft-mount and spring pack causing uneven loading situations as seen in Figure 6. This can be negated by allowing the aft-mount to slide axially on the skid while the turbine expands, thereby maintaining a uniform load on the mount. Also, the radial forces caused by the radial expansion of the turbine are less effective on the now more linearly deflecting machined spring, which may be more accurate with maintaining shaft centerline during operation. A CAD diagram of the proposed machined spring can be seen in below in Figure 31.

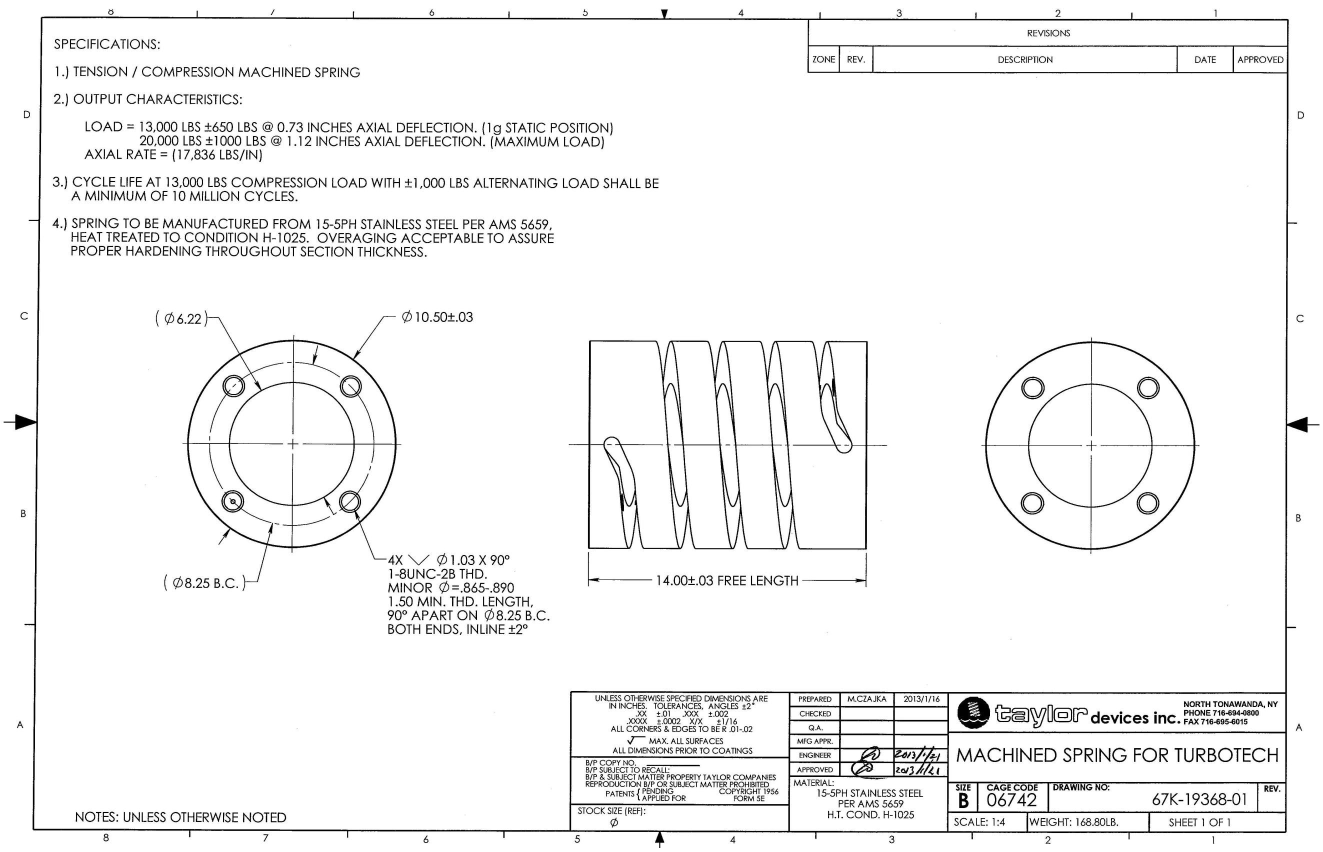


Figure 31. Detail drawing of machined spring produced by Taylor Devices.

In order to accomplish the task of sliding the aft-mount axially along the skid, a slider system, Figure 32, from PBC Linear has been chosen to be mounted between the base of the aft-mount the skid. The low friction, dual rail slider system is rated up to 18,750 pound load as well as a wide range of temperatures. Mounting the rail slider system to the skid should not require extensive redesign of the skid or mounting pedestal with its low profile of 5.875 inches in height and 18 inch width at its base and standard rail length of 24 inches. The mounting pad on the rail slider is 18 inches by 18 inches which is more than enough area to mount the 13 inch by 18 inch aft-mount base plate securely to the slider by either a permanent weld or by using fasteners.

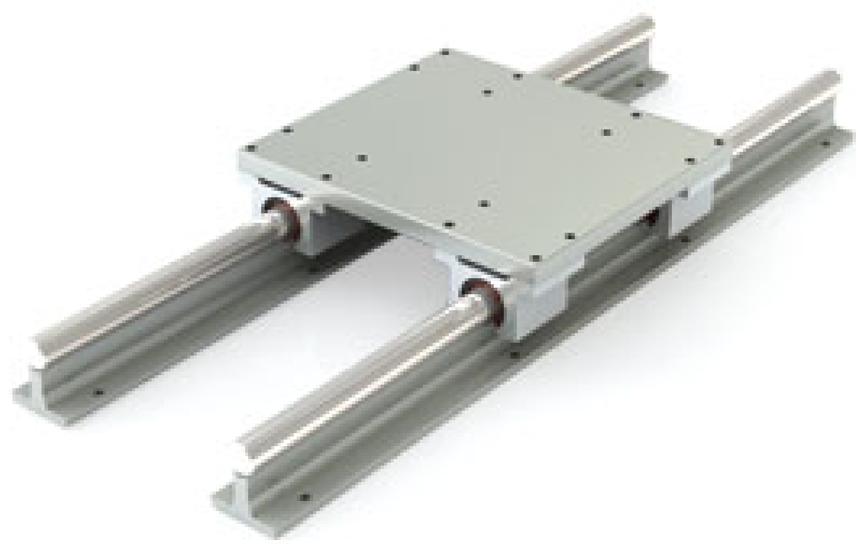


Figure 32. PBC Linear low profile LRPS rail system.

The main purpose of the aforementioned design and its selected features is to enhance the beneficial qualities of the current aft-mount design and its spring pack, but also to eliminate or substantially decrease its undesirable qualities as well. Belleville springs are useful when a system needs to handle heavy loads, but cannot maintain the consistent spring rate required of a highly repeatable system during situations involving uneven loading. Thermal growth causes uneven loading and a large moment force on the current aft-mount, and springs in harsh conditions can crack and fail. All of these issues can be corrected by utilizing the correct machined spring. With a more linear spring constant and high cycle life, a machined spring can be fully loaded and withstand the vibrations of the Titan 130 turbine in operation and maintain the necessary centerline alignment up to its overhaul period of 30,000 hours. At which point, it can then be reloaded and adjusted again without issue. The linear slider system allows for slight aft-mount movements to correct for the axial thermal expansion of the turbine.

Design II – Fluid Concept

For a fluid design, it would be important to incorporate the fluid properties, be it air, water, or some hydraulic fluid, to maintain the shaft alignment during the engine's operating life of 30,000 hours. A hydraulic cylinder supported by a pin and trunnion system at the base, and a clevis pin at the top where the mount attaches to the turbine was used for this design. A model of the design can be seen in Figure 33. Each of the pins would be supported by type E heavy duty roller bearings to accommodate the axial expansion of the engine. Finally, a pressure transducer could be used to measure the load that the mount would be withstanding in operation for repeatability in the mounting of the engine.

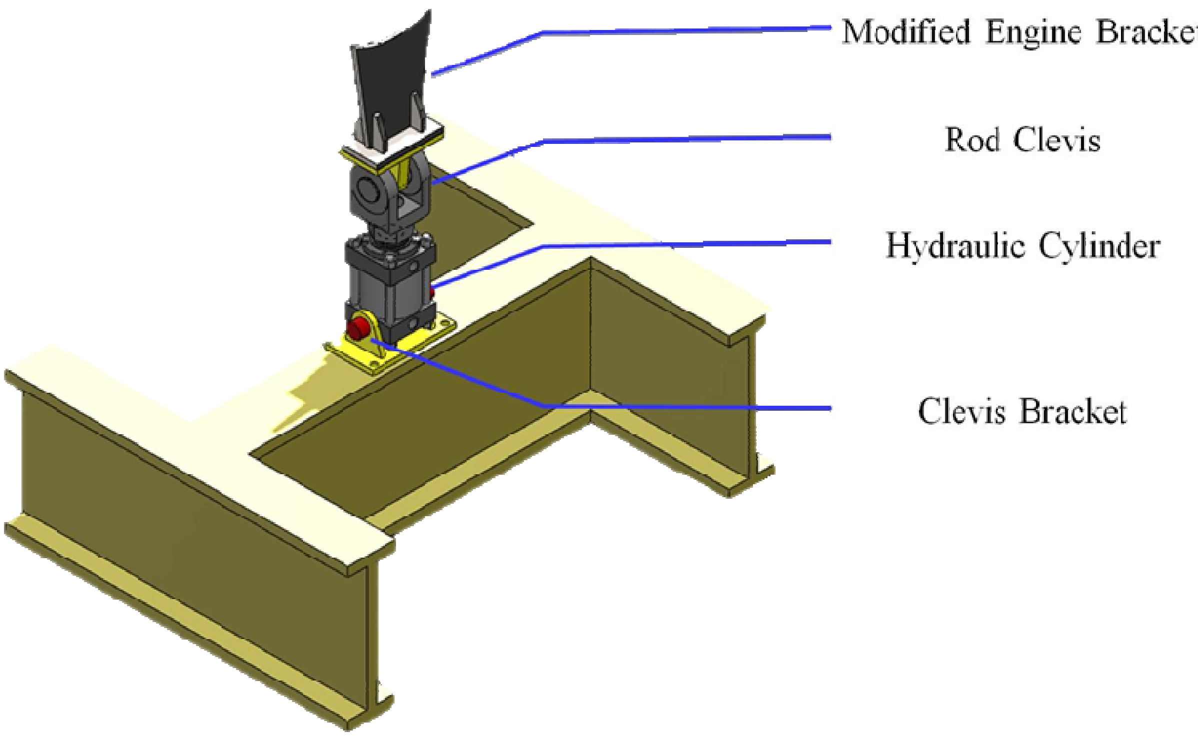


Figure 33. Mechanical concept design layout showing the motor and turn counter.
pushed away from the jack for clarity.

Beginning with the most important component, the main method for holding up the load that the engine puts on the aft-mount would be supported by a heavy-duty hydraulic cylinder. Hydraulics are used throughout industry to lift very large loads including cars for a mechanic's shop, a stage in an opera house, and even bridges. A cylinder with an adequate bore size would easily lift the expected static and dynamic loads that are expected from the Titan 130. The SoLoNOx version has an expected static load of approximately 15,500 pounds to get the shaft into proper alignment. When the dynamic vibration is taken into account, a load factor of about

1,500 pounds can be added onto the static loading condition. With this taken into account, the bore size was determined by using a load of 18,000 pounds for added factor of safety. The following equation used for pressure calculations is the driving method for determining the necessary bore size of the cylinder.

$$F = PA$$

An iterative process would eventually be used to determine an adequate bore size for the cylinder. The equation can be rearranged for area (A) and by assuming an initial value for P to be 500 *psi*, the diameter of the bore could be calculated from that. For this application, it would be necessary to decide what style of hydraulics would be used. The selection was narrowed to either using a single-acting or double-acting cylinder. The difference being that for the single-acting cylinder, it would only push in one direction and not apply any pull force. It was decided that a double-acting cylinder would be most beneficial in this application so that the cylinder could always maintain the initial loading condition. For the end with the rod protruding, it would be necessary to determine the working surface area by subtracting the area of the rod from the total area found by rearranging the pressure equation. This eventually resulted in a starting bore size of 8 *in* and a rod size of 3 *in*. In moving forward with this design, the bore size would be optimized using the above described iterative process; however this was used as a starting value for calculations.

Several companies were researched to buy this component of the mount from. This included companies such as Milwaukee Cylinders and Parker Hannifin. They provide the most complete list of cylinder types, end conditions, mounting techniques, and system accessories that are necessary when designing a system such as this. From their respective websites, it was determined that it was possible to buy heavy-duty cylinders with bore sizes ranging from 1 *in* to 12 *in* with rod sizes ranging from 0.5 *in* all the way to 8 *in* depending on the size of the cylinder's bore diameter. There are over 15 different methods for mounting the cylinder to the existing equipment that the hydraulic system would be used with. It was decided that the best configuration would be an MT2 type pin and trunnion system seen in Figure 34 below would be ideal for the aft-mount application. At the top where the rod would connect to the engine, a clevis and pin could be attached to the rod using a KK2 type thread on the rod so that once the cylinder was in place, it could act as a pinned two force member. This mounting configuration would allow for the mount to be set initially out of alignment, and as the engine reaches the heat soaked condition, the mount would finally rest in the vertical position due to axial expansion.

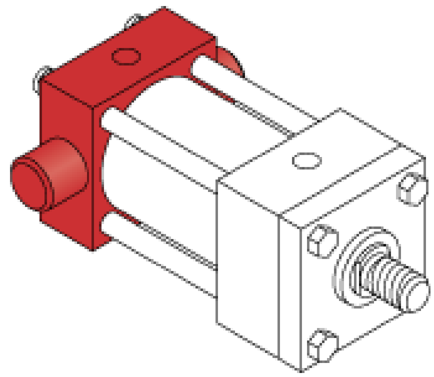


Figure 34. This represents a cylinder with a MT2 blind end pin and trunnion style mount. Courtesy of Milwaukee Cylinders.

In order to get the rotation that is necessary in this concept, the pins at either end of the cylinder would need to be on some sort of bearing system. It was determined that the best action for this would be to apply pillow block style tapered roller bearings at the skid and engine connection points. Given the very large radial loading characteristics that the bearings would have to endure, type E heavy duty roller bearings would be the most ideal and could definitely withstand the loading at the expected low spin speed. Timken provides the most comprehensive spreadsheet for determining the safe radial loading conditions for this type of bearing. Given that our bearings would have pins going through the inner race of 3 *in* at the base of the mount and 4 *in* at the engine connecting point. Based on the data that can be observed in Table 5, it was determined that the maximum load per bearing at the base could be 12,300 *lb.* and for the engine connection it would be 26,900 *lb*. In this design, there would be two radial bearings at the base straddling the sides of the cylinder, and a single bearing at the top to connect with the engine. Given the values obtained from the table mentioned above, a bearing pattern as described above will be more than adequate. The bearings could be mounted to the engine and skid by either using a two bolt or four bolt style bearing. It was determined that to get the most stiffness in the axial plane of the engine to use a four bolt pattern bearing.

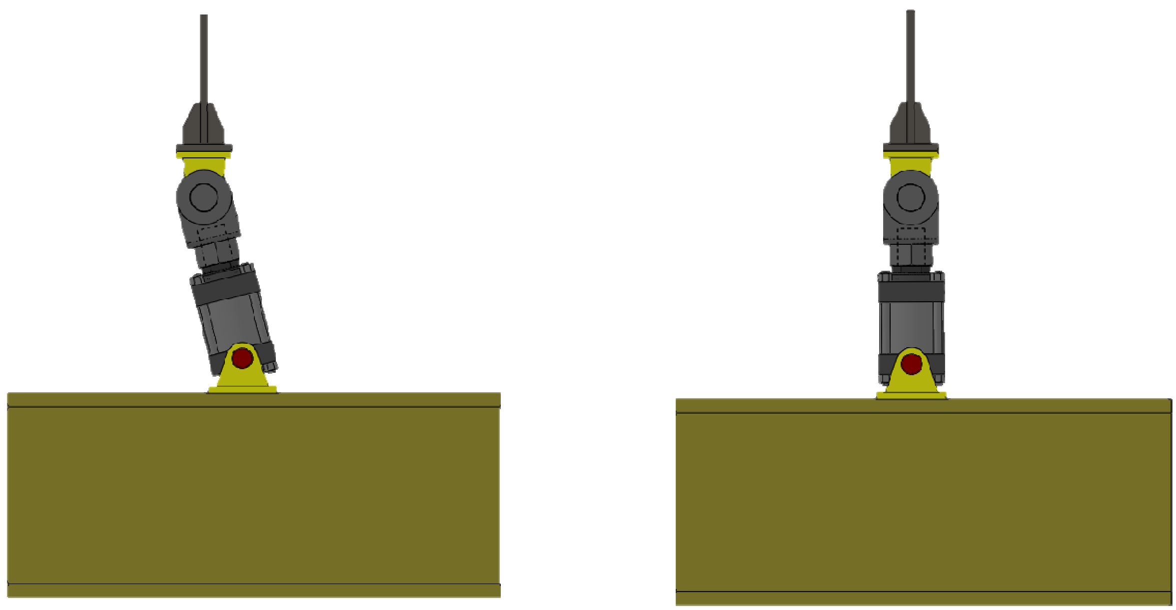
Table 5. This is an abbreviated table provided from Timken that shows the radial loading for type E heavy duty roller bearings. For a bearing with a 3” shaft, the maximum radial load for the bearing is 12,300 *lb.* and for a 4” shaft, the maximum radial load for the bearing is 26,900 *lb*.

Shaft Diameter	Basic Dynamic Load Rating C_{90}	Max Speed Timken Double-Lip Seal	Life L_{10}	Equivalent Radial Loads Allowed, P_r at Various Speeds, RPM																		
				50	100	150	250	500	750	1000	1200	1360	1530	1640	1750	2060	2420	2730	3050	3320	3820	4490
in. mm	lbs.	RPM	Hrs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.	lbs.
2 11/16			10000	17102	13891	12300	10552	8571	7590	6962	6591	6348	6128	6002	5886	5605						
2 3/4			30000	12300	9991	8846	7590	6165	5459	5007	4741	4566	4407	4317	4233	4031						
2 15/16	12300	2060	40000	11283	9165	8115	6962	5655	5007	4593	4349	4188	4043	3960	3883	3698						
3			60000	9991	8115	7186	6165	5007	4434	4067	3851	3709	3580	3506	3439	3274						
70			100000	8571	6962	6165	5289	4296	3804	3489	3304	3182	3071	3008	2950	2809						
75																						
			10000	37401	30379	26900	23078	18745	16598	15226	14415	13884	13402									
3 15/16			30000	26900	21850	19347	16598	13482	11938	10951	10368	9986	9639									
4	26900	1530	40000	24676	20043	17747	15226	12367	10951	10045	9511	9160	8842									
100 mm			60000	21850	17747	15715	13482	10951	9697	8895	8421	8111	7829									
			100000	18745	15226	13482	11566	9395	8319	7631	7225	6959	6717									

In order to maintain the shaft alignment of then engine, it would be necessary for the fluid, once pressurized within the cylinder, to be locked into place by using some sort of shutoff valve for each side of the cylinder. This would act as a method to maintain the pressure that is initially set when the engine is set in place at the shaft center line. This also means that the sealing technology used in the cylinder would need to be as high of a quality and redundant as possible. There are two main types of connectors that are generally used for hydraulics, NPT and SAE O-ring style connections. The NPT type connector uses mechanical deformation of the threads to act as a seal and tends to leak with larger diameter openings. The SAE O-ring uses an O-ring to make perform the seal and would therefore be the optimal connection choice. There are also several options for the materials of the seals that can be used to prevent leakage in the engine. The most commonly used seal types that work with the largest range of hydraulic fluids include Viton style seals and the Buna-N style seals. The Buna-N seals, where N stands for Nitrile, have a rather large range of operating temperatures from -65°F to 275°F. Given that there are some environmental conditions that the engines will be placed in that can reach -40°F, these particular style seals may be ideal. The Viton seals provide many of the same characteristics as the Buna-N seals, with the main differences being higher temperature operating range and more chemical resistance characteristics.

Finally, in order to meet the initial loading condition that is described in the engineering documents for both the conventional and SoLoNOx engines, it will be necessary to have a load measuring device. For the hydraulic cylinder, the best method for accomplishing this would be with a pressure transducer. Using this technology, the pressure within the cylinder could be adjusted very accurately to obtain the specific load conditions for each engine.

This fluid concept design manages to meet the design requirements that have been previously outlined. The hydraulic cylinder can easily lift the required loads that are expected through the life of the engine including the static load and the dynamic loading from engine vibration. This can be done in a repeatable manner by using the pressure transducer described above so that each time the engine is set, the pressure from the pump can be read, and then the cylinder can be locked in. Although liquids are generally considered as incompressible, they actually do compress somewhat which can be calculated using the bulk modulus of the fluid. From this calculation, the spring constant of the mount can be found. Because of this, the fluid can be selected so that the bulk modulus will produce a stiffness that will be ideal for either the conventional or SoLoNOx engines. Finally, the design accounts for the axial thermal expansion of the engine in the design of the mounting method for the cylinder. With a pin and trunnion system at the base and engine connections, it will allow the mount to be set initially out of alignment, and as the engine reaches the heat soaked condition, the mount will be able to rotate into a vertical position as seen in Figure 35.



(a) (b)

Figure 35. (a) Shows the initial setup of the mount. It would start out angled so that as the engine starts heating up, it would be able to expand into the final position shown in (b).

Design III – Mechanical Concept

The mechanical design uses a power screw jack to support the load of the turbine with a load cell and turn counter in order to achieve a repeatable setting for the engine. A rail and carriage sliding system account for the axial expansion of the turbine while vibration dampening material under the jack will provide a way to dampen the vibrations through the aft-mount. A layout of the design can be seen below in Figure 36.

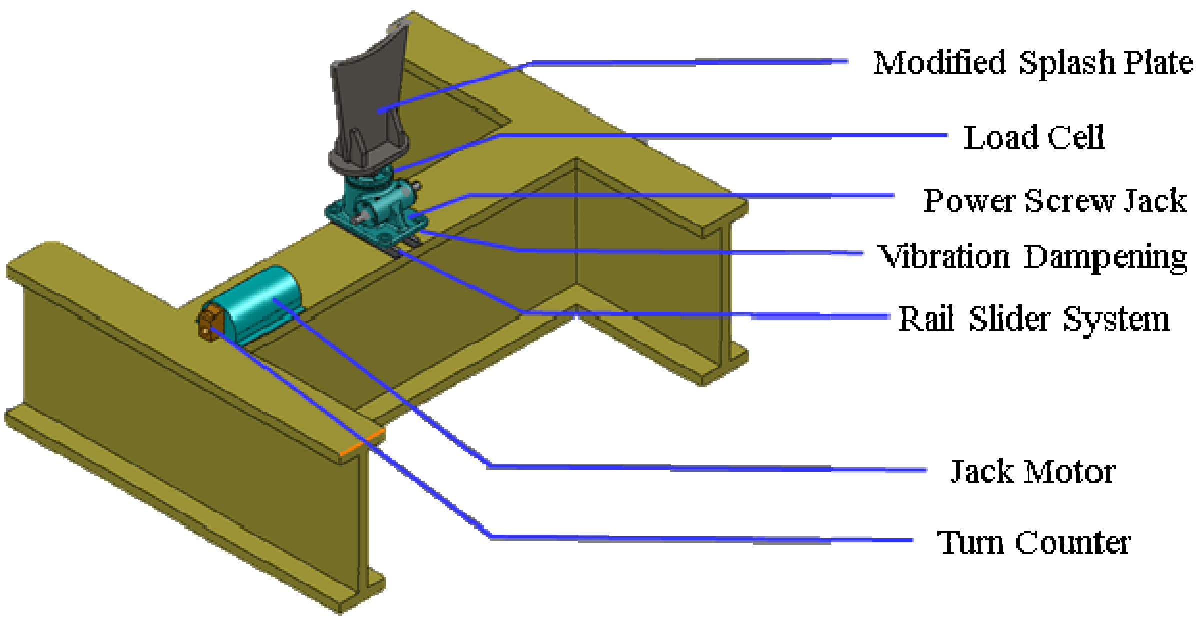


Figure 36. Mechanical concept design layout showing the motor and turn counter pushed away from the jack for clarity.

The main component of the system is the power jack. In this design, it is crucial for the jack to be able to fully support the load and maintain position over the life of the engine. Several manufacturers provide jacks capable of supporting the load including Joyce-Dayton, Duff-Norton, Power Jacks, and Zimm Screw Systems. The decision was made to use Joyce-Dayton as the jack provider since they are a leading manufacturer based in the United States and they provide complete systems including motors and turn counters. Joyce offers jacks with a worm gear ratio of 25:1 which can give 0.01" of screw rise for every one turn of the input shaft which fits into the tolerance specified at Solar Turbines. Also, several end conditions are available for the end of the lifting screw including a load pad option that would be capable of holding the load cell in place. Finally, there are many additional options including anti-backlash devices, turn counters, hand cranks, or motors for the jack system.

The next crucial component to this design is the load cell. Futek, Flintec, Omega, and Honeywell produce a range of compression load cells that are capable of handling the loads experienced on the aft-mount. The biggest design challenge is that Solar Turbines does not want to have the load cell installed for the life of the mount. They would prefer that the load cell be used as a field tool for installation purposes only. In order to get around this design challenge, the idea of a strain gage on the engine bracket was brought to the table. If a strain gage was applied to each splash plate, then the power source and data acquisition system could be used as the field tool. There were several problems with this idea however. Even though the strain gages are inexpensive, they are quite difficult to install and calibrate. Additionally, the strain gages need to be protected in order to operate properly. The decision was made to stick with load cells for this reason. Solar Turbines has a Honeywell Model 41 precision low profile load cell with data acquisition system in house so this is the model chosen in the design.

The mechanical design is the stiffest aft-mount of the three. In order to reduce vibration, several vibration dampening materials were examined. Farrat makes a high grade nitrile rubber seen in Figure 37 that is used on shipboard machines, pumps and compressors, injection molding machines, and blow molders just to name a few. Another option was Industrial Noise Control's D306 PVC vinyl based material. In addition to the vibration dampening characteristics, this material can withstand temperatures between 0°F and 225°F. More research will have to be done for the long term life characteristics of these materials if this option is to be utilized.

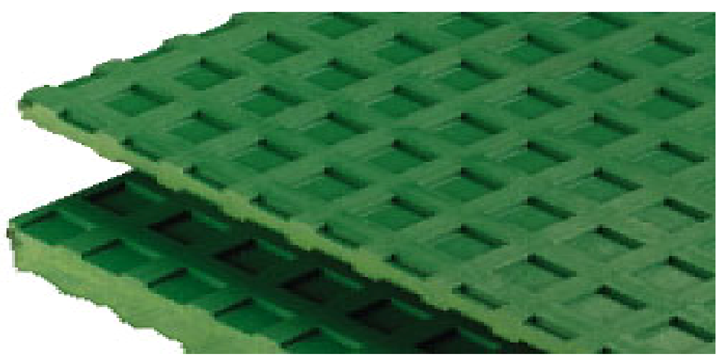


Figure 37. Farrat high grade rubber machine vibration dampening pad.

The final design hurdle was dealing with the axial expansion of the engine. Since this design uses a power jack, it is critical that the load remain directly over the jack so that the lifting shaft remains vertical. Bearing slider systems were the obvious answer but finding a system that could handle the high load and variety of environments was a challenge. There were two main types of linear motion solutions that were examined. The first uses roller technology on the side of the load and a C-channel to allow for linear motion. The second uses a set of rails underneath the load and a carriage to provide for the motion. The decision was made to use the rail and

carriage system since there is support directly under the load. The first option allows for deflection to occur in the base plate which could lead to the slider system not working properly. PBC Linear provides a linear actuator technology carriage and rail assembly, seen in Figure 32, that is capable of handling 18,750lbs which exceeds the load requirement. It will take about a thousand pounds of force in the axial direction of the engine in order to break the static friction and slide with the thermal growth. Since this system can handle the load and can operate in a wide range of temperatures, this is the best option for meeting the axial expansion design requirement.

Since repeatable engine alignment is the main goal of this project, the mechanical design utilizes the combination of the power jack, load cell, and turn counter in order to achieve the repeatability of installation. At the factory, the jack is installed on the skid and the load cell is attached, the jack can be raised to the approximate position by using the turn counter and a specified setting. Once the jack is in position, the engine with shortened splash plate is lowered onto the jack and load cell assembly. The jack is then adjusted to meet the specified load on the load cell. Once this position is recorded on the turn counter, the load cell is removed and the jack is raised the appropriate amount to account for the removed load cell. Once the package arrives onsite, the recorded turn counter position can be used to set the engine in the field. If the load cell does not match the specifications, the jack can be adjusted to fix any preload conditions present from the exhaust bellows. Then, the final position from the load cell is recorded and stays with the package for the remainder of its life at that location. Through this process, the engine can be set reliably every time. Since the design is rigid, it will maintain this position for the life of the engine.

Selection Method

In order to bring forward a leading design, a weighted matrix, see Table 6, was set up in order to judge the concept designs against the current mount. The mechanical, fluid, and elastic designs were given either a positive value if the characteristic was better than the original mount, a zero if comparable to the datum, or a negative value if the concept is not as strong as the current mount. Each of the ratings was multiplied by the weighted importance factor and summed to get a total value. The design with the highest final value is the best theoretical concept.

A set of design characteristics were used in order to compare the three concept designs. Repeatability and ability to expand axially received the highest weight factor because they are the number one goals for this redesign. Vibration absorption capabilities and repeatability of the mount received the next highest weighted rating because they are important to consider but are secondary to the main design objectives. Mount installation time, maintenance in operation and after the 30,000 hour operating life, as well as time to set up the alignment of the engine were also considered in the matrix. Finally, manufacturability and cost were ranked with the lowest weighted factor because, while important, they are not as limiting to the final design as the previous design considerations.

The benefits of each design have been laid out previously in the consolidated concepts section however each design has their own respective weaknesses. For the elastic concept, the cost of the spring is very large compared to the other designs, which makes it somewhat prohibitive in that regard. The initial estimate came in to be \$10,000 for the purchase of the first spring which would eventually be reduced to \$7,000 for each subsequent purchase. In contrast, the other designs will cost around a few thousand including the supporting material and machining costs. In the fluid concept, maintenance is a huge concern. There is a significant possibility that a leak in the hydraulic loop could form, which would likely lead to a catastrophic failure of the engine. Finally, the mechanical design's rigidity is a concern for other areas in the skid. Essentially, it eliminates any isolator characteristics that the previous mount had and all vibrational dampening is lost. Since there is no dampening from the engine to the skid, other components in the package could be affected by the vibration, and therefore be damaged.

Table 6. Decision matrix used to evaluate the three leading concept designs.

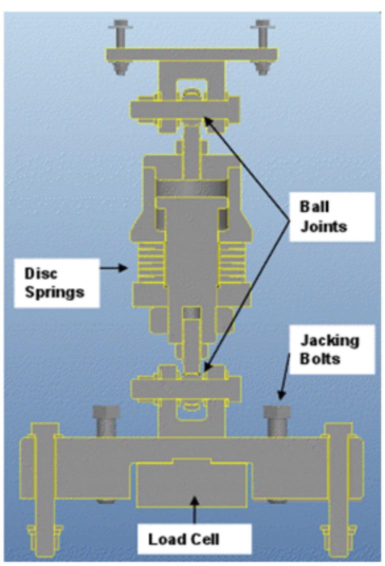
Design Considerations	Weight Factor	Mechanical Concept Rating	Mechanical Concept Wt'd Rating	Fluid Concept Rating	Fluid Concept Wt'd Rating	Elastic Concept Rating	Elastic Concept Wt'd Rating	Datum Rating
Repeatability Components	5	1	5	1	5	1	5	0
Axial Expansion	5	1	5	1	5	1	5	0
Setup Time	2	-1	-2	-1	-2	0	0	0
Mounting Installation Time	3	1	3	1	3	0	0	0
Maintenance in Operation	3	0	0	-1	-3	0	0	0
Maintenance During Overhaul	1	1	1	1	1	1	1	0
Manufacturability	1	0	0	0	0	0	0	0
Reliability	4	0	0	-1	-4	0	0	0
Cost	1	-1	-1	-1	-1	-1	-1	0
Vibration	4	-1	-4	0	0	0	0	0
Total			7		4		10	

From the decision matrix above, the elastic concept received the highest rating. Even though the mechanical and fluid concepts meet the design requirements, they take a serious hit in vibration dampening and maintenance respectively. The elastic concept is so similar to the current mount that it is equal in many ways. One thing that is not shown in the matrix is that the cost of the elastic concept is much more expensive than the other designs. Each concept received negative scores in the cost category but the cost to produce the machine spring is an order of magnitude more than the mechanical or fluid design. In our opinion, with higher risk comes with higher reward. Even though the fluid and mechanical concepts have not been used in this industry before, they offer a more complete solution to the problem. The mechanical concept design is the leading option that will be brought to Solar Turbines.

FINAL DESIGN

After the Set Base Engineering design presentation, each design was given feedback from Solar Turbines. Cost was a major concern with the elastic concept since the spring is over ten times the cost of the current aft-mount. Another concern is that the spring will deflect before the sliders have a chance to correct for the axial expansion. For the fluid design, the sponsor was mainly concerned with the possibility of catastrophic failure and the fact that this practice has not been used in industry before. With the mechanical design, vibration effects were a concern since the design is so rigid. Solar Turbines also made it clear that even though there is plenty of complexity in the skid, supporting equipment such as this mount should remain simple and be designed to take a beating in the field.

Since no one design was able to fully meet all the design requirements, the team at Solar made the decision to present two internal designs to our team. The first design is for the Taurus 65 which is a smaller engine than the Titan 130. A cross section of this design can be seen in Figure 38. Another concept that is being developed is for the Titan 130 which is also in Figure 38. This concept uses a hydraulic jack with a load cell to lift the engine into place. Once alignment is set, two wedges are placed to maintain the position of the engine. This design is rigid, so another design, Figure 39, uses the same idea for setting the engine but retains the original spring pack to dampen the vibration. The decision was made to give these concepts to Turbotech with the goal to blend designs in order to achieve a quality final product that meets all the design requirements.



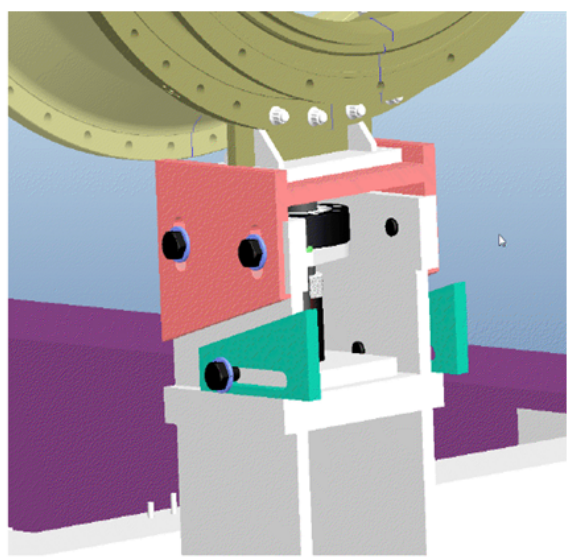


Figure 38. Left, Solar Turbines concept drawing for a Taurus 65 engine mount. Right, Solar Turbines proposed aft-mount for the Titan 130. This concept design does not have a provision for axial or radial expansion and is ridged. A hydraulic jack and load cell is used to set the alignment of the engine.

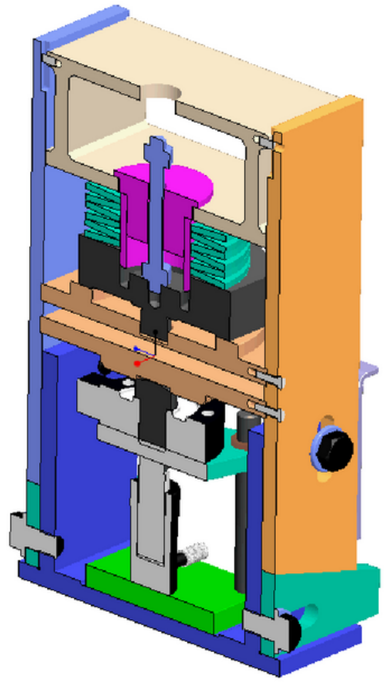


Figure 39. Cross section of Solar Turbines proposed aft-mount for the Titan 130. This concept design includes a spring pack which can account for radial expansion and vibration dampening.

Once our team was given Solar's internal design, another brainstorming session was held in order to determine the best option for the final aft-mount design. The first change was to rotate the proposed aft-mount design in Figure 39 by 90 degrees. This allows for better access to the lifting mechanism and load cell from the side of the skid. Next, one of the two wedges was turned around giving the mount another fail safe feature without any additional machining cost. In addition, a slider mechanism was proposed to accommodate for the axial growth of the engine. Finally, a power jack will replace the hydraulic concept in the ultimate design for the new aft-mount.

Another possible solution that was not implemented in the final design was a machine spring concept. As stated in the elastic design section, the inclusion of a machined spring utilizes a linear spring constant as opposed to the nonlinear stiffness of the Belleville spring pack that is used in the current mount. Research has been conducted on the manufacturability of a machined spring and the cost is prohibitive and ultimately resulted in its omission from the final design.

The final aft-mount design as seen in Figure 40 incorporates the best features of all the proposed designs into one consolidated design. The only feature in the final design that is carried over from the original design is the Belleville spring pack which is the exact same as is

used on the current aft-mount design. The original splash plate is modified to be shorter to allow for the taller components of the new design and also reduce the moment caused from a longer bracket. The interface between the new bracket and the turbine was left unaltered since this was one of the design constraints. The key feature of this final design is that it integrates a mechanical screw jack in conjunction with a load cell in order to accurately set the engine into alignment The Belleville spring pack sits securely on the top plate assembly which in turn rests atop the base plate assembly. The top plate assembly's wedged side faces are coincident with the outer plates of the base plate assembly so that two wedges, facing opposite directions, can be used to support the upper plate assembly once the aft-mount has been adjusted to the proper height using the mechanical jack and load cell. The base plate is mounted via studs tapped into the linear rail slider system and secured with lock washers and lock nuts. Finally, the top plate assembly and wedges are bolted into place and the jack and load cell are removed from the aft-mount.

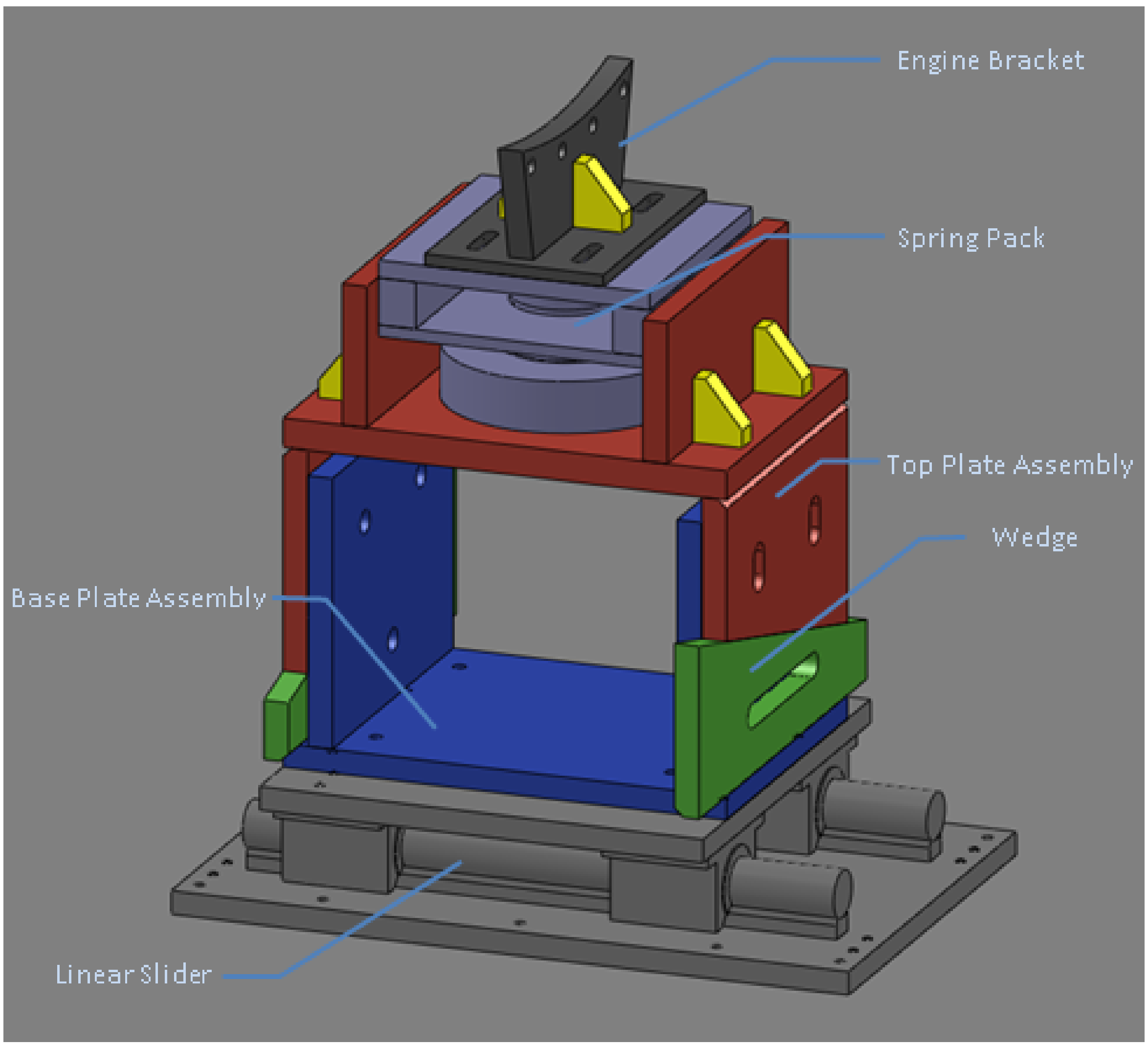


Figure 40. Final aft-mount design proposed for the Titan 130.

The most obvious difference in the final aft-mount design compared to the current model is the support structure that transfers the load from the engine through the spring pack and into the rail slider. The structure of the top plate and base plate allows the engine bracket to be shorter, places the spring pack closer to the engine, and allows the jack and load cell to be removable. The base plate has three one inch bolts on each side that will allow the wedges and top plate to be fastened to the base once alignment is set. Chamfers in the plates will allow for a deeper penetration weld and yet still have the mating surfaces between the wedge, baseplate, and top plate ground smooth. The top plate has two slots on each side that will allow for vertical movement of the mount. Each wedge also has a slot that runs horizontally across the part. A side view of the mount which shows how the slots and holes align can be seen in Figure 41.Once alignment is set, a bolt can be dropped through the slot and into the hole of the base plate. The top plate also has a 0.25" depression in the center that will accommodate the base of the spring pack. In addition, the top plate assembly has two panels that border the spring pack. During axial expansion and contraction of the engine, the spring pack will push or pull on the panel and transfer the axial force down into the slider system. Braces give additional support to the panel during the expansion and contraction stages.

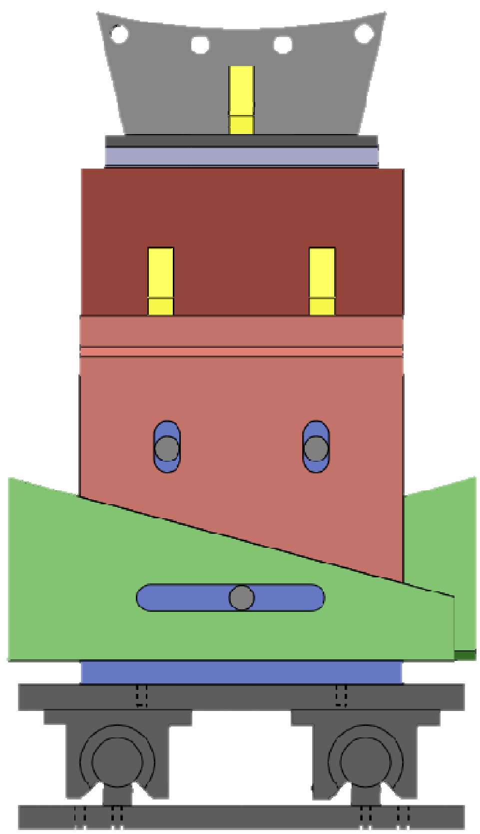


Figure 41. Side view of the final aft-mount design showing the alignment between the slots in the wedge and top plate compared to the holes in the base plate.

The spring pack used in this design uses the same set of springs, however the supporting pieces have been altered. The base pad is shorter than the current design and it has a recession for the tensioning bolt so that it will sit on the base plate. The top plates have also been altered to fit in between the spring constraint plates of the top plate assembly. The engine bracket is shorter than the original splash plate but retains the engine bolt pattern as specified in our design constraints section. The bracket also features a set of slots in the bracket base plate which will allow for the engine to be moved left or right during the assembly phase. A top view of these slots can be seen in Figure 42. This feature will give the installation crews at Solar Turbines even greater control over the final position of the engine centerline.

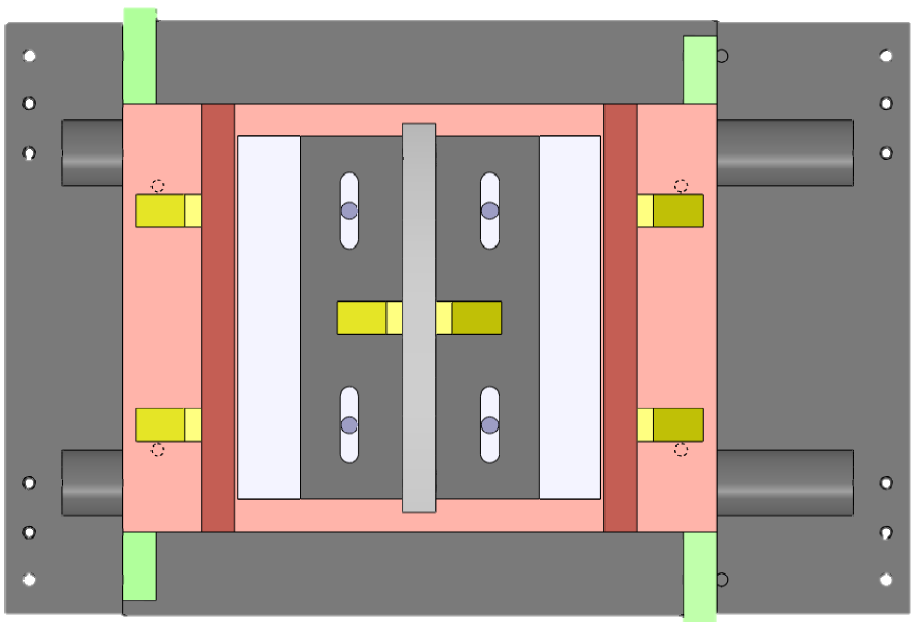


Figure 42. Top view of the final aft-mount design showing the slots in the bracket base which will provide left and right motion of the engine for additional alignment precision.

In order to set the alignment of the engine using the new aft-mount design, a jack and load cell can be inserted into the cavity of the mount. The power jack will be used by shop or field technicians to lift the mount in order to set the alignment of the engine. The jack ordered from Joyce Dayton has a 10 ton capacity worm gear machine screw. A 25:1 gear ratio provides precision positioning of 0.01" per revolution of the input shaft. The jack is upright and keyed allowing for the pad to rise and lower without rotation. The screw has 2 inches of rise and a load pad end condition. Special features include extended length input shafts to give the operators better access to the jack. There is an 8 inch hand wheel on one end and a hex input on the other which allows for manual or automatic operation of the jack with the use of a wrench or pneumatic wrench. The load on the mount will be recorded using a Honeywell Model 41

precision low profile load cell that Solar Turbines has on site. Once the specified load is achieved, the bolts on the mount can be fastened and the jack and load cell can be removed.

The rail slider used in the aft-mount redesign is a plain bearing linear slide produced by PBC linear. Minarik Controls and Automation, a San Diego based distributor, is the company that will be providing the system for this project. The low profile linear slider is capable of supporting up to 18,750lbs and will begin to slide after about 2,400lbs of load is applied along the direction of the rails.

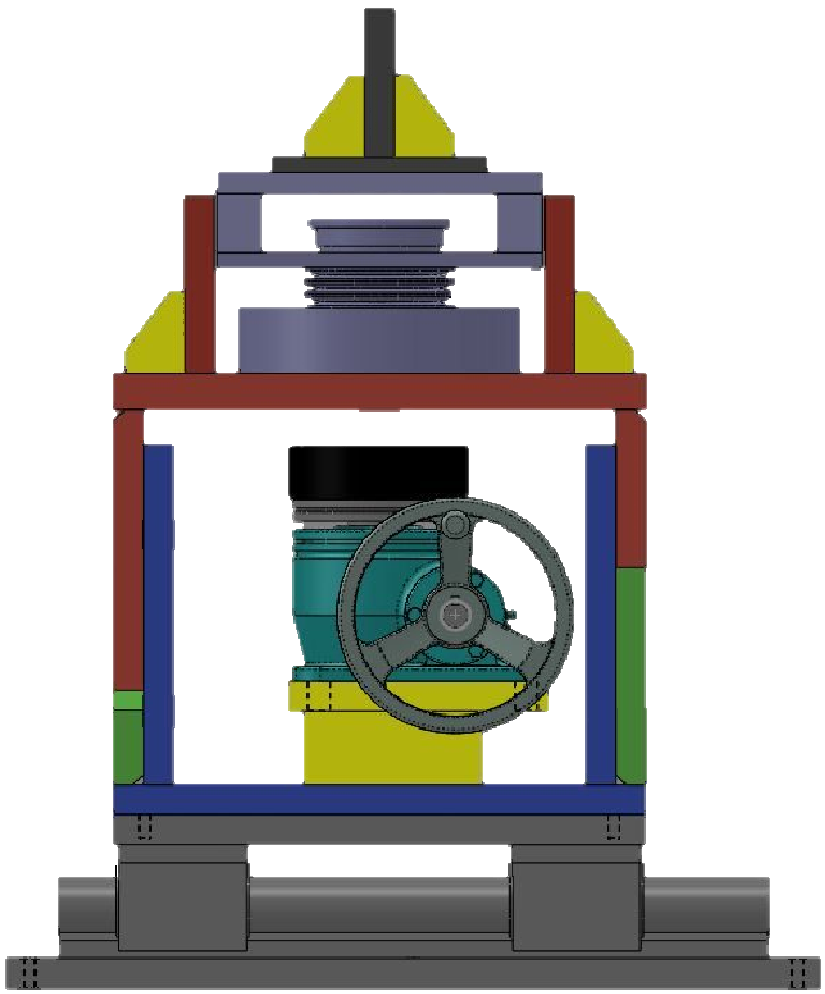


Figure 43. Side view of the final aft-mount design with the jack and load cell still in place.

The first design requirement for this project is to be able to accurately set the alignment of the engine using the aft-mount. During the initial placement of the engine, the power jack is supporting the base of the spring pack. Once alignment is achieved by reading the load from the load cell, the bolts are secured and the jack is then removed from under the spring pack.

The ability of the new aft-mount design to account for axial expansion from the engine is the second most important design requirement behind repeatability. The use of the PBC linear slider system at the base of the mount should allow the entire aft-mount to move along with the axial thermal growth. In order for the mount to begin sliding, there will need to be a certain amount of force applied in order to break the static friction in the slider. It was determined that a thousand pounds of force in the axial direction is needed to break the static friction in the bearings and move the slider. These calculations can be found in Appendix F under the title "Sliding Friction."

The static weight of the engine must be supported by the mount, as well as the dynamic loading that will be applied during normal operation. Two wedges are used to support the top plate when the alignment of the engine is set. The angle of these wedges was selected based on the friction between the two wedges as well as the overall height change that the wedges could provide. An angle of 15º was selected which will provide enough vertical height change and yet provide enough friction so as to maintain position for the life of the mount. The detailed calculations for the angle selection can be found in Appendix F. The new aft-mount design also requires bolts to hold the plates in position during the operating life of the engine. There are four bolts in the top plate which are designed to fully support the static and dynamic load of the engine. In the unlikely event that all four of these bolts fail during operation, there are two additional bolts that will hold the bottom wedges in place. These bolts are also designed to be able to fully withstand the loading of the turbine. Under each loading condition, the bolts were designed to hold up a maximum loading condition of 18,000 pounds of force. The detailed calculations for the bolt characteristic selection can be found in Appendix F under the title "Bolt Selection."

The main components of the mount will be made from carbon steel. This material choice was made based on the availability of carbon steel and its relative cost to other material. In its raw form carbon steel is fairly inexpensive compared to other structural material. An estimate for a square foot of low carbon steel is about $140 per square foot of one inch thick stock. Any corrosion or rust issues that are associated with using bare carbon steel in harsh environments are negated by painting all the carbon steel aft-mount components with protective enamel after machining and welding is completed. All of the machining for this project will be handled internally through Department 45 of Solar Turbines. Machining costs came to $27,794.12. The spring pack and engine bracket included in design can be modified from current Solar parts. The cost of the stainless steel used to manufacture the main structural components of the aft-mount

came to be $3,091.00. In addition, the rail and slider system purchased through Minarik will cost $2302.58 with tax and shipping included. The cost of the power jack is $2216.80; however, this cost is associated with the tooling for the mount and is therefore not included in the total cost of the aft-mount.

The assembly procedure for the aft-mount is explained in detail in Appendix G.

Once the mount has been set and is under compression loading, it is important to note that the friction of the wedges, as well as the preload applied in the bolts, is all that holds up the turbine. Technicians should employ caution when working underneath the engine when the mount is under static loading conditions. After the expected engine lifespan of 30,000 hours, the aft-mount should require minimal servicing. The hypothesis is that the slider system will allow the turbine to grow axially and the mount will move with it. Because of this additional feature, the spring pack will be loaded uniformly. When the engine is overhauled, the mount should be checked that the springs have not cracked, and are otherwise in working condition. Once this has been confirmed, it should be able to be placed back into service. It would be prudent to replace the six bolts that were used to hold the plates in position during the overhaul.

DESIGN REALIZATION

Manufacturing

Several different options were considered for the manufacture of the prototype Titan 130 aft-mount. Because of the fact that the mount would eventually be tested on an actual engine, there was a necessity for the manufacturing of the mount to be ISO certified. This precluded the members of TurboTech from manufacturing any structural parts in Mustang ’60 or the Hangar. Different machine shops and weld shops were considered in the San Luis Obispo area for the manufacturing to reduce shipping cost and time. The alternative to local machine shops was for Solar Turbines’ Department 45, experimental manufacturing and tooling, to machine and weld all of the parts. For convenience and quality control purposes, it was concluded that Department 45 would complete the manufacturing requirements.

Drawings for each of the parts were drafted and reviewed by Donovan Vick, a manufacturing engineer in Department 45. Several iterations were generated and the final set of drawings can be found in Appendix C. Donovan procured 316 stainless steel to construct the mount and used the final drawings to begin machining each detail. The details were roughly cut using a high pressure water jet and then finished using both an EDM wire and 3-axis mill. Each of the assemblies were welded together using MIG and TIG processes.

Figure 44. Machining of one of the wedge details on a 3-axis mill at Department 45.

Figure 45. Verifying dimensions on a wedge detail at Department 45.

Figure 46. Welding the top assembly at Department 45.

Once the parts were finished and assembled, Oscar Lopez, a manufacturing engineer in Department 45, packaged and shipped the parts via Con-way Freight to San Luis Obispo. The parts were received on Monday, April 29, 2013. All parts were accounted for at that time.

In order to do a full part fit test and assembly procedure check, it was necessary to order bolts, nuts, and washers for the mount. Fastenal was the company used to purchase the six 1”-14 x 3 inch bolts, four ½”-20 x 2¾ inch bolts, six 1”-14 nuts, four ½”-20 nuts, twelve 1” washers, and eight ½” washers. The order was placed on April 17, 2013 and part of the order was received on May 6, 2013 and the rest of the order was received on May 14, 2013. With all of the assembly components acquired, the fits for all of the mating parts were checked and verified.

In order to connect the Honeywell load cell to the jack, a separate connector plate had to be manufactured. A 1½”-12 x 2” bolt was purchased from Fastenal for the threaded rod that bears the load on the load cell. A 6” x 8.625” x 0.75” 316 stainless steel plate and ½”-13 rod was purchased from McCarthy Steel.

In order for the 316SS plate to attach to the jack, it was necessary to CNC a hole pattern into the steel. To accomplish this, a new ½”-13 tap spiral flute plug tap was purchased to tap the holes. Using a 3-axis CNC mill, the bolt hole pattern was made based on a center hole that was also tapped with the ½”-13 plug tap. A hole was drilled into the end of the 1½”-12 bolt and then tapped with a ½”-13 tap so that a piece of the ½”-13 rod could be threaded into it. Once the hole had been tapped and drilled, the head of the bolt was removed using a horizontal saw. Finally, the cut face was faced using a lathe and then the threads were cleaned up using a 60º cutting tool.

Figure 47. Setup for the 316 stainless steel plate on the CNC mill.

Figure 48. The program has reached the first hole in the cutting process and is beginning to drill. Coolant is used to reduce the effect of work hardening the material.

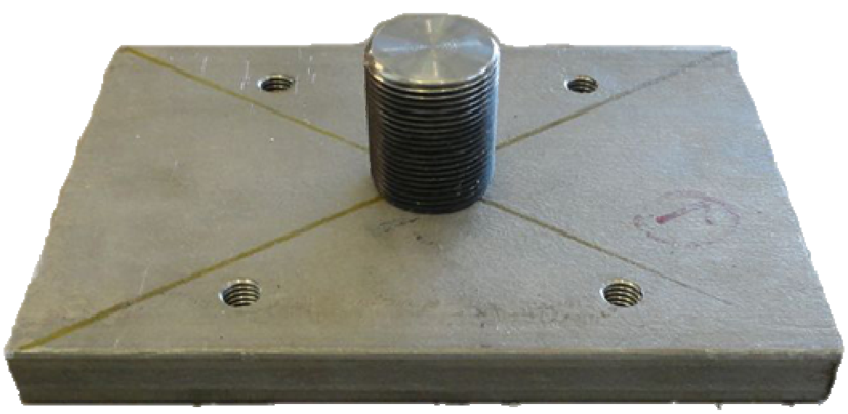


Figure 49. This is the finished product with the 1½”-12 bolt in place. ½”-13 rod will be cut and placed in the 4 holes which will locate in the jack loading pad.

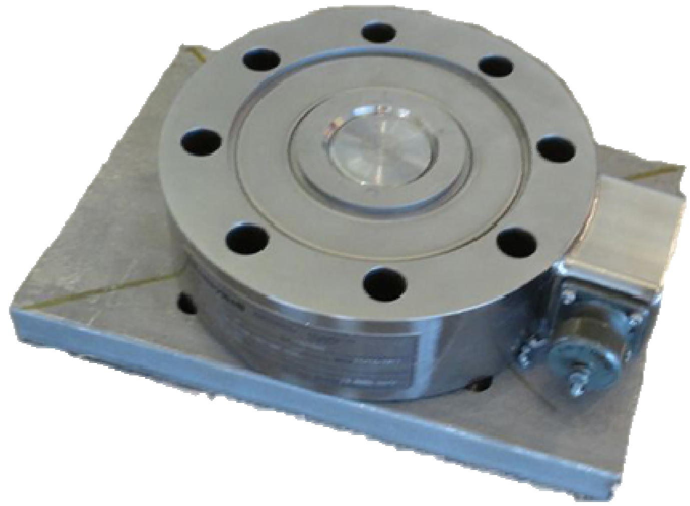


Figure 50. This picture shows how the load cell will sit on top of the connecting plate. The bolt threads through the center of the load cell and bears the load on its threads.

All of the critical dimensions that were specified in the drawings to Solar were met. Once the parts arrived in San Luis Obispo however, it was determined that the center hole in the spring pack base needed to be expanded to accommodate the ¾” tie-rod. To fix this, a one half inch diameter reduced-shank TiN-coated cobalt steel 0.875” drill bit was purchased to cut the hole in the stainless steel part. The spring pack base pad was clamped onto a 3-axis mill in the Cal Poly Hangar and the hole was drilled out to meet the new required diameter.

Figure 51. This shows the setup of the spring pack base pad on the 3-axis mill before the center hole is opened up to the new diameter of 0.875 inches.

Figure 52. A power z control was used so that the tool would move continuously through the part to reduce the effect of work hardening the material.

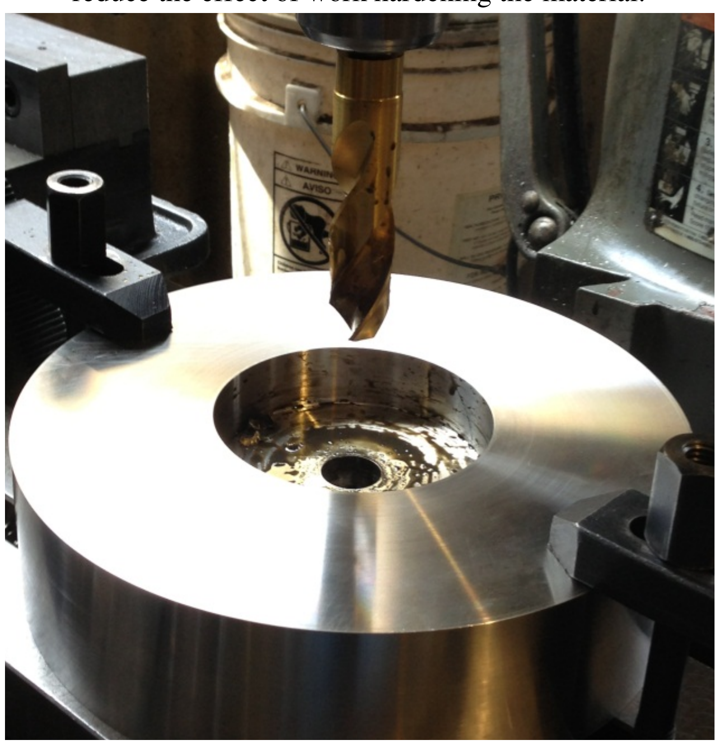


Figure 53. Finished cut in the spring pack base pad.

Finished Mount and Recommendations

On May 8th, all the parts needed to begin testing were on hand. The final aft-mount assembly seen below is composed of rail sliders from PBC Linear, a jack from Joyce-Dayton, a load cell from Honeywell, and the parts from Department 45 at Solar Turbines.

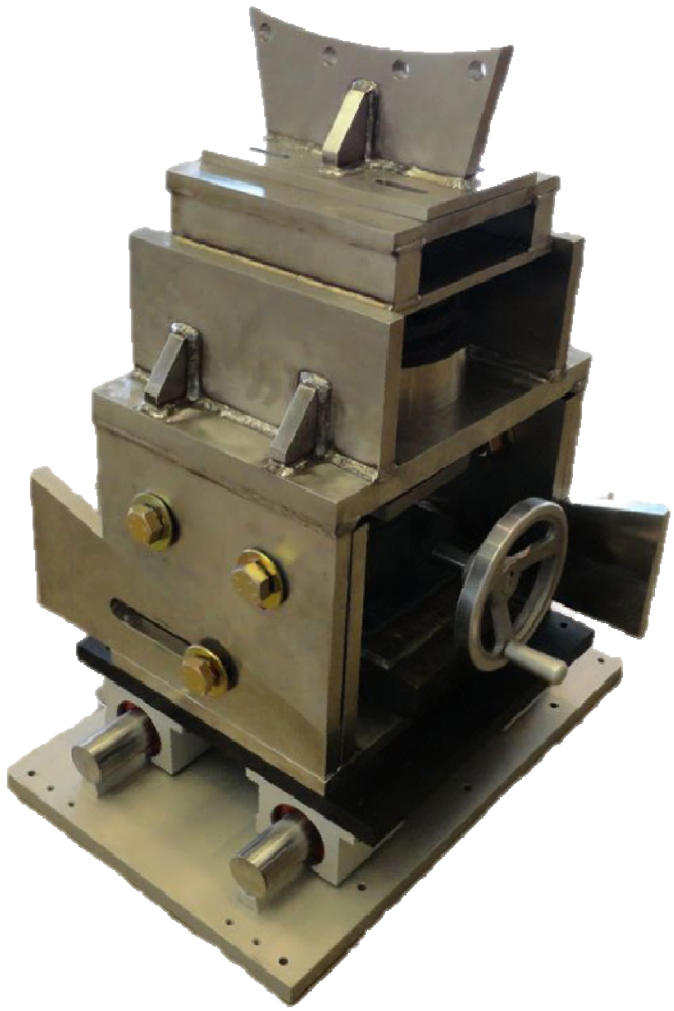


Figure 54. This is the final assembly of all of the parts from Solar Turbines, PBC Linear, Joyce Jacks, and Fasntenal.

Most of the features for the Titan 130 aft-mount prototype resemble the original design on SolidWorks, but there are a few differences that are significant enough to mention. The main difference was the material choice for the mount details. Originally, carbon steel was specified for the material choice and calculations were done based on that assumption. However, when it came to purchasing the material for the mount and the manufacturing process, 316 stainless steel was selected and used for the parts. This change in material actually resulted in several

improvements on the design. First, the coefficient of thermal expansion is higher for stainless steel that it is for carbon steel. This means that as the mount heats up during operation, and the metal expands, it will actually expand more than the grade 8 carbon steel bolts used to hold the mount together. This means that the bolts will actually hold the parts together tighter than they were at the initial setup, thereby increasing the friction between the wedge plates and decreasing the likelihood of failure. Also, stainless steel has better corrosive properties than carbon steel, making it an ideal choice for use out in the field. The change in material from carbon steel to 316 stainless steel was a very positive change from the original design. The different characteristics that stainless steel offers over carbon steel outweigh any increase in cost and greatly improve the overall quality of the mount.

Another significant difference between the original design and the prototype is for all the parts that were originally called out for 1 inch stock material thickness, the steel vendor delivered 0.9 inch thick material. This mistake was only noticed after significant work had been done on each of the parts. It was determined that this change in thickness would not be detrimental to the design and could be accepted for the prototype. Similarly, during testing, the thickness of the top plate that the spring pack assembly sits on, it was found that the thickness of that plate was also thinner than originally specified. In the detail drawing for the top plate, it calls out for 1.5 inch thick steel but in actuality, the thickness of the top plate is 1.2 inches. This change resulted in greater deflection of that cross beam under loaded conditions.

Finally, the hole in the spring pack base pad was originally designed to be 0.563 inches in diameter. However, it was necessary to open the hole up to be 0.875 inches in diameter in order to accommodate the ¾ inch tie-rod that runs through the center of the spring pack assembly to hold the assembly together. Similarly, there is a hole on the bottom of the spring pack base pad to allow a nut to be threaded into the recession of the base pad. This hole is specified to be 1.5 inches in diameter; however a 2.5 inch hole would make it so that a socket wrench could be used to hold the nut in place during tightening. Also, there is a through hole in the top plate designed for the same purpose. This hole would also be better being 2.5 inches in diameter. Finally, the pillars are slightly too short for the tie-rod once the spring pack assembly is in compression. Increasing their overall height approximately 1 inch would make for a better fit.

For the future manufacturing of the mount, it would be ideal if the entire mount was analyzed using some kind of finite element (FEA) software. This type of analysis would result in the most efficient use of material in the mount and would prove where certain parts could use less material and still maintain the overall structural integrity.

Design Verification

Compression Testing

Several tests were run in order to validate the design of the new engine aft-mount. The mount was assembled on an MTS 322 Test Frame, and all deflection data was collected using a data acquisition system that directly exported deflection, load, and time elapsed into excel spreadsheets. Using the proposed assembly procedure, the mount was assembled on the test platform as depicted in Figure 55 below.

Figure 55. Fully assembled mount with wedges bolted into place at a mock loading condition. The entire mount is placed so that its center sits directly below the piston of the test machine.

In order to begin testing, it was necessary to initially compress the spring pack so the spring length was 2.5 inches, as specified by ES-2314. By measuring the free length of the spring, it was determined how much the springs needed to be compressed to accommodate the engineering specification. Once the load was applied to the spring pack, as seen in Figure 56, the nut on the tie rod placed to hold the spring pack together was tightened down to hold the spring pack at the required length.

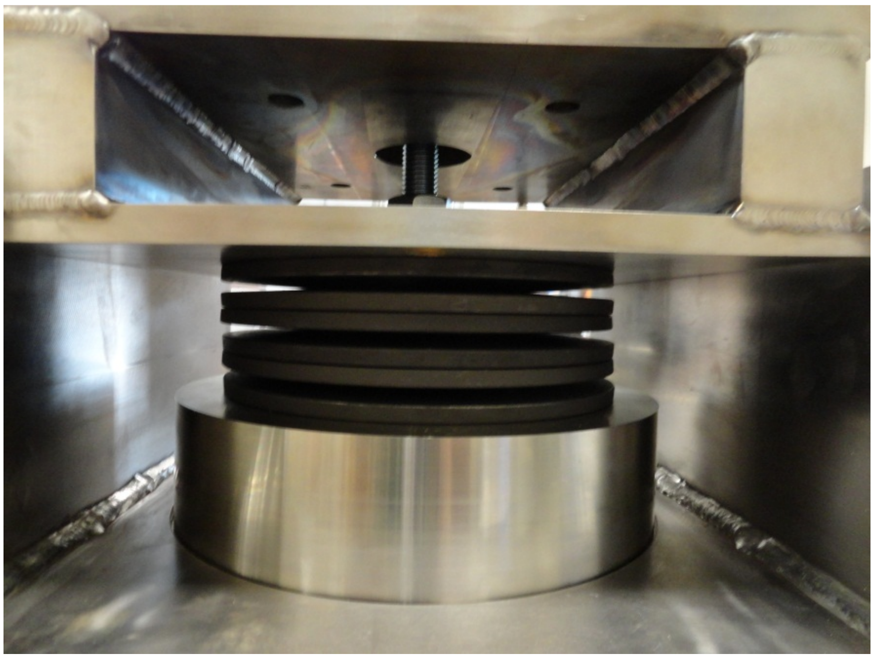


Figure 56. The spring pack is first loaded in compression and eventually the nut on the tie-rod is tightened so that the spring maintains a length of 2.5 inches.

The first test that was conducted on the mount was a static load test. This entailed applying a varying load on the mount ranging from zero pounds of force all the way up to 18,000 pounds of force. The test machine was programmed to move the hydraulic piston to compress the mount at an approximate rate of 0.004 inches per second for all of the tests. Once 18,000 pounds of compression had been reached, the load was removed, and the test was reset. This test was run for a total of 10 repetitions in order to obtain several data points for each engine loading condition.

The next test involved how the mount would respond to expected normal engine vibrations during operation. The MTS 322 Test Frame has the capability to apply a sinusoidal change in load at a maximum frequency of 100 Hz. From figure B4 in appendix B, we can see that the frequency at running speed for a Titan 130 is 187 Hz and has an acceleration of 0.1g. Using the following formula, the appropriate deflection at 100 Hz could be determined.

$$D = \frac{(70.4 \times 10^6)A}{CPM^2}$$

$$A \equiv Vibrational\ Acceleration\ in\ g's$$

$$CPM \equiv Cycles\ per\ Minute$$

$$D \equiv Deflection$$

For a conventional engine, the test was performed so that the engine weight, 12,927 pounds, was applied to the mount with an amplitude of 600 pounds which corresponded to a piston displacement of 0.008 inches. Similarly, the SoLoNOx engine was performed so the engine weight of 15,484 pounds was applied to the mount with an amplitude of 600 pounds, corresponding to a piston displacement of 0.039 inches. Each test was run for 30 minutes which is the longest amount of time that could be spent for time allotted using the test equipment. Using the equation above, the displacement for the SoLoNOx engine equivalent for 100 Hz was found to be 0.028 inches. This displacement is significantly less than the first SoLoNOx cyclic loading test that was performed using a displacement of 0.039 inches. A third and final cyclic loading test was performed using an amplitude of displacement at 0.028 inches for an additional 30 minutes to simulate actual engine conditions.

Based on calculations for friction between the plates of steel, it was determined that friction alone between the plates could support the load of the engine. In no instance during engine operation should the bolts be removed, however this test proves that there is another level of safety redundancy in the design. By loosening each of the 6 bolts, the mount was slowly loaded in compression to simulate a boltless condition. At a maximum expected loaded condition of 18,000 pounds, the mount did not move. This test proves that the friction between the steel plates is high enough to hold the mount in place once the bolts have been set to the specified torque setting.

After each test was performed, a final engine mounting trial run was performed to determine if the 10 ton jack was capable of setting the alignment of both engines. Before the jack was tested, the Honeywell load cell needed to be calibrated and a conversion factor between volts and pounds force needed to be determined. A load was directly applied to the load cell using the test frame and several data points were plotted in order to find the conversion factor. The plot of the data can be seen in Figure 57.

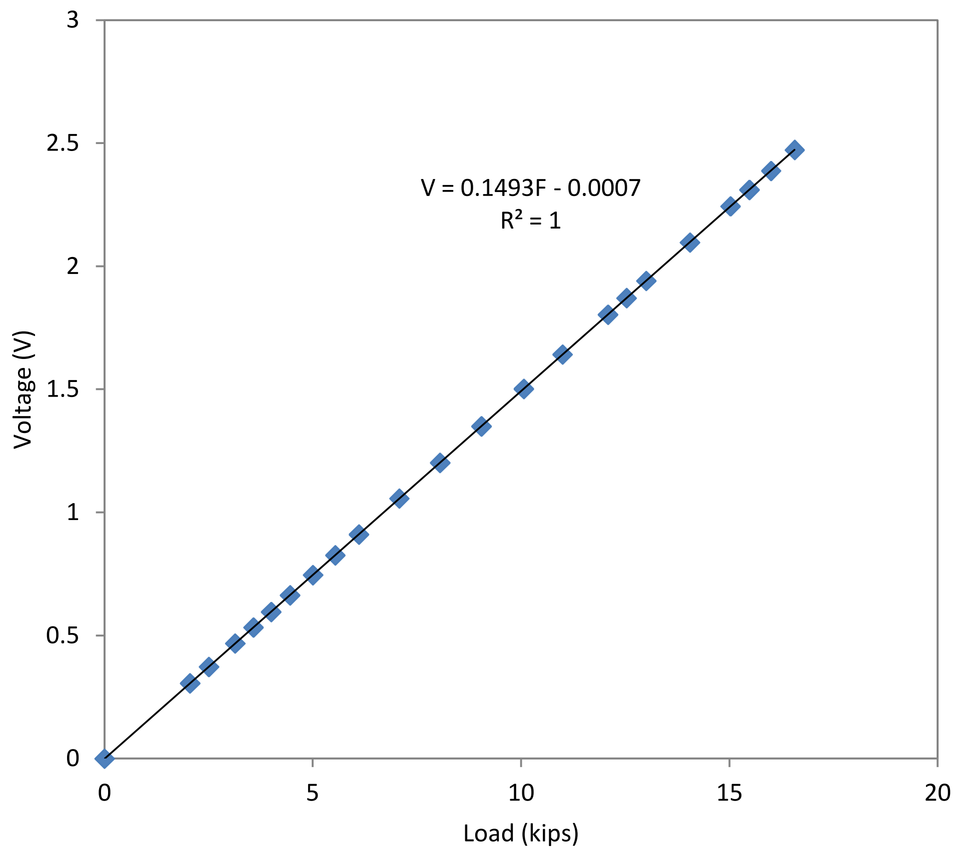


Figure 57. By applying a load directly to the load cell at 1000 pound increments, a trend line was generated to convert pounds force to voltage. Extra data points were collected surrounding the engine loads of 12,927 lbs. and 15,484 lbs. to ensure accuracy at those levels. The plot clearly shows a linear trend in the range required for its operation. The conversion ratio is 0.1493 Volts per kip.

Figure 58. This device is used for observing data from the Honeywell load cell. The readout displays changes in load in volts.

After the load cell had been calibrated, the MTS 322 Test Frame was used to simulate the weight of each engine type sitting on top of the mount. The jack was then used in combination with the Honeywell load cell to lift the “engine” to the specified position. Once the engine was in position, the wedges were moved into place and the bolts were torqued to the specified settings. Then the jack and load cell were removed and the process is finished. Based on this experience, it is expected that the engine will be able to be aligned in less than 10 minutes. Using a pneumatic torque wrench to apply the torque to the 6 bolts on the mount is expected to expedite the process.

Rail Slider Testing

In order to determine whether or not the rail sliders will move axially with the expansion of the engine, the coefficient of friction was determined experimentally. An Omega Instruments load cell and data acquisition system were used to determine the force applied to the rail sliders. During the test, the load on the slider was incrementally increased and a force measurement was recorded. In order to determine the static coefficient of friction, the force was recorded just as the slider began to move. After the slider was moving at a constant velocity, a second force was recorded to find the kinetic coefficient of friction. The slider was placed on a level surface and the force was measured so that the cable was perpendicular to the slider and parallel to the table surface in order to get an accurate measurement.

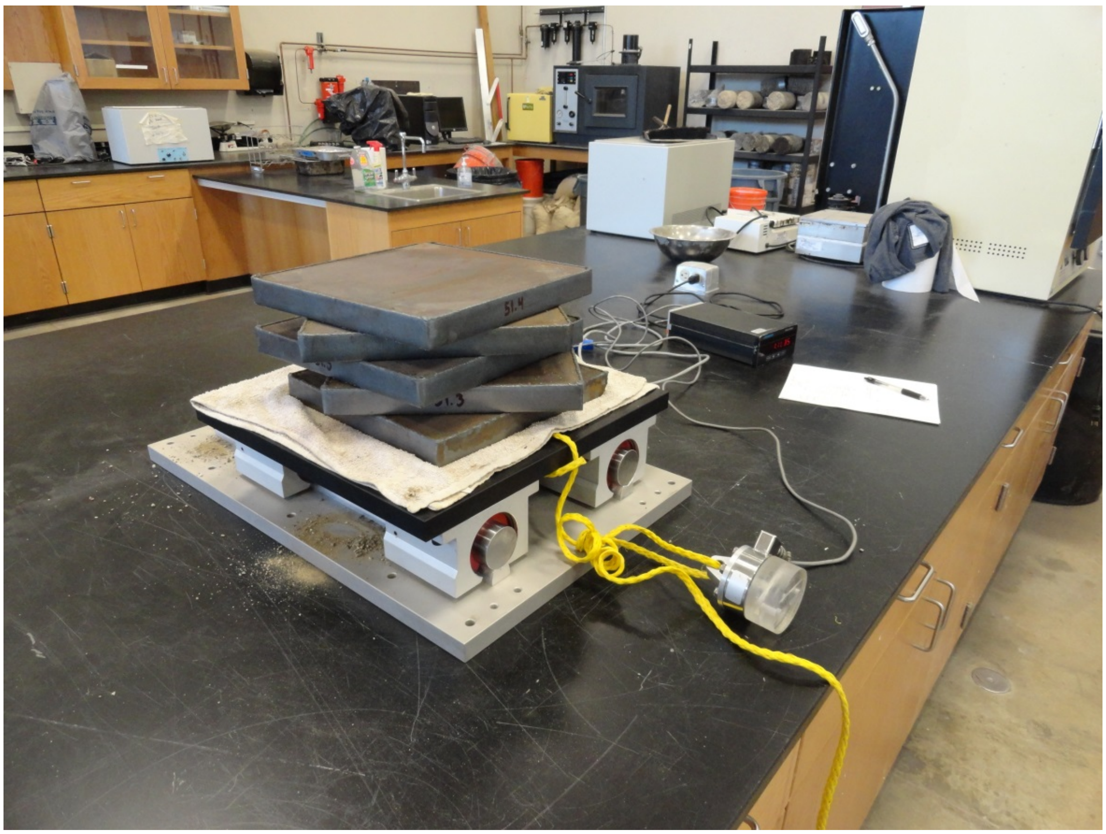


Figure 59. Rail slider coefficient of friction test set-up. The load cell is connected to the mount by a cable located at the midsection of the slider. The DAQ readout is in the background.

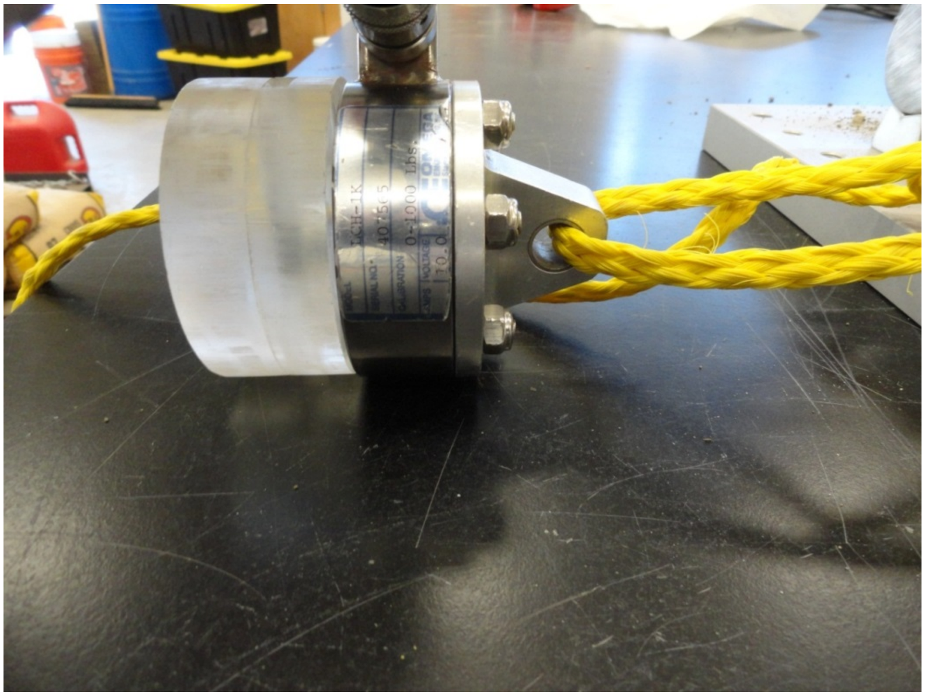


Figure 60. Omega load cell attached to the cable by a clevis. Force was applied by pulling on the acrylic handle on the left of the load cell.

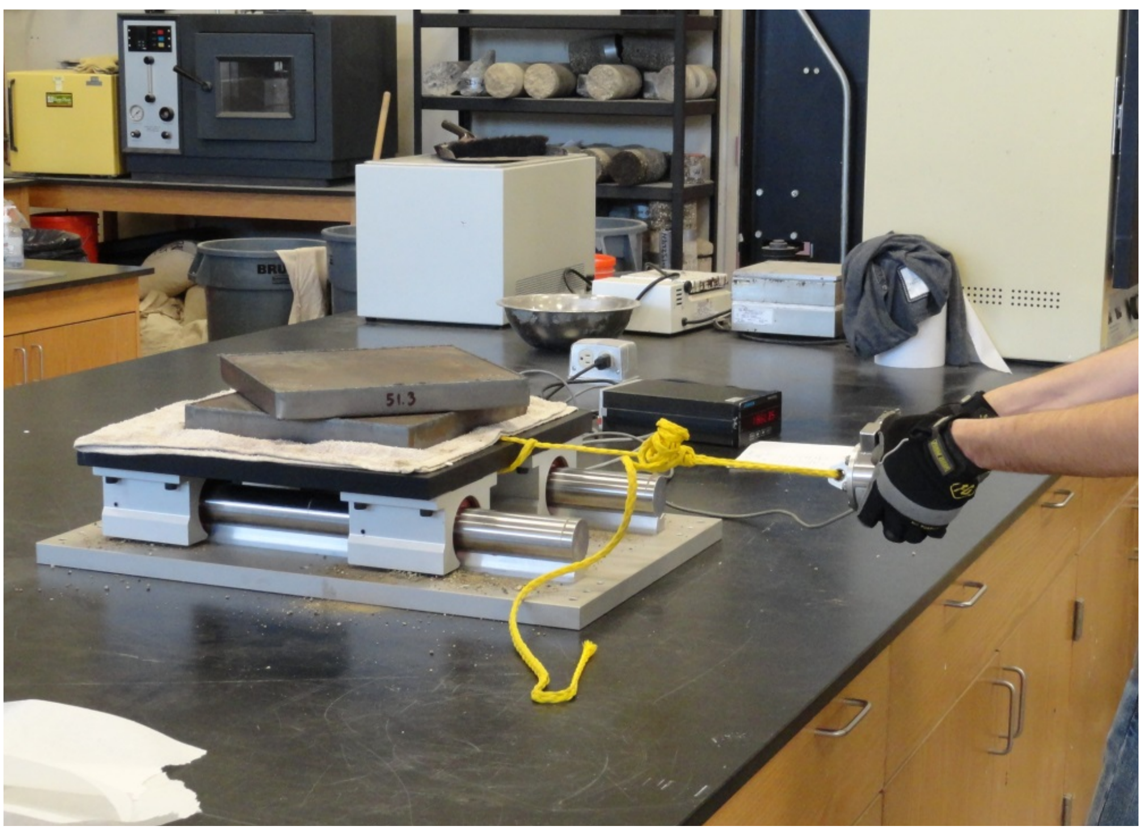


Figure 61. Test run with 100lbs on top of the rail slider assembly.

Detailed Results

Compression Tests

The first of the compression tests for the mount was designed to test the repeatability of the system for both engine loading conditions. Ten tests were run in a similar fashion, and each of the data sets that were collected for each run were plotted against one another to view any possible differences between each test. There is very little noticeable difference between the plot for spring pack deflection over a varying load and the overall deflection of the mount over a varying load. The reasoning behind this similarity is that the LVDT measuring the spring pack displacement takes into account the deflection of the top plate in addition to the spring pack deflection. The LVDT measuring the overall deflection takes both of those changes in height with the addition of any deflection in the top plate of the spring pack assembly. There is very little deflection in the top plate of the spring pack assembly and therefore, the two plots look almost identical. The reason for the slope change in the two plots comes from the spring pack being initially compressed to an approximate spring length of 2.5 inches. Up until the load on the mount reaches about 13.5 kips, the spring pack does not compress more than the initial compression due to the tie-rod bearing the tension load from the spring. The change in deflection read from the 0.5” LVDTs is the deflection of the top plate that the spring pack assembly sits on up until the load reaches about 13.5 kips. Once the load continues past 13.5 kips, the tension force in the tie-rod reaches zero and the spring pack begins to compress and the deflection read by the LVDT is now predominantly the change in height of the spring pack.

Figure 62. 0.1” LVDTs placed to measure symmetric displacement of the top plate.

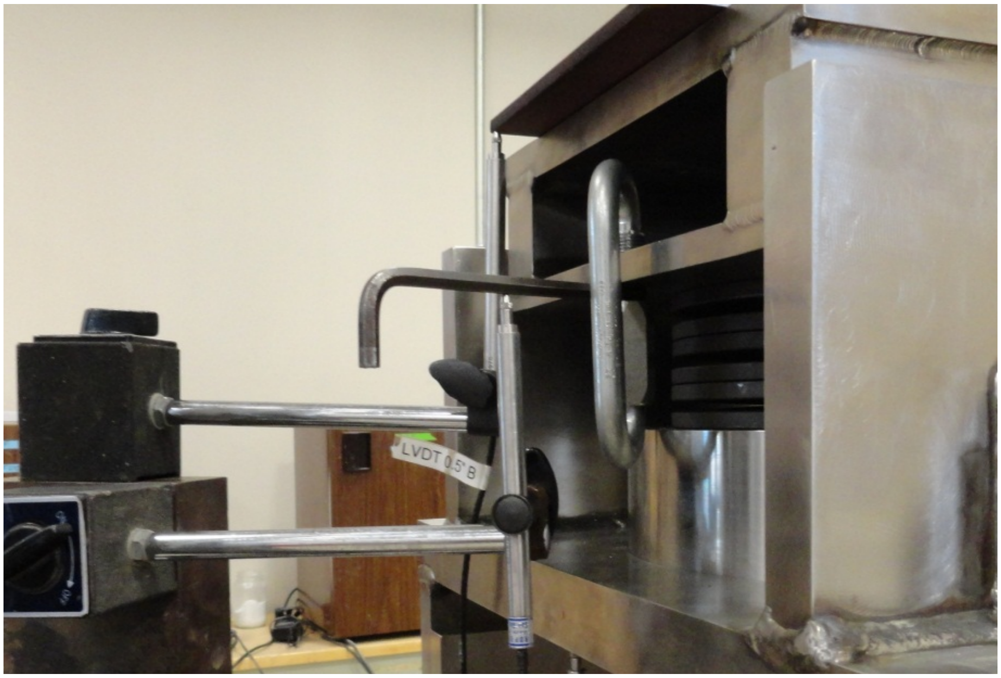


Figure 63. 0.5” LVDTs placed to measure displacement of the spring pack and the overall displacement of the mount.

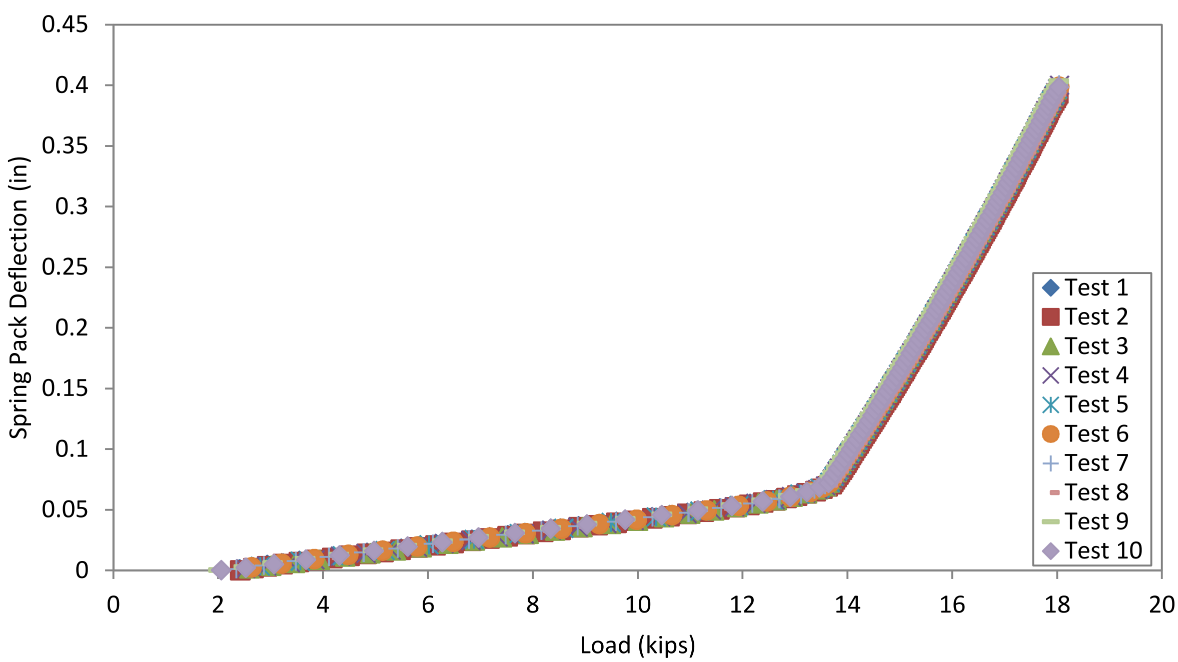


Figure 64. Spring pack compression data from 0.5” LVDT A for tests 1 through 10.

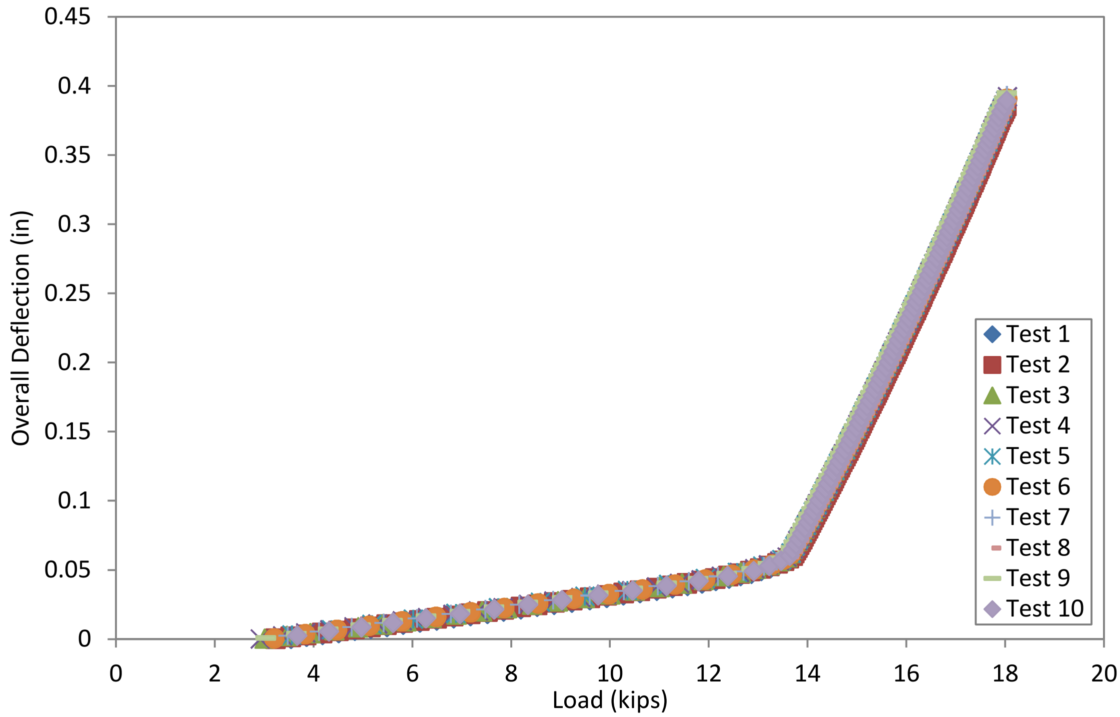


Figure 65. Overall mount compression data from 0.5” LVDT B for tests 1 through 10.

Measurements were also taken on both sides of the top plate assembly on which the spring pack assembly sits on. This data shows over a varying load, how much the top plate deflects. 0.1” LVDTs were placed on both sides to compare symmetrical data for the top plate. Knowing that the error associated with each of the LVDTs is 0.2%, it was concluded that the data acquired from the 0.1” LVDT B is flawed. It is believed that there was some kind of malfunction with the equipment possibly due to being dropped.

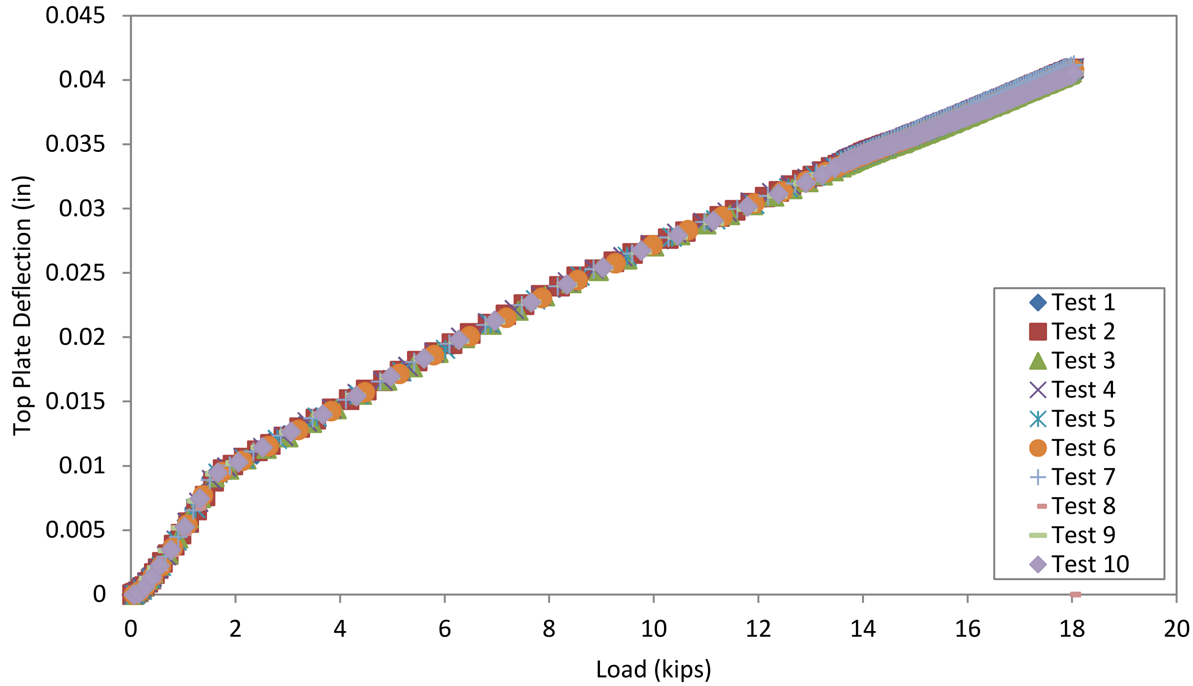


Figure 66. Top plate deflection data from 0.1" LVDT A for tests 1 through 10.

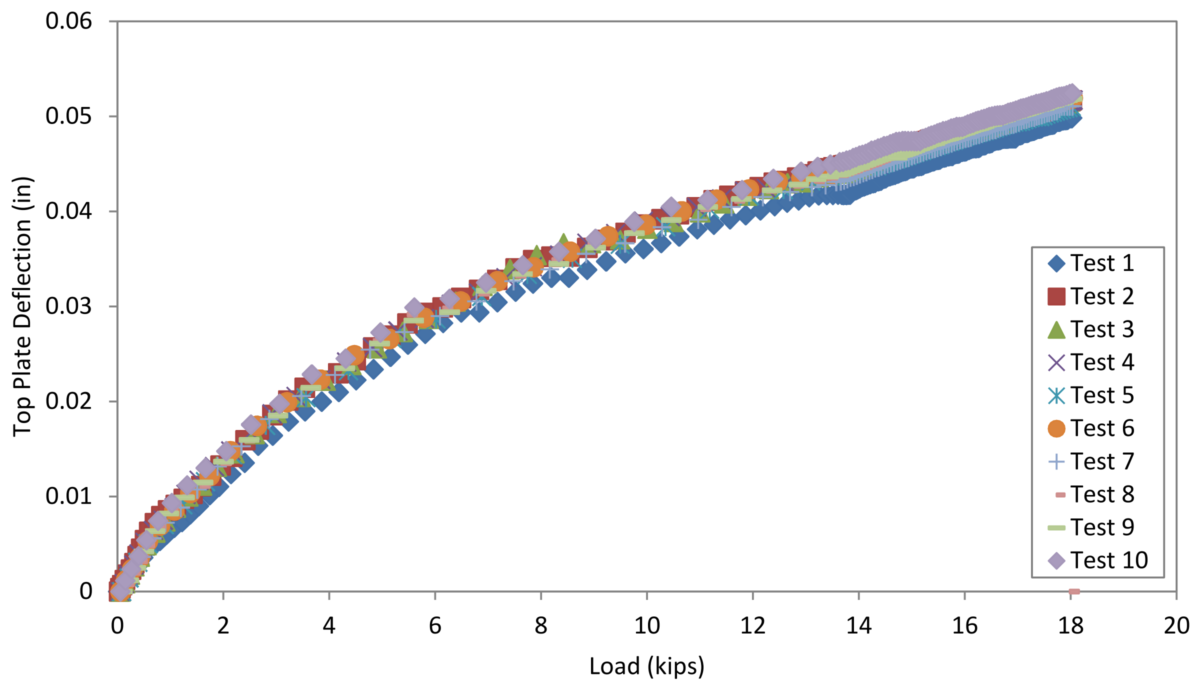


Figure 67. Top plate deflection data from 0.1" LVDT B for tests 1 through 10. Data is possibly corrupted from equipment malfunction.

The next set of tests that were conducted involved cyclic loading of the mount to simulate normal engine running operations. Due to limited time on the test machine, the duration of these tests were limited to 30 minutes each. The test equipment was set to record data at 1 second intervals. Because of this setting, the data acquired is limited in what was actually tested. For the conventional engine, the target load was 12,927 pounds ± 600 pounds. Since the frequency of data collection was set so low, the results only show a change of about ±100 pounds. This is true for the conventional test and both SoLoNOx tests that were conducted. For the SoLoNOx tests the target load was 15,484 pounds ± 600 pounds, but the results only show a change of approximately ± 100 pounds of load. For the conventional engine test, there is a distinct trend in the plot that shows as the load is decreased, the mount decompresses. Similarly, as the load increases, the mount compresses. There is a very small change in the overall change in height of the mount as the load is fluctuated; however, speculation on the mounts change in deflection over time is impossible to extrapolate given the relatively short duration of the test. For the SoLoNOx engine tests, it is believed that the LVDTs were unable to keep up with the large change in amplitude of the deflection at such a high frequency. This explains why there is no apparent correlation in the data for any of the SoLoNOx plots. Vibration data results can be viewed in Appendix J.

Rail Slider Tests

The first test was conducted with the sliders in an as-delivered condition with clean, non-lubricated rails. The weight of the top plate of the slider was neglected for the testing and plotted data is based on payload, rather than total weight. Five tests were conducted, each time adding about fifty pounds to the sliders. The static coefficient of friction was found to be 0.1849 and the kinetic coefficient of friction was found to be 0.1678.

Table 7. Summary of data taken during the clean slider test.

Clean Slider Test		
Load (lb)	Static Force (lb)	Kinetic Force (lb)
0	16.2	15.7
52.1	29.8	28.9
103.4	41.6	38.1
154.7	48.1	45.6
206.1	54.6	49.1
257.5	66.6	62.5

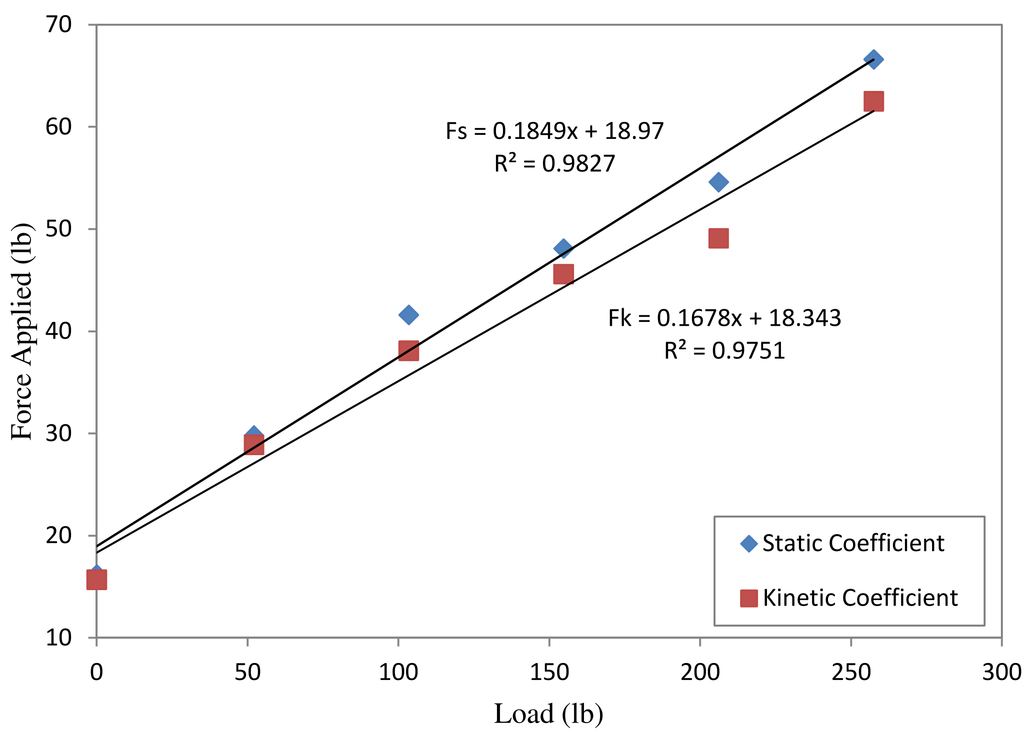


Figure 68. As load on the slider increased, the load required to move the slides increases in a linear fashion. The coefficient of friction is the slope of the trendline.

The second test was conducted simulating dirt and grime build-up on the rails. Sand was poured over the rails during the test to replicate worst case scenario to get coefficients of friction. The weight of the top plate of the slider was neglected for the testing and plotted data is based on payload rather than total weight. Five tests were conducted, each time adding about fifty pounds to the sliders. The static coefficient of friction was found to be 0.1861 and the kinetic coefficient of friction was found to be 0.1836.

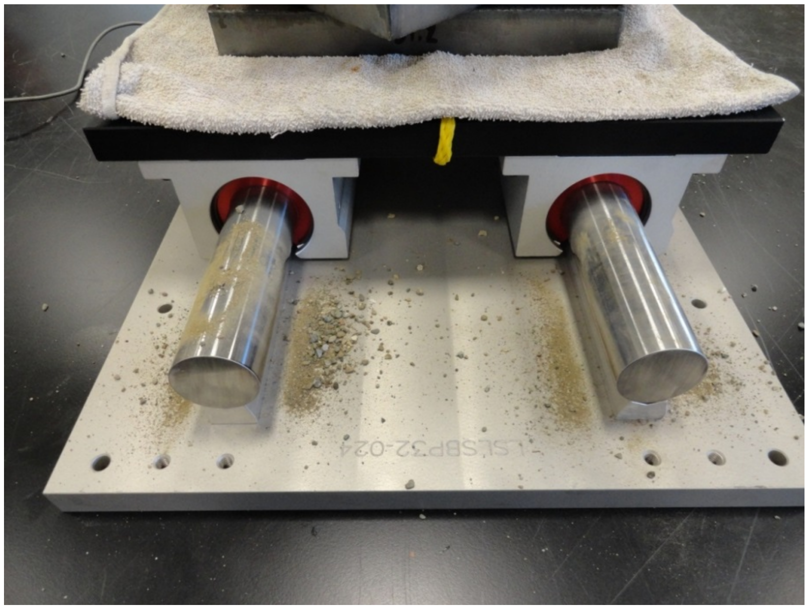


Figure 69. Condition of sliders during the dirty rail test.

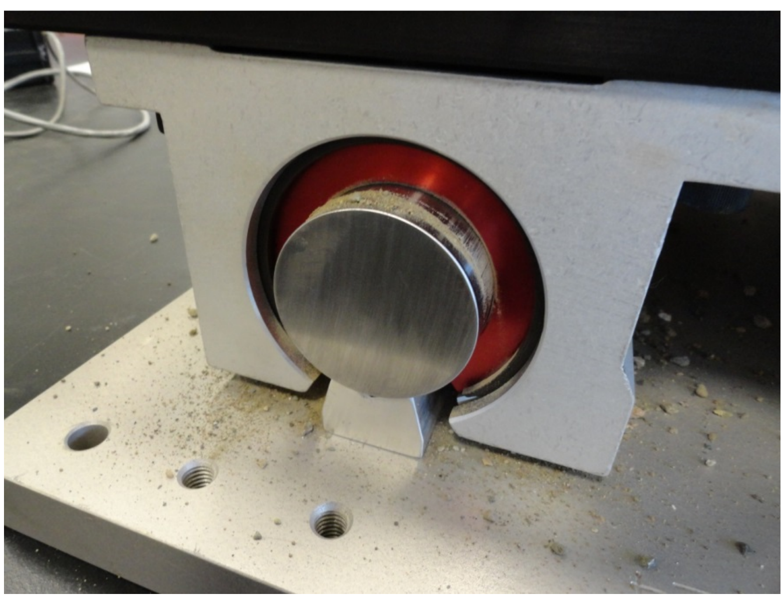


Figure 70. Sand was used to simulate a worst case scenario when getting the coefficients of friction.

Table 8. Summary of data taken during the dirty slider test. Sand war poured over the rails in order to simulate dirt and grime build up that could happen in the field.

Dirty Slider Test		
Load (lb)	Static Force (lb)	Kinetic Force (lb)
0	27.4	20.1
52.1	38.6	35.2
103.4	53.6	44.1
154.7	58.1	53
206.1	61.1	54.1
257.5	80	73.1

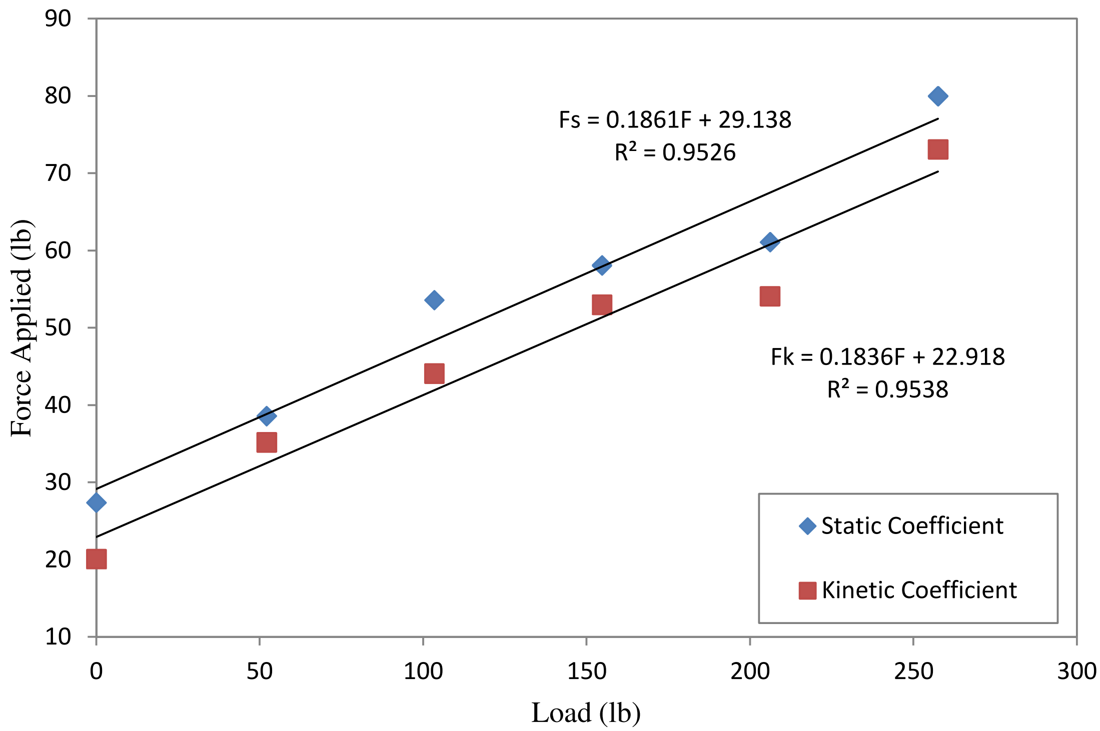


Figure 71. As load on the slider increased, the load required to move the slides increases in a linear fashion. The coefficient of friction is the slope of the trendline.

The top plate of the slider cannot be easily removed, so the total weight or the slider assembly was recorded and found to be 152.2lbs. Using SolidWorks, the volumetric percentage of the top plate was found to be 43.4% of the total volume, making the top plate weigh approximately 66lbs.

Table 9. Summary of data taken during the clean slider test including the estimate for how much the top plate of the slider weighs.

Clean Slider Test		
Load (lb)	Static Force (lb)	Kinetic Force (lb)
66.0548	16.2	15.7
118.155	29.8	28.9
169.455	41.6	38.1
220.755	48.1	45.6
272.155	54.6	49.1
323.555	66.6	62.5

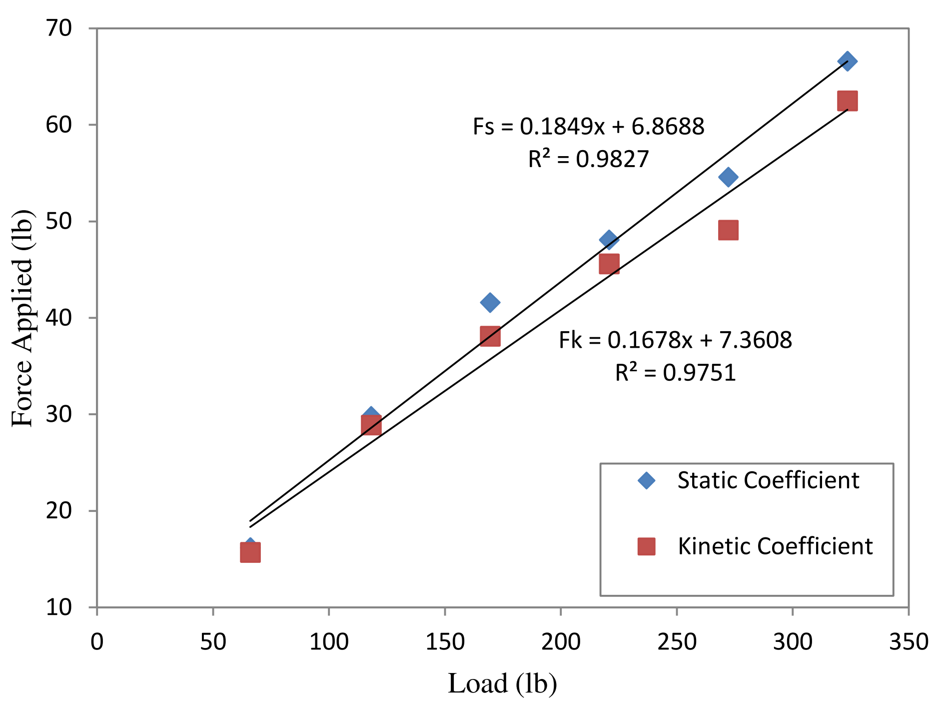


Figure 72. As load on the slider increased, the load required to move the slides increases in a linear fashion. The coefficient of friction is the slope of the trendline. The test was not perfect so the trendline does not pass through the origin.

Table 10. Summary of data taken during the dirty slider test. Sand war poured over the rails in order to simulate dirt and grime build up that could happen in the field. This data includes the estimate for how much the top plate of the slider weighs.

Dirty Slider Test		
Load (lb)	Static Force (lb)	Kinetic Force (lb)
66.0548	27.4	20.1
118.155	38.6	35.2
169.455	53.6	44.1
220.755	58.1	53
272.155	61.1	54.1
323.555	80	73.1

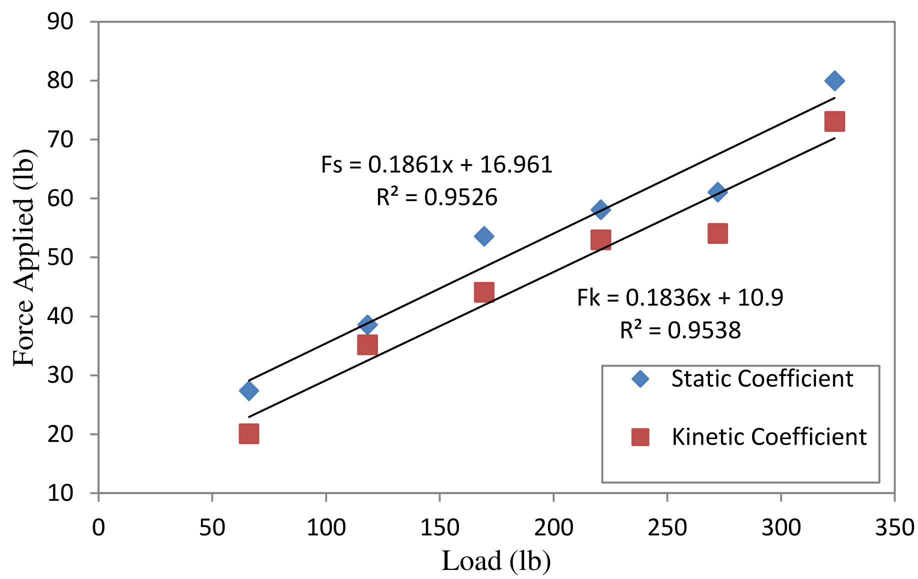


Figure 73. As load on the slider increased, the load required to move the slides increases in a linear fashion. The coefficient of friction is the slope of the trendline. The test was not perfect so the trendline does not pass through the origin.

After conducting the tests, the force required to overcome the friction for both static and kinetic coefficients for both clean and dirty sliders is recorded in Table 11. The test results seem to make logical sense because the clean rails require less force than the dirty rails and the coefficient of kinetic friction was less than that of static friction.

Table 11. Summary of static and kinetic coefficients of friction for both clean and dirty rails.

	Coefficients of Friction	
	μ_{static}	$\mu_{kinetic}$
Clean	0.18	0.17
Dirty	0.19	0.18

Table 12. Summary of force required by the axial expansion of the engine to begin and maintain sliding for both the clean and dirty rail sliders.

To Break Static Friction	
Axial Force Needed by Conventional Engine (lb)	
Clean	2390
Dirty	2405
Axial Force Needed by SoLoNOx Engine (lb)	
Clean	2863
Dirty	2882

To Maintain Kinetic Friction	
Axial Force Needed by Conventional Engine (lb)	
Clean	2169
Dirty	2373
Axial Force Needed by SoLoNOx Engine (lb)	
Clean	2598
Dirty	2843

Specification Verification

After testing, the list of design objectives was reviewed in order to make sure our design met expectations. Table 12 describes how each specification was fulfilled.

Table 13. Summary of how each objective was achieved in the design.

Objective	Brief Summary
Repeatability	Through the use of the jack and load cell, precise alignment of the engine is possible. Once the desired load is set, wedges and fasteners in the mount secure the position of the engine.
Alignment	Engine shaft positioning height is provided by the jack. Slots in the top plate allow for vertical motion of the mount. Slots in the engine bracket allow for horizontal positioning during assembly.
Axial Expansion	From testing, it was determined that the coefficient of static friction ranged from 0.18 to 0.19 depending on the condition of the rails. A contact with Solar Turbines is confident that the coefficient of friction should be low enough for the engine to overcome the static friction to move the sliders.
Radial Expansion	Radial expansion is absorbed using the same method as the previous design, which is through the use of the spring pack.
Vibration	Tests were conducted for 30 minutes which showed that the mount can withstand normal engine vibration. However, duration of testing is limited and for a conclusive result, longer tests are recommended.
Reliability	Based on testing, it is believed that the mount can maintain alignment. Similar to vibration, longer tests are recommended for dependable results.
Installation Time	The installation time for the aft-mount has been significantly reduced due to the elimination of the shimming procedure. During a simulation of the engine setup procedure, the alignment was able to be set within ten minutes.
Maintenance	The only maintenance required would be periodic cleaning of the rails on the slider system. This would ensure that the coefficient of friction would remain as low as possible to allow for any engine movement.
Manufacturability	The mount redesign requires more fabrication than the previous mount.

CONCLUSION AND RECOMMENDATION

There are several aspects that can be taken away as a result of the tests that were conducted on the new aft-mount design. First, preliminary prototype testing was an overall success. The repeatability of the mount's deflection over several tests is very accurate. In addition, the overall structural integrity of the mount was verified and determined to be sound. Given the tightening torque settings that were determined for safe operation, the triple redundant design to prevent mount failure was also deemed a success. In regard to the normal engine running vibrations, the mount is expected to withstand running conditions; however length of testing was somewhat limited due to test equipment access. Finally, rail friction testing proved that as the engine expands, it would have to overcome approximately 2400 pounds of force for the conventional engine, and 2900 pounds of force for the SoLoNOx engine.

As tests were being conducted, the deflection of the top plate was noticeably greater than original calculations showed. Based on the calculations defined in Appendix F, deflection analysis, the top plate was expected to deflect only about 0.010 inches. After testing, a maximum of 18,000 pounds of load was applied, and the deflection of the top plate was just over 0.040 inches. This difference between calculations and prototype testing was found to be a flaw in the model that was originally used to calculate the expected top plate deflection. The original model assumed a fixed-fixed end condition, where-as a pinned-pinned end condition was a more accurate model. Once new calculations were performed, viewable in Appendix F, the expected deflection of the top plate was approximately 0.039 inches. This number coincides with testing much more than the original 0.010 inches. Based on this new model, in order to decrease the deflection to be within tolerance limits as specified by Solar Turbines, it would be necessary to increase the thickness of the top plate by approximately 0.57 inches. An increase to make the top plate 2 inches thick would theoretically decrease the deflection of the top plate to under 0.010 inches.

For the duration of testing, the mount did not experience any structural issues. This leads to the conclusion that the overall design is very durable and can withstand the immense loading conditions experienced throughout engine operation. The tightening torque settings for each of the bolts are expected to provide enough frictional force between the plates to keep the mount in place once it has been set to the proper height. A test was also conducted to observe the wedge angle and whether or not friction alone could keep the mount in place at a desired height. The 15° wedge angle proved to be a shallow enough angle, that even when loaded to 18,000 pounds, the plates did not slip due to friction. This proves that the triple redundant design is expected to hold up the engine at the originally set height for the total operating life of the engine.

As for the rail slider system, several data points were collected in order to find the average static coefficient of friction and the average kinetic coefficient of friction for the rails

when they are both clean and dirty. Based on these tests, the largest static coefficient of friction expected for dirty rails is about 0.19. As it is tabulated in Table 11, the largest force the expansion of the engine would have to generate for the mount to move would be 2400 pounds for a conventional and 2900 pounds for a SoLoNOx engine. A source from Solar Turbines believes the experimentally determined coefficients of friction are low enough for the engine's thermal expansion to overcome the static friction.

The results from vibration testing on the mount were somewhat limited. Based on the information that was gathered, the vibration characteristics of the mount appear to be good. Each test was run for 30 minutes each. This very short amount of time limits the conclusions that can be made based on the data that was collected. Also, the equipment only recorded data once per second. This also significantly reduces the accuracy of any observations that can be made. For future testing it would be ideal to increase data collection frequency for vibration tests. This would allow for plots to show a full spectrum of loading over time. Also, more accurate gages should be used that can follow the change in height quickly would be a good upgrade. For the SoLoNOx engine test, the gage was not capable of keeping up with changing deflection and therefore, no correlation could be found in the data. As it stands, the data shows that as the load increases, the spring compresses and as the load decreases, the spring pack decompresses. This test should be run for a much longer period of time to really determine whether or not the mount is affected by continuous exposure to high load vibration.

It would also be beneficial for rail slider testing to improve the DAQ system. For the tests conducted on the Cal Poly campus, the DAQ did not have the ability to save maximum load values, so the tester had to determine and record the highest load from the readout. This introduces error that could be avoided by a system that records values straight to a computer or only displays the highest value achieved. Also, it would be better to have more data points to generate a curve. The increased data would provide a more accurate trend line for the coefficients of friction. With these improvements, a more accurate number can be achieved for the static and kinetic coefficients of friction for the clean and dirty rails.

Cost Summary

Table 14. Component parts and cost summary for the construction of the engine aft-mount. This is not including any costs associated with the previous design including the Belleville washers.

Part	Quantity	Cost	Brief Summary
Stainless Steel Metal Stock		$3091.00	Material used to assemble and manufacture main components.
Manufacturing		$27,794.12	Time spent during fabrication of main components.
1”-14 x 3” Bolts	6	$11.50	Used in main assembly. Located in top plate and lower wedges.
1”-14 Nuts	6	$3.39	Used on 1” bolts.
1” Washers	12	$3.42	Used on 1” bolts.
0.5”-20 x 1.75” Bolts	4	$1.37	Used in turbine assembly. Located where engine bracket is attached to mount.
0.5”-20 Nuts	4	$0.65	Used on 0.5” bolts.
0.5” Washers	8	$0.20	Used on 0.5” bolts.
Rail-Sliding System	1	$2,012.58	Used in main assembly. First component to attach to turbine skid.
Total		$33,064.59	

Table 15. Tooling components and cost summary for each engine aft-mount. This is not including the cost required for jack support material or interface plate material.

Part	Quantity	Cost	Brief Summary
Mechanical Jack	1	$2,132.80	Used during main assembly. Used to raise the assembly to proper shaft alignment.
Honeywell Load Cell	1	$1,239.00	Used during main assembly. Used to determine proper height of mount.
Interface Plate and jack support hardware	1	TBD	Used to attach load cell to the jack as well as give the jack a flat surface to rest on.
Total		$3,371.80 +	

Appendix A: Contact Information

Group Email: cpturbotech@gmail.com

Group Contact Info:

Harrison (Woody) Newell
hnewell@calpoly.edu
619.672.7118

Michael Haworth
mhaworth@calpoly.edu
951.505.0076

Dylan Perry
drperry@calpoly.edu
916.704.7268

Sponsor Contact Info:

Solar Turbines
9330 Sky Park Court
San Diego, CA 92123

Ismael DePaz
depaz_ismael_a@solarturbines.com
858.715.2077

Varadharaja Sampathkumar
sampathkumar_varadharaja@solarturbines.com
858.292.3158

Donovan Vick
Vick_donovan_r@solarturbines.com
858.505.8549

Oscar Lopez
lopez_oscar_t@solarturbines.com
619.948.4653

Advisor Contact Info:

California Polytechnic State University: San Luis Obispo
1 Grand Avenue
San Luis Obispo, CA 93407

Dr. Joseph Mello
jdmello@calpoly.edu
805.756.1356

Testing Contact Info:

Dr. Daniel Jansen
djansen@calpoly.edu
805.756.2264

Jim Gerhardt
jgerhard@calpoly.edu
805.756.1278

Cal Poly Shipping Contact Info:

Frank Limon, Jr.
flimon@calpoly.edu
(805) 756-6648

Appendix B: Summary of Important Figures

This appendix provides a quick reference guide to important figures and tables used in this report. The following only provides a brief description on topics that are discussed in detail in the above report.

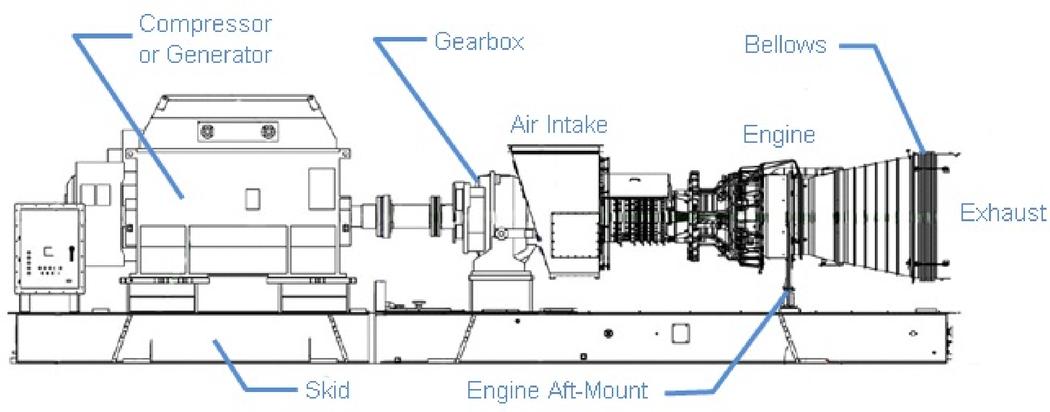


Figure B1. Side view drawing of the Titan 130 package showing relevant features of the platform.

Table B1. Summary of background into the engine aft-mount redesign.

Background	Brief Summary
Current Problem	Solar Turbines needs to be able to accurately set the engine on the mount every time regardless of engine, skid, and environment.
Focus on Aft-Mount	Even though there are other issues in that could affect the mount, such as exhaust position and the gearbox-to-engine interface, our scope is just on the mount assembly itself.
Driving Force	The mount is primarily load driven even though Solar Turbines currently uses deflection to set the mount.
Belleville Springs	Washer springs are great under high load and have low creep characteristics, however like all springs, the deflection becomes less predictable with time
Tall Splash Plate	The large engine bracket puts an uneven load on the spring pack during axial expansion of the engine, which puts excess wear on the washer springs .
Engine Types	There are two different engines that will sit on the aft-mount. First, the conventional engine applies a load of 13,000lbs to the mount. Second, the SoLoNOx engine applies a load of 15,500lbs to the mount due to the added weight of the environmentally friendly engine.
Skid Types	There are three different skids that the mount must be incorporated into. The Oil and Gas package has a very stiff frame with three feet that secure the skid to the deck. The PG onshore package has eight tie down pads and a smaller skid frame. Both of those models utilize trunnions to add support to the engine. The TBM package is built by TurboMach and has six tie down pads and no trunnions.

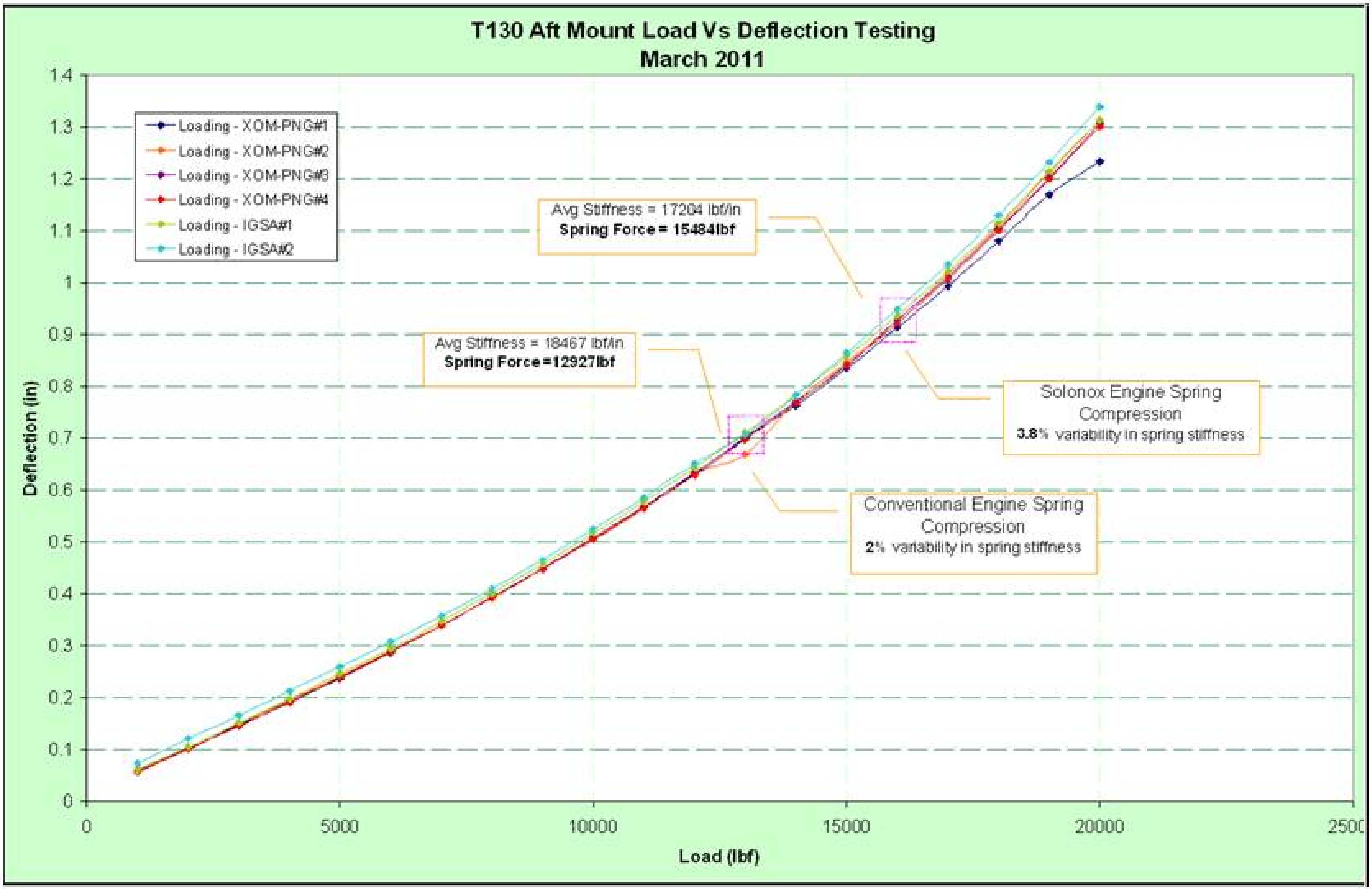


Figure B2. Load – Deflection curve for six different identical spring packs. Belleville springs do not have a linear spring constant as seen above. Notice the variability in the spring stiffness at the engine loading positions.

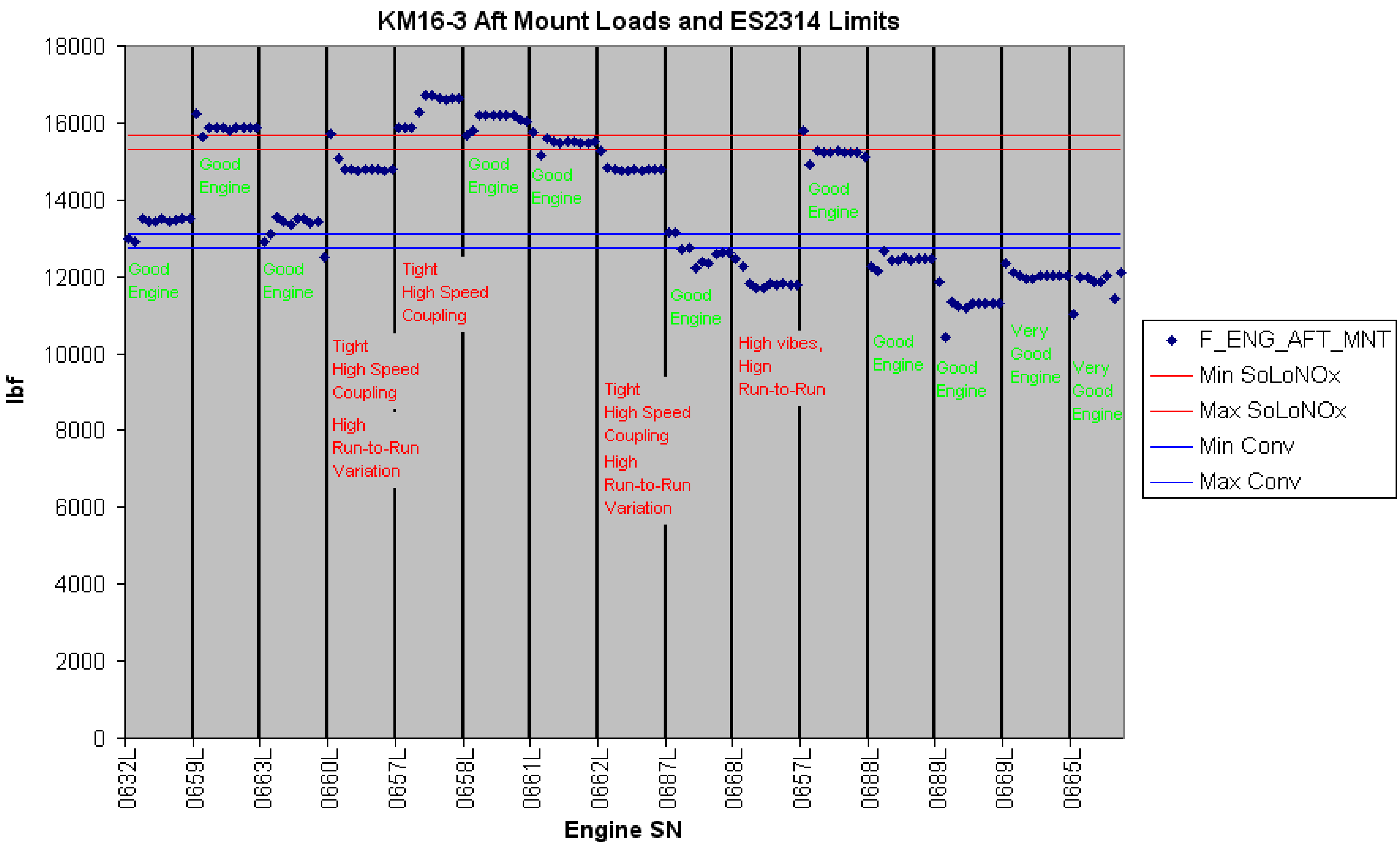


Figure B3. This diagram shows aft-mount loading for fifteen total engine packages (both conventional and SoLoNOx variations) through various stages of engine run up. The condition of the engine is shown in green for a good engine and red for a problem engine.

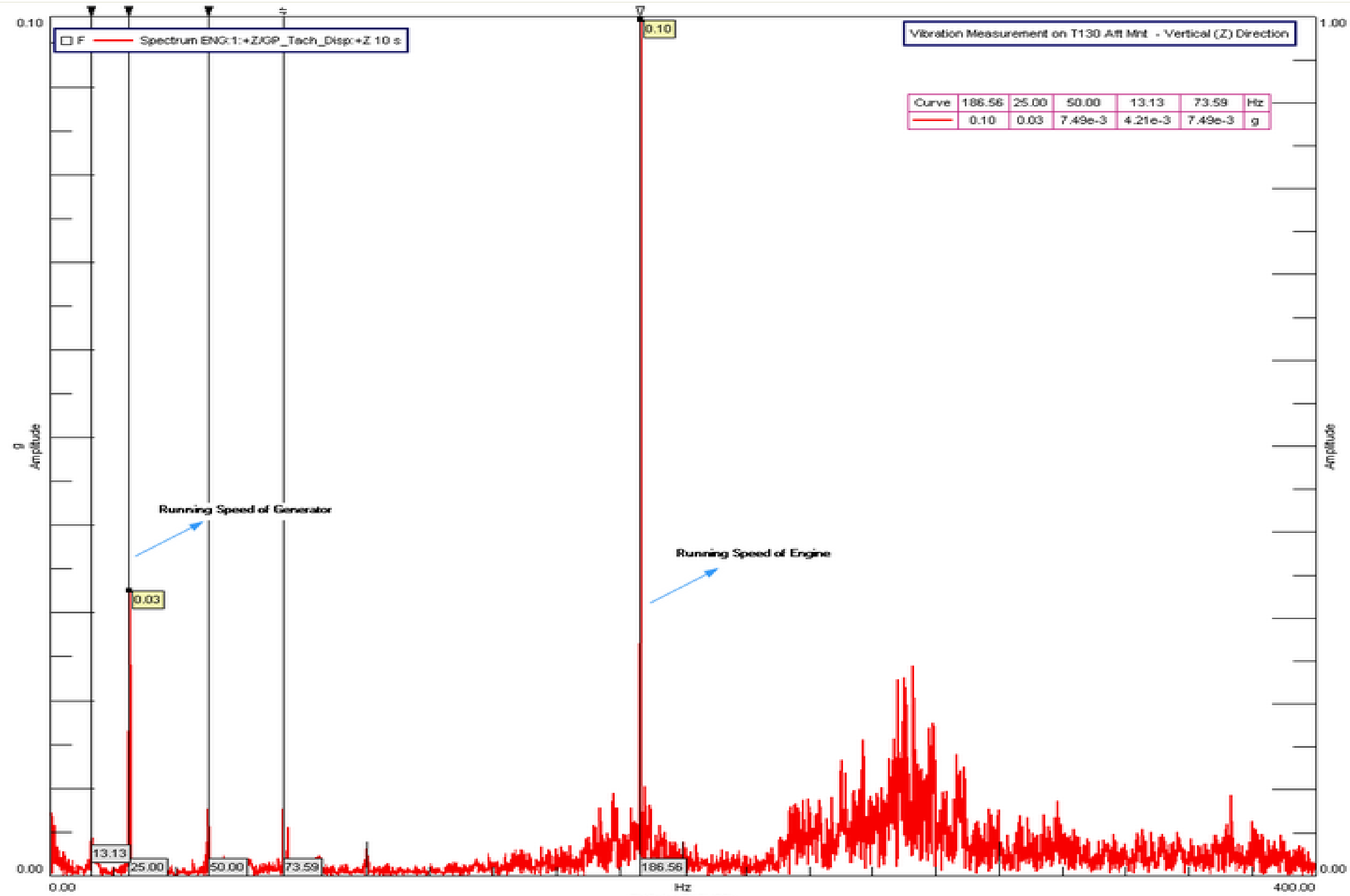


Figure B4. Vibration data taken over a range of operating speeds. At the running speed of the engine, the maximum vibration amplitude is 0.1g which is used to calculate the dynamic load factor.

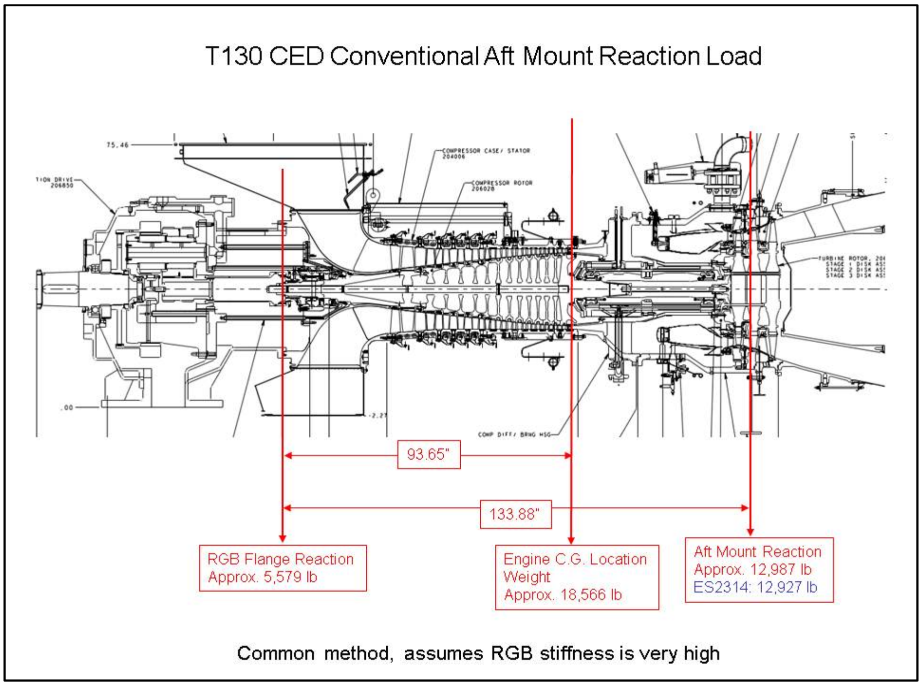


Figure B5. Diagram showing the reaction forces for a conventional Titan 130 engine at the gearbox-engine interface and the mount location.

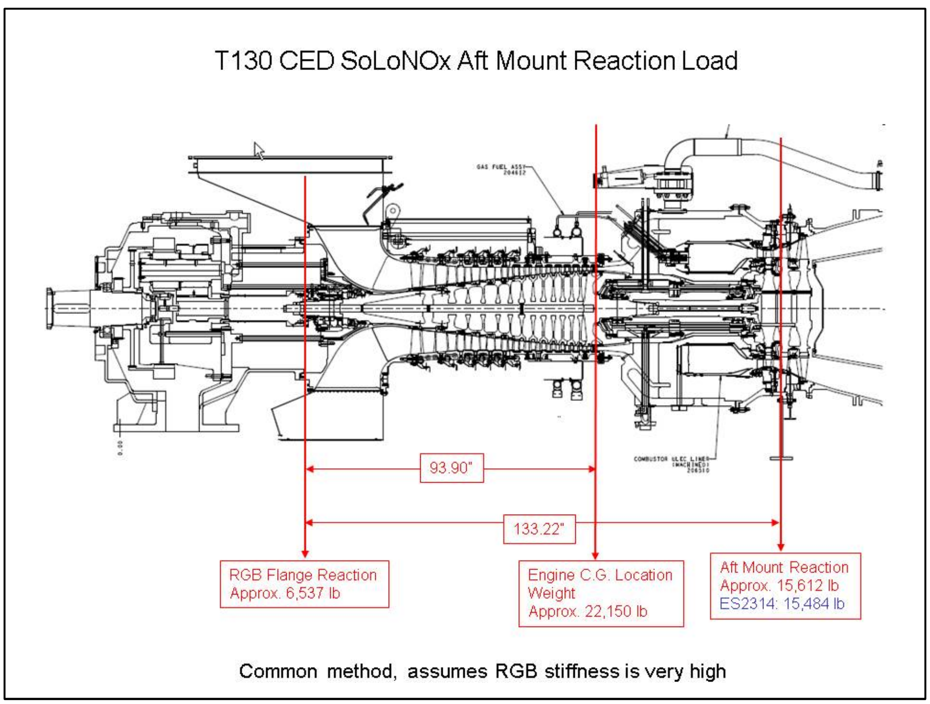


Figure B6. Diagram showing the reaction forces for a SoLoNOx Titan 130 engine at the gearbox-engine interface and the mount location.

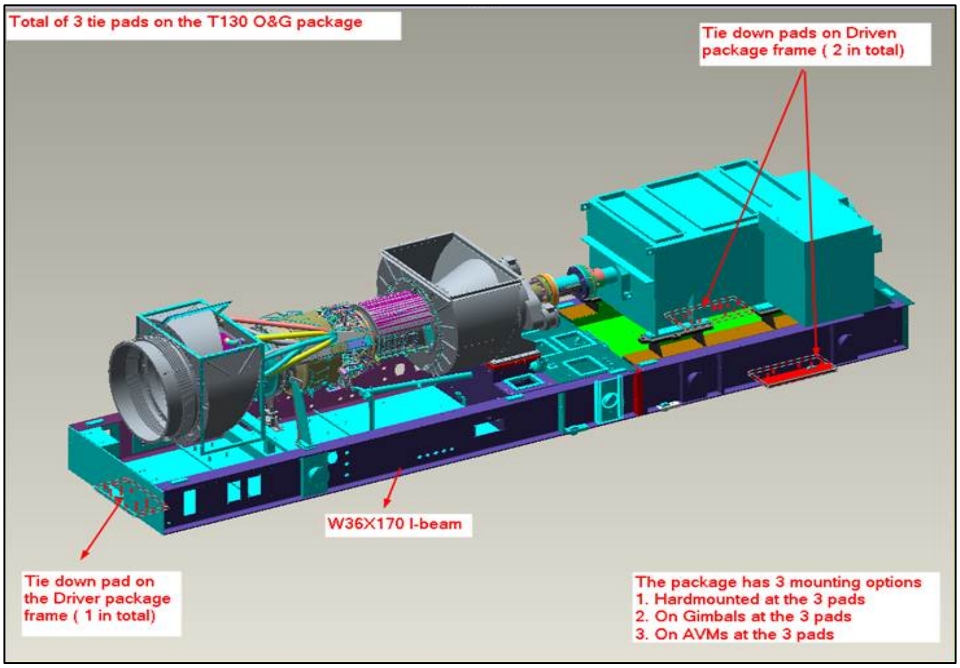


Figure B7. Solid model of the Titan 130 Oil and Gas package for offshore applications. This model features the stiffest frame of the three, and is mounted in three locations to the foundation.

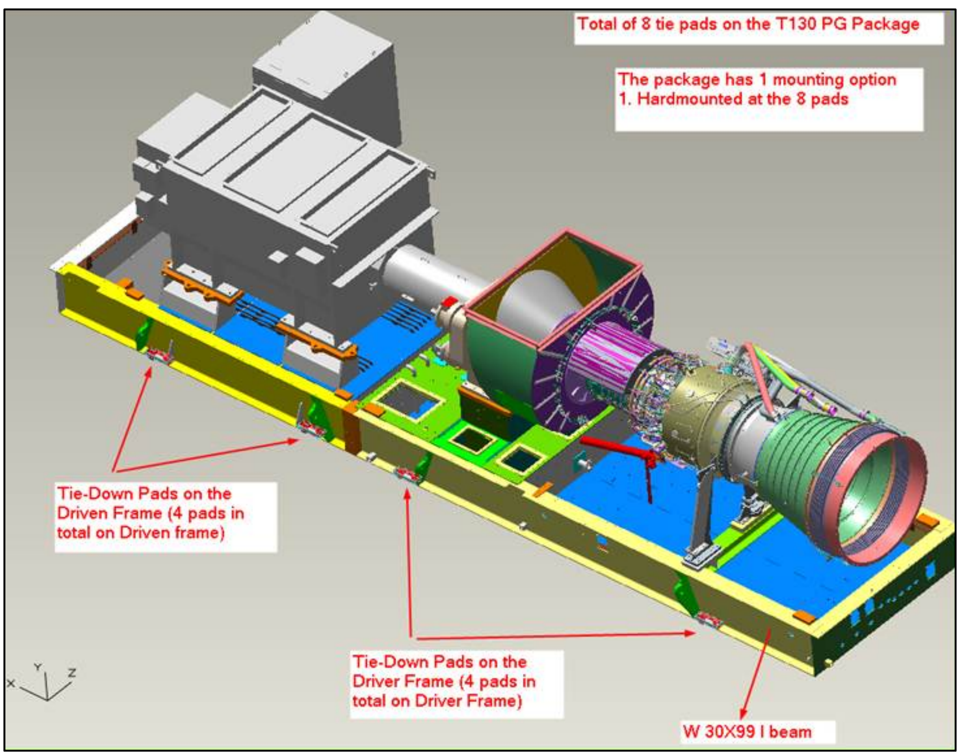


Figure B8. Solid model of the Titan 130 PG Onshore package. This model features eight tie down pads and a smaller frame than the offshore package.

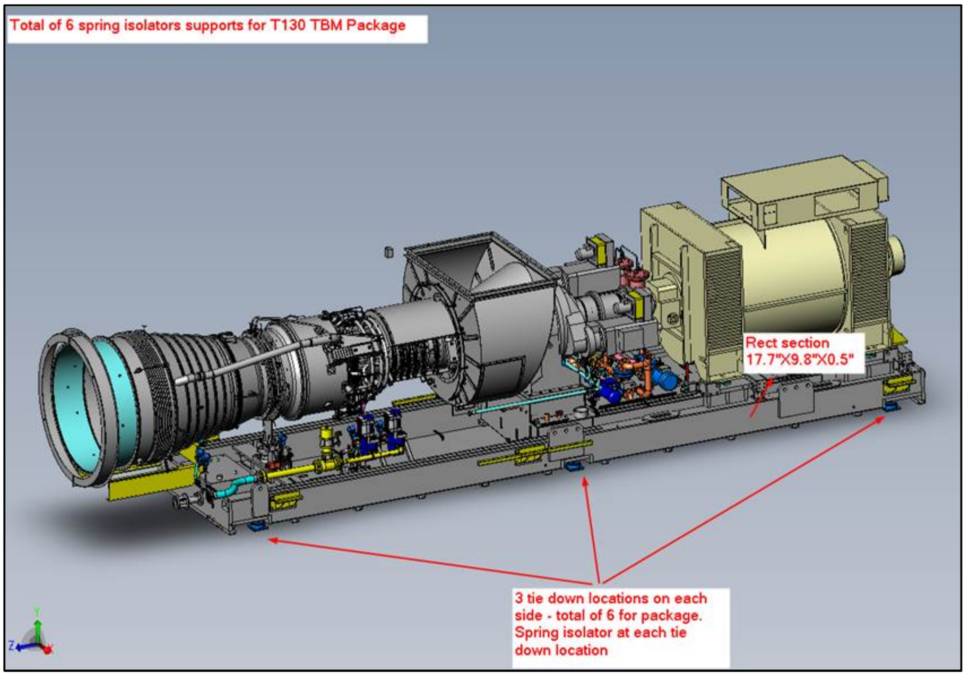
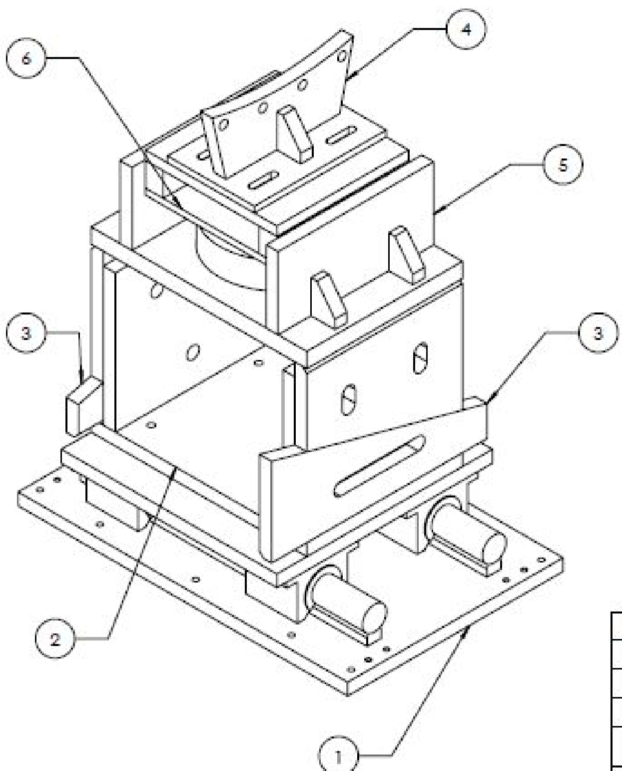


Figure B9. Solid model of the Titan 130 TBM package. This model is built by TurboMach in Europe and has a different frame design than the Solar Turbines counterparts. The six tie down pads feature spring isolators. Also, no trunnions are used on this model.

Table B2. Summary of engineering requirements and objectives for the aft-mount redesign.

Objective	Brief Summary
Repeatability	Assembly staff or field installation team must know how to accurately set the alignment of the engine. Engine must be set into proper alignment regardless of the age, condition, or load on the mount.
Alignment	Alignment of the shaft relies on up and down positioning as well as left to right movement of the engine.
Axial Expansion	Mount must accommodate motion in the axial direction to account for thermal expansion.
Radial Expansion	Mount must accommodate motion in the radial direction to account for thermal expansion.
Vibration	Resonant frequency of 2050Hz must be avoided.
Reliability	Must not fail in operation and last for over 30,000 hours
Installation Time	Shorter than or equal to current design of one hour to assemble and an hour and a half to level the engine
Maintenance	Must be accessible and safe to work on
Manufacturability	Must be manufactured in a time effective manner or available to purchase from a vendor
Cost	Not a limiting factor considering failure of the mount leads to a $600,000 gearbox replacement, but the mount is support equipment to the engine so cheaper is always better

Appendix C: Final Drawings



RAIL ASSEMBLY IS A VENDOR PART

THE RAIL SLIDERS AT THE BASE CANNOT BE PAINTED. ALL OTHER PARTS ARE OKAY TO PAINT

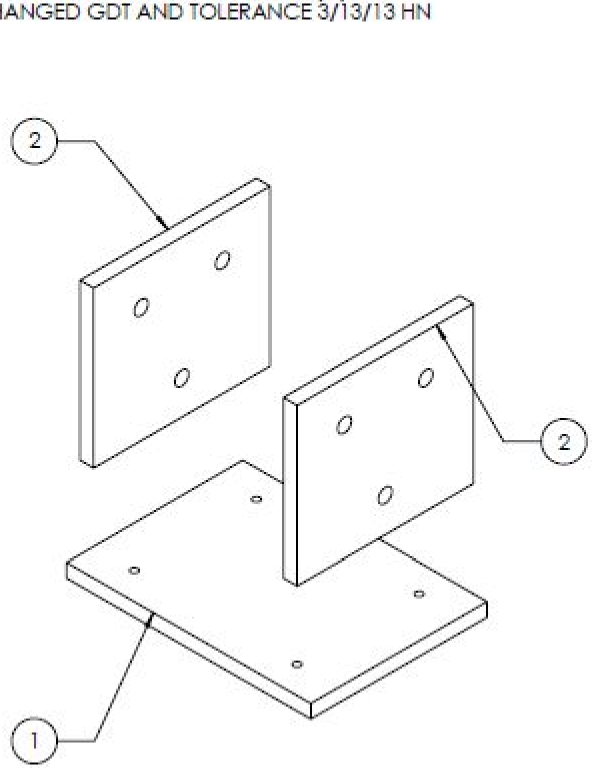
ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	2005	RAIL ASSEMBLY	1
2	2004	MOUNT BASE ASSEMBLY	1
3	1004	WEDGE	2
4	2003	ENGINE BRACKET ASSEMBLY	1
5	2007	MOUNT TOP ASSEMBLY	1
6	2006	SPRING PACK ASSEMBLY	1

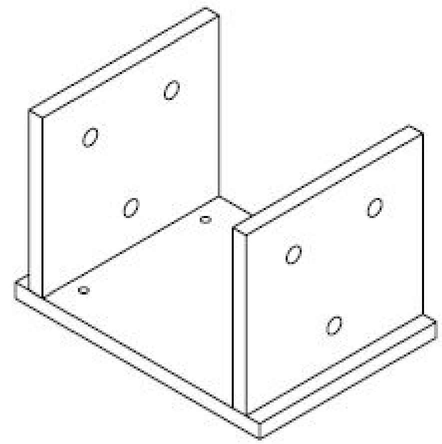
TURBO TECH

DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE:	MATERIAL:	
NEXT ASSY:	UNITS: INCHES	TITLE: AFT-MOUNT ASSEMBLY	
DWG #: 3000	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

REVISIONS: ADDED WELDING INSTRUCTIONS 3/7/13 HN
CHANGED GDT AND TOLERANCE 3/13/13 HN





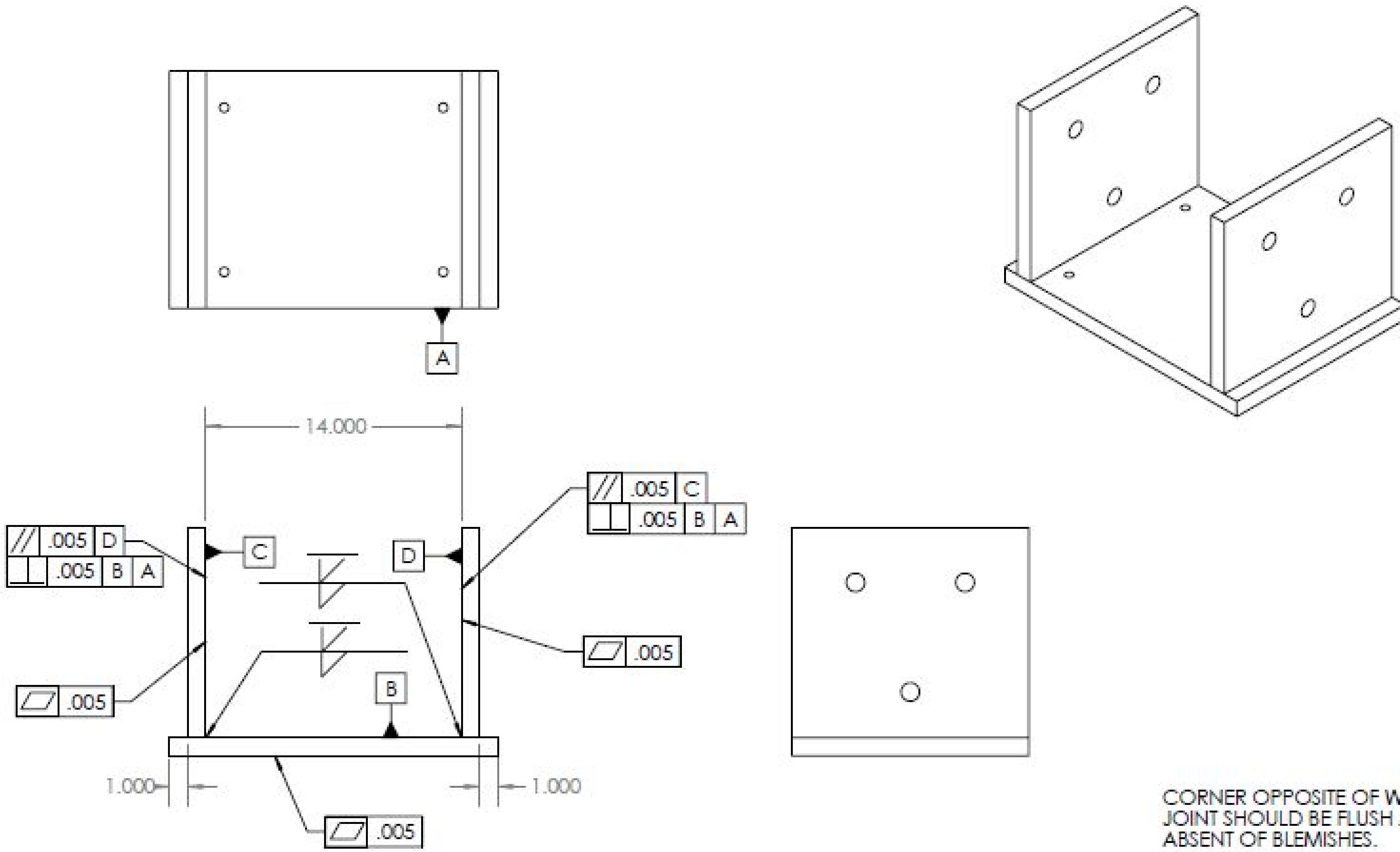
EACH PART IS TO HAVE A FILLET WELD AT EACH CORNER JOINT.

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	1013	BASE PLATE	1
2	1014	BASE PLATE SIDE	2

TURBO TECH

DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL:	
NEXT ASSY: 3000	UNITS: INCHES	TITLE: MOUNT BASE ASSEMBLY	
DWG #: 2004	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

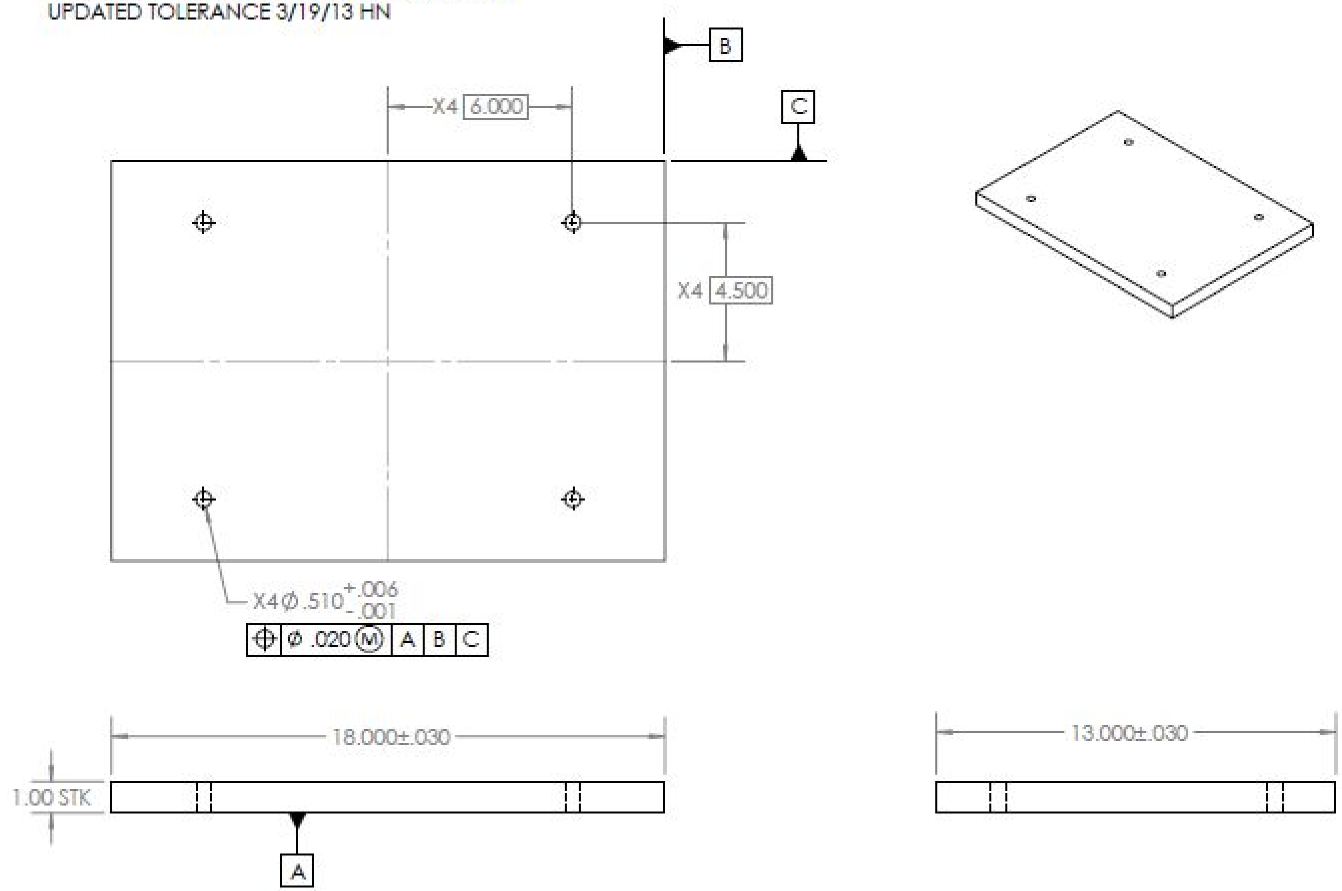


CORNER OPPOSITE OF WELDED JOINT SHOULD BE FLUSH AND ABSENT OF BLEMISHES.

TURBOTECH				
DRAWN BY: HARRISON NEWELL	INIT:		CKD BY: MICHAEL HAWORTH	INIT:
DATE: 3/7/2013	TOLERANCE: ±0.06		MATERIAL: STEEL	
NEXT ASSY: 3000	UNITS: INCHES		TITLE: MOUNT BASE ASSEMBLY	
DWG #: 2004	SCALE: 1:8		CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

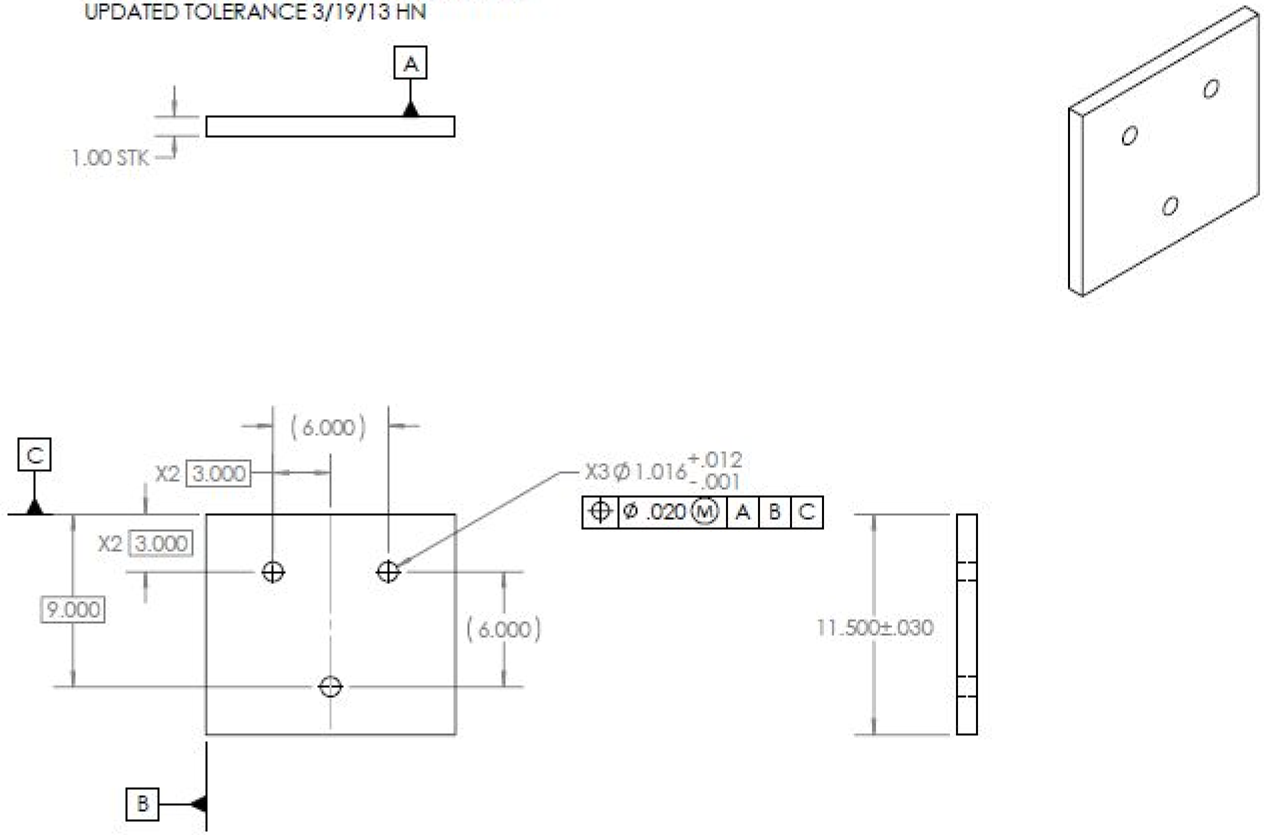
REVISIONS: CHANGED GDT AND TOLERANCE 3/13/13 HN
UPDATED GDT AND TOLERANCE 3/14/13 HN
UPDATED TOLERANCE 3/19/13 HN



DRAWN BY: HARRISON NEWELL		INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06		MATERIAL: STEEL	
NEXT ASSY: 2004	UNITS: INCHES		TITLE: BASE PLATE	
DWG #: 1013	SCALE: 1:5		CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

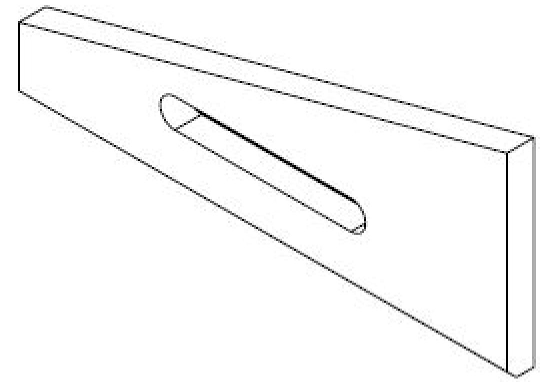
REVISIONS: CHANGED TOLERANCE 3/13/13 HN
UPDATED GDT AND TOLERANCE 3/14/13 HN
UPDATED TOLERANCE 3/19/13 HN

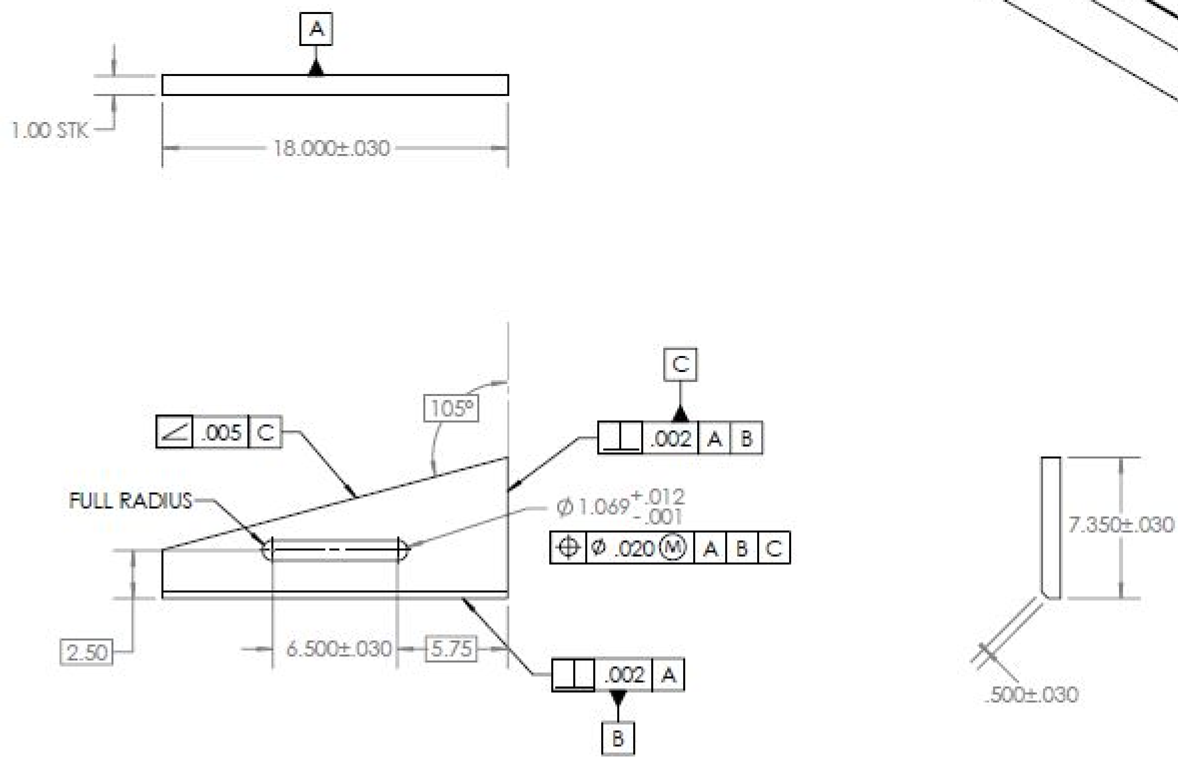


DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.01	MATERIAL: STEEL	
NEXT ASSY: 2004	UNITS: INCHES	TITLE: BASE PLATE SIDE	
DWG #: 1014	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

REVISIONS: ADDED CHAMFER 3/13/13 HN
CHANGED GDT AND TOLERANCE 3/13/13 HN
UPDATED GDT AND TOLERANCE 3/14/13 HN
UPDATED TOLERANCE 3/19/13 HN

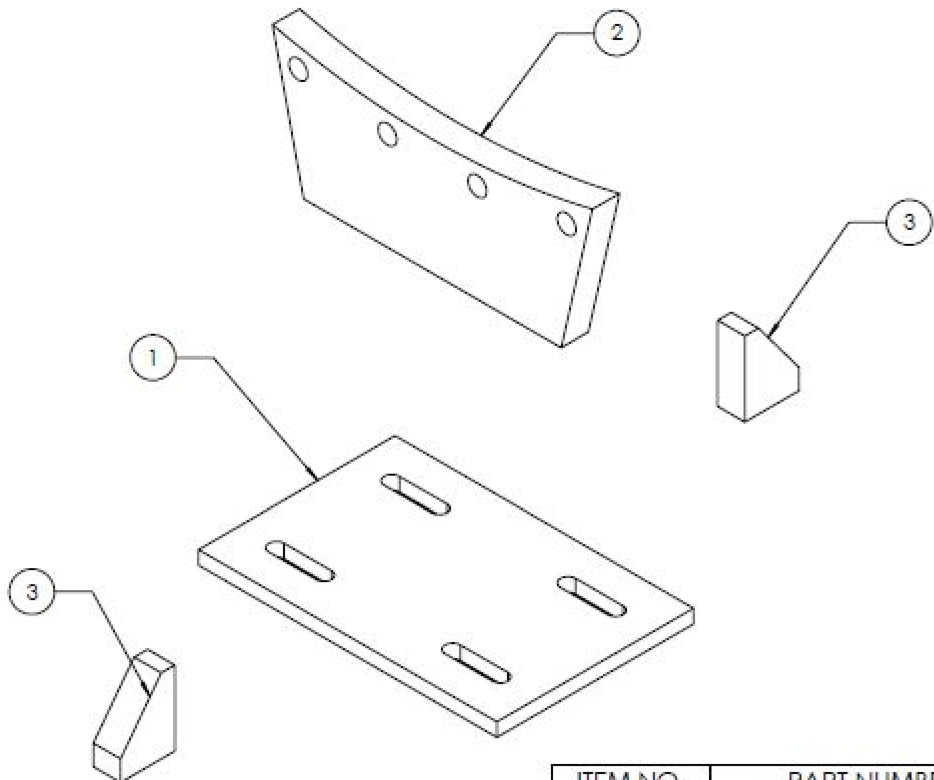


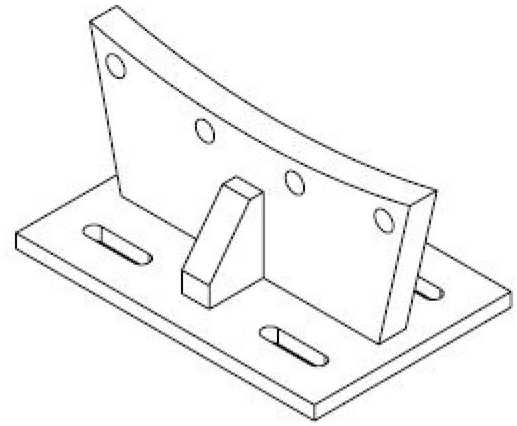


DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 3000	UNITS: INCHES	TITLE: WEDGE	
DWG #: 1004	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

REVISIONS: ADDED WELDING INSTRUCTIONS 3/7/13 HN
CHANGED TOLERANCE 3/13/13 HN





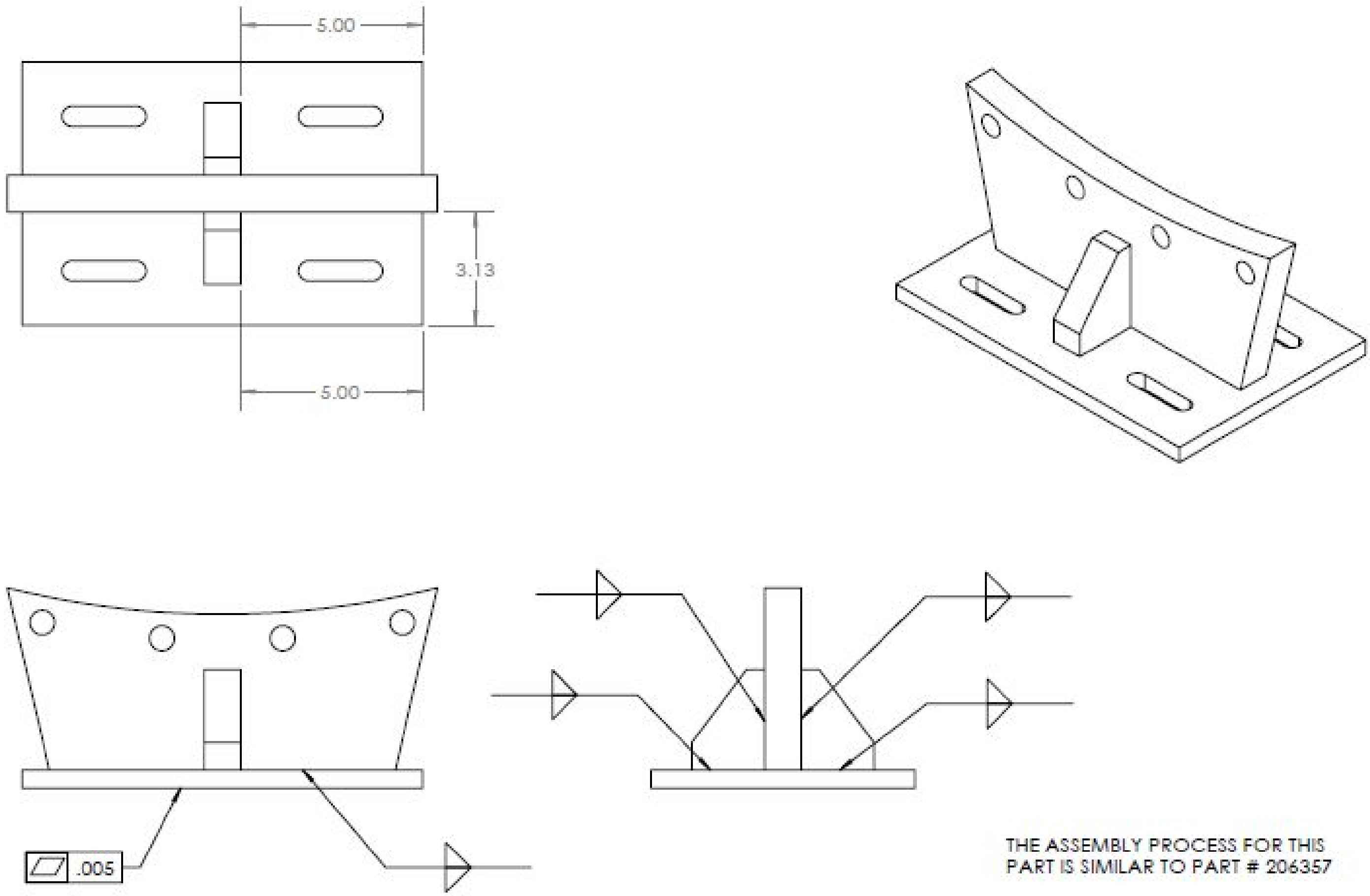
EACH PART IS TO HAVE A FILLET WELD AT EACH CORNER JOINT.

PART #1010 CAN BE MADE FROM SOLAR PART #206357

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	1012	ENGINE BRACKET BASE	1
2	1010	ENGINE BRACKET	1
3	1011	BRACE	2

DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL:	
NEXT ASSY: 3000	UNITS: INCHES	TITLE: ENGINE BRACKET ASSEMBLY	
DWG #: 2003	SCALE: 1:4	CP Senior Project - Engine Aft-Mount	

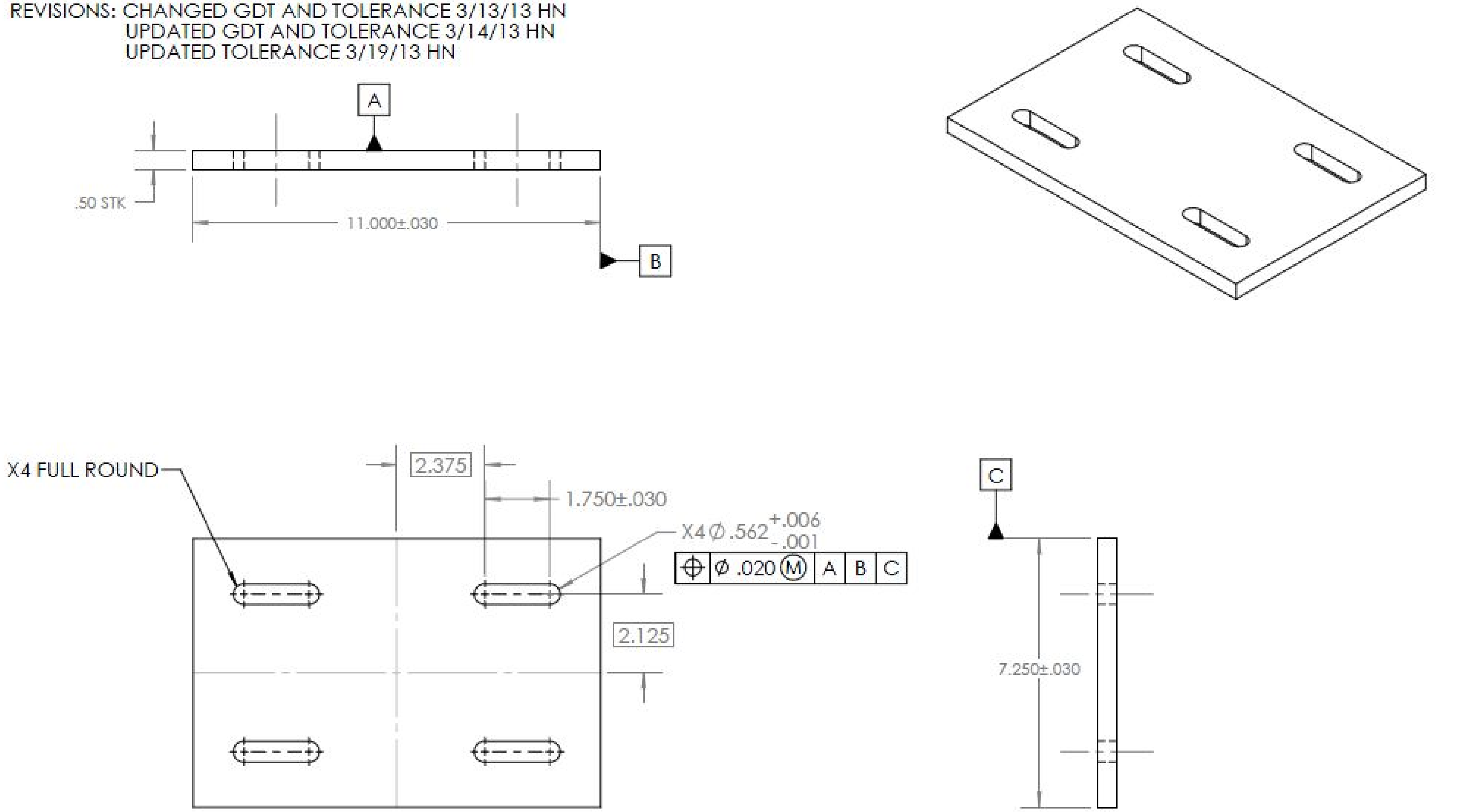
5 4 3 2 1



DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 3/7/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 3000	UNITS: INCHES	TITLE: ENGINE BRACKET ASSEMBLY	
DWG #: 2003	SCALE: 1:4	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

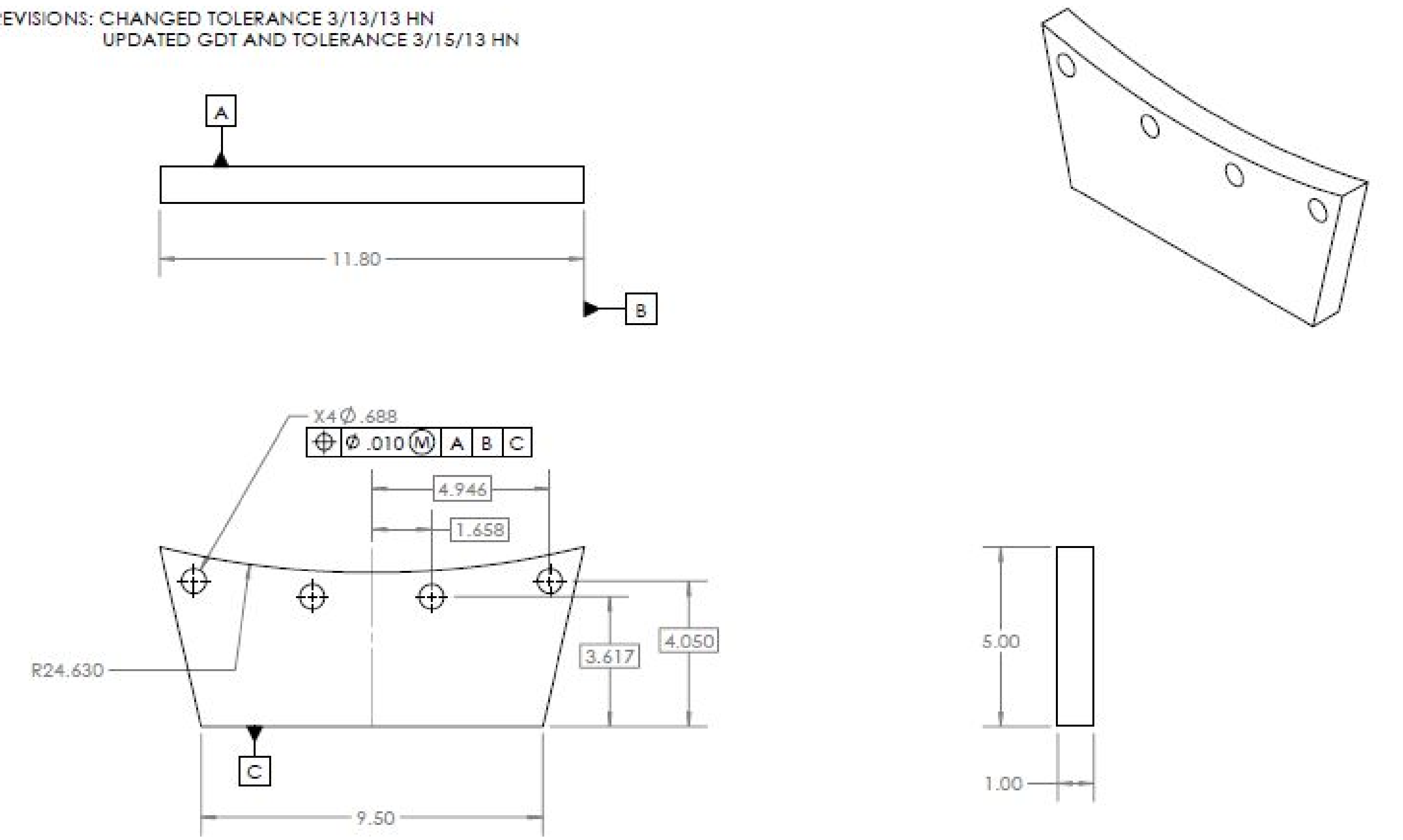
REVISIONS: CHANGED GDT AND TOLERANCE 3/13/13 HN
UPDATED GDT AND TOLERANCE 3/14/13 HN
UPDATED TOLERANCE 3/19/13 HN



DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 2003	UNITS: INCHES	TITLE: ENGINE BRACKET BASE	
DWG #: 1012	SCALE: 1:4	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

REVISIONS: CHANGED TOLERANCE 3/13/13 HN
UPDATED GDT AND TOLERANCE 3/15/13 HN

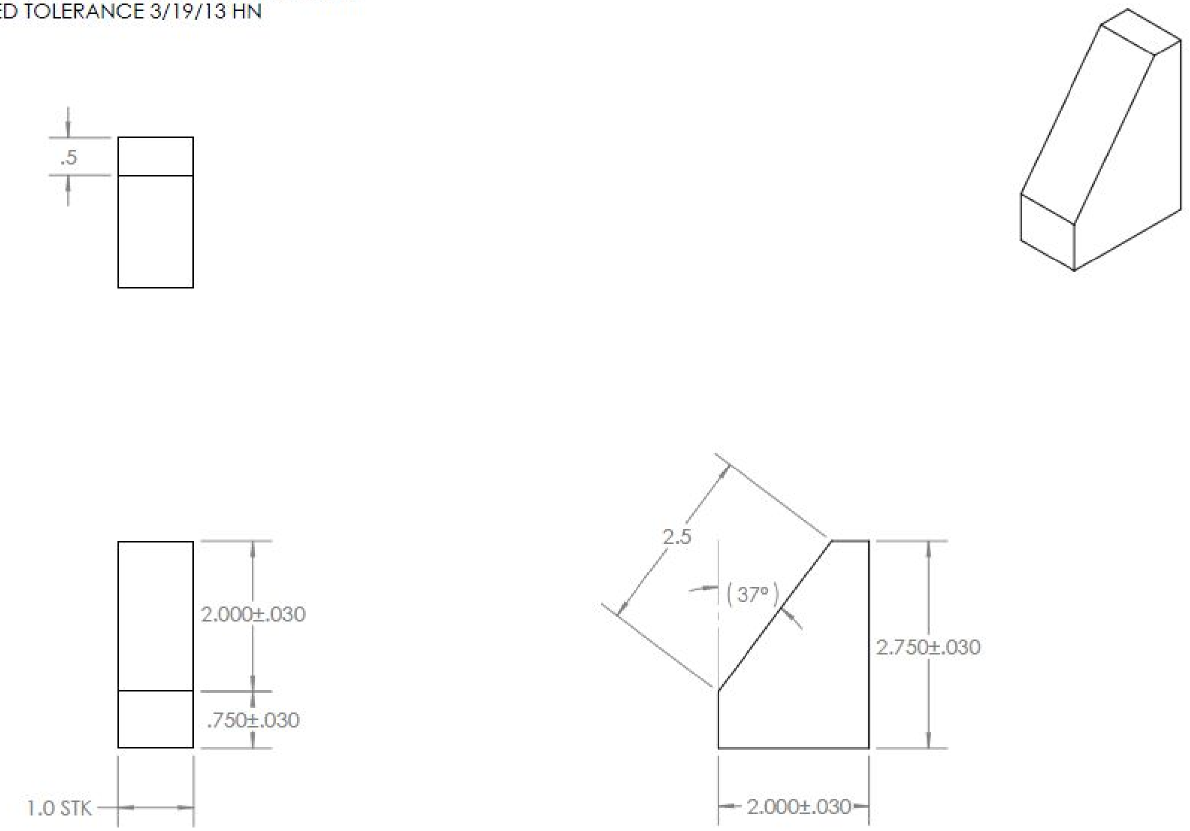


MAKE FROM SOLAR PART# 206357

DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.01	MATERIAL: STEEL	
NEXT ASSY: 2003	UNITS: INCHES	TITLE: ENGINE BRACKET	
DWG #: 1010	SCALE: 1:4	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

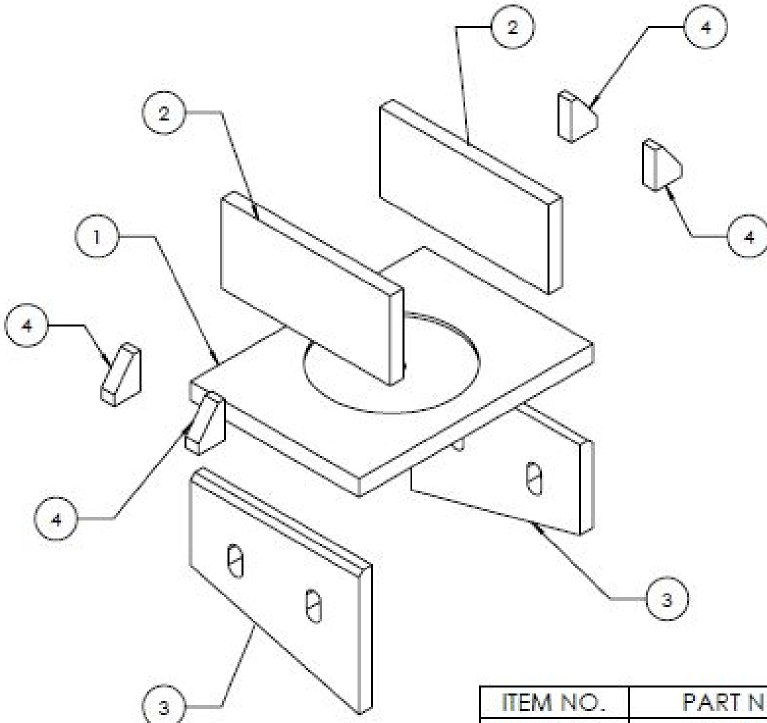
REVISIONS: CHANGED TOLERANCE 3/13/13 HN
ADDED ANGLE REFERENCE DIM 3/14/13 HN
UPDATED TOLERANCE 3/19/13 HN

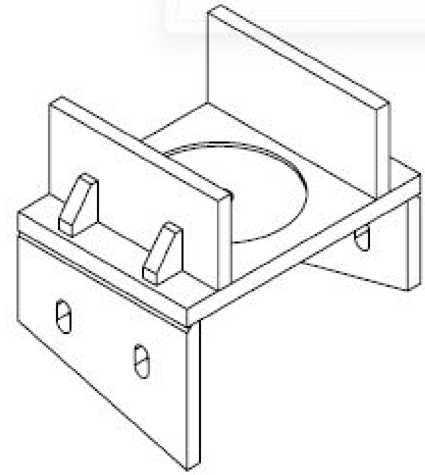


DRAWN BY: HARRISON NEWELL		INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06		MATERIAL: STEEL	
NEXT ASSY: 2003	UNITS: INCHES		TITLE: BRACE	
DWG #: 1011	SCALE: 1:2		CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

REVISIONS: ADDED WELDING INSTRUCTIONS 3/7/13 HN
CHANGED GDT AND WELD SYMBOLS 3/13/13 HN



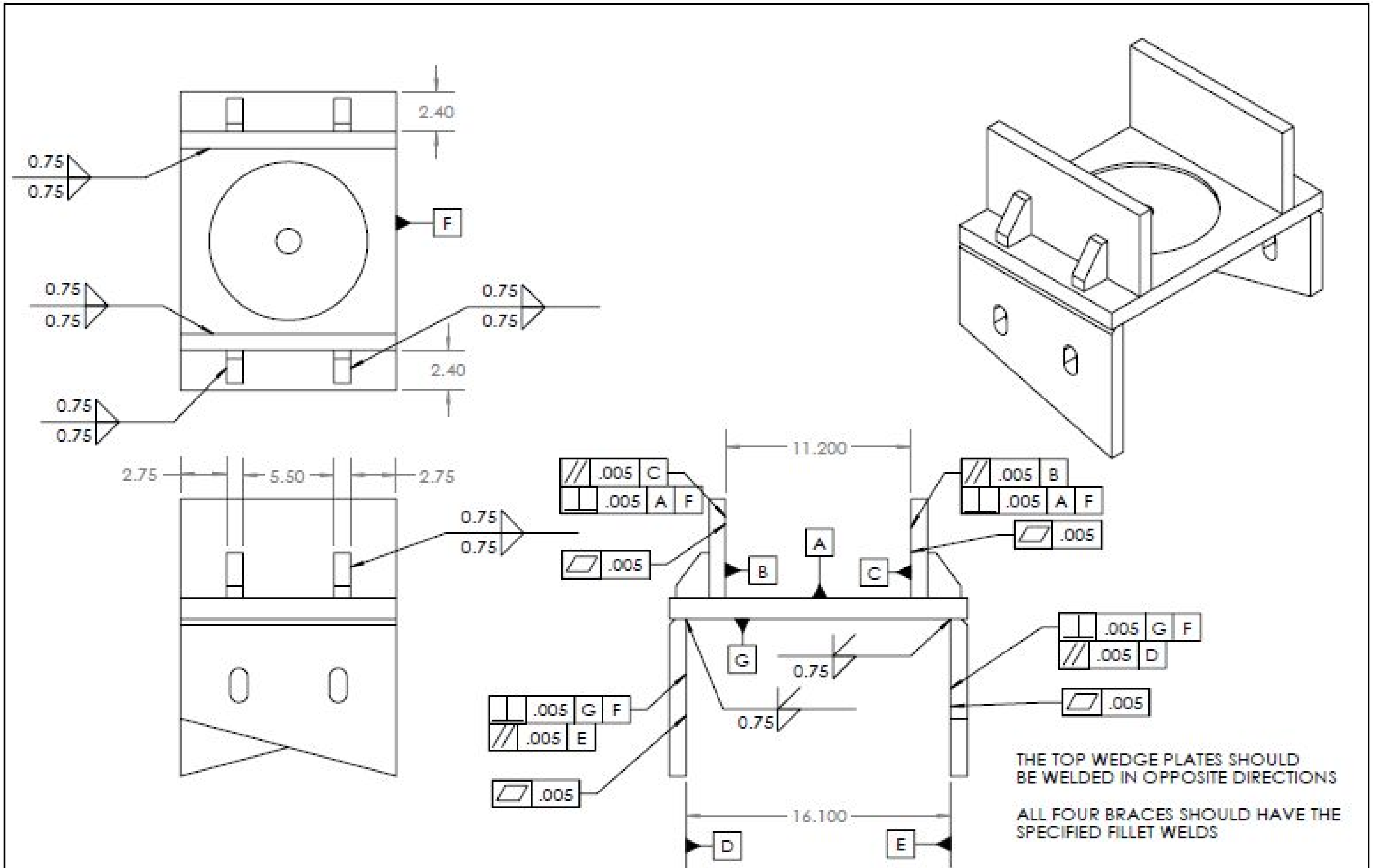


EACH PART IS TO HAVE A FILLET WELD AT EACH CORNER JOINT.

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	1006	MAIN TOP PLATE	1
2	1007	SPRING CONSTRAINT	2
3	1005	TOP WEDGE	2
4	1011	BRACE	4

TURBOTECH

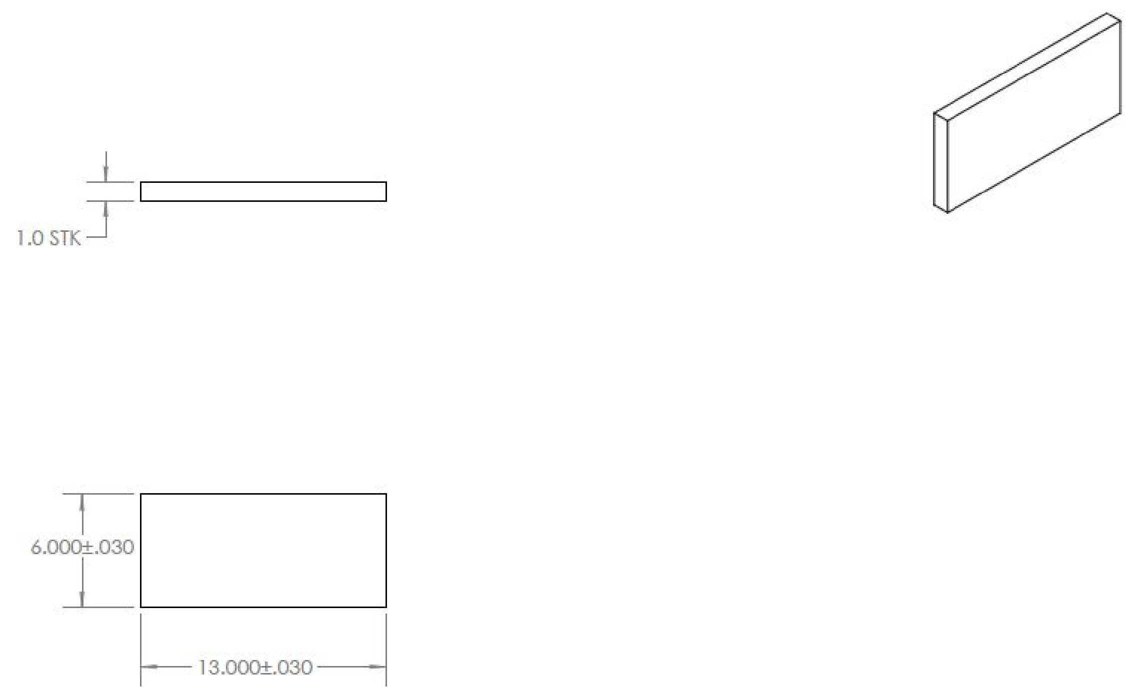
DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL:	
NEXT ASSY: 3000	UNITS: INCHES	TITLE: MOUNT TOP PLATE ASSEMBLY	
DWG #: 2007	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	



TURBOTECH	DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
	DATE: 3/7/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
	NEXT ASSY: 3000	UNITS: INCHES	TITLE: MOUNT TOP PLATE ASSEMBLY	
	DWG #: 2007	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

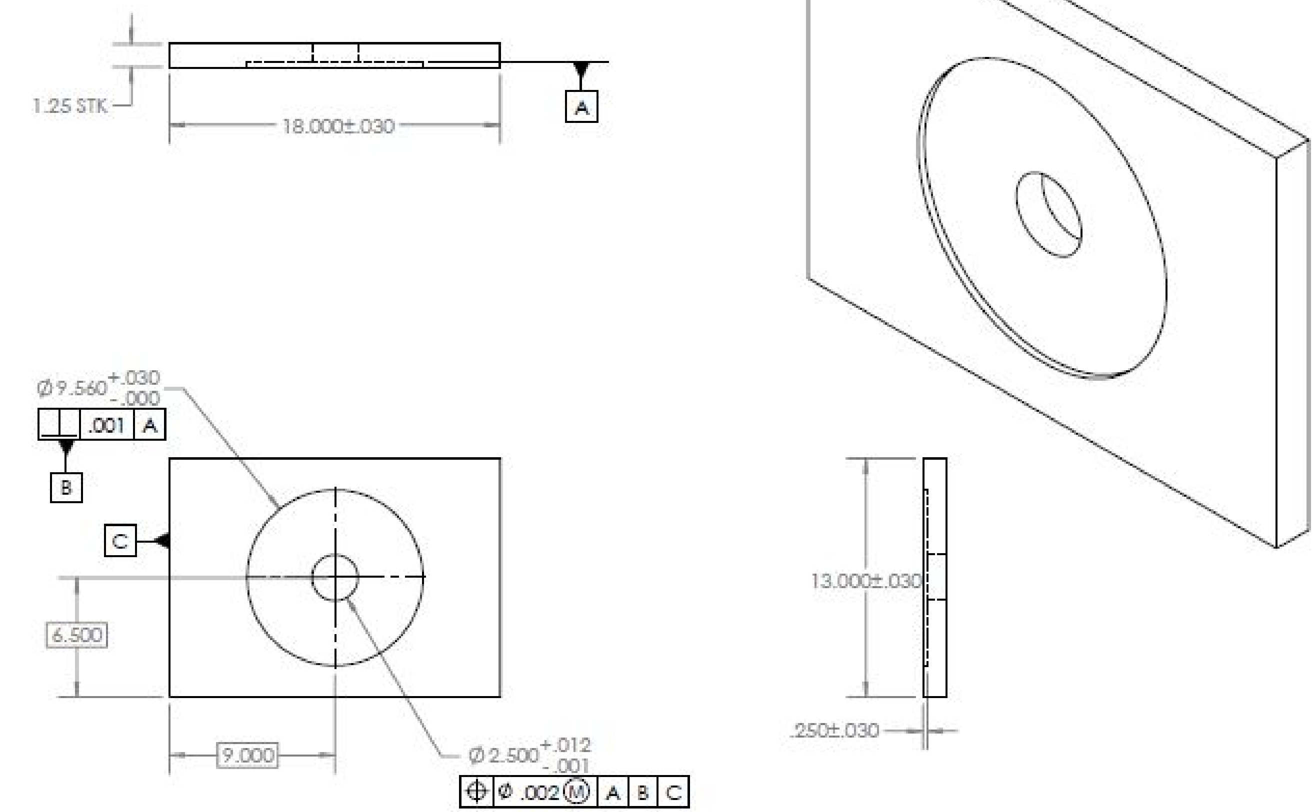
REVISIONS: CHANGED TOLERANCE 3/13/13 HN



DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 2007	UNITS: INCHES	TITLE: SPRING CONSTRAINT	
DWG #: 1007	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

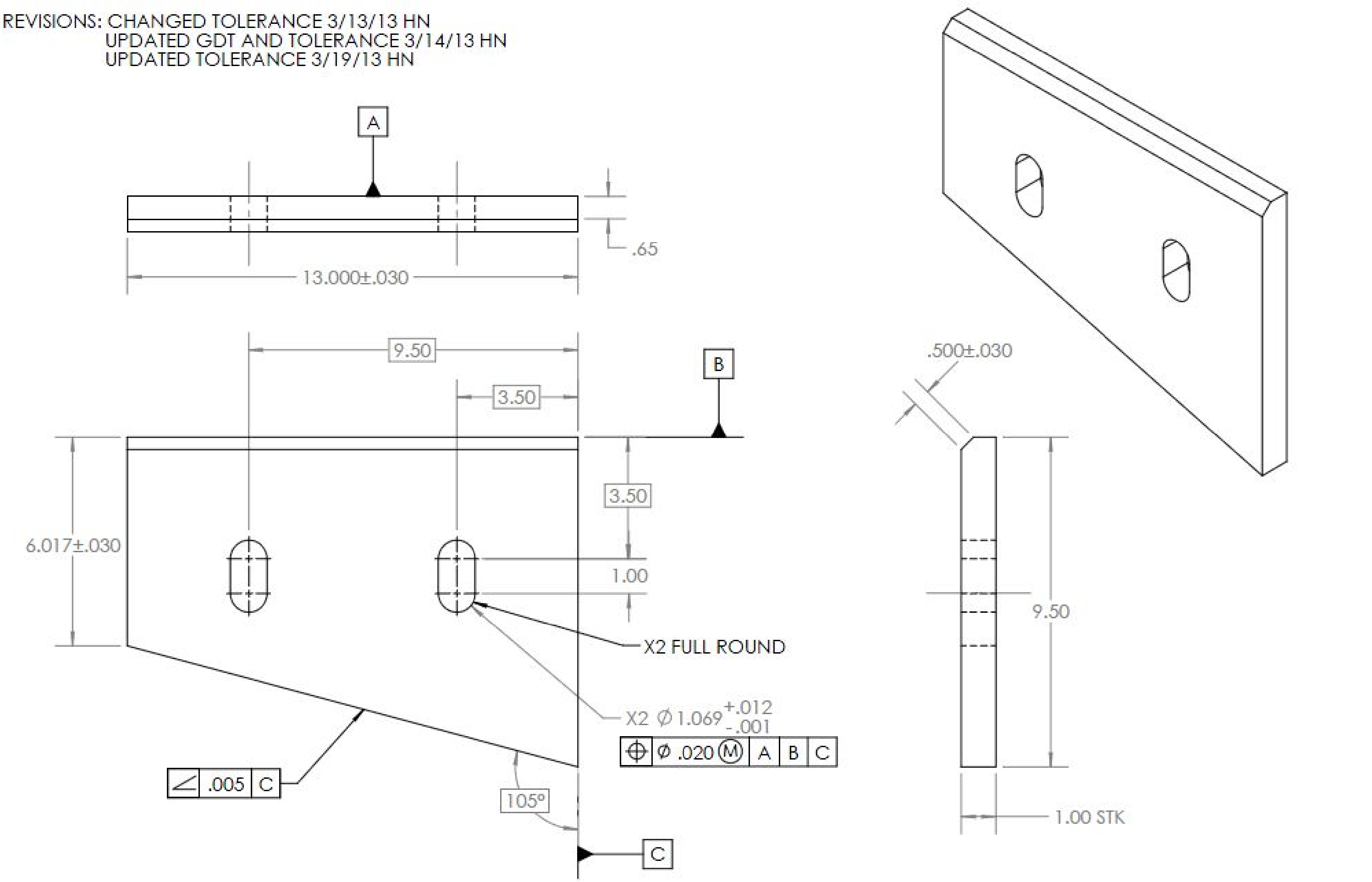
REVISIONS: CHANGED TOLERANCE AND GDT 3/13/13 HN
UPDATED GDT AND TOLERANCE 3/14/13 HN
UPDATED TOLERANCE 3/19/13 HN
UPDATED TOLERANCE 3/20/13 HN
CHANGED HOLE DIAMETER 5/27/13 HN



DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 2007	UNITS: INCHES	TITLE: MAIN TOP PLATE	
DWG #: 1006	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

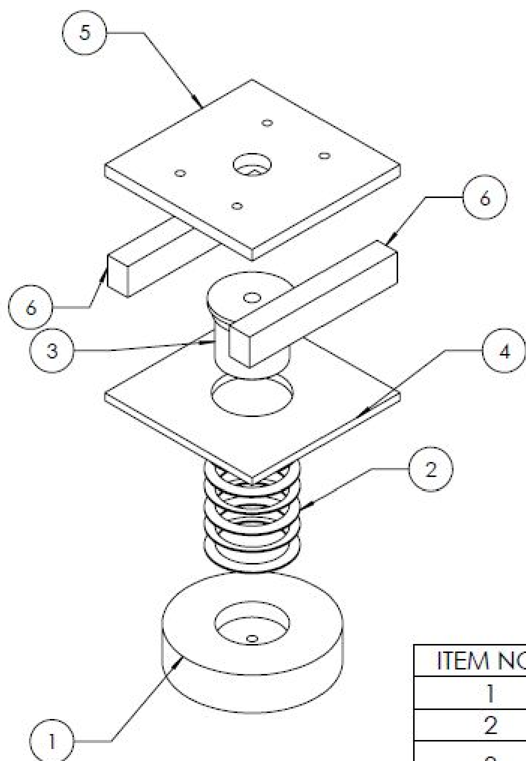
REVISIONS: CHANGED TOLERANCE 3/13/13 HN
UPDATED GDT AND TOLERANCE 3/14/13 HN
UPDATED TOLERANCE 3/19/13 HN

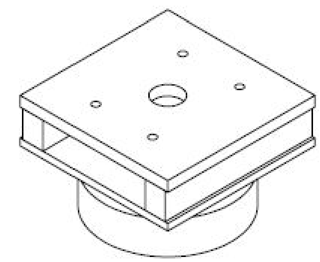


DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 2007	UNITS: INCHES	TITLE: TOP WEDGE	
DWG #: 1005	SCALE: 1:4	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

REVISIONS: ADDED WELDING INSTRUCTIONS 3/7/13 HN
CHANGED GDT AND TOLERANCE 3/13/13 HN
UPDATED WELD INFORMATION 4/18/13 HN



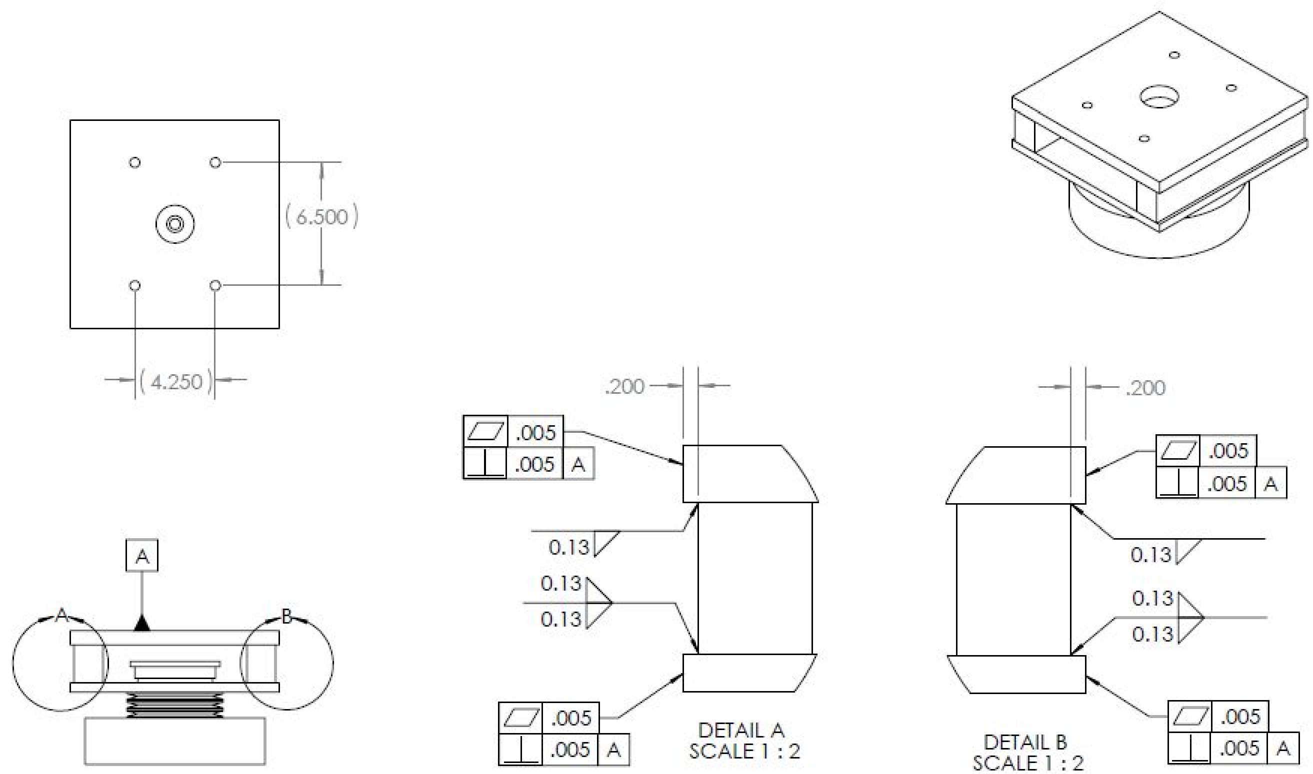


THE PARTS WASHER SPRING (1037343) AND CYLINDER GUIDE (1036343-2) ARE STOCK PARTS.

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	1008	SPRING BASE PAD	1
2	1037343	WASHER SPRING	6
3	1036343-2	CYLINDER GUIDE	1
4	1018	BOTTOM PLATE	1
5	1015	TOP PLATE	1
6	1017	PILLAR	2

TURBOTECH	DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
	DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL:	
	NEXT ASSY: 3000	UNITS: INCHES	TITLE: SPRING PACK ASSEMBLY	
	DWG #: 2006	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

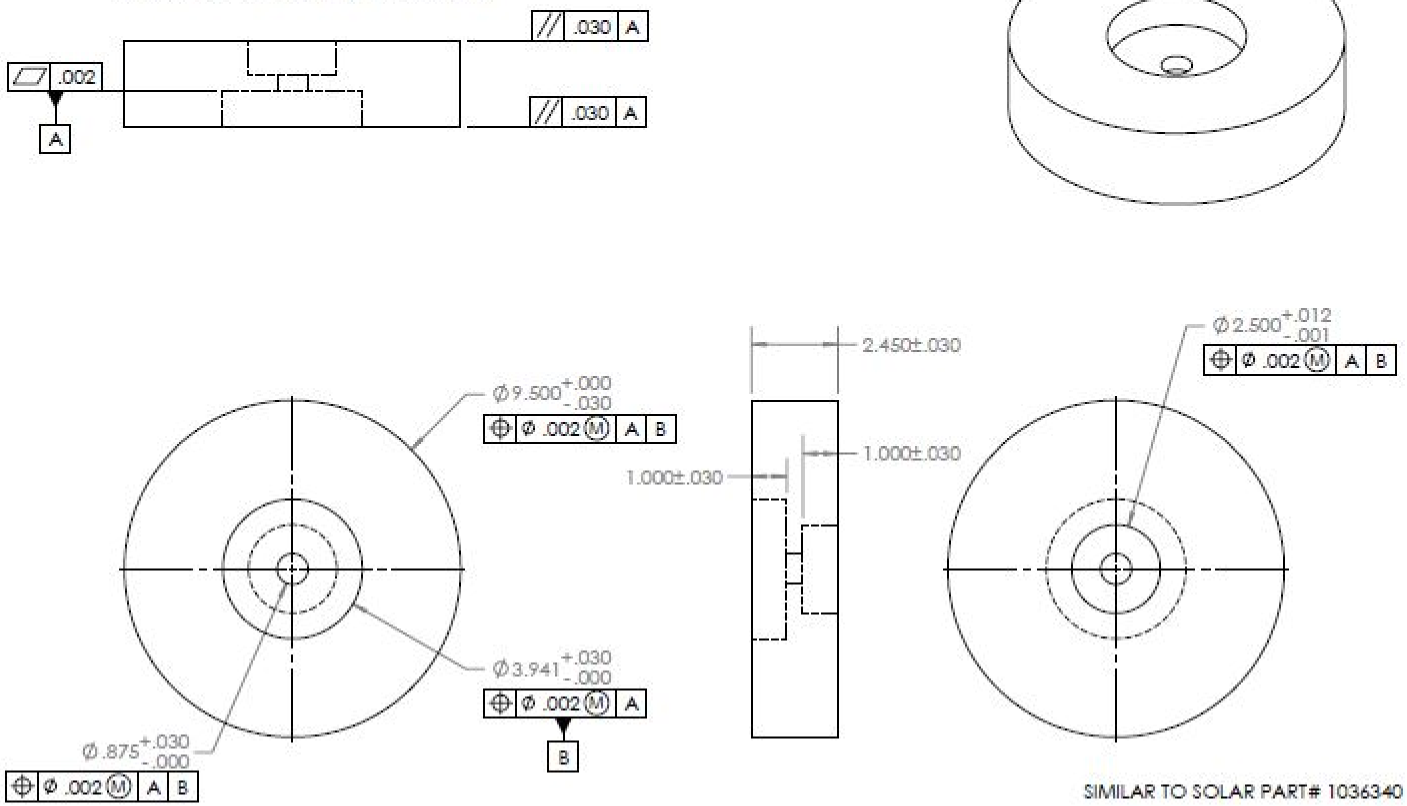


THE ASSEMBLY PROCESS IS MODIFIED
FROM PART # 1037548-1

DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 3/7/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 3000	UNITS: INCHES	TITLE: SPRING PACK ASSEMBLY	
DWG #: 2006	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

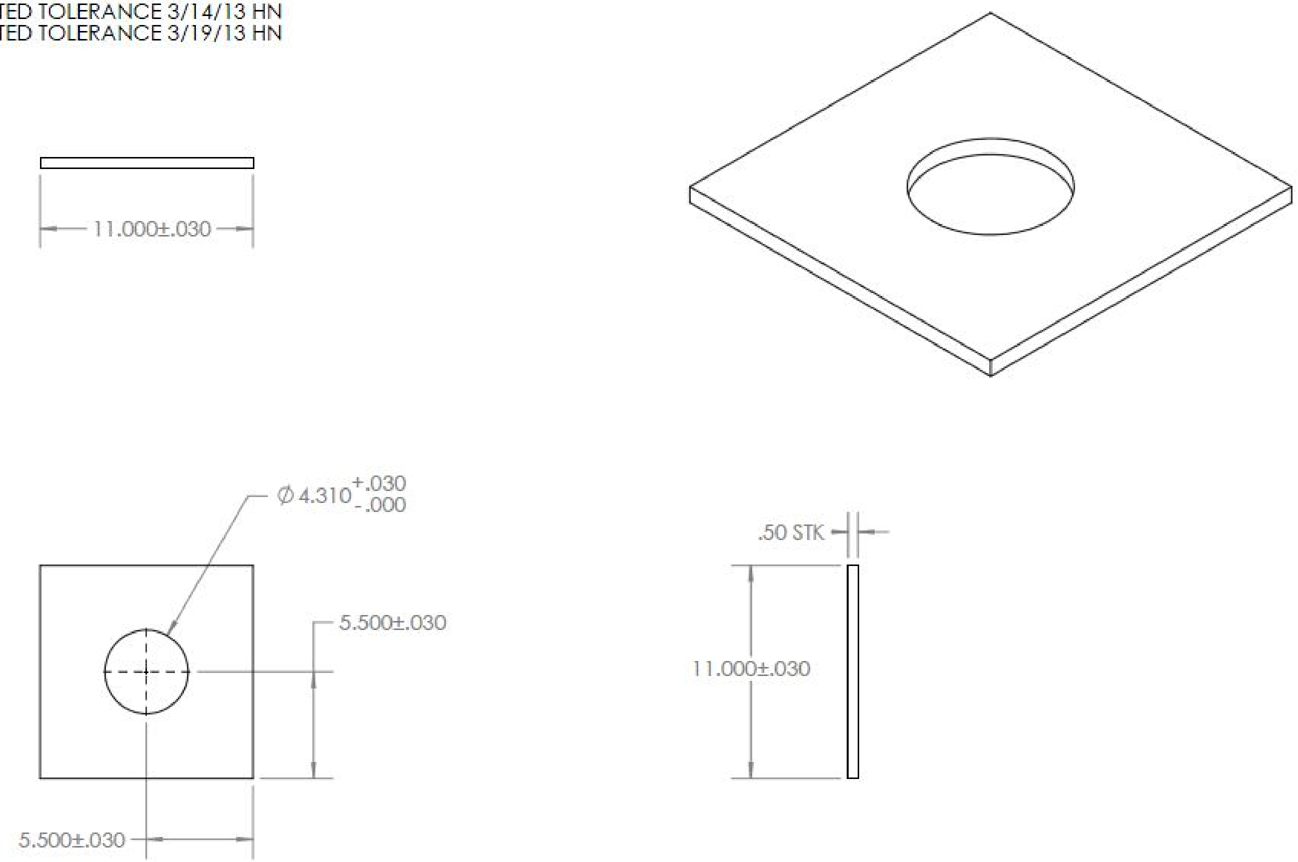
REVISIONS: CHANGED GDT AND TOLERANCE 3/13/13 HN
UPDATED GDT AND TOLERANCE 3/14/13 HN
UPDATED TOLERANCE 3/19/13 HN
CHANGED HOLE DIAMETERS 5/27/13 HN



DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 2006	UNITS: INCHES	TITLE: SPRING BASE PAD	
DWG #: 1008	SCALE: 1:4	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

REVISIONS: CHANGED GDT AND TOLERANCE 3/13/13 HN
UPDATED TOLERANCE 3/14/13 HN
UPDATED TOLERANCE 3/19/13 HN

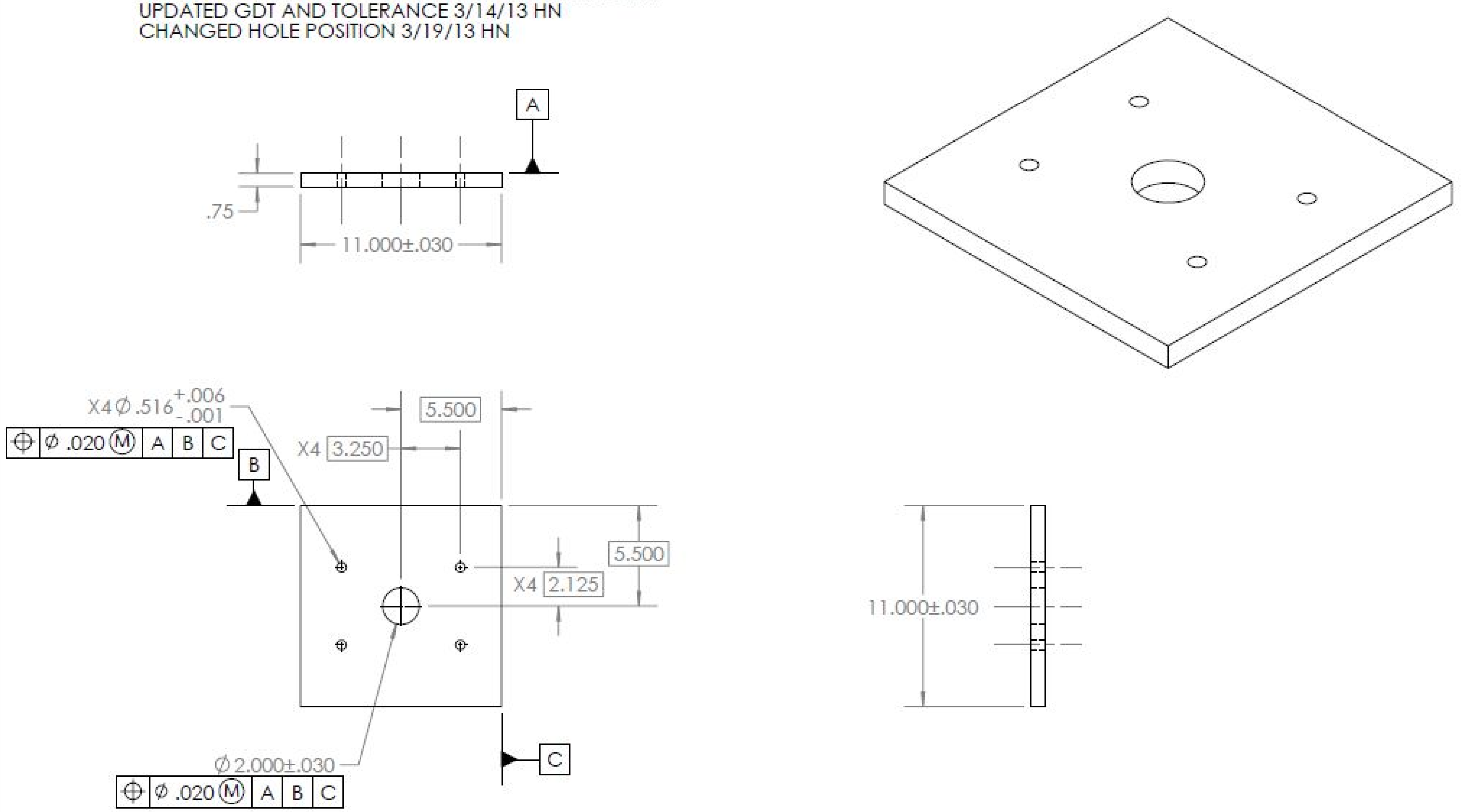


SIMILAR TO SOLAR PART# 1037548

DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 2006	UNITS: INCHES	TITLE: BOTTOM PLATE	
DWG #: 1018	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

REVISION: CHANGED GDT AND TOLERANCE 3/13/13 HN
CHANGED TAPPED HOLES TO THRU HOLES 3/13/13 HN
UPDATED GDT AND TOLERANCE 3/14/13 HN
CHANGED HOLE POSITION 3/19/13 HN



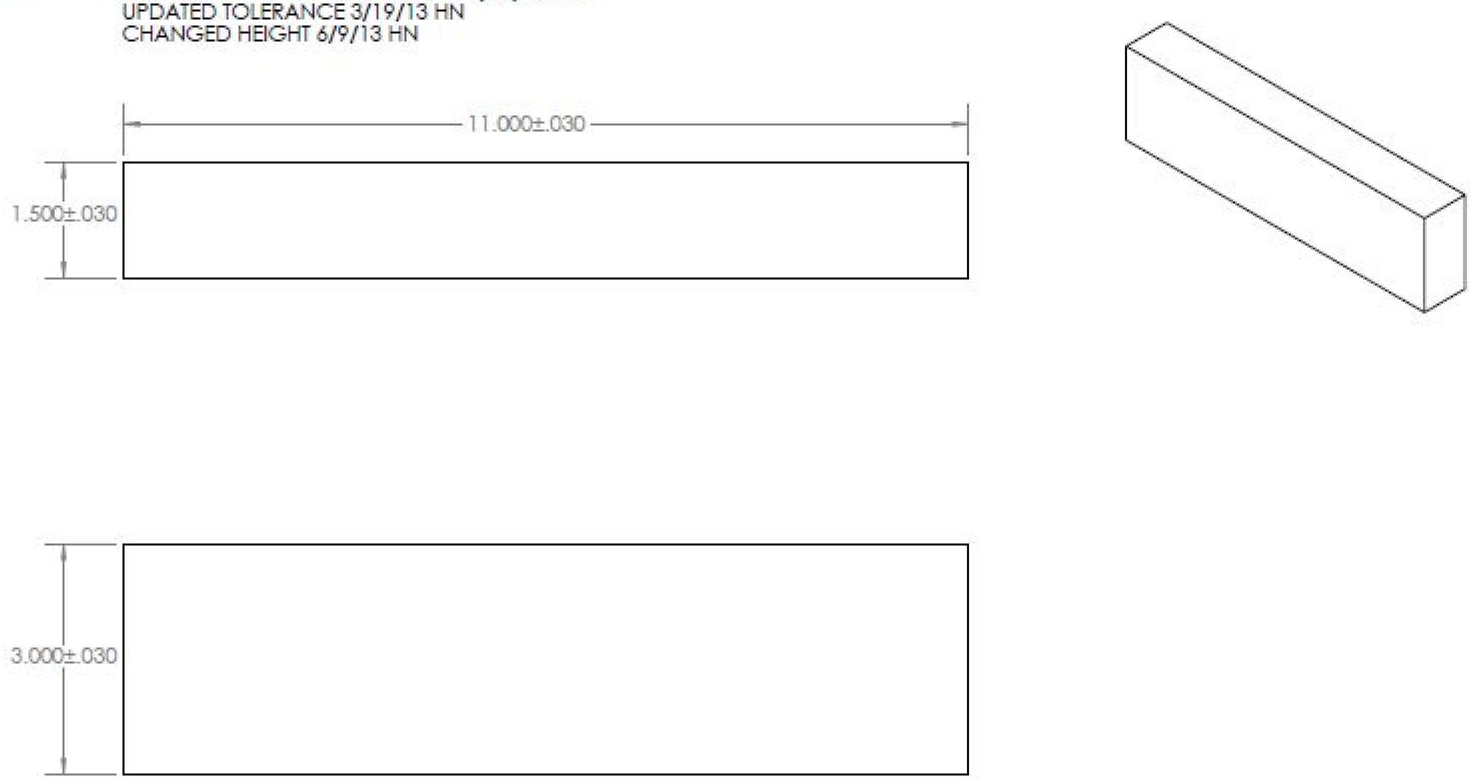
SIMILAR TO SOLAR PART# 1037548

TURBOTECH

DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 2006	UNITS: INCHES	TITLE: TOP PLATE	
DWG #: 1015	SCALE: 1:8	CP Senior Project - Engine Aft-Mount	

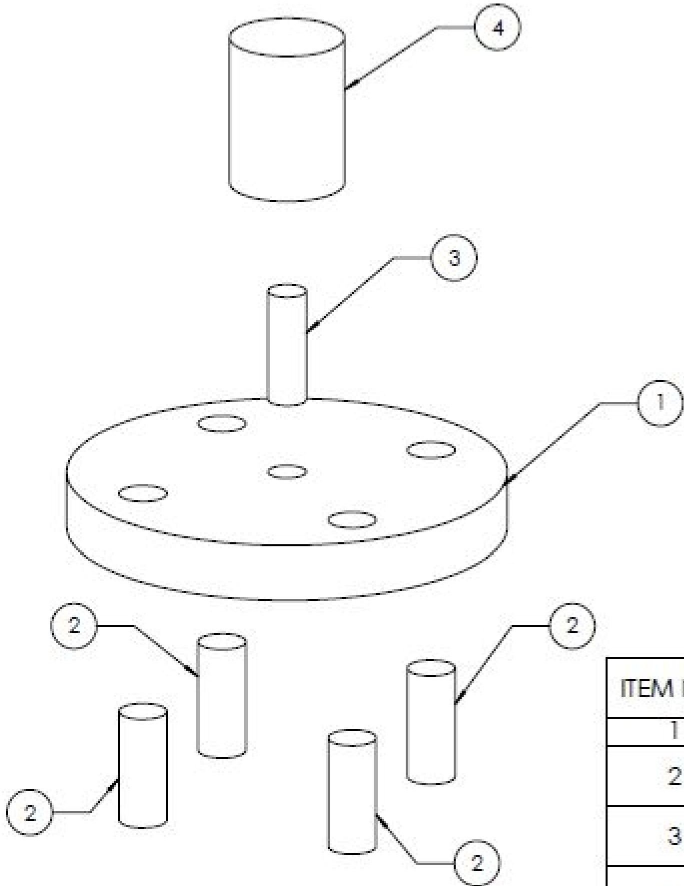
5 4 3 2 1

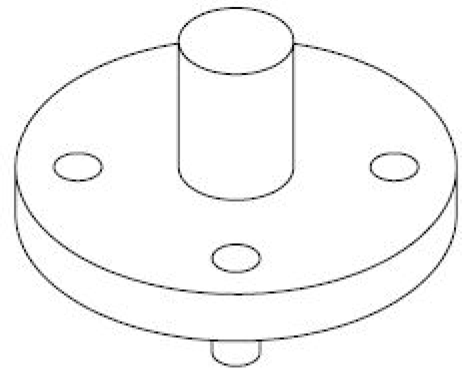
REVISIONS: CHANGED LENGTH 3/10/13 HN
CHANGED GDT AND TOLERANCE 3/13/13 HN
UPDATED TOLERANCE 3/19/13 HN
CHANGED HEIGHT 6/9/13 HN



DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 2/27/2013	TOLERANCE: ±0.06	MATERIAL: STEEL	
NEXT ASSY: 2006	UNITS: INCHES	TITLE: PILLAR	
DWG #: 1017	SCALE: 1:2	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1



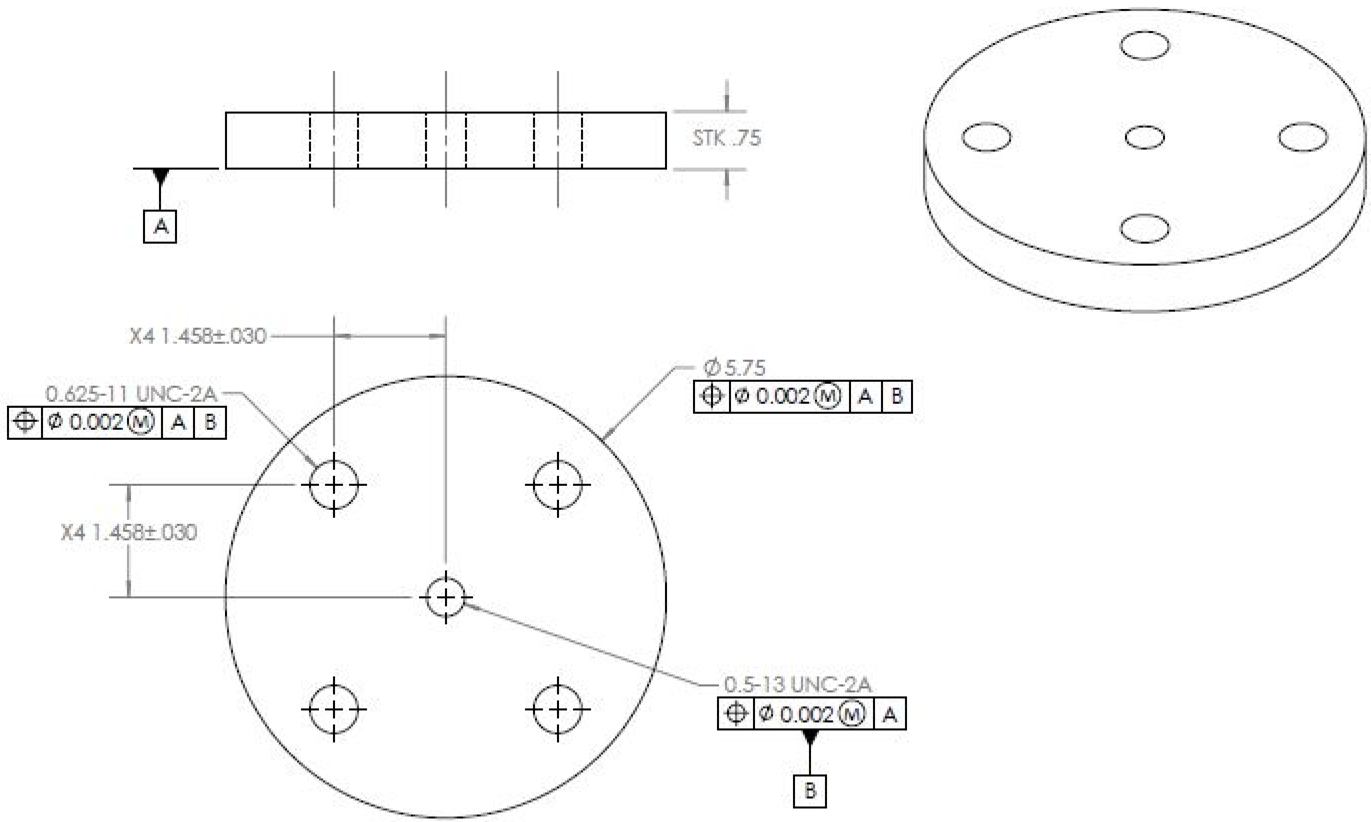


ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	1020	Load cell interface plate	1
2	Threaded Rod	0.625"-11 x 1.5"	4
3	Threaded Rod	0.5"-13 x 1.5"	1
4	Threaded Rod	1.5"-12 x 2"	1

TURBOTECH

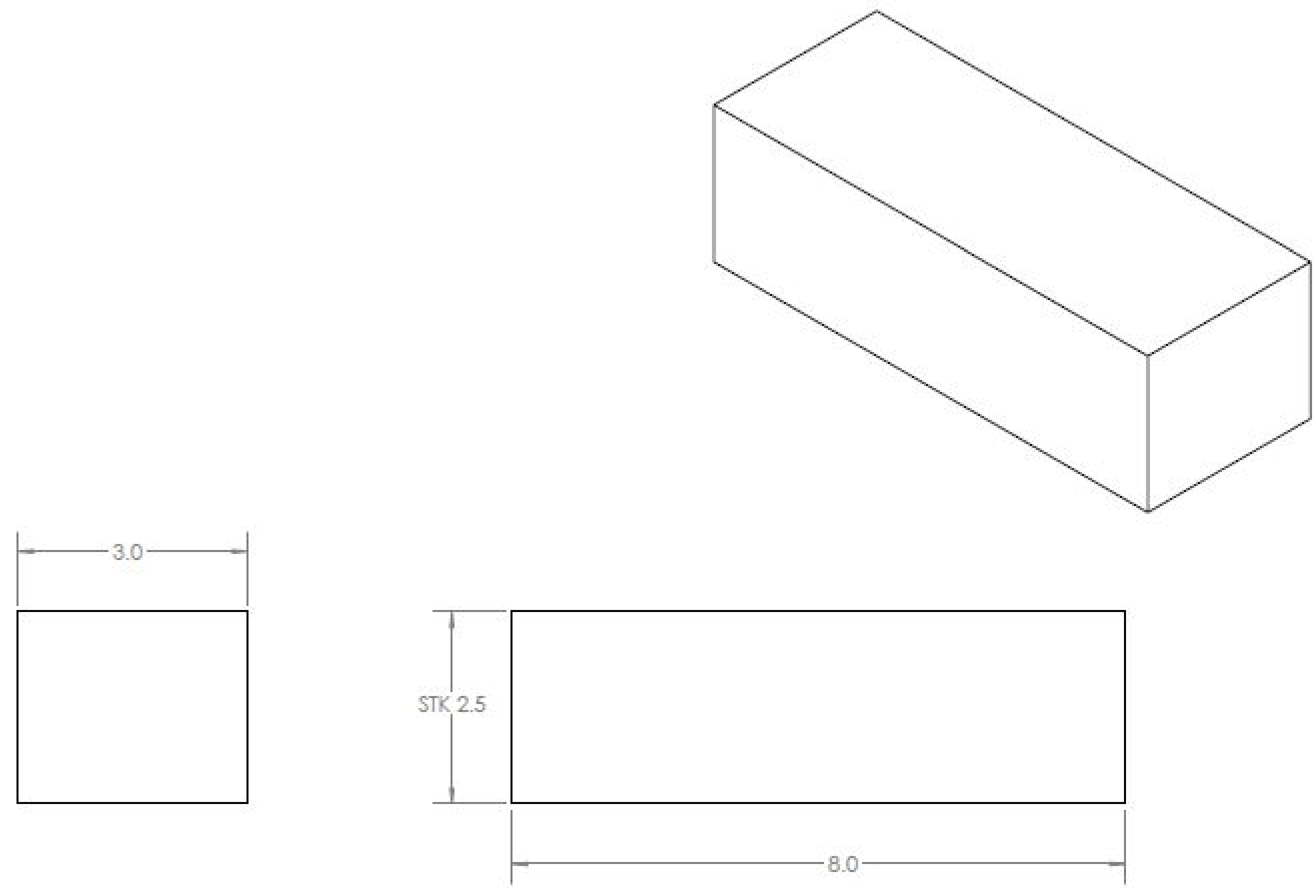
DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
DATE: 5/27/2013	TOLERANCE:	MATERIAL: STEEL	
NEXT ASSY:	UNITS: INCHES	TITLE: LOAD CELL INTERFACE PLATE ASSEMBLY	
DWG #: 2010	SCALE: 1:2	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1



TURBOTECH	DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
	DATE: 2/27/2013	TOLERANCE: ±0.060	MATERIAL: STEEL	
	NEXT ASSY: 2010	UNITS: INCHES	TITLE: LOAD CELL INTERFACE PLATE	
	DWG #: 1020	SCALE: 1:2	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1



TURBOTECH	DRAWN BY: HARRISON NEWELL	INIT:	CKD BY: MICHAEL HAWORTH	INIT:
	DATE: 2/27/2013	TOLERANCE: ±0.1	MATERIAL: STEEL	
	NEXT ASSY:	UNITS: INCHES	TITLE: JACK SUPPORT	
	DWG #: 1021	SCALE: 1:2	CP Senior Project - Engine Aft-Mount	

5 4 3 2 1

Appendix D: Vendor Information

Power Jack: Joyce/Dayton Corp.
www.joycedayton.com
3300 South Dixie Drive
Kettering, OH 45439
800.523.5204

Mark Burleson
mburleson@joycedayton.com
Application Engineer

Patty Deppen
pdeppen@joycedayton.com
Sales Associate

Rail Slider: Minarik
www.minarik.com
9530 Chesapeake Drive #501
San Diego, CA 92123

Adam Reay
adam.reay@minarik.com
Sales Engineer

Catherine McHugh
Catherine.mchugh@minarik.com
Customer Service Representative

Fasteners: Fastenal
www.fastenal.com
2001 Theurer Blvd.
Winona, MN USA 55987
877.507.7555

Phillip Albrecht
phalbrec@fastenal.com
Sales Representative
507.313.7417

Appendix E: Vendor Component Specifications

Power Jack

The power jack ordered from Joyce Dayton has a 10 ton capacity worm gear machine screw. A 25:1 gear ratio provides precision positioning of 0.01" per revolution of the input shaft. The jack is upright and keyed allowing for the pad to rise and lower without rotation. The screw has 2 inches of rise and a load pad end condition. Special features include extended length input shafts to give the operators better access to the jack. There is an 8 inch hand wheel on one end and a hex input on the other which allows for manual operation of the jack or automatic with the use of a wrench or pneumatic wrench. The specification sheets and the quote for the system are below.

Table E1. Specification summary for the jack that will be used in the aft-mount redesign..

MODEL	CPTY	SCRW DIAM (INCH)	THRD PITCH/ LEAD	WORM GEAR RATIO	WORM SHAFT TURNS FOR 1" TRAVEL	TARE TORQUE (INCH LBS.)	STARTING TORQUE (INCH LBS.)	OPERATING TORQUE (INCH LBS.)	EFFICIENCY RATING % APPROX.	SCREW TORQUE (INCH LBS.)	BASIC JACK WEIGHT (LBS).	JACK WEIGHT PER INCH TRAVEL (LBS.)
WJ2510	10 ton	2	.250 pitch ACME 2C	25:1	100	20	.024W*	.014W* @ 200 RPM	11.3	.161W*	43	1.3

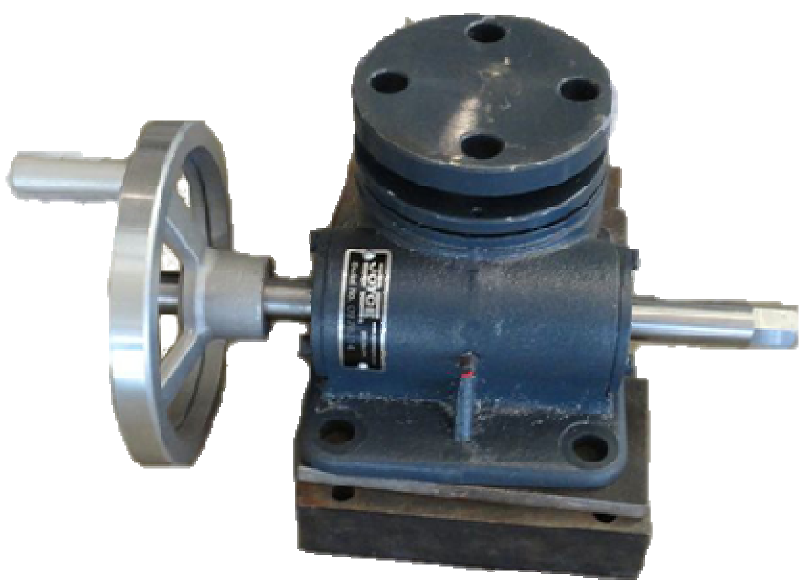


Figure E1. Power jack as delivered from Joyce Dayton.

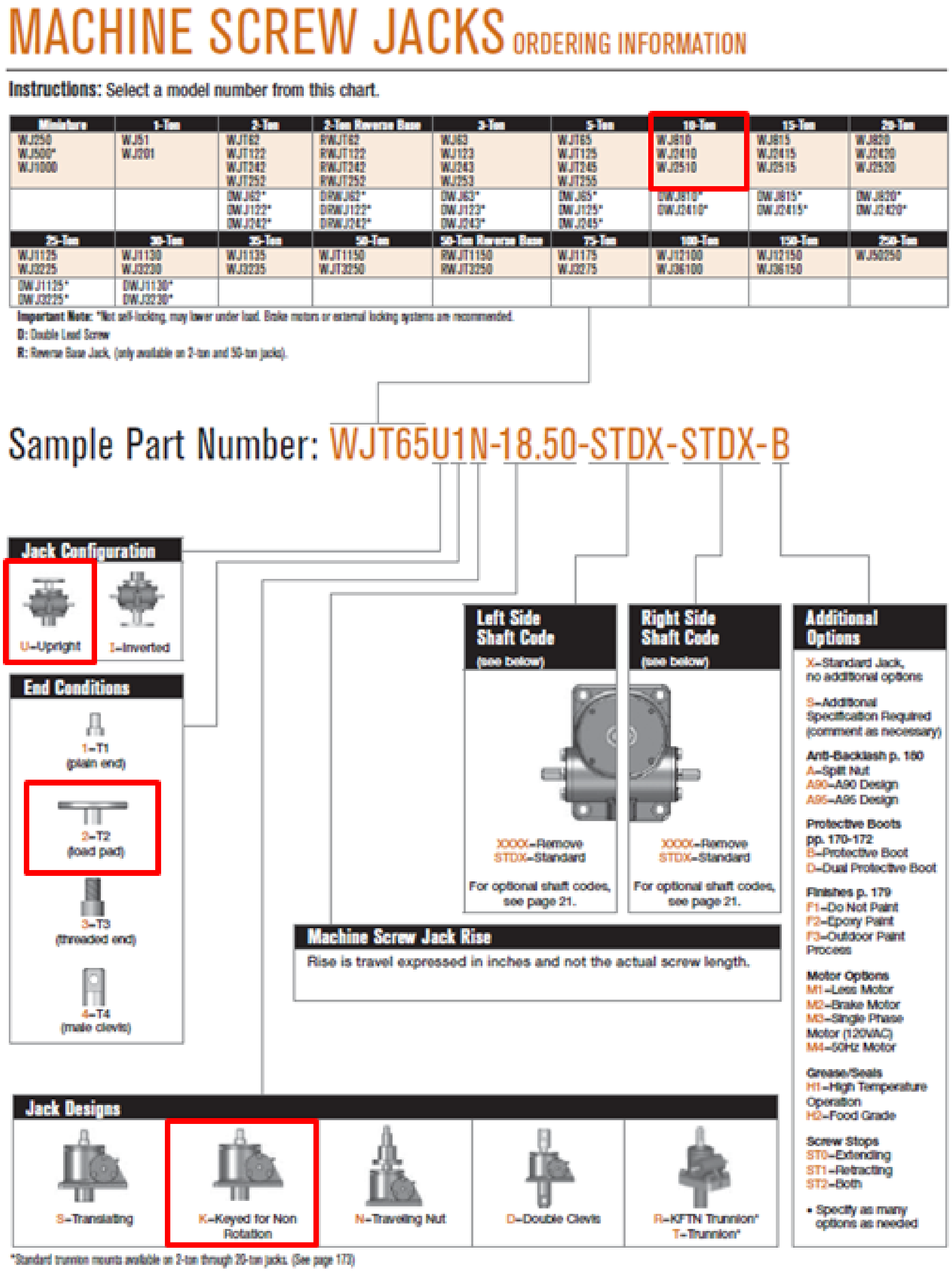


Figure E2. List of available options for Joyce Dayton power jacks. The jack used for the aft-mount redesign is upright, keyed, has a T2 load pad end condition, and has an 8" hand wheel on the left hand input shaft and a hex cut on the right input shaft.

JOYCE

Joyce/Dayton Corp.
PO Box 1630
Dayton, OH 45401
Tel: 937-294-6261
800-523-5204
Fax: 937-297-7173

Your local representative:
Duncan MacDonald
Engineered Industrial Products, Inc.
16604 Edwards Rd.
Cerritos, CA 90703

Tel: 951-907-3727
Fax: 562-404-8060
dmacdonald@eip-inc.com

3/13/2013
Screw Jack Quotation #99010-90404-4

Prepared for...

Ismael DePaz		Telephone:	8587152077
Solar Turbines		Fax:	
9330 Sky Park Ct		E-Mail:	depaz_ismael_a@solarturbines.com
San Diego, CA 92123-4304			

Please forward all purchase orders directly to sales@joycedayton.com or fax to 937-297-7173, referencing quotation number 99010-90404-4.

Quantity	Description	Net Price Each	Extended Net Price	Lead Time** (business days)
1	WJ2510U2K-2-HW08-STDX-S The selected jack is a 10 Ton capacity* worm gear machine screw jack, 25:1 ratio gearset, Upright, Keyed, T2 (load pad) end condition. This jack has 2 inches of screw travel (rise). A 8" diameter hand wheel is on the left hand input shaft. The right hand input shaft is standard. This jack has the following additions/modifications: *Right Hand Worm - Hex *Left HandWorm - Extension to 2X Standard *Right HandWorm - Extension to 2X Standard	$2,132.80	$2,132.80	20
1	Prepaid Freight Charges Prepaid freight charges for shipment via UPS Ground for quantity 1of: WJ2510U2K-2-HW08-STDX-S	$84.00	$84.00	5

Figure E3. Final quote for the power jack used in the aft-mount redesign.

Rail Slider

The rail slider used in the aft-mount redesign is a plain bearing linear slide produced by PBC linear. Minarik Controls and Automation, a San Diego based distributor, is the company that will be providing the system for this project. The low profile linear slider is capable of supporting up to 18,750lbs and will begin to slide after about 2,400lbs of load is applied along the direction of the rails. The specification sheets and the quote for the system are below.

Table E2. The following specification gives the rail size based upon loading characteristics of the system. Our design calls for the size 32 rail which is approximately 2” diameter. The right half of the table is the manufacturer’s suggested center of gravity for the load placed on the slider. For this application, we will be ignoring these specifications.

RECOMMENDED SAFE LOADING*

SIZE	F MAX. (lbs.)	F MAX. (N)	X (in.)	Y (in.)	Z (in.)
08	1450	2402	4.00	2.37	3.00
10	2200	3381	4.75	2.76	3.50
12	2850	3737	5.00	2.85	4.00
16	5275	4671	5.50	3.37	4.50
20	7750	7784	6.75	4.05	5.50
24	10600	9341	7.86	4.90	6.50
32	18750	14679	10.75	6.00	9.00

*Load ratings for Simplicity plane bearings only.

Table E3. List of options available from PBC linear. The slider used for the aft-mount redesign will be made from alloy steel, have Freelon bearings, and be 24” in length.

PART NUMBER

SERIES				SIZE			L		LEAD
2LRPS	X	XXX	-	08	XX	-	XXX.XXX	-	YYY
2LRPS	X	XXX	-	10	XX	-	XXX.XXX	-	YYY
2LRPS	X	XXX	-	12	XX	-	XXX.XXX	-	YYY
2LRPS	X	XXX	-	16	XX	-	XXX.XXX	-	YYY
2LRPS	X	XXX	-	20	XX	-	XXX.XXX	-	YYY
2LRPS	X	XXX	-	24	XX	-	XXX.XXX	-	YYY
2LRPS	X	XXX	-	32	XX	-	XXX.XXX	-	YYY

MATERIAL:
Aluminum Alloy - Top Plate, Bottom Plate, Rail Supports, Pillow Blocks
Shafting - Alloy Steel, Chrome Plated, 440 or 303 Stainless Steel

ORDERING EXAMPLE:
To order a slide with a 0.625 diameter alloy steel shaft, 28" rail length, .200" right hand select ball screw - specify part number, 2LRPS-10-028.000-AR1

Substitute standard or select lead code from table for “YYY”

Enter rail length in inches XXX.XXX (EX: 28” = 028.000)

Shafting material: **00** = Alloy Steel (Standard) **CC** = Ceramic Coated **CR** = Chrome Plated
SS = 440 Stainless Steel **XX** = 303 Stainless Steel
NOTE: Ball bearings not suitable with stainless steel or chrome plate shafting

Shaft diameter dimension in 1/16" increments

Internal Lubrication: **Blank** = None (Standard) **JKM** = Thru hole and internal felt wick

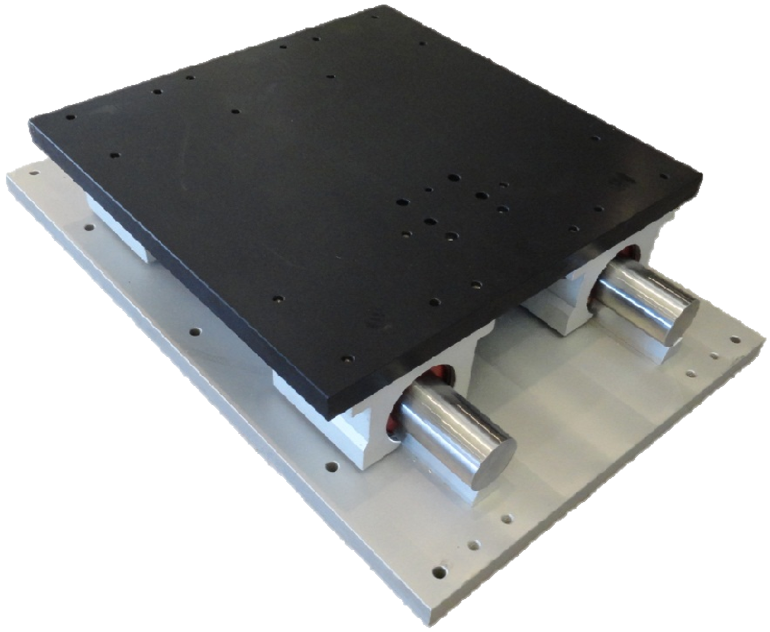


Figure E4. Rail sliders as delivered from Minarik Controls and Automation.

Minarik Automation & Control

Minarik, Anaheim
4975 East Landon Dr.
Anaheim, CA 92807
714-696-3750
Fax: 714-696-3760

Quotation: 0201828

Attn: Mr. Michael Haworth

Cal Poly State University
San Luis Obispo CA 93407
Phone: 951-505-0076

Account ID: A804282DD
Terms: 0%/,Net 30 days

Effective Date: 02/28/2013
Expire Date: 03/30/2013
Prepared By: Catherine McHugh
Catherine.McHugh@minarik.com

Sales Person: Adam Reay

Line	PART Number	Description	Lead Time Days	Qty	Unit Price	Extension
1	LRPS-3200-024.000	PBC Low Profile Linear Slides Rail Mounted & Plate Supported Slide Assembly Alloy Steel Shaft, 2 Inch Diameter, 24 Inch Length Approx. 15 working day factory lead time plus shipping Non-Cancelable/Non-Returnable		1	2,012.58	2,012.58
2	Freight	Estimated ground shipping to 92123		1	120.00	120.00
3	Tax	Estimated sales tax		1	170.00	170.00
					Total in USD :	**$2,302.58**

Figure E5. Final quote for the power jack used in the aft-mount redesign.

Fasteners

The bolts used in the aft-mount redesign are yellow zinc coated grade 8 carbon steel. The nuts and washers are fashioned from the same material type as each of the bolts. There are six 1”-14 UNF bolts in the wedges and four ½”-20 UNF bolts that attach the engine bracket to the mount. Each bolt will have two washers and a single nut to fasten everything together.

Order Header

Order Status : Received at Store
Customer Account No : ECOMMCASH
Cust. PO :
Release/Job No :
Creation Date : 2013-04-18
Confirmation Number : WZNTJJWQ42G

Servicing Store

Store Code: **ECOMM**
2001 Theurer Blvd.
Winona, MN USA 55987
P:1-877-507-7555
ECOMM@stores.fastenal.com

Shipping Address

Turbo Tech
1 Grand Avenue
Attention: Woody Newell
San Luis Obispo, CA USA 93407

Billing Address

Turbo Tech
1 Grand Avenue
ATTN: Woody Newell
San Luis Obispo, CA USA 93407

Payment Method

Visa
Card Number: 4246-XXXX-XXXX-0940
Expiration Date: 07/2013

Order Comments

Preferred shipping method: UPS Ground Service;

Line No.	Qty	Description	Qty/Pkg	Ext Qty	Price	Total
1	6	1"-14 x 3" Yellow Zinc Finish SAE J429 Grade 8 Hex Cap Screw Made In USA 0152348 Status: Received at Store	1	6	$11.50	$69.00
2	6	1"-14 Yellow Zinc Finish Grade 8 Finished Hex Nut 36470 Status: Received at Store	1	6	$3.39	$20.34
3	12	1" Grade 9 Yellow Zinc Finish USS Flat Washer 0160892 Status: Received at Store	1	12	$3.42	$41.04
4	4	1/2"-20 Yellow Zinc Finish Grade 8 Finished Hex Nut - Made In USA 0147917 Status: Received at Store	1	4	$0.6486	$2.59
5	4	1/2"-20 x 2-3/4" Yellow Zinc Finish SAE J429 Grade 8 Hex Cap Screw Made In USA 0152494 Status: Received at Store	1	4	$1.83	$7.32
					Subtotal	**$140.29**
					Estimated Freight Cost (UPSG)	**$14.25**
					State Tax	**$10.03**
					County Tax	**$1.55**
					City Tax	**$0.77**
					Total (USD)	**$166.89**

Figure E6. Final quote for the fasteners used in the aft-mount redesign.

Appendix F: Supporting Analysis

Wedge Angle Selection

The angle that was chosen for the wedge was 15°. This decision was made based on an iterative design process that took into consideration the length of the wedge, the overall height change of the mount, and the force required to break the static friction of the mount if there were no bolts holding the system in place. The calculations used for the wedge were subsequently used to determine the amount of pre-load that would be necessary to apply to the bolts holding the wedges in place. The following calculations show the final iteration of the wedge angle as 15°. These calculations were repeated several times before deciding on the optimal angle for the wedge.

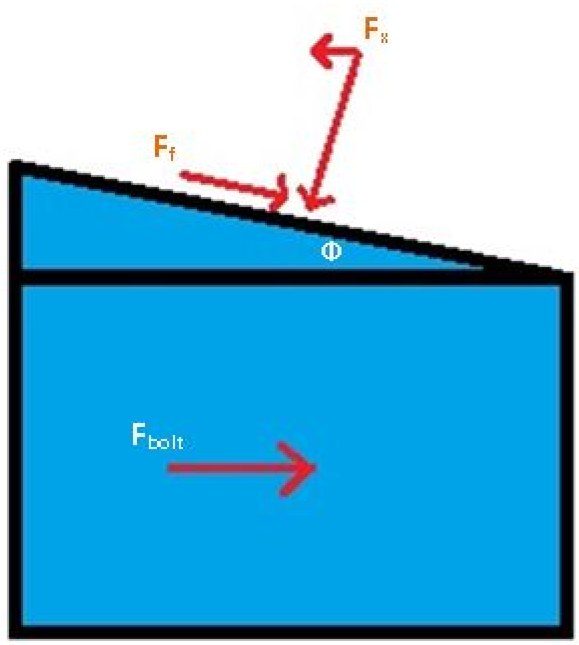


Figure F1. This figure shows how the wedge angle affects the amount of load the bolts will have to resist.

First it is necessary to find the friction force between the top and bottom wedge:

$$F_f = 18{,}000 \times \mu_f$$

$$F_f = 18{,}000 \times 0.10$$

$$F_f = 1{,}800\ lb_f$$

Next, using the wedge angle, the components of the force in the x-direction that determine the pre-load in the bolt can be calculated. First there is the x component of the overall load:

$$F_x = 18{,}000\sin(\phi)$$

$$F_x = 18{,}000\sin(15°)$$

$$F_x = 4{,}658\ lb_f$$

Then, the opposing friction force between the plates in the x-direction:

$$F_{fx} = F_f\cos(\phi)$$

$$F_{fx} = 1{,}800\cos(15°)$$

$$F_{fx} = 1{,}738\ lb_f$$

The load that the bolts will have to resist comes from the difference between these two x-component values:

$$F_{bolt} = F_x - F_{fx}$$

$$F_{bolt} = 4{,}658 - 1738$$

$$F_{bolt} = 2920\ lb_f$$

As the angle of the wedge is changed, there are several components of the design that are affected. First of all, it will affect the change in height of the mount. The height change capacity will increase as the angle is increased. However, as the angle is increased it will also increase the x-component of force that the engine has on the bottom wedge. Therefore, keeping this angle as shallow as possible will decrease the overall force that the bolts in the wedge will have to resist. This is why the wedge angle was eventually decided on to be 15°.

Bolt Selection

The bolts that were chosen for this design are 1-8 UNC grade 8 hex head bolts. This size bolt was chosen based on several design considerations when determining the expected loading conditions that the aft-mount is expected to sustain. One such consideration is the static weight of the engine. Depending on the engine type, the weight is approximately either 13,000 or 15,500 pounds. Also, during operation, there is some dynamic loading that occurs due to the gyroscopic moment incurred from the spinning shaft of the turbine. This adds a dynamic load factor to the overall calculation of the bolt diameter. As, seen in Figure B4, the vibration at operating speed is 0.1g. From this information, it was determined that a dynamic load factor of 1500lb_f, would be sufficient for the load calculations. Finally, it is important to add a factor of safety in the weight. With all of these considerations in mind, the bolt calculations were performed with a conservative assumption that the maximum weight that the mount will have to withstand is 18,000 pounds of force. The following equations were used in the final determination for the bolt diameter and torque setting.

First, the pre-load force on the top plate bolts is calculated. The total force of the engine is divided up between the 4 bolts in the top plate assembly:

$$F_{PL} = \frac{18{,}000}{(\#bolts)\mu_f}$$

$$F_{PL} = \frac{18{,}000}{4 \times 0.1}$$

$$F_{PL} = 45{,}000\ lb_f$$

From this pre-load value, the necessary torque value can be calculated. This is based on a K factor (coating of the bolts) that is assumed to be a zinc coating. This gives a K value of 0.2:

$$T = KD_{bolt}F_{PL}$$

$$T = 0.2 \times 1.0\ in\left(\frac{1ft}{12in}\right) \times 45{,}000lb_f$$

$$T = 750\ ft - lb_f$$

Once the diameter of the bolt was determined, it was important to determine the grade of the bolts used. This was done based on the shear and bending stresses in the bolts before the load of the engine is applied:

$$\tau = \frac{T}{JG}$$

$$J = \frac{\pi D_{bolt}^4}{32}$$

$$J = \frac{\pi \times (1\ in)^4}{32}$$

$$J = 0.0982\ in^4$$

$$\tau = \frac{9000\ in - lb_f}{(0.0982\ in^4) \times (1.15 \times 10^7)}$$

$$\tau = 0.008 psi$$

$$\sigma = \frac{F_{PL}}{A}$$

$$\sigma = \frac{45{,}000 lb_f}{0.785 in^2}$$

$$\sigma = 57{,}295\ psi$$

$$\sigma_e = \sqrt{\tau^2 + \sigma^2}$$

$$\sigma_e = \sqrt{0.008^2 + 57295^2}$$

$$\sigma_e = 57{,}295 psi$$

The same calculations were done for after the load of the engine is applied:

$$\tau = \frac{V}{A}$$

$$\tau = \frac{18{,}000\ lb_f}{4 * 0.785\ in^2}$$

$$\tau = 5{,}729 psi$$

$$\sigma = \frac{F_{PL}}{A}$$

$$\sigma = \frac{45{,}000 lb_f}{0.785 in^2}$$

$$\sigma = 57{,}295\ psi$$

$$\sigma_e = \sqrt{\tau^2 + \sigma^2}$$

$$\sigma_e = \sqrt{5{,}729^2 + 57295^2}$$

$$\sigma_e = 57{,}581 psi$$

These stress values are well below the yield stress of a grade 8 bolt which has a yield strength of 130ksi. Next it was necessary to do the same calculations for the bottom wedge bolts to find the best bolt diameter size to hold them in place. The force acting on the bolt was determined through the wedge angle selection:

$$F_{bolt} = 2{,}920 lb_f$$

$$F_{PL} = \frac{F_{bolt}}{(\#bolts) \times \mu_f}$$

$$F_{PL} = \frac{2{,}920 lb_f}{2 \times 0.1}$$

$$F_{PL} = 14{,}600\ lb_f$$

From this pre-load value, the necessary torque value can be calculated. This is based on a K factor (coating of the bolts) that is assumed to be a zinc coating. This gives a K value of 0.2:

$$T = KD_{bolt}F_{PL}$$

$$T = 0.2 \times 1.0\ in \left(\frac{1ft}{12in}\right) \times 14{,}600 lb_f$$

$$T = 243\ ft - lb_f$$

The stress in the bolts for the bottom wedge also needs to be verified that it is below the yield stress of the grade of bolt decided on. It was determined that keeping a uniform bolt throughout the entire mount would be ideal and result in minimal confusion during the assembly process. The stress calculation is as follows

$$\tau = \frac{T}{JG}$$

$$J = \frac{\pi D_{bolt}^4}{32}$$

$$J = \frac{\pi \times (1\ in)^4}{32}$$

$$J = 0.0982\ in^4$$

$$\tau = \frac{2{,}920\ in - lb_f}{(0.0982\ in^4) \times (1.15 \times 10^7)}$$

$$\tau = 0.003psi$$

$$\sigma = \frac{F_{PL}}{A}$$

$$\sigma = \frac{14{,}600lb_f}{0.785in^2}$$

$$\sigma = 18{,}589\ psi$$

$$\sigma_e = \sqrt{\tau^2 + \sigma^2}$$

$$\sigma_e = \sqrt{0.003^2 + 18{,}589^2}$$

$$\sigma_e = 18{,}589psi$$

The same calculations were done for after the load of the engine is applied:

$$\tau = \frac{V}{A}$$

$$\tau = \frac{2{,}920\ lb_f}{2 * 0.785\ in^2}$$

$$\tau = 1{,}858psi$$

$$\sigma = \frac{F_{PL}}{A}$$

$$\sigma = \frac{14{,}600lb_f}{0.785in^2}$$

$$\sigma = 18{,}682\ psi$$

$$\sigma_e = \sqrt{\tau^2 + \sigma^2}$$

$$\sigma_e = \sqrt{1{,}858^2 + 18{,}589^2}$$

$$\sigma_e = 18{,}683psi$$

These values are significantly below the yield stress of a grade 8 bolt. For this reason, 1-12 UNF grade 8 bolts have been chosen to hold the top and bottom wedges in place.

Deflection Analysis

The first design requirement for this project is to be able to accurately set the alignment of the engine using the aft-mount. During the initial placement of the engine, the power jack is supporting the base of the spring pack. Once alignment is set and the bolts secured, the jack is removed under the spring pack. The following calculation was used to determine how much deflection would occur after the mount has been removed.

As can be seen below, the plate that the spring pack sits on is 1 ¼" thick carbon steel with a ¼" recession for the spring pack to sit in. For the purpose of this calculation, the minimum material thickness of one inch will be analyzed. The alignment of the engine is set to a tolerance of ±0.01" so a deflection of less than ten thousands is acceptable for this worst case scenario.

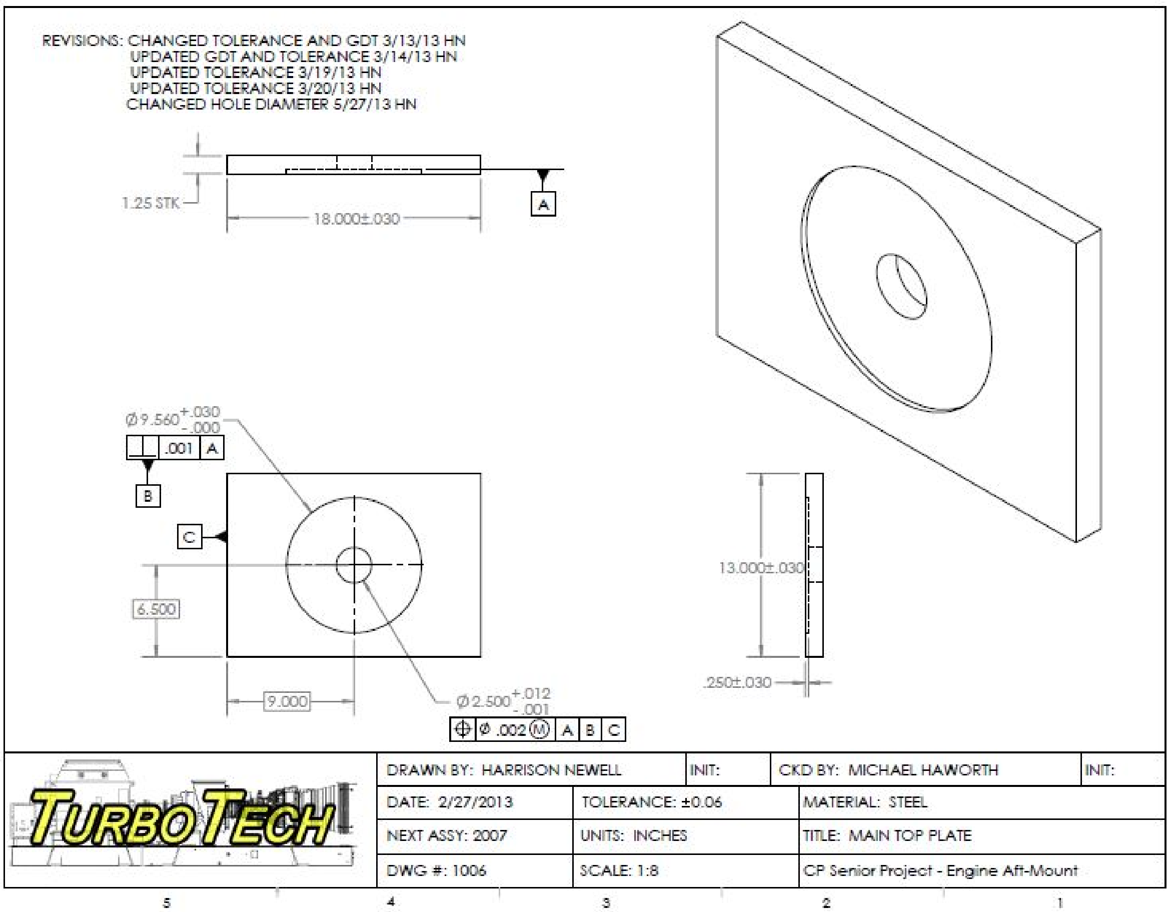


Figure F2. The spring pack sits in the recession of the part which is supported on two sides. The area of interest is the deflection along the long side of the plate.

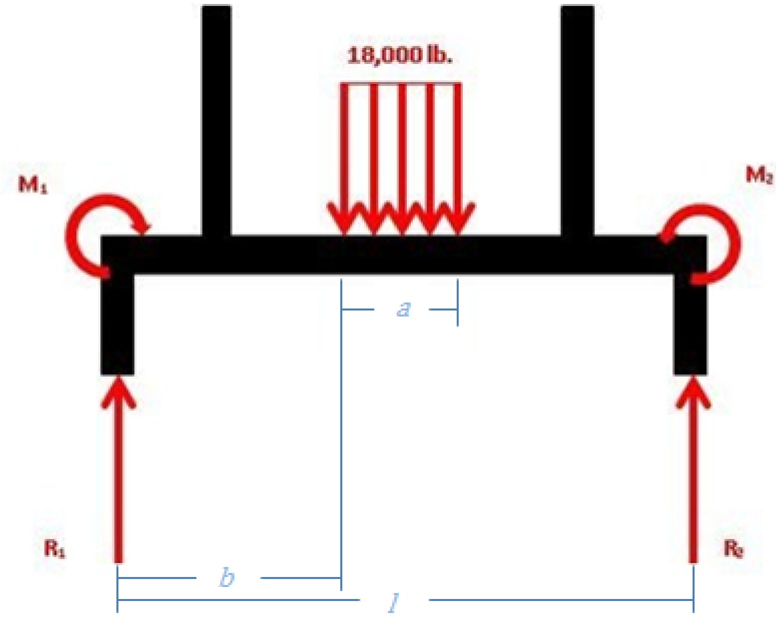


Figure F3. The beam bending model above for a fixed-fixed end condition shows the distributed load of the spring pack on the long side of the top plate. Reaction R_1 and R_2 represent the supporting plates from the base.

The following calculation was performed in order to determine the static deflection at the center of the beam:

(1)

(2)

(3)

(4)

(5)

Given the boundary conditions:

(6)
(7)

$$\theta = 0 \text{ at } x = 0 \qquad (8)$$
$$\theta = 0 \text{ at } x = 0 \qquad (9)$$

By solving equation 4 using boundary condition 8, C_1 is found to be zero. Similarly, by solving equation 5 using boundary condition 6, C_2 is found to be zero. Also, because the system is in static equilibrium, by setting equation 3 equal to zero and solving at $x = 0$, it is determined that $M_1 = M_2$. Finally by solving equation 5 using boundary condition 9, it is found that:

$$M = \frac{R_1}{3} l - \frac{\omega}{12l^2}(l-b)^4 + \frac{\omega}{12l^2}\big(l-(b+a)\big)^4 \qquad (10)$$

Based on symmetry and statics, R_1 is found to be:

$$R_1 = R_2 = \frac{\omega a}{2}$$

By plugging in the values for R and M and rearranging equation 5, the deflection equation for the system can be found:

$$y = \frac{\omega}{24EIl^2}\left[2ax^2l^2(x-l) + (l-b)^4x^2 - \big(l-(b+a)\big)^4x^2 - l^2 < x-b >^4 + l^2 < x-(b+a) >^4\right] \qquad (11)$$

In order to find ω, the total load of 18,000 *lbs* was divided by the length of the distributed load. The total length l was 16.2 inches from R_1 to R_2. A modulus of elasticity for carbon steel was assumed to be $30\text{x}10^6$ *psi*. Using system parameters, the deflection of the top plate can be determined:

$$l = 16.2\ in$$
$$a = 9.5\ in$$
$$b = 4.05\ in$$
$$x = 8.1\ in$$
$$h = 1\ in$$
$$\omega = \frac{18000 lbs}{9.5\ inches}$$
$$\omega = 1894.7\frac{lbs}{in}$$

$$I = \frac{lh^3}{12}$$

$$I = \frac{(16.2in)(1in)^3}{12}$$

$$I = 1.35in^4$$

$$y_{max} = \frac{(1894.7\frac{lbs}{in})}{24(30E6)(1.35in^4)(16.2in)^2}[2(9.5in)(8.1in)^2(16.2in)^2(8.1in - 16.2in) + (16.2in - 4.05in)^4(8.1in)^2 - \left(16in - (13.55in)\right)^4(8.1in)^2 - (16.2in)^2(8.1in - 4.05in)^4]$$

$$y_{max} = -0.0096\ in$$

According to this calculation, there will be a little over nine thousandths deflection after the jack is removed. It was later found that a using a fixed-fixed end condition model was not accurate. It was concluded that a pin-pin end condition is a more accurate model, given the system's ability to pivot somewhat on the base plate assembly. A new deflection calculation was done based on this new model.

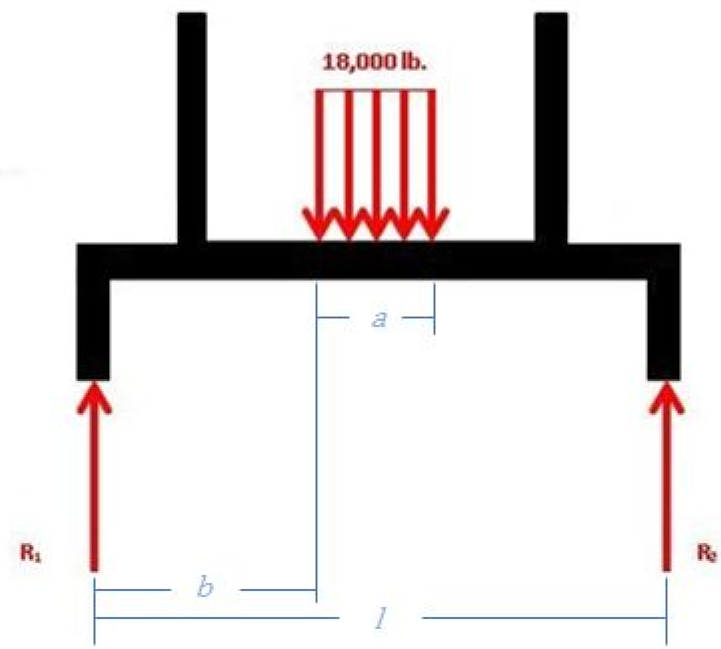


Figure F4. The beam bending model above for a pinned-pinned end condition shows the distributed load of the spring pack on the long side of the top plate. Reaction R_1 and R_2 represent the supporting plates from the base.

$$q = R_1 < x >^{-1} - \omega < x - b >^0 + \omega < x - (b + a) >^0 + R_1 < x - l >^{-1} \tag{12}$$

$$V = R_1 < x >^0 - \omega < x - b >^1 + \omega < x - (b + a) >^1 + R_1 < x - l >^0 \tag{13}$$

$$M = R_1 < x >^1 - \frac{\omega}{2} < x - b >^2 + \frac{\omega}{2} < x - (b + a) >^2 + R_1 < x - l >^1 \tag{14}$$

$$EI\theta = \frac{R_1}{2} < x >^2 - \frac{\omega}{6} < x - b >^3 + \frac{\omega}{6} < x - (b + a) >^3 + \frac{R_1}{2} < x - l >^2 + C_1 \tag{15}$$

$$EIy = \frac{R_1}{6} < x >^3 - \frac{\omega}{24} < x - b >^4 + \frac{\omega}{24} < x - (b + a) >^4 + \frac{R_1}{6} < x - l >^3 + C_1 x + C_2 \tag{16}$$

Given the boundary conditions:

$$y = 0 \text{ at } x = 0 \tag{17}$$

$$y = 0 \text{ at } x = l \tag{18}$$

By solving equation 16 using boundary condition 17, C_2 is found to be zero. Now by solving equation 16 using boundary condition 18, it is found that:

$$C_1 = \frac{-R_1}{6} l^2 + \frac{\omega}{24l}(l - b)^4 - \frac{\omega}{24l}\big(l - (b + a)\big)^4 \tag{19}$$

Based on symmetry and statics, R_1 is found to be:

$$R_1 = R_2 = \frac{\omega a}{2}$$

By plugging in the values for R and C_1, and rearranging equation 16, the deflection equation for the system can be found:

$$y = \frac{\omega}{24EIl}[2axl(x^2 - l^2) + (l - b)^4 x - \big(l - (b + a)\big)^4 x - l < x - b >^4 + l < x - (b + a) >^4] \tag{20}$$

In order to find ω, the total load of 18,000 *lbs* was divided by the length of the distributed load. The total length l was 16.2 inches from R_1 to R_2. A modulus of elasticity for carbon steel was assumed to be 30×10^6 *psi*. Using system parameters, the deflection of the top plate can be determined:

$$l = 16.2\ in$$
$$a = 9.5\ in$$
$$b = 4.05\ in$$
$$x = 8.1\ in$$
$$h = 1\ in$$

$$\omega = \frac{18000lbs}{9.5\ inches}$$

$$\omega = 1894.7\frac{lbs}{in}$$

$$I = \frac{lh^3}{12}$$

$$I = \frac{(16.2in)(1in)^3}{12}$$

$$I = 1.35in^4$$

$$y_{max} = \frac{(1894.7\frac{lbs}{in})}{24(30E6)(1.35in^4)(16.2in)}[2(9.5in)(8.1in)(16.2in)((8.1in)^2 - (16.2in)^2) + (16.2in - 4.05in)^4(8.1in) - \left(16in - (13.55in)\right)^4(8.1in) - (16.2in)(8.1in - 4.05in)^4]$$

$$y_{max} = -0.0384\ in$$

With the new pin-pin end condition model, the calculation more accurately represents the numbers that were found during testing. In order to decrease this number to be within tolerance limits as specified by Solar Turbines, it would be necessary to increase the thickness of the top plate by approximately 0.57 inches. This increase, based on the model, would decrease the deflection of the top plate to under 0.010 inches.

Sliding Friction

The ability of the new aft-mount design to account for axial expansion from the engine is the second most important design requirement behind repeatability. The use of the PBC linear slider system at the base of the mount should allow the entire structure to move along with the axial thermal growth. In order for the mount to begin sliding, there will need to be a certain amount of force applied in order to break the static friction in the slider. The following calculation was used to determine that it will take about a thousand pounds of force in the axial direction to move the slider.

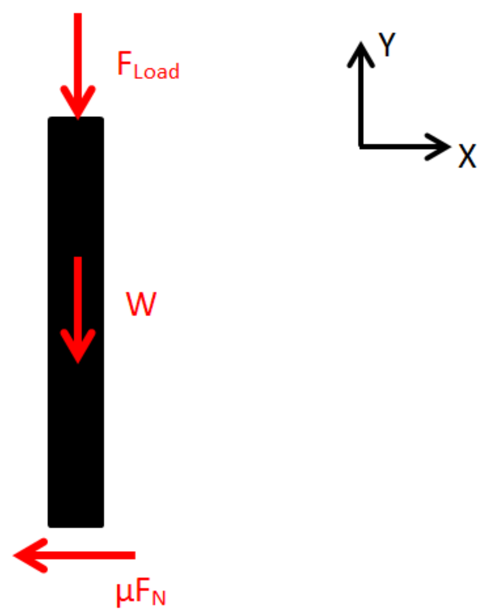


Figure F5. Free body diagram of the entire mount structure showing the load on the mount as well as the friction from the slider at the base of the mount. The black body represents a simplified model of the aft-mount.

The required force to break the static friction of the slider was found using the following calculation. The force from the load was assumed to be 16,000lbs and the weight of the mount was assumed to be 200lbs. A coefficient of static friction was assumed to be 0.25, which is conservative considering the final system should be around 0.1.

$$\sum F_y = 0$$

$$F_{Load} + W = F_{Normal}$$

$$16{,}000lbs + 200lbs = F_{Normal}$$

With 2 rails and 2 bearings per rail,

$$F_{Rail} = \frac{F_{Normal}}{2}$$

$$F_{Rail} = \frac{16{,}200 lbs}{2}$$

$$F_{Rail} = 8{,}100 lbs$$

$$F_{To\ Move} = F_{Rail} * \mu * \left(\frac{1\ rail}{2\ bearings}\right)$$

$$F_{To\ Move} = 8{,}100 lbs * 0.25 * \left(\frac{1\ rail}{2\ bearings}\right)$$

$$F_{To\ Move} = 1{,}012.5 lbs$$

Appendix G: Assembly Description

As with any new tool or component, assembly procedures and instructions are crucial to the successful implementation of the product. In order to begin the mount's assembly process, it is first important to have each of the sub-components fully assembled. This entails welding together each of the sub-assembly parts that are specified in drawing number 3000B. Specific components isometric views can be seen in Figures G1 through G14.

Figure G1. The base plate assembly is to be welded together per instructions denoted in sub-assembly drawing 2004C.Base Plate fastened to rail

Figure G2. The top plate assembly is to be welded together per instructions denoted in sub-assembly drawing 2007C.

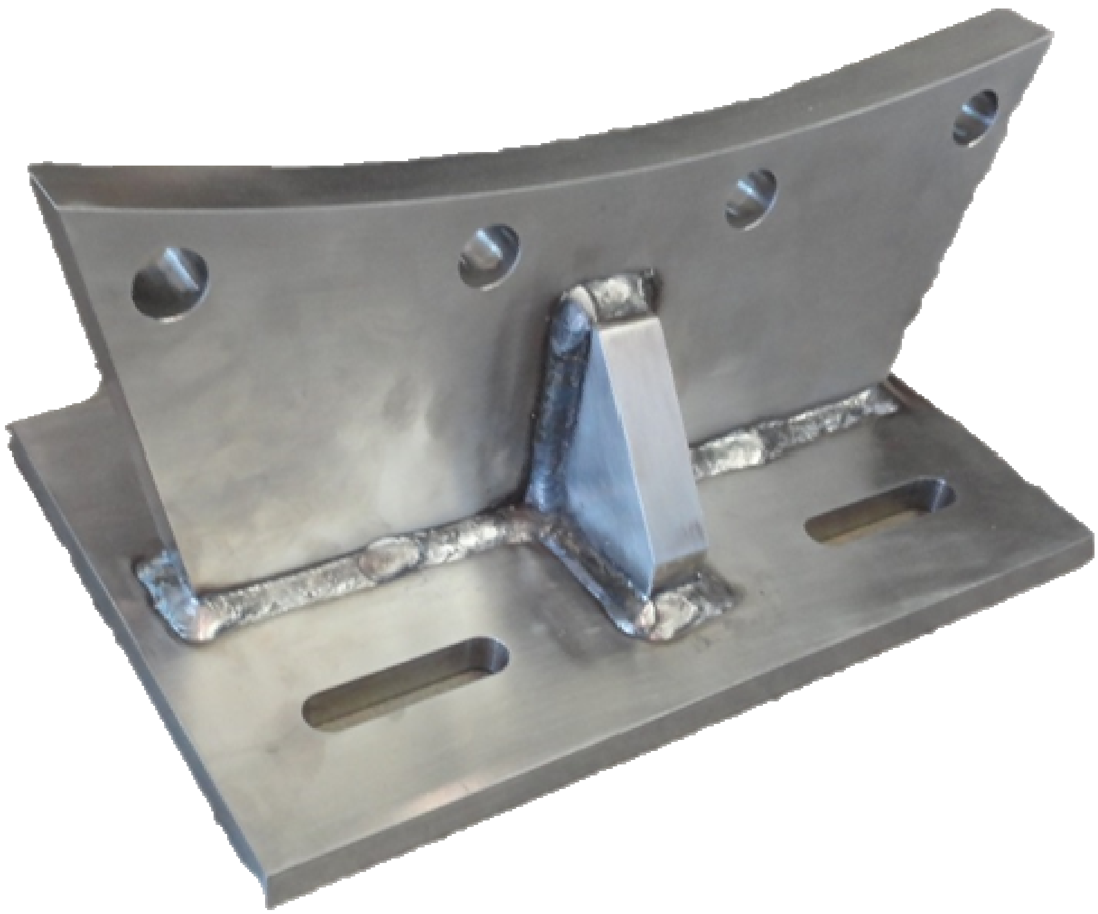


Figure G3. The engine bracket assembly is to be welded together per instructions denoted in sub-assembly drawing 2003C. Slots allow for left-right alignment of engine.

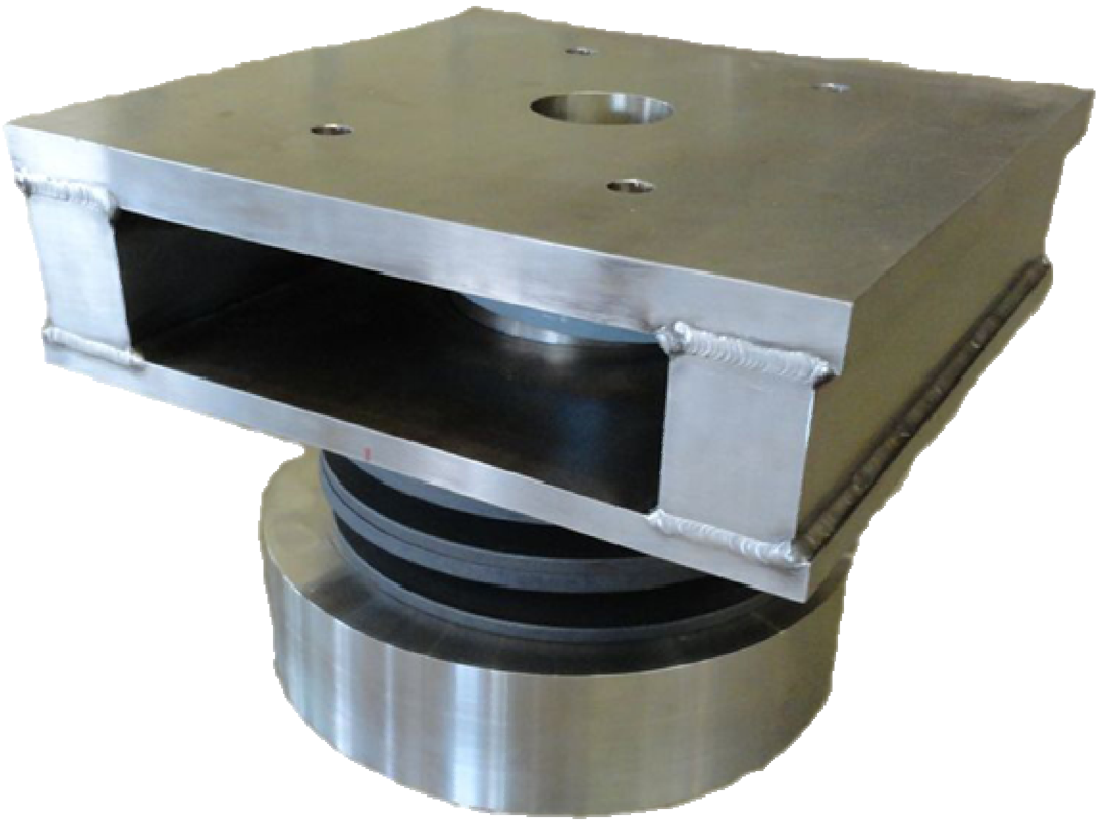


Figure G4. The spring pack assembly is to be welded together per instructions denoted in sub-assembly drawing 2006C.

To begin the mount setup, the first step would be to attach the rail and slider system onto the skid. Depending on the skid design, this would require either placing the slider system directly atop a pedestal, or on a crossbeam of the skid. For the purpose of this document, the skid has been omitted.

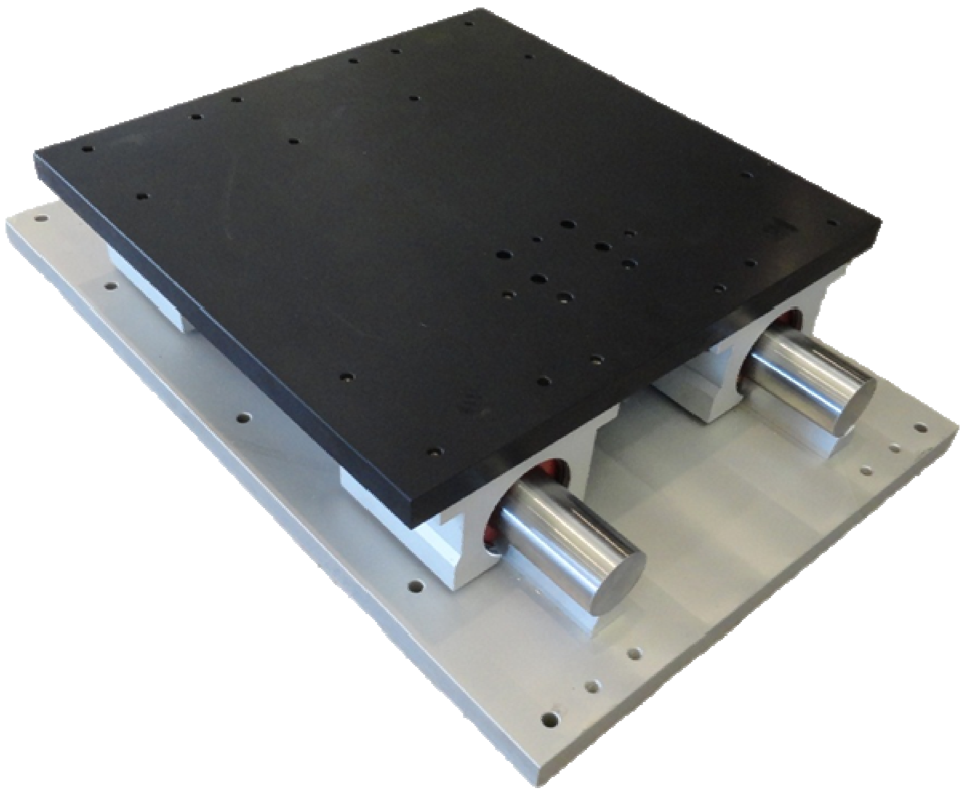


Figure G5. The rail and slider system is placed on top of the skid and bolted in place.

Next, part number 2004 is dropped into place on top of the slider system. There will be four studs protruding from the slider plate which the base plate will mate to. Once the base plate is in position, the washers and nuts will be screwed on, effectively locking the mount in place on top of the slider system.

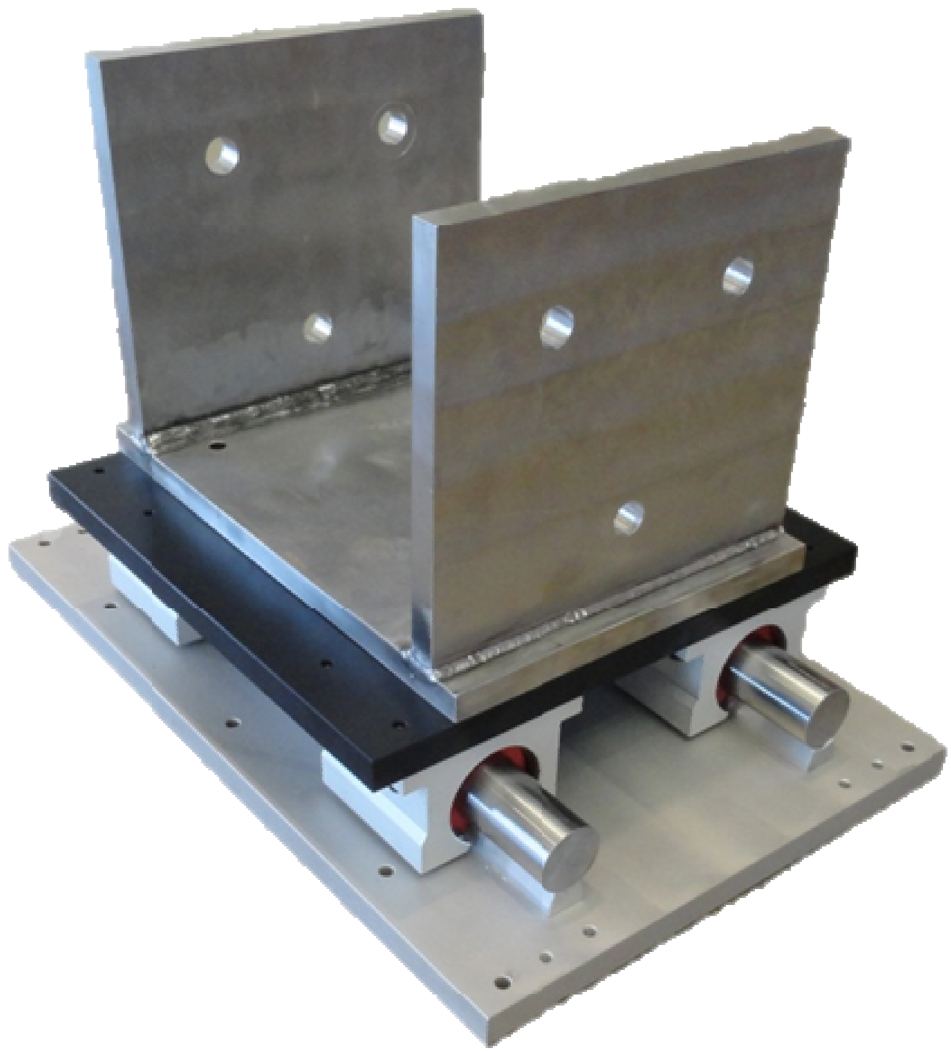


Figure G6. The base plate assembly is placed on top of the rail and slider system.

Now, the mount's top assembly can be set on top of the base plate. Once the slots in the top assembly are lined up with the holes in the side plates of the base, four 1 inch diameter bolts can be dropped into position. The washer and nut that holds the bolt in place should not be tightened down until after the engine is in its final position. Until then, the bolts will prevent the top of the mount from moving left and right during the assembly process.

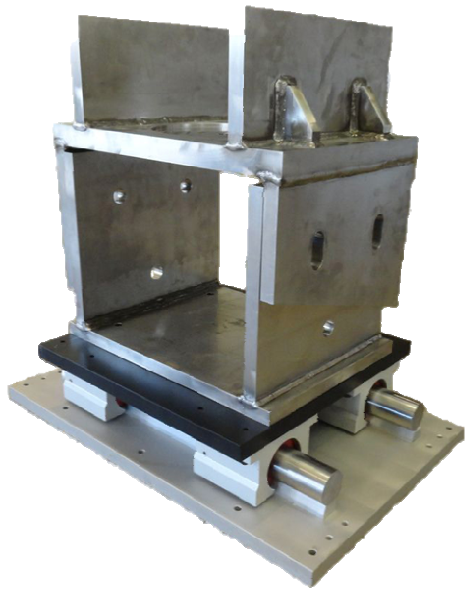


Figure G7. Top plate assembly rested on top of the base plate assembly.

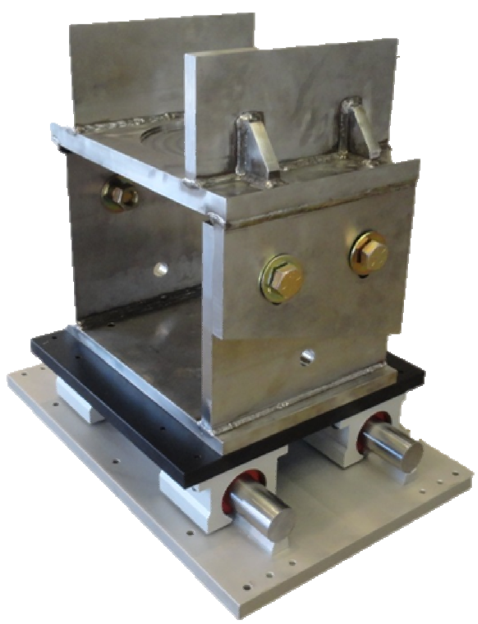


Figure G8. Bolts are then put in place between the top plate and side of the base plate but are loose to allow for vertical movement.

The wedges can be put into place at this point so that when the alignment is achieved, they can be slid into place and secured. Again, the bolts are loose.

Figure G9. Wedges put in place. Bolts are loose to allow for horizontal movement.

The spring pack assembly can now be placed on top of the mount. It should sit in the recessed hole within the top plate assembly. The long side of the pillars, part number 1017, should be parallel with the side of the top plate, part number 1007. This step can be viewed in Figure G10.

Figure G10. The spring pack is then placed on top of the top plate assembly.

The mechanical jack with the load cell can now be placed in the cavity of the mount. This will be used to apply the required load to set the engine's shaft centerline. The engine, which should already have the engine bracket, part number 2003, attached, is lowered onto the mount, and bolted in place. The jack can be cranked up using a pneumatic hand tool to get the approximate required load, and then a wrench can be used to acquire a more precise load number.

Figure G11. Load cell attached to jack through interface plate. Data acquisition system records voltage that corresponds to load on the cell.

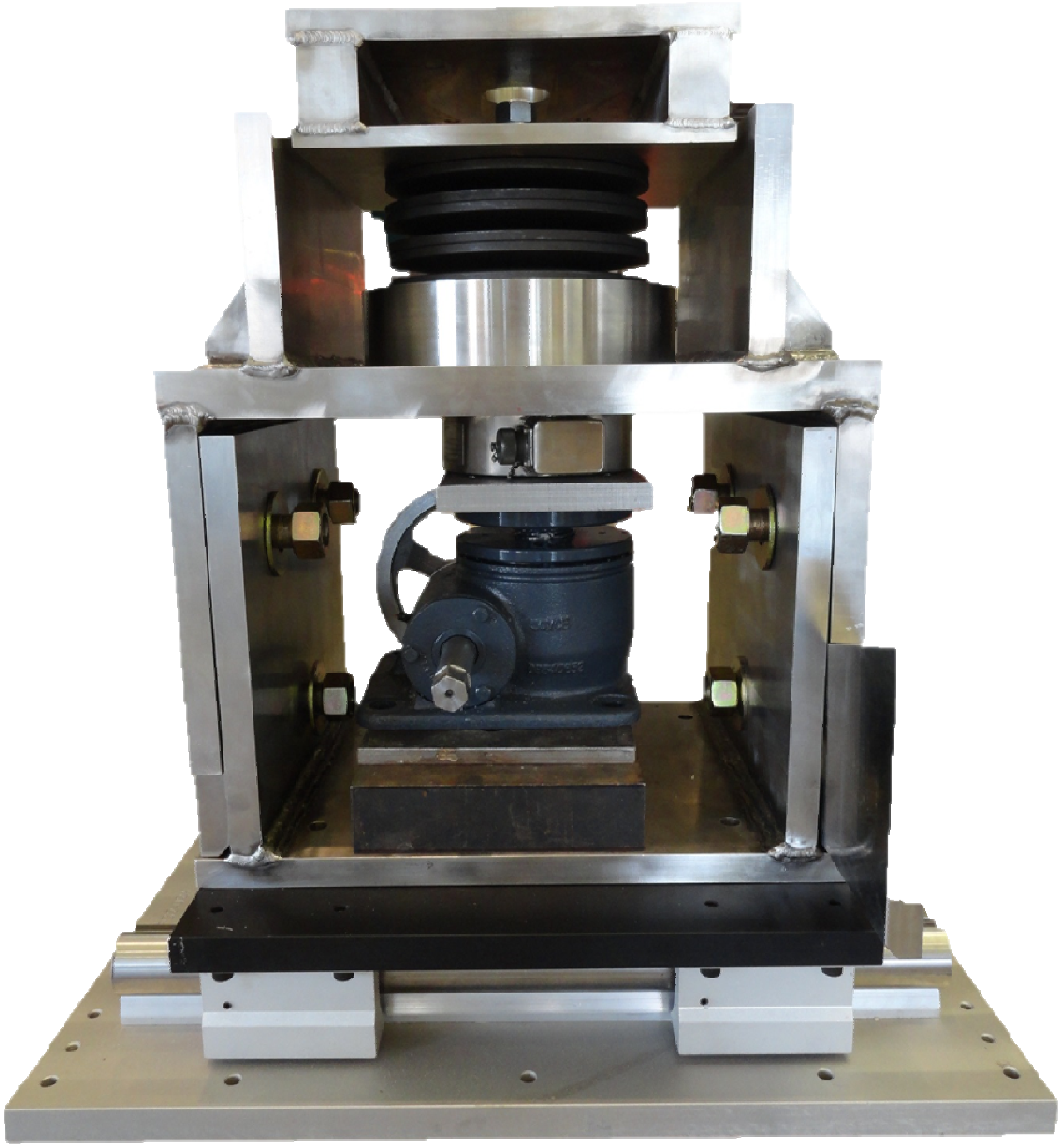


Figure G12. Jack is sitting on supports. Load cell is attached to the jack through the interface plate.

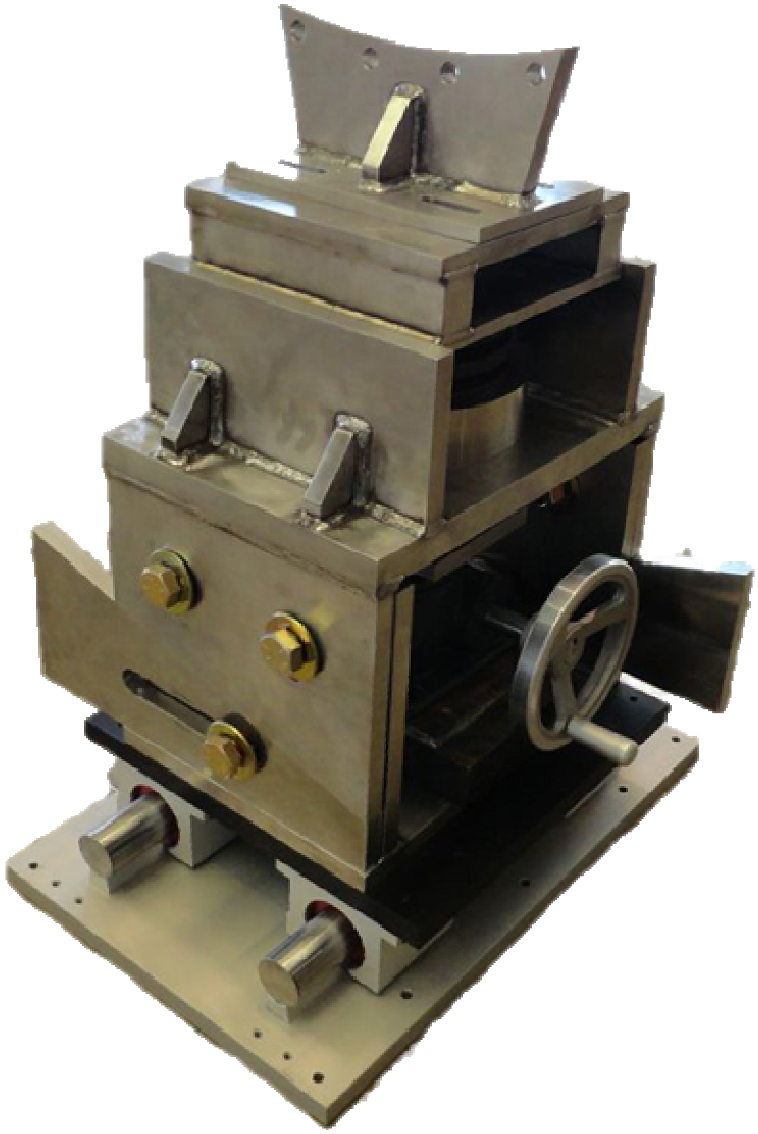


Figure G13. One side of the jack has a hand wheel while the other has a hex connection for a pneumatic wrench. At this point, the engine alignment is ready to be set.

Once the proper load setting is achieved, the bolts previously placed in the top plate assembly can be tightened to a torque setting of 750ft-lb. The wedges can be slid into place and tightened to a torque setting of 243ft-lb. This configuration can be viewed in Figure G14. Now that the mount has been locked into place, the jack can be released and removed from the cavity. The bolts in the mount will now be supporting the static load of the engine. This is the final step in the assembly process; caution should be exercised when operating underneath the engine in this final state.

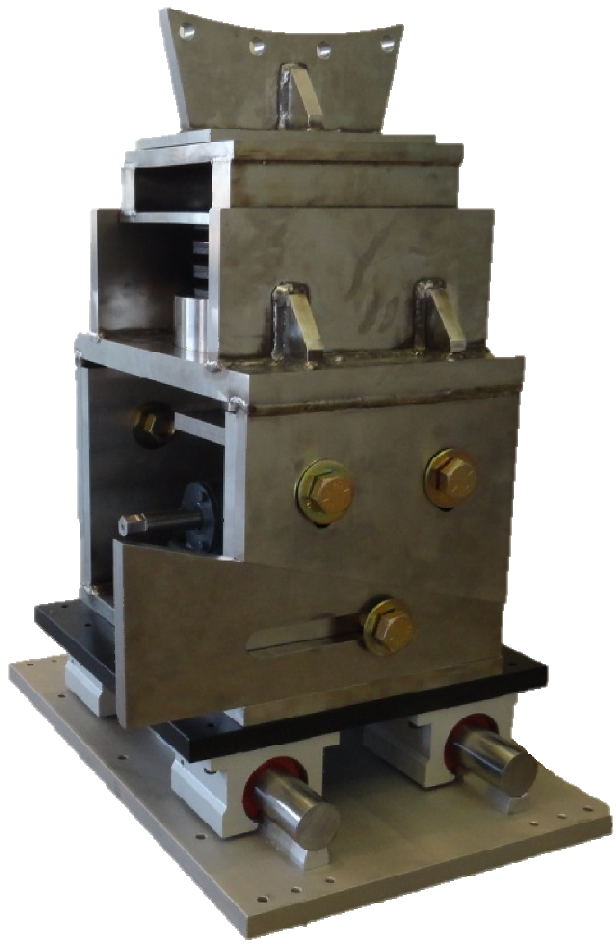


Figure G14. After alignment is achieved, bolts are tightened and the jack and load cell are removed.

Appendix H: Assembly Procedure

Parts Checklist:

1.0 Titan 130 Aft-Mount Components
- 1.1 Base Plate sub-assembly
- 1.2 Top Plate sub-assembly
- 1.3 2x Wedge
- 1.4 Spring Pack sub-assembly pre-compressed to ES2314 specifications
- 1.5 Engine Bracket sub-assembly
- 1.6 Rail Slider buy-out item
- 1.7 Fasteners
  - 1.7.1 6x 1” - 14 x 3” bolts
  - 1.7.2 6x 1” - 14 nuts
  - 1.7.3 12x 1” washers
  - 1.7.4 4x ½” - 20 x 2 ¾” bolts
  - 1.7.5 4x ½” – 20 nuts
  - 1.7.6 8x ½” washers

2.0 Titan 130 Aft-Mount Alignment Tooling
- 2.1 Joyce Jack
- 2.2 Honeywell Model 41 Load Cell with DAQ
- 2.3 2x jack support bars
- 2.4 Jack-load cell interface plate

Aft-Mount Assembly

3.0 Secure rail slider to skid

4.0 Secure base plate to rail slider using studs

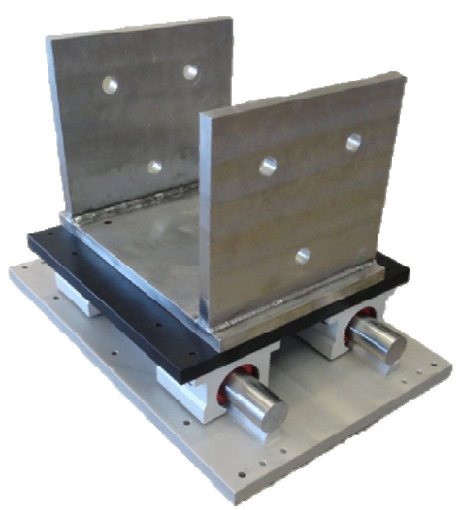


Figure H1. Orientation of base plate on rail sliders.

5.0 Rest top plate on base plate
  5.1 Insert 4x 1” fasteners into top plate holes
    5.1.1 Bolts will remain loose at this point just to keep vertical alignment
6.0 Place 2x wedge on base plate, chamfered edge facing the weld
  6.1 Insert 2x 1” fasteners into top plate holes
    6.1.1 Bolts will remain loose at this point just to keep horizontal alignment
  6.2 Slide wedge as far to the left as possible in order to leave gap between wedge face and top plate.
7.0 Place spring pack assembly into the recessed cut out of the top plate. The tie rod nuts should be accessible from the side of the mount.

Figure H2. Orientation of spring pack assembly on aft-mount.

Engine Alignment Procedure:

8.0 Insert alignment tooling into the cavity of the aft-mount
  8.1 2x jack support plates resting flush on bottom of base plate
  8.2 Place jack on top of support plates making sure that the jack load pad is centered
  8.3 Thread load cell onto interface plate
  8.4 Secure interface plate with load cell on top of jack. Alignment pins should fit into the holes on the load pad.
  8.5 Attach data acquisition system to load cell
9.0 Attach engine bracket to the engine
10.0 Set engine onto aft-mount adjusting for left-right positioning at this time

11.0 Secure engine bracket to mount using ½" bolts
12.0 Begin to raise jack using either the hand wheel or pneumatic wrench
13.0 Begin to raise jack until
    13.1 A voltage of 1.929V is achieved for a conventional engine
    13.2 A voltage of 2.311V is achieved for a SoLoNOx engine
14.0 Once the proper voltage is achieved on the DAQ, slide wedges in so that the wedge face is in contact with the top plate.
15.0 Tighten the bolts of the assembly to the specified torque settings
    15.1 Torque for 4 top plate bolts → 750 ft-lbs
    15.2 Torque for 2 wedge bolts → 243 ft-lbs
16.0 Release jack and remove alignment tooling from aft-mount cavity

Appendix I: Test Plan Procedure

Static Load Test:

1.0 Assemble mount on test machine
- 1.8 Place sliders onto test machine
- 1.9 Set base plate on sliders
- 1.10 Place top plate on base plate
  - 1.10.1 Loosely place 1”-14 bolts in 4 slot positions
- 1.11 Lift top plate to mock loading height 1” above base plate and place wedges securely underneath the top plate
  - 1.11.1 Place 1”-14 bolts into slots of wedges and tighten all six bolts to specified torque setting
    - 1.11.1.1 Torque for 4 top plate bolts → 750 ft-lbs
    - 1.11.1.2 Torque for 2 wedge bolts → 243 ft-lbs
- 1.12 Set uncompressed spring pack assembly on the top plate
  - 1.12.1 Apply load until the required initial deflection of the spring is achieved
    - 1.12.1.1 Total spring length of 2.5”
  - 1.12.2 Tighten the nuts on the top side of the spring pack assembly to hold the spring pack in the initial deflection position
- 1.13 Place the engine bracket simulation plate over the bolt holes on top of the spring pack assembly

17.0 Turn on the MTS 322 Test Frame to begin static load testing of the mount

18.0 Begin running the static load test program for Titan 130 engine test
- 18.1 Load the mount up to maximum weight expectation of 18,000 lbs
  - 18.1.1 Measure center deflection of top plate using 0.1” dial gages on both sides of plate
    - 18.1.1.1 Record deflections in data table
  - 18.1.2 Measure spring pack deflection using a 0.5” dial gage
    - 18.1.2.1 Record deflections in data table
  - 18.1.3 Measure overall mount deflection using a 0.5” dial gage
    - 18.1.3.1 Record deflections in data table

19.0 Repeat step 3.0 for a total of 10 repetitions

Loaded condition with engine vibration simulation:

20.0 With the mount assembled on the test machine, begin the frequency test program for the conventional engine on the MTS 322 Test Frame.

21.0 Before the frequency is applied, loosen the nuts holding the spring pack to a maximum length of 2.5” so that there is a 0.050” clearance above the cylinder guide.

22.0 Begin frequency generation for the conventional engine for 30 minutes.

22.1 Measure center deflection of top plate using 0.1” dial gages on both sides of the plate

22.1.1 Record deflections in data table

22.2 Measure spring pack deflection using a 0.5” dial gage

22.2.1 Record deflections in data table

22.3 Measure overall mount deflection using a 0.5” dial gage

22.3.1 Record deflections in data table

23.0 Once the conventional engine test is concluded, reset the spring pack assembly nuts to hold the spring length at 2.5”

24.0 Begin the frequency test program for the SoLoNOx engine on the MTS 322 Test Frame.

25.0 Before the load is fluctuated, loosen the nuts holding the spring pack to a maximum length of 2.5” so that there is a 0.050” clearance above the cylinder guide.

26.0 Begin frequency generation for the SoLoNOx engine for 30 minutes.

26.1 Measure center deflection of top plate using 0.1” dial gages on both sides of the plate

26.1.1 Record deflections in data table

26.2 Measure spring pack deflection using a 0.5” dial gage

26.2.1 Record deflections in data table

26.3 Measure overall deflections using a 0.5” dial gage

26.3.1 Record deflections in data table

27.0 Once the SoLoNOx engine test is concluded, reset the spring pack assembly nuts to hold the spring length at 2.5”

Jack and load cell:

28.0 Assemble mount on the test machine

28.1 Place sliders onto test machine

28.2 Set base plate on sliders

28.3 Place top plate on base plate

28.3.1 Loosely place 1”-14 bolts in 4 slot positions

28.4 Place wedges securely underneath the top plate

28.4.1 Loosely place 1”-14 bolts into slots of wedges

28.5 Set uncompressed spring pack assembly on the top plate

28.5.1 Apply load until the required initial deflection of the spring is achieved

28.5.1.1 Total spring length of 2.5”

28.5.2 Tighten the nuts on the top side of the spring pack assembly to hold the spring pack in the initial deflection position

28.6 Place the jack and load cell assembly into the jack cavity

29.0 Apply conventional load expectation of 12,927 lbs to the mount with jack in place

30.0 With the load applied, lift the top plate assembly so that the load cell gives a reading of 12,927 lbs or 1.929 V
31.0 Tighten the bolts of the assembly to the specified torque settings
   31.1 Torque for 4 top plate bolts → 750 ft-lbs
   31.2 Torque for 2 wedge bolts → 243 ft-lbs
32.0 Lower the jack and remove from the assembly
33.0 Release load from the mount
34.0 Replace the jack and load cell assembly back into the jack cavity so that it supports the weight of the upper assembly
35.0 Loosen bolts in the mount and return to resting position
36.0 Apply SoLoNOx load expectation of 15,484 lbs to the mount with jack in place
37.0 With the load applied, lift the top plate assembly so that the load cell gives a reading of 15,484 lbs or 2.311 V
38.0 Tighten the bolts of the assembly to the specified torque settings
   38.1 Torque for 4 top plate bolts → 750 ft-lbs
   38.2 Torque for 2 wedge bolts → 243 ft-lbs
39.0 Lower the jack and remove from the assembly
40.0 Release load from the mount
41.0 Replace the jack and load cell assembly back into the jack cavity so that it supports the weight of the upper assembly
42.0 Loosen bolts in the mount and return to resting position

Bolts removed from assembly:

43.0 Assemble mount on test machine
   43.1 Place sliders onto test machine
   43.2 Set base plate on sliders
   43.3 Place top plate on base plate
      43.3.1 Loosely place 1”-14 bolts in 4 slot positions
   43.4 Lift top plate to mock loading height 1” above base plate and place wedges securely underneath the top plate
      43.4.1 Loosely place 1”-14 bolts into slots of wedges
   43.5 Set uncompressed spring pack assembly on the top plate
      43.5.1 Apply load until the required initial deflection of the spring is achieved
         43.5.1.1 Total spring length of 2.5”
      43.5.2 Tighten the nuts on the top side of the spring pack assembly to hold the spring pack in the initial deflection position
44.0 Slowly increase the load up to 18,000 lbs on the mount
   44.1 Calculations show that due to the designed wedge angle and friction, the mount will hold its position at this load

44.1.1 This is worst case scenario where all 6 bolts lose tension

Rail and Slider Friction Test:

45.0 Place rail-slider system on flat and level ground

45.1 Attach load cell to g a cable wrapped around the sliding plate

45.2 Pull until static friction is broken between the plate and rails

45.2.1 Record required force in data table

45.3 Continue to pull so that slider is barely moving

45.3.1 Record required force in data table

46.0 Add 50 lb plate on top of the sliding plate

47.0 Repeat step 30.2 through 31.0 for a total of 4 more times or until 250 lbs has been reached and recorded

48.0 Using these data points, the static coefficient of friction can be determined

49.0 Repeat steps 30.0 through 33.0 after applying dirt and grit to the rails to simulate a worn condition for the slider system after many hours of use.

50.0 Record both "clean" and "worn" static coefficients of friction

Appendix J: Testing Data and Analysis

The first of the compression tests for the mount was designed to test the repeatability of the system for both engine loading conditions. Ten tests were run in a similar fashion, and each of the data sets that were collected for each run were plotted against one another to view any possible differences between each test. There is very little noticeable difference between the plot for spring pack deflection over a varying load and the overall deflection of the mount over a varying load. The reasoning behind this similarity is that the LVDT measuring the spring pack displacement takes into account the deflection of the top plate in addition to the spring pack deflection. The LVDT measuring the overall deflection takes both of those changes in height with the addition of any deflection in the top plate of the spring pack assembly. There is very little deflection in the top plate of the spring pack assembly and therefore, the two plots look almost identical. The reason for the slope change in the two plots comes from the spring pack being initially compressed to an approximate spring length of 2.5 inches. Up until the load on the mount reaches about 13.5 kips, the spring pack does not compress more than the initial compression due to the tie-rod bearing the tension load from the spring. The change in deflection read from the 0.5” LVDTs is the deflection of the top plate that the spring pack assembly sits on up until the load reaches about 13.5 kips. Once the load continues past 13.5 kips, the tension force in the tie-rod reaches zero and the spring pack begins to compress and the deflection read by the LVDT is now predominantly the change in height of the spring pack.

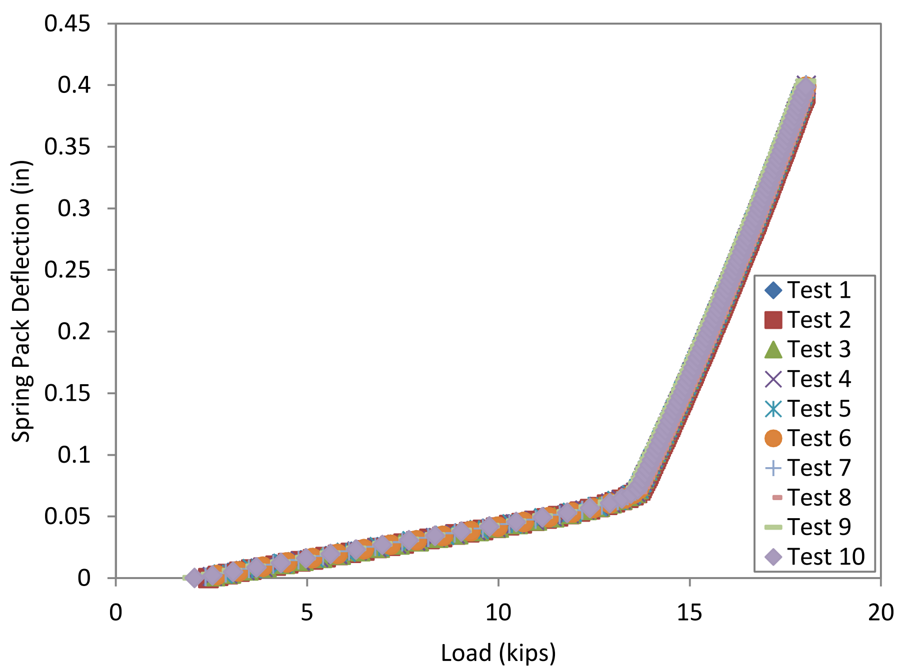


Figure J1. Spring pack compression data from 0.5” LVDT A for tests 1 through 10.

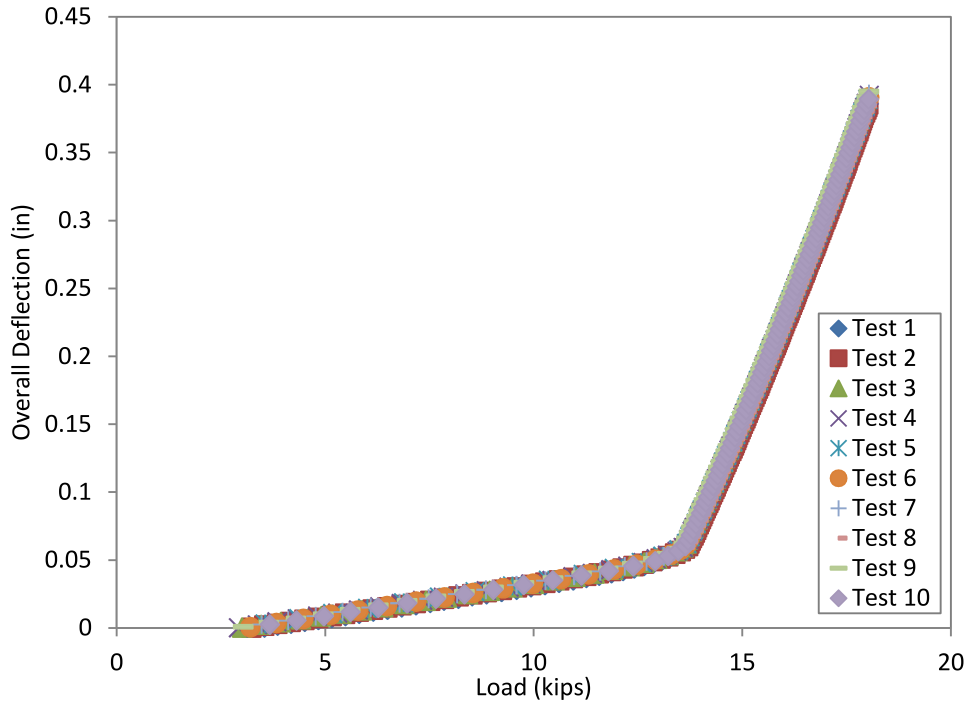


Figure J2. Overall mount compression data from 0.5” LVDT B for tests 1 through 10.

Measurements were also taken on both sides of the top plate assembly on which the spring pack assembly sits on. This data shows over a varying load, how much the top plate deflects. 0.1” LVDTs were placed on both sides to compare symmetrical data for the top plate. Knowing that the error associated with each of the LVDTs is 0.2%, it was concluded that the data acquired from the 0.1” LVDT B is flawed. It is believed that there was some kind of malfunction with the equipment possibly due to being dropped.

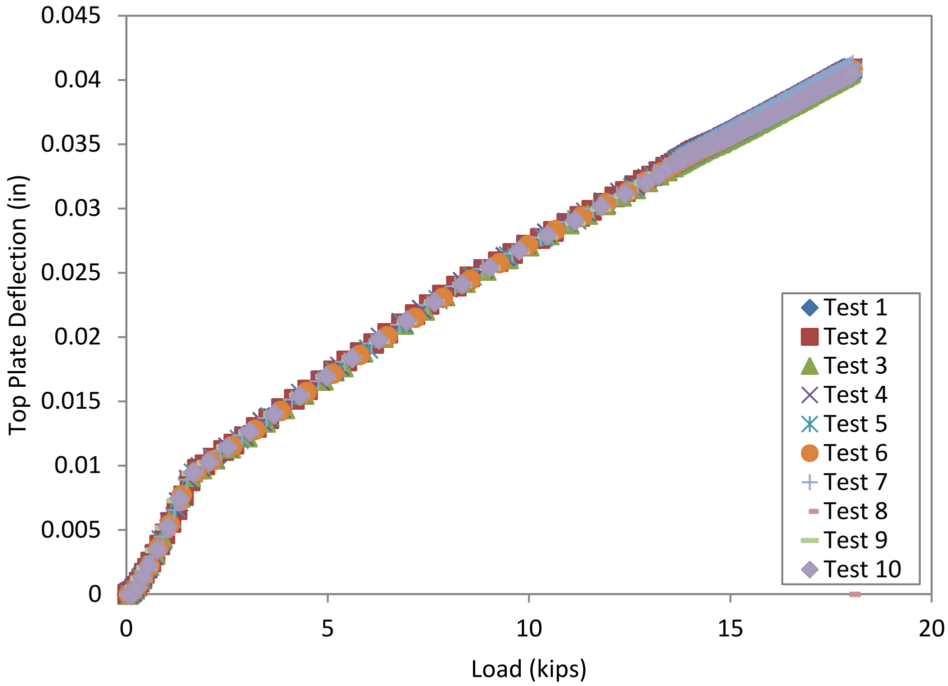


Figure J3. Top plate deflection data from 0.1” LVDT A for tests 1 through 10.

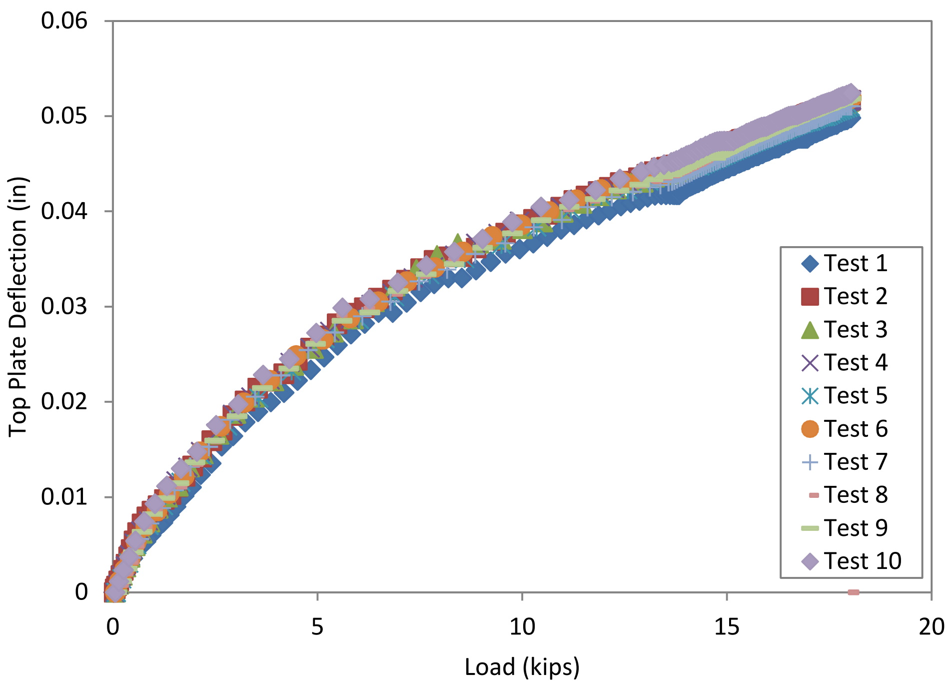


Figure J4. Top plate deflection data from 0.1” LVDT B for tests 1 through 10. Data is possibly corrupted from equipment malfunction.

The next set of tests that were conducted involved cyclic loading of the mount to simulate normal engine running operations. Due to limited time on the test machine, the duration of these tests were limited to 30 minutes each. The test equipment was set to record data at 1 second intervals. Because of this setting, the data acquired is limited in what was actually tested. For the conventional engine, the target load was 12,927 pounds ± 600 pounds. Since the frequency of data collection was set so low, the results only show a change of about ±100 pounds. This is true for the conventional test and both SoLoNOx tests that were conducted. For the SoLoNOx tests the target load was 15,484 pounds ± 600 pounds, but the results only show a change of approximately ± 100 pounds of load. For the conventional engine test, there is a distinct trend in the plot that shows as the load is decreased, the mount decompresses. Similarly, as the load increases, the mount compresses. There is a very small change in the overall change in height of the mount as the load is fluctuated; however, speculation on the mounts change in deflection over time is impossible to extrapolate given the relatively short duration of the test. For the SoLoNOx engine tests, it is believed that the LVDTs were unable to keep up with the large change in amplitude of the deflection at such a high frequency. This explains why there is no apparent correlation in the data for any of the SoLoNOx plots.

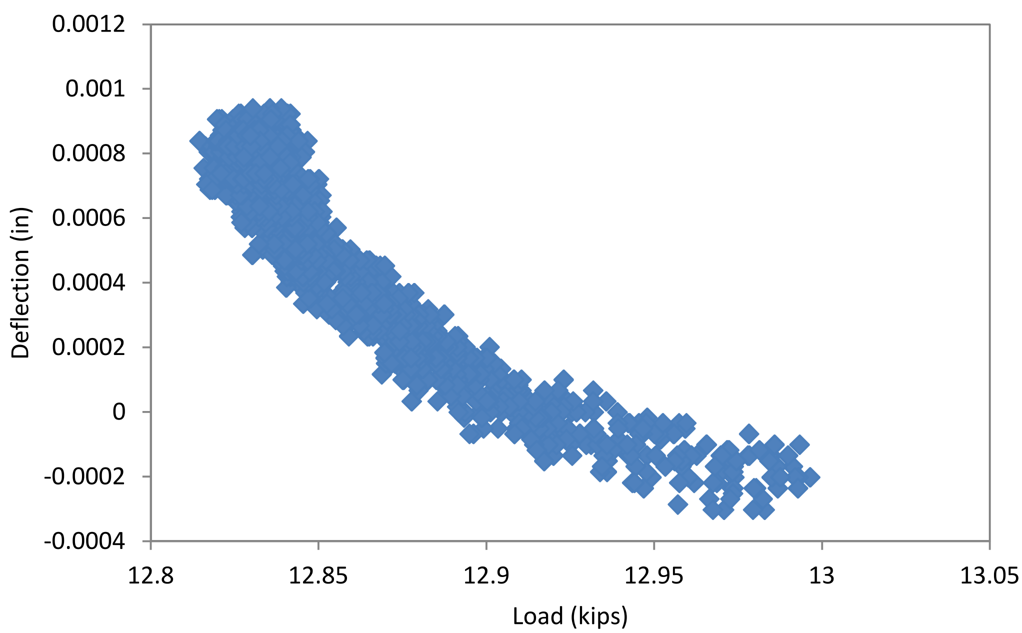


Figure J5. Cyclic loading test for conventional engine where the target load is 12,927 pounds ± 600 pounds. Measurements taken from the spring pack.

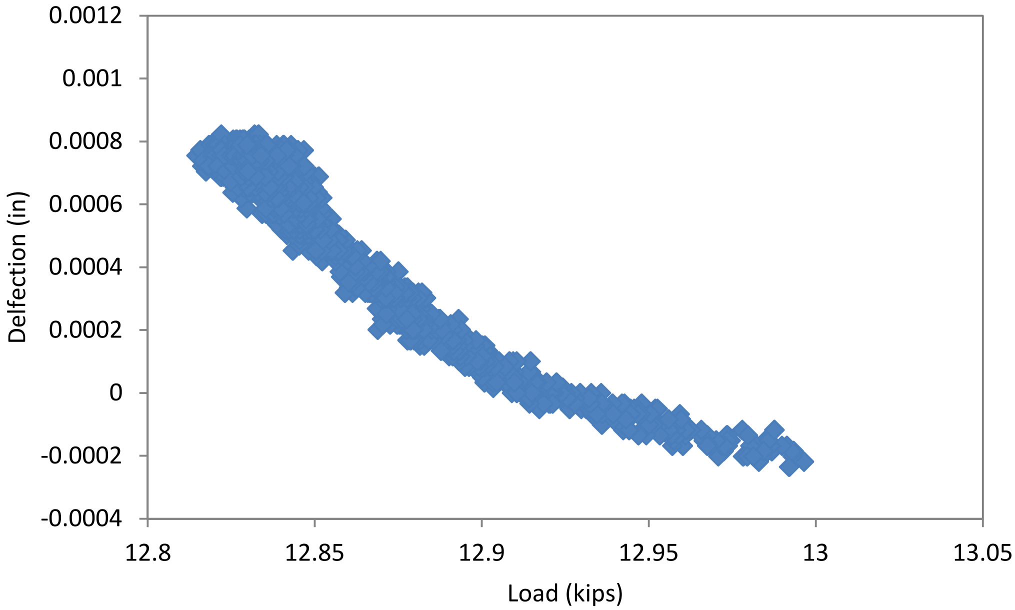


Figure J6. Cyclic loading test for conventional engine where the target load is 12,927 pounds ± 600 pounds. Measurements taken from the top of the spring pack assembly.

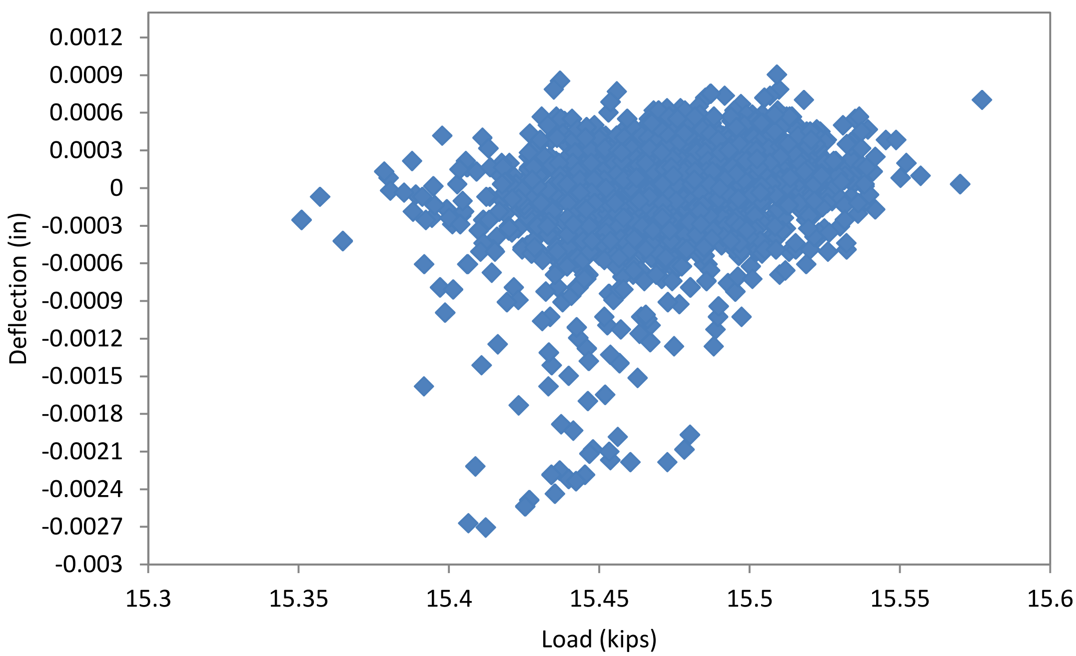


Figure J7. Cyclic loading test for SoLoNOx engine where the target load is 15,484 pounds ± 600 pounds. Measurements taken from the spring pack.

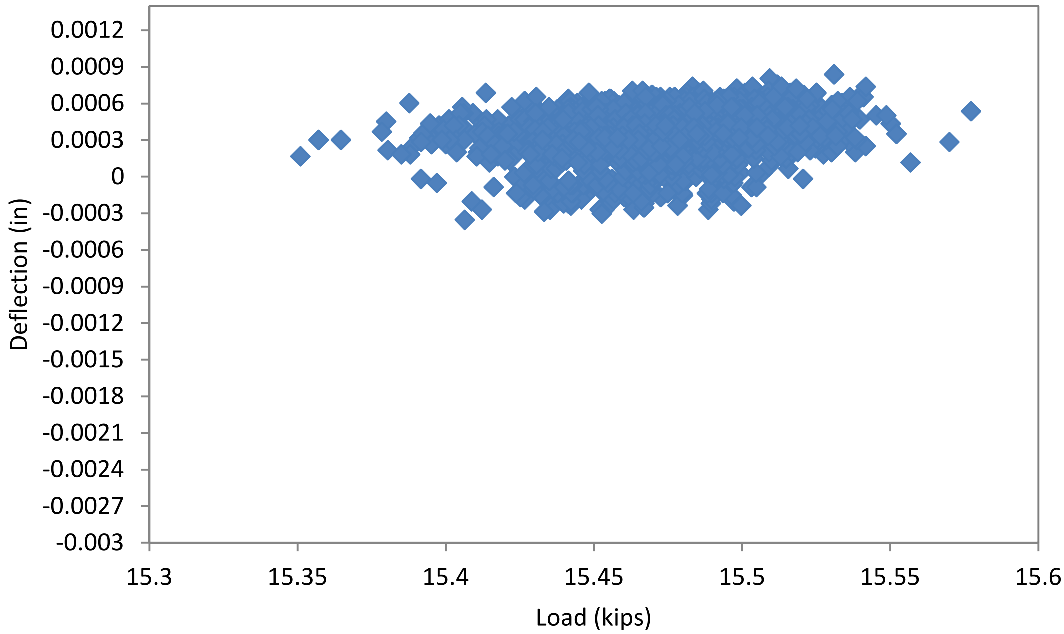


Figure J8. Cyclic loading test for SoLoNOx engine where the target load is 15,484 pounds ± 600 pounds. Measurements taken from the top of the spring pack assembly.

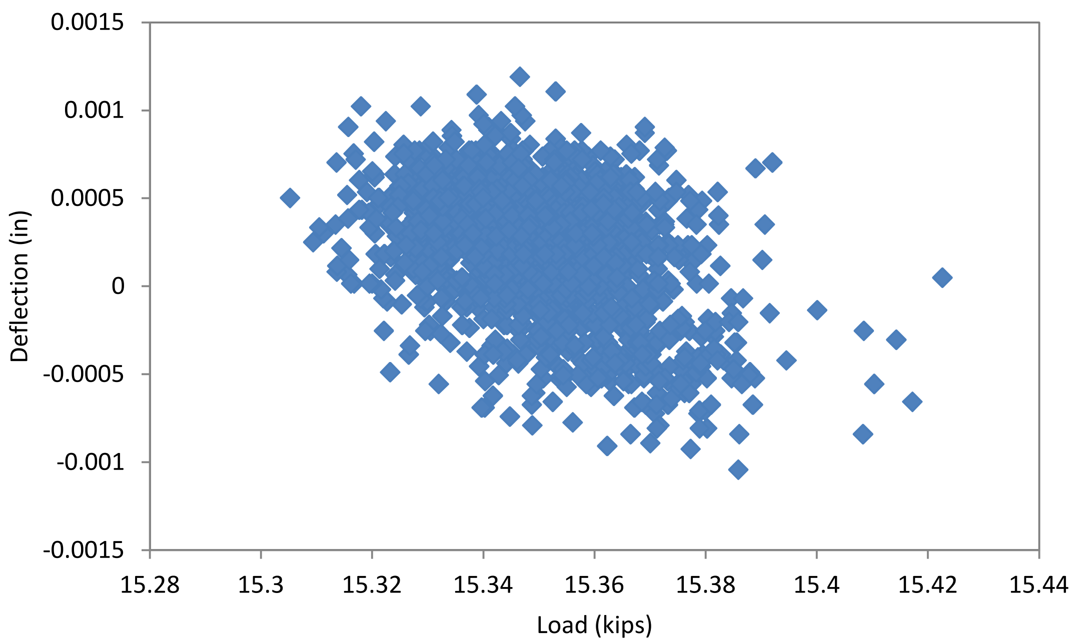


Figure J9. Cyclic loading test for SoLoNOx engine based on the actual engine conditions adjusted for 100Hz operating frequency. Measurements taken from the spring pack.

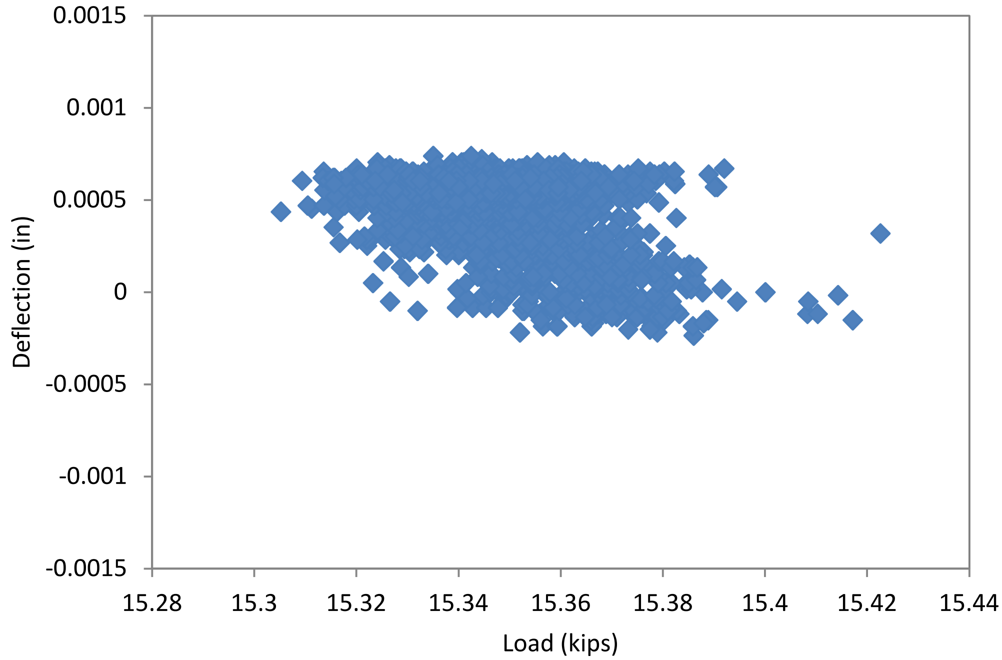


Figure J10. Cyclic loading test for SoLoNOx engine based on the actual engine conditions adjusted for 100Hz operating frequency. Measurements taken from the top of the spring pack assembly.

The first test was conducted with the sliders in an as-delivered condition with clean, non-lubricated rails. The weight of the top plate of the slider was neglected for the testing and plotted data is based on payload, rather than total weight. Five tests were conducted, each time adding about fifty pounds to the sliders. The static coefficient of friction was found to be 0.1849 and the kinetic coefficient of friction was found to be 0.1678.

Table J1. Summary of data taken during the clean slider test.

Clean Slider Test		
Load (lb)	Static Force (lb)	Kinetic Force (lb)
0	16.2	15.7
52.1	29.8	28.9
103.4	41.6	38.1
154.7	48.1	45.6
206.1	54.6	49.1
257.5	66.6	62.5

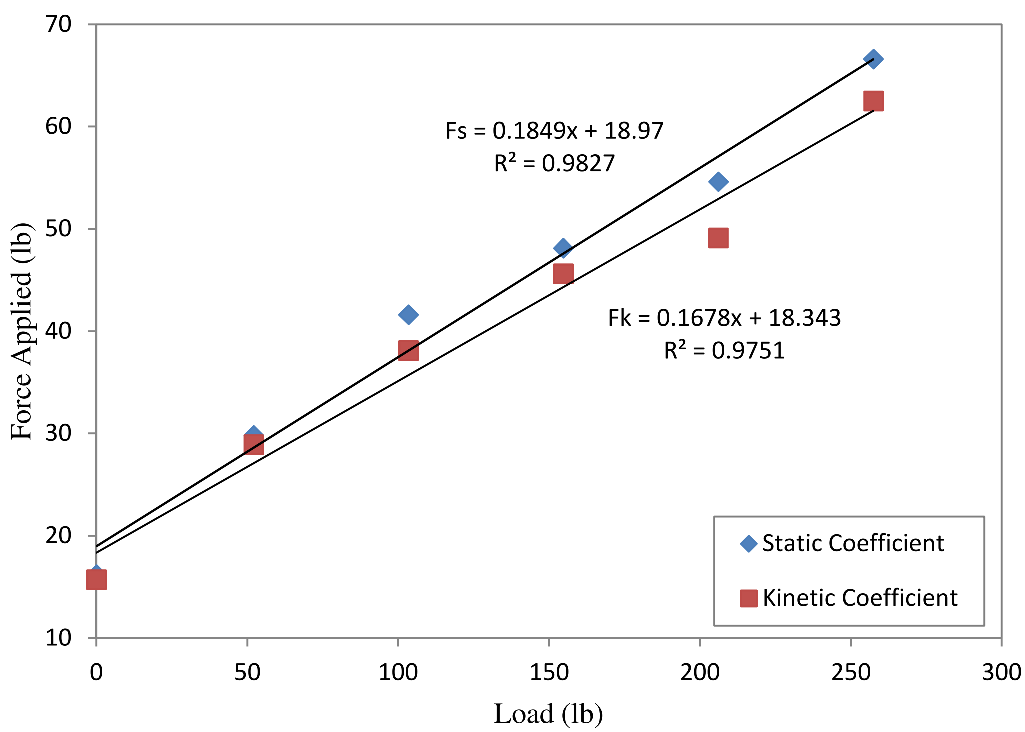


Figure J11. As load on the slider increased, the load required to move the slides increases in a linear fashion. The coefficient of friction is the slope of the trendline.

The second test was conducted simulating dirt and grime build-up on the rails. Sand was poured over the rails during the test to replicate worst case scenario to get coefficients of friction. The weight of the top plate of the slider was neglected for the testing and plotted data is based on payload rather than total weight. Five tests were conducted, each time adding about fifty pounds to the sliders. The static coefficient of friction was found to be 0.1861 and the kinetic coefficient of friction was found to be 0.1836.

Table J2. Summary of data taken during the dirty slider test. Sand war poured over the rails in order to simulate dirt and grime build up that could happen in the field.

Dirty Slider Test		
Load (lb)	Static Force (lb)	Kinetic Force (lb)
0	27.4	20.1
52.1	38.6	35.2
103.4	53.6	44.1
154.7	58.1	53
206.1	61.1	54.1
257.5	80	73.1

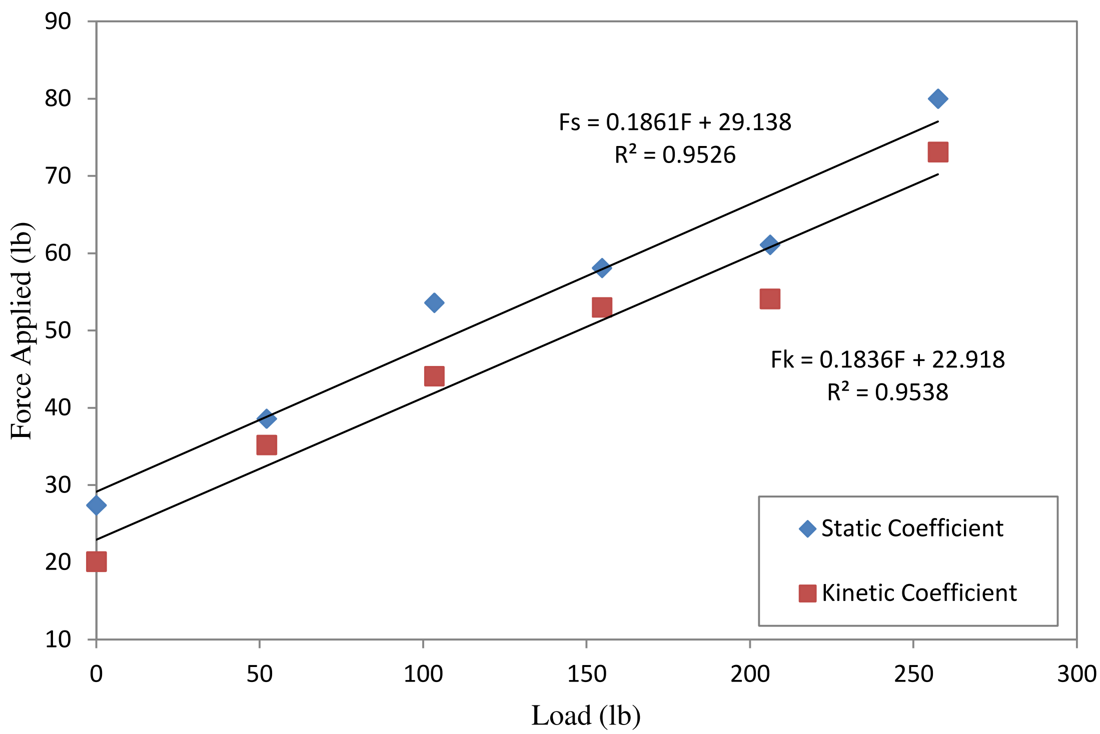


Figure J12. As load on the slider increased, the load required to move the slides increases in a linear fashion. The coefficient of friction is the slope of the trendline.

The top plate of the slider cannot be easily removed, so the total weight or the slider assembly was recorded and found to be 152.2lbs. Using SolidWorks, the volumetric percentage of the top plate was found to be 43.4% of the total volume, making the top plate weigh approximately 66lbs.

Table J3. Summary of data taken during the clean slider test including the estimate for how much the top plate of the slider weighs.

Clean Slider Test		
Load (lb)	Static Force (lb)	Kinetic Force (lb)
66.0548	16.2	15.7
118.155	29.8	28.9
169.455	41.6	38.1
220.755	48.1	45.6
272.155	54.6	49.1
323.555	66.6	62.5

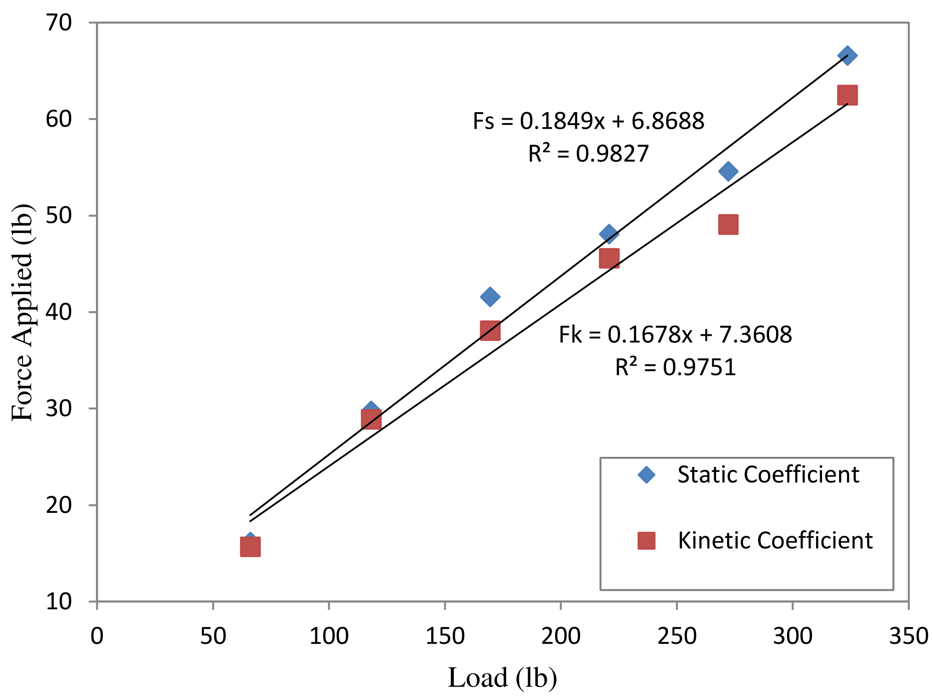


Figure J13. As load on the slider increased, the load required to move the slides increases in a linear fashion. The coefficient of friction is the slope of the trendline. The test was not perfect so the trendline does not pass through the origin.

Table J4. Summary of data taken during the dirty slider test. Sand war poured over the rails in order to simulate dirt and grime build up that could happen in the field. This data includes the estimate for how much the top plate of the slider weighs.

Dirty Slider Test		
Load (lb)	Static Force (lb)	Kinetic Force (lb)
66.0548	27.4	20.1
118.155	38.6	35.2
169.455	53.6	44.1
220.755	58.1	53
272.155	61.1	54.1
323.555	80	73.1

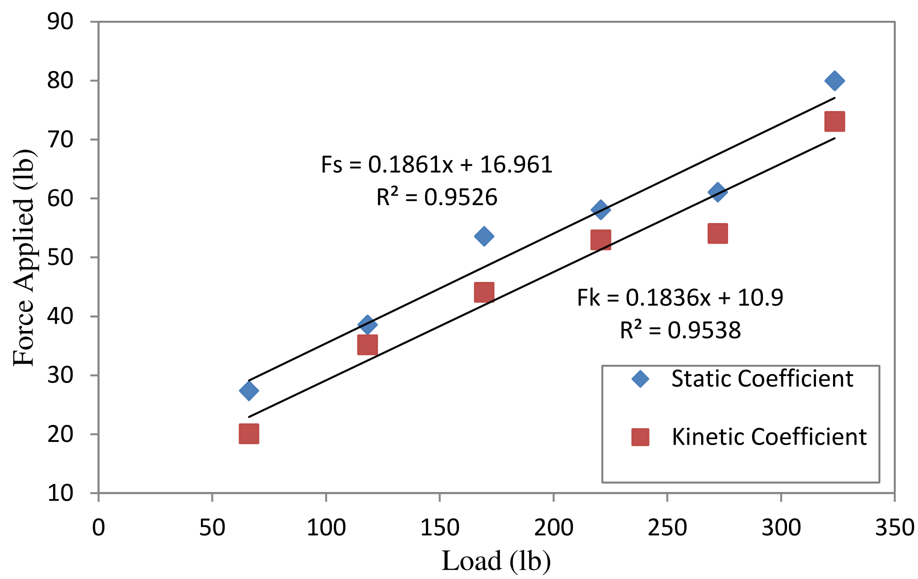


Figure J14. As load on the slider increased, the load required to move the slides increases in a linear fashion. The coefficient of friction is the slope of the trendline. The test was not perfect so the trendline does not pass through the origin.

After conducting the tests, the force required to overcome the friction for both static and kinetic coefficients for both clean and dirty sliders is recorded in Table J5. The test results seem to make logical sense because the clean rails require less force than the dirty rails and the coefficient of kinetic friction was less than that of static friction.

Table J5. Summary of static and kinetic coefficients of friction for both clean and dirty rails.

	Coefficients of Friction	
	μ_{static}	$\mu_{kinetic}$
Clean	0.18	0.17
Dirty	0.19	0.18

Table J6. Summary of force required by the axial expansion of the engine to begin and maintain sliding for both the clean and dirty rail sliders.

To Break Static Friction	
Axial Force Needed by Conventional Engine (lb)	
Clean	2390
Dirty	2405
Axial Force Needed by SoLoNOx Engine (lb)	
Clean	2863
Dirty	2882

To Maintain Kinetic Friction	
Axial Force Needed by Conventional Engine (lb)	
Clean	2169
Dirty	2373
Axial Force Needed by SoLoNOx Engine (lb)	
Clean	2598
Dirty	2843