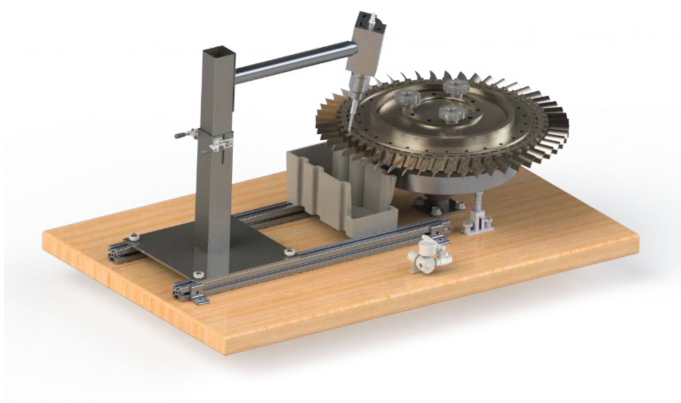
PROJECT:	Medium Duty Gas-Turbine Engine Blade Removal Tool				**PART:**	Removal Tool
TITLE:	Final Design Report				**SECTION:**	ME 430
ORIGINATOR:	J. Syage, E. Greb, K. Field		**DATE:**	11/30/2017	**REVISION:**	A
Reviewer:	J. Hernandez				**PAGE: 1 of**	178

Medium Duty Gas-Turbine Engine Blade Removal Tool



Project Sponsored by: Solar Turbines Incorporated

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Statement of Disclaimer

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1 Introduction

Solar Turbines Incorporated (Solar) designs, manufactures, and overhauls gas turbine engines used in the oil/gas and power generation industries worldwide. The current overhaul and remanufacturing process of engines received from the field occurs at Solar's DeSoto, Texas facility which encompasses the entirety of engine overhauling across all product lines. The overhaul of an industrial gas turbine includes complete disassembly of the engine, inspection and testing of items to be reused for a subsequent service interval, reassembly of the engine, and a thorough post-overhaul run test.

During the disassembly of the engine, compressor and turbine blades must be removed from their respective rotor disk assemblies. In some engines operated in severe environments, hot-section turbine blades become difficult to dislodge from the rotor disk due to corrosion that occurred during the item's service interval. The current process for removing the seized blades involves using hand tools to unseat the blade from the fir-tree bore. A rubber mallet and a punch are used to provide a swift impulse on the blade root; however, occasionally the disk assembly is struck instead of the blade root. The damage caused by accidental disk strike can render a reusable turbine disk unsuitable for future use. This hinders the part's ability to be reinstalled in the engine, requiring the disk to be repaired if possible or scrapped, leading to significant waste product every year.

Increased down time during overhaul, as well as increased interest in reducing the number of disks being scrapped initiated an investigation by Solar Turbines into the process that was most frequently causing disk damage. Collaboration and analysis between engineers and technicians at Solar's DeSoto, Texas and San Diego, California reached the conclusion that a process improvement to eliminate the human factor from disk damage was worth exploring in an effort to reduce disk scrap rates. A lack of uniformity in the blade removal process, as well as a lack of precision tooling across Solar facilities encouraged engineers to ponder possible solutions to the problem at hand. With no in-house solution readily available, Solar determined the project could be contracted to a 3rd party.

To develop a solution, Solar Turbines Incorporated sponsored a team of three Cal Poly Undergraduate Mechanical Engineering students to design, manufacture, prototype, and test a tool for the removal of turbine blades from their respective disk assembly without damaging the disk itself. The tool shall be developed for Solar's industrial gas turbine engines. The tool must comply with all OSHA and Solar Turbines safety and engineering constraints, and will be designed for use in a shop facility. The tool must be adaptable for geometric variances in multiple turbine rotor disk stages, and should require no more than one technician to operate using a maximum of two shop resources. The tool must stand alone on its own fixture, with ease of portability and disassembly in mind. A discrete process shall be developed to assist in the transition from the current blade removal process to the new blade removal process. In doing so, future disk damage can be prevented by developing a tool and process used to replace the current rudimentary techniques for blade removal.

1.1 Management Plan

Each member of the team has an equal say and authority in the project; however, each member has assigned duties to ensure smooth project development. These duties are listed below in Table 1.1.1 for each team member. Each member is responsible for completing these tasks; however; they can be changed as new issues arise.

Table 1.1.1. Team Member Designated Responsibilities

Team Member	Responsibility
Kellen Field	• Maintain the team's travel budget • Maintain the team's material budget (2nd quarter) • Keeps a day-to-day log of expenses and expendable budget (stores receipts) • Places orders for required materials • Head of manufacturing
Elliott Greb	• Maintain information repository for team (e.g. team binder, google docs site, etc.) • Team note takers during relevant meetings with sponsor and advisor • Verifies logbook information
Jacob Syage	• Be the main point of communication with sponsor • Facilitate meetings with sponsor • Manages campus resources • Maintains communication with outside experts/consultants • Direct point of contact for advisor • Head of testing

2 Background

Solar Turbines Incorporated is a subsidiary company of Caterpillar Incorporated. Headquartered in San Diego, California, Solar Turbines produces mid-scale industrial gas turbine engines that are used primarily in the oil and gas industries as well as the power generation industry. Solar has multiple engine family lines that range from the smallest Saturn 20 engine which produces 1590 hp, to the largest Titan 250 engine that produces 30,000 hp.

The bulk of Solar's engineering and design work is completed at the Harbor Drive campus in San Diego, which also houses the majority of the manufacturing for the six engine lines. The Kearny Mesa location in east San Diego produces Solar's gas compressors and is the location of final engine assembly and testing before shipment. The current overhaul location for engines received from the field is in DeSoto, Texas, where Solar has a large facility specializing in the disassembly, reassembly, and distribution of previously used turbine engines.

When an engine is received by the DeSoto facility, the engine is axially disassembled starting with the hot section, known as the turbine and exhaust section, and ends with the cold section, known as the compressor and inlet section. As the engine is disassembled, each turbine rotor stage is removed, placed on a pallet, and tagged for proper organization and tracking throughout the overhaul period. The turbine disks of interest are worked on as an independent assembly separate from the engine casing itself. Technicians lay the disk assembly on a table or jig to affix the disk to a flat surface for ease of working. Turbine rim seals are removed first, if equipped, followed by the removal of the first damper to begin the blade removal process. Once the rim seals and dampers are removed, the only force holding a blade in place is the interference of fit due to fatigue or corrosion. The severity of the friction in the turbine blade fir tree is primarily dependent on the operating conditions of the engine throughout its service interval. Frequent restarts, thermal cycling, and corrosive environments lead to unfavorable operating conditions throughout a service interval. This can cause the blade root to distort under rotational stresses leading to permanent deformation over time. Deformation, along with corrosion from combustion gasses and extreme heat, cause blades to remain bound in the disk assembly during the overhaul process [1].

Most stage one and two turbine blades are exposed to the most extreme operating environment within the engine. Although constructed from enhanced materials to operate under harsher conditions than the downstream gas producer (GP) and power turbine (PT) stages. However, the rotor disks can be reused if they are undamaged, with no deformation, and no apparent fatigue wear or cracking. The rotor disks have high manufacturing costs compared to the blades and it is desirable to reuse a disk for a subsequent service interval in an overhauled engine.

The current process for removing stuck blades in a turbine disk assembly at Solar Turbines is very rudimentary and improvisational. Technicians use common hand tools such as a rubber mallet and punch to impact the stuck blade along its root, attempting to free the blade from the disk. If the object doing the impulsive loading slips or is misaligned, the rotor disk can be struck and damaged. The damage can range from a small required repair to the scrapping of an entire disk, potentially costing Solar Turbines hundreds of thousands of dollars per year in avoidable costs. There is currently a standardized process at Solar for removing stuck blades but it is subject to variation and human error. Additionally, there was minimal knowledge on the force required to remove the blades that become lodged in the fir tree slot. Techniques for removal vary from technician to technician as they feel fit, and is largely based on technician preference.

After completing research on current industry solutions, two recent applicable patents from competing corporations were found that address a similar situation at their respective companies. Patent US2015/0218948 was published August 6, 2015 on behalf of Siemens Energy Incorporated and describes a “Turbine Engine Blade Removal Apparatus and Method” [2]. This device is a mechanical punch with varying angle and large flat tip used to impact stuck blades while protecting the disk assembly. This solution tackles a problem very similar to Solar's and requires minimal assembly and setup time. The tool holds the punch at the desired broach angle within a guided bore to prevent the punch from wallowing or changing impact direction which could allow it to strike in an undesirable place. From the limited drawings in the patent, this tool appears to focus on the removal of compressor blades rather than turbine section blades; however, the product design appears to work in either case.

A second patent, US2015/0128417 was published May 14, 2015 on behalf of Mitsubishi Hitachi Power Systems Americas Incorporated. The patent outlines a "Turbine Blade Removal Tool and Method" [3] that is used to press out seized turbine blades from a gas turbine engine. This device mounts to the rotor disk itself using a set of bolts along a curved flat surface that mates to the disk. It uses pneumatic power to drive a piston with an end contoured to an exact fir tree geometry into the blade root, removing it from the rotor disk. The tool appears to have varying broach angle capability and can remove multiple blades before being uninstalled and reinstalled at a different position on the disk. Minimal human input is required, and the device is fixed to prevent the striker from impacting the incorrect location which prevents disk damage.

The Cal Poly students visited Solar Turbines in San Diego on February 10, 2017 to perform analysis on a second stage turbine disk assembly that included many seized blades in order to understand the current process for removal and the desires of Solar Turbines for project development. The test rotor disk assembly was received after one service interval of use. It is unknown if the disk or blades performed a service interval prior to the most recent service. Interviews with engineers and shop technicians were performed to understand the full scope of the current approach, and use the desires of all possible tool users to guide the design process.

Testing during the visit included the disassembly of the test rotor from start to finish. Beginning with the removal of the rotor's blade retention wire, it became obvious that the tools used near the disk posed great threat to the disk and blades during use. Rubber mallets and punches were used to pound the retention wire out of place. The wire is a single use item that is used to prevent lateral translation of blades and dampers once installed. It acts similar to a snap ring, but does not maintain tension during use. Rather, it is clamped in place. Once removed, the blades and dampers could be freed from the disk. It was found that approximately every third blade became seized in the fir tree broach and could not be removed by hand.

The seized blades were removed using a half-pound ball peen hammer and center punch with repeated light taps. Significant force was not necessary; instead, repetitive light force helped free the stuck blade from the broach slot. Blade removal rate was rapid during testing. Countless strikes to the disk occurred, demonstrating the ease of damaging the disk using current techniques. The group learned that it is unlikely that all of the blades in a rotor disk will be seized; however, those that are seized did not need to be impacted with great force.

Discussions with shop technicians and engineers led to conclusions that rapid blade removal is critical if shop technicians are expected to adopt the new tool and process. Ease of use and adaptability to multiple engine stages will encourage users to refrain from using old methods and embrace the specialty tool for blade removal. It was determined that the tool shall be used in a shop facility, not in the field, and that the device should stand alone on its own fixture. Solar engineers and the Cal Poly team agreed that the device should support the entire weight of the rotor disk assembly being disassembled, and that the tool should require minimal user input force. Concern of injury caused by repeated motion guided the discussion of using a shop resource such as pressurized air and electrical power to provide mechanical advantage to the tool. This would allow for a reduction in required user force application in the removal process.

The Cal Poly group reviewed the tests that were completed alongside Solar engineers during the visit and began an ideation process for tool development. A down selection procedure followed initial brainstorming and led to the foundation of the selected device. Project advisors at Cal Poly reviewed the ideas that were generated and agreed that the final idea selected will meet and exceed the expectations of Solar Turbines and the project problem statement requirements.

3 Requirements and Specifications

Prior to any detailed design of the tool, the group established a set of specifications for the tool that will ensure the fulfilment of the needs of Solar Turbines Incorporated. The process of specifying customer needs and desires into distinct specifications was accomplished using a Quality Function Deployment (QFD) matrix that is found in Appendix A.

Although the QFD matrix looks complex, it is an intuitive and objective way to translate the customer's requirements into quantitative engineering specifications that can be tested and evaluated. One of the strengths of the QFD matrix is that it also takes into consideration competing products in the marketplace. Although there are no marketed competitors, there are current patents for blade removal tools from global competitors that were included in the matrix.

Due to the highly variable nature of operating conditions, the design loads cannot be defined theoretically without testing. Therefore, the group conducted testing on a rotor assembly with seized blades to determine the force magnitude required to free a seized blade. The group found that the blades needed short, small, impacts to shake the blade loose rather than the initially assumed continuous pressing force

The majority of the damage to the turbine rotor assembly from the current blade removal process originates from the lack of directional tool guidance for the item that impacts the blade root. In order to eliminate this risk, the tool will not rely on the user for directional constraint during impacting. Force exerted by the user will remain under 50 pounds to prevent operator injury, and will be done in a manner that can be easily controlled through the use of a lever, wrench point, or trigger. The use of a shop floor resource for mechanical advantage, such as the use electrical or pneumatic power to provide the necessary rapid impact in a controllable and guided manner, will be used in the tool. To prevent user injury from lifting, the tool will be designed to be lifted by overhead cranes which have a safe load limit of 4000 lb.

Although no explicit safety regulations have been outlined by Solar Turbines Incorporated, the group will design the tool to be in compliance with applicable OSHA regulations, especially those governing repetitive tasks or heavy lifting from the operator [4]. The safety and use specifications shown in the specification matrix were derived from analysis of the estimated tool requirements placed on operators. By establishing expected tool requirements, the group was able to correlate these to existing OSHA regulations to define the engineering use specifications.

The QFD matrix was used to develop specifications that define tool operation, such as blade removal rate and required tool maintenance. Although not explicitly defined by Solar Turbines, values used were determined by the group to be reasonable design goals to shape the tool development process. It was found that the current blade removal method can remove 100-140 blades per hour because most of the blades do not require additional attention during removal and can be freed from the broach slot by hand. Due to the infrequent nature of bound blades, the tool will be designed to remove only one blade per impacting cycle in order to limit complexity and possible damage to the disk. Designing the tool to remove more than one blade per cycle was deemed not necessary because the seized blades are not usually in continuous segments, but instead, scattered around the entire perimeter of the disk. Removing blades faster than the current method will encourage more frequent use of the specialty blade removal tool by technicians. This will also help to reduce downtime during overhaul, increasing company output. The group determined that tool maintenance intervals be kept at a minimum to prevent disrupting the blade removal process. The tool will be designed to operate for one complete rotor disassembly process before requiring any maintenance or tool disassembly. The tool will be designed so that regular maintenance intervals will take less than one hour.

Upon visiting the Solar Turbines facility, the group surveyed the shop floor technicians who will be the primary tool operators to verify that their desires align with Solar Turbines' Mechanical Engineers, as well as those of the Cal Poly student group. In doing so, the group will be able to develop a tool that reduces the amount of required training needed to teach technicians to operate the device, while also developing a product that the operators desire to use. In conjunction with a low frequency maintenance schedule, the tool life will be designed to achieve a life corresponding to the disassembly of one hundred rotor disk assemblies. Simplicity of the tool design is key, and tool manufacturing time will be designed to last less than one standard eight-hour shift. Any items on the tool that will require scheduled replacement will be designed to use off-the-shelf parts that can be easily ordered in large quantities. All of these factors will work together to minimize shop down time, improve the efficiency of overhaul compared to current procedures, and will encourage adoption by shop floor employees.

The specification matrix below has been altered from its original form in the project proposal to reflect changes the group made in conjunction with Solar Turbines.

Table 3.1 Specification matrix that organizes design criteria and assess risk.

Spec. #	Parameter Description	Requirement or Target	Tolerance	Risk	Compliance
1	Rotor Damage	None	Go / No-Go	H	A, T, I
2	Blade Damage	Minimal	Go / No-Go	H	A, T, I
3	Human Driven Force	50 lbf	Max	M	A, T
4	Tool Cost	$4,000	Max	M	A
5	Blade Removal Rate	140 blades/hr	Min	M	A, T
6	Tool Assembly Weight	4000 lbf	Max	L	A, I
7	User Approval	80%	Go / No-Go	L	T
8	Service Interval	1 Disk Overhaul	Min	L	T
9	Tool Lifetime	100 Disk Overhauls	Min	L	A, T
10	Service Time	1 Hour	Max	L	T
11	Shop Resources	2	Max	M	A
12	Training Time	1 Hour	Max	M	A
13	Manufacturing Time	8 Hours	Max	M	A, T

The specification matrix above outlines the individual specifications the group has determined to be the most critical in guiding the design process. The risk column denotes which specifications are low (L), medium (M), or high (H) risk of not being met or exceeded in the design phase and will require significant refinement. The compliance column outlines how the group will determine if the design has met the specification, through analysis (A), testing (T), or inspection (I). The highest risk objectives are damage to the rotor and blades due to the forces they will be subjected to.

4 Design Development

4.1 Idea Generation

Idea generation began with group brainstorming sessions in the senior project laboratory. Due to our project's complexity, brainstorming a single solution in its entirety was not a realistic approach to generate multiple ideas. Instead, the problem solution was broken down into different, specific aspects of the final solution that would be considered separately. Brainstorming took form in multiple ways. Each team member brainstormed one aspect of the project individually without communication to prevent a loss of "train-of-thought". Then, each member rotated ideas and built off the other's. All members collaborated and discussed after each member had an opportunity to consider everyone else's ideas.

First, the team considered the task of fixing the blade removal device to a table, and how the turbine disk would be rigidly mounted to allow for a well-supported work-station. Second, the team considered what the driving force for blade removal would be. Lastly, the team considered how the tool would ensure repeatability in the removal process. Precision and accuracy of the strike for removing blades is of utmost priority to ensure no misstrikes occur that could damage the disk. During this time, the team focused on the *quantity* of ideas for each aspect of the project. Later, ideas from each aspect were combined to allow for a wide range of solutions where the best solutions were chosen and developed upon.

Each member drew their respective ideas on a white board with rough geometry to visually demonstrate their idea to other members to help clarify sizing and logistics. Again, each member built upon each other's ideas to further refine the potential solutions.

After completing the brainstorming phase, a large number of ideas for each individual project component were created. Next, each selected solution related to a specific project area was combined with ideas from another area to allow for the creation of solutions that may not have been previously considered. This allowed for the analysis of the viability of each idea based upon their relationship to other parts of the tool. Conflicts between ideas that would prevent further development were also found, helping in the down selection process. From these, the *quality* of each was considered based upon the required specifications and feasibility outlined previously by the Cal Poly students and Solar Turbines engineers.

4.2 Ideas Generated

After the idea generation phase was complete, the final solution ideas were built upon based on their *quality.* Five prominent ideas were generated that required additional consideration before continuing the down selection process for the final design.

4.2.1 Table and Punch Design

A design that proved to be a viable concept during the ideation phase was a table and punch design. The table would act as the fixture for the disk assembly and would support the disk by using guide pins through the rim seal mounting holes. The turbine disk would be vertically lowered and fixed to the table before removing the blades. The forcing mechanism to remove the blade in this iteration incorporated a purely mechanical system, a pneumatic system, or hydraulic system to drive a punch into the blade root, forcing the blade from the fir tree slot. The punch was designed to rotate around the table while the disk remained fixed in order to reach each fir tree slot. No adaptability to different sized rotor assemblies was possible with this design. The punch guide on this system would rotate to multiple angles to account for the variability in broach angles across different engine rotor stage configurations.

Figure 4.1. Table and punch design isometric view.

4.2.2 Rim Seal Design

The clamping press shown below would use the rim seal bolt holes to locate the tool, and sandwich the disk between the upper and lower plates (green and yellow respectively). The lead screw and pressing head (red) would move down by turning a hex or square head at the top of the shaft. This would enable the technician to use either a wrench or impact driver for mechanical advantage. The wider head and mechanical advantage provided by a lead screw would enable this design to press out multiple blades at a time, increasing the speed of the removal process. Unfortunately, because most of the blades do not require significant force during removal, and only a small percentage of the blades are unable to be removed by hand, this design would require an excessive amount of set-up time for each use, drastically slowing the process. A benefit of this design is that it could be used in the field; however, discussions with Solar Turbines engineers determined that the development of a tool that worked in the shop facilities and the field was not necessary.

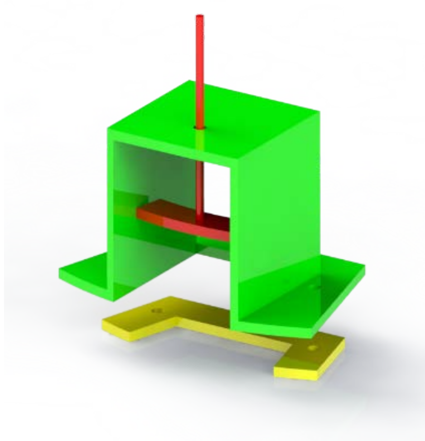

Figure 4.2a: Rendering model of the Rim Seal Mounted Clamping Tool.

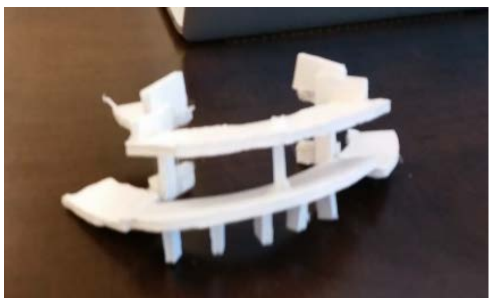

Figure 4.2b: Basic foam core model of the Rim Seal Mounted Clamping Tool.

4.2.3 Hand Guided Punch

The Handheld User Guided Punch design was developed with extra consideration for portability and speed. The user only needs to have access to standard shop air or electric power and can quickly aim and trigger the captive bolt to strike out individual blades rapidly. The non-marring tip made of a polymer or soft metal would help to limit possible damage from misaimed actuation. The downsides to this design are similar to the current method in that there is still a significant possibility of missing the blade root during activation, leading to rotor disk damage. Additionally, without precise control over the angular orientation of the tool, the blade can bind in the broached slot, prevent removal, and cause damage to both blade and rotor.

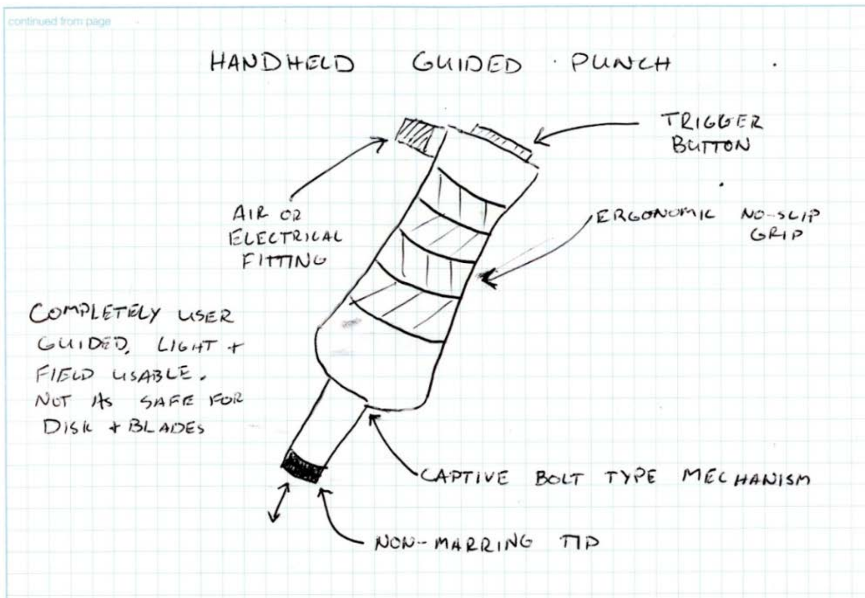

Figure 4.3: Hand drawing of the Handheld User Guided Punch design.

4.2.4 Vertical Table and Separate Clamp Tool

Growing on the turret-style punch idea mentioned previously, a space-constrained variant was developed that turned the horizontal style punch into a vertically mounted assembly. The table mounted punch was removed, and instead, this device would be combined with the rim seal mounted press discussed previously. This variant had the potential to save significant shop space with easy access to the blades as they are removed, however, there were significant drawbacks to the design. Safety and part damage concerns relative to lifting a rotor disk vertically, reduced the viability of this design. Not only is it difficult to lift a rotor vertically, but if lifting equipment failed, the rotor would be suspended significantly higher in the air than with the horizontal model. Supporting the structure of the disk mounting hub would require significant strength increase to support the assembly in a cantilever setup compared to a horizontally mounted setup. Although space could be saved with a vertically mounted model, the drawbacks compared to the horizontal model removed this design from the potential pool of final designs.

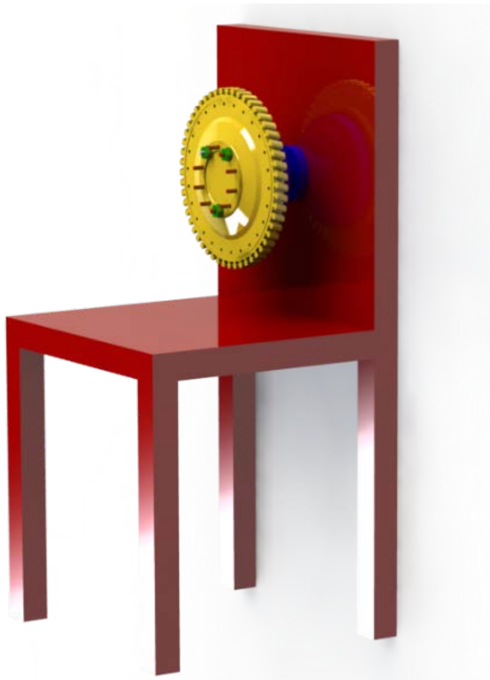

Figure 4.4: Vertical rotor mount design for space-constrained shop area.

4.2.5 Guided Hand Punch

Further developing the table and punch design, a disk mounted precision punch guide was developed to help guide a hand punch used for blade removal. Rather than using a pneumatic system or table-mounted punch, the punch-guide would sit on the tie bolt guide pins and would have holes machined to a specific broach angle that reside over each individual blade root. The user would insert a center punch into these holes and the punch tip would be guided precisely to the blade root. This would constrain the punch to impacting only the blade, eliminating the risk of impacting the disk. This design would require significant user input, where the operator must physically impact each blade using a punch and hammer. There would be no adjustability with this design, and a specific punch guide would have to be constructed for every stage rotor disk for every engine model. The amount of user input required, lack of uniformity in the process, potential blade damage from impacting, and the lack of adaptability to multiple engine families reduced the viability of this design. Although relatively simple to manufacture, this tool would not improve the current blade removal process significantly enough to warrant development of this design.

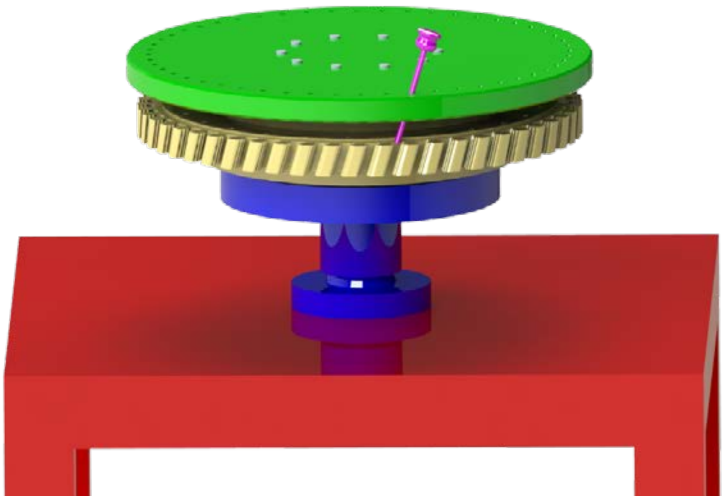

Figure 4.5: Assembly model of the disk-mounted guided punch design.

4.3 Selection Process

The down selection process to determine a viable final design was greatly dependent on the objectives outlined in the specification matrix in Table 3.1 as well as the Pugh Matrix in Appendix B. The Pugh matrix helped *quantify* the viability of each semi-final design to find the best possible solution. The matrix was created by placing general key criteria for a successful design in a vertical column that were rated based upon importance, placing the current solutions and the different design choices in a horizontal row, and then rating them based upon relative importance in a matrix fashion. Each design was then summarized and compared to find the best path to move forward with. Additionally, a few parameters contained the highest priority with a Go/No-Go tolerance including ability to avoid disk damage and user approval.

In the beginning of the section process, the group believed that a driving design factor for the project would be its adaptability and versatility among multiple engine configurations. However, Solar Turbines later requested the design efforts be specifically focused towards their medium-sized, industrial gas turbine engines. The advantage to all five potential designs is they are easily redesigned to be used on multiple engine configurations.

User approval remained a high priority during down selection. It is expected that the tool to be developed will be primarily used by shop technicians in the overhaul facilities. To satisfy the users, blade removal speed and ease of use had a significant impact on product viability. The team discovered that overhaul technicians can remove blades at a more rapid pace than anticipated. It was determined that if the selected design did not meet or exceed current removal speeds, the design would not be well-liked by users and would reduce productivity. For this reason, the clamp tool design outlined in Section 4.2.2 was ruled unfavorable due to the repeated repositioning of the tool required during blade removal.

An initial product goal during ideation was to have a tool that could be used in the shop as well as in the field for on-site repairs. This drove a design that was lightweight and small for ease of portability. Initial designs such as the hand guided punch and rim seal clamp were preferred before removing the need to develop a tool that worked in both locations. Once it was decided that focusing the development efforts on a design that performed the necessary requirements strictly within the overhaul facilities, the team pursued a design similar to that outlined in Section 4.2.1.

4.4 Final Preliminary Design

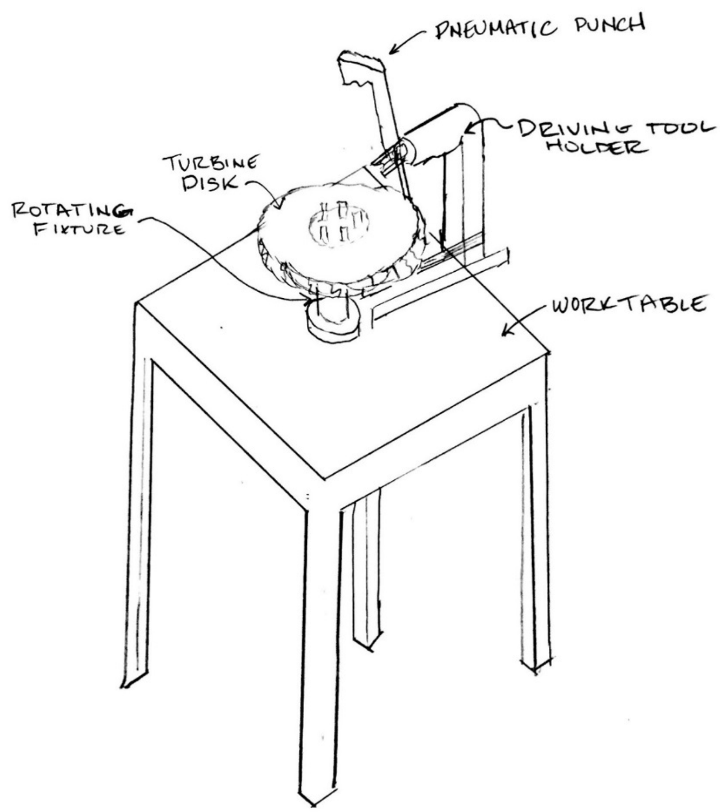

Figure 4.6a: A preliminary sketch of the table based tool with rotating disk mount.

Figure 4.6b: A preliminary rendering of the table based tool with rotating disk mount.

Following the down selection process outlined in the Pugh matrix seen in Appendix B, the table based tool with rotating disk mount and adjustable driving tool was selected. In the rendering above, the disk assembly (yellow) sits on a rotating base and table (red) while the driving tool holder (green) can slide on rails to accommodate varying diameters of different rotor stages. The head of the tool holder also rotates to accommodate varying broach angles. In this preliminary design, the driving tool is a standard air hammer (grey) fitted with a non-marring tip to provide rapid impacts at a precise location.

To use this tool, a shop crane will lift the disk assembly off the engine being disassembled and onto the hub support plate of the tool mounted to the table. The hub will have guide pins used to locate the disk using the tie bolt holes specific to each engine type. The curvic teeth on the rotor disk will be protected by a rubber pad that sits on top of the hub plate where the teeth will lay during blade removal. Once set onto the hub, simple wing nuts will be used to secure the disk to the hub plate using the threaded guide pins. A non-marring washer between the disk and wing nut will protect the disk assembly from damage during affixation to the hub. The driving tool holder (green) will be moved linearly to the correct radial location of the blade root, and then the head rotated to the correct broach angle. Both will be secured by splined locks to prevent movement during the impacting process. The disk will be hand checked for loose blades that can be easily removed without the use of the tool prior to beginning the impacting process. The table will have a cushioned top to protect blades that fall from the disk to ensure they can be captured without damage.

For any seized blades, the disk will be rotated using the hub fixture into the desired location below the pneumatic punch for blade removal. The rotating hub will be lockable at any location to prevent the rotor from moving out of angular position once set by the user. This will ensure the pneumatic punch will impact the desired point without risk of disk movement during the removal process. Once all blades are removed, whether by hand or pneumatic punch, the disk will be removed from the work table to enter further overhaul stages. The pneumatic punch will be operated by a simple switch mechanism and will employ industry standard replaceable tips. Using an off-the-shelf standard size and geometry for punch tips will ensure Solar can easily source tips as needed.

Construction of the tool will focus on simplicity using Cal Poly and American Society of Mechanical Engineers (ASME) guidelines for designing for manufacturability and assembly (DFMA). The base table will be made from an off-the-shelf steel work table to ensure robustness and long term durability. Using a steel table will allow for welding to the table as well as using threaded fasteners to attach any components needed to the base. Steel also has high resistance to chemical degradation. Solar Turbines uses harsh solvents throughout their shop facilities which can react to and destroy composites. A composite table would pose imminent safety threats and could be compromised in the presence of many common shop chemicals.

The rotating hub assembly will be constructed from steel and machined to proper geometry. The base of the hub assembly will be welded or bolted to the table, and will rise to suspend the disk mounting plate slightly above the table surface. The hub will be supported by two bearings and a center bearing cup. The disk mounting plate will rotate freely from the base and will be affixed to the upper bearing and hub shaft by an axial retention flange nut.

The disk mounting plate will be made from aluminum stock for ease of manufacture and will be machined to fit the curvic tooth diameter of the engine of greatest interest in Solar's lineup. The mounting plate will be lined with an EPDM rubber surface to protect the curvic teeth during rotor disassembly by preventing contact damage from the aluminum surface. Threaded guide pins will be installed at opposing locations to locate and affix the turbine rotor disk to the mounting plate. The pins will be threaded into precise locations in the mounting plate that correspond to the bolt pattern of the tie bolts on the rotor disk. Different mounting plates will be used for different engines as the tie bolt configuration changes, and the plates can be easily installed by removing the single bearing retention nut. The guide pins will be fully threaded, allowing for the installation of a non-marring washer and wing nut on top of the disk assembly to fix the disk to the plate.

The pneumatic punch assembly used to drive the blades out of the rotor disk will be mounted on guide rails, allowing the punch to slide radially relative to the disk. This will enable the user to slide the punch head out of the way when mounting or dismounting the disk to the hub. It will also allow the tool to adapt to differing diameters of disks and accommodate tolerance stack in manufacturing.

The punch head unit will be a splined, variable angle device, allowing for adaptation to different broach angles with positive stops for the user to lock the punch into. This will allow for securing the punch at the correct broach angle for operation on different rotor disks. Using splines rather than set screws will allow for positive lock engagement that will prevent wandering of the device from the set angle. This will prevent misalignment during blade root striking.

The pneumatic hammer will be fixed so that the user must only control a switch to operate the device- no manual input needed. This will improve ergonomics for the user as it will reduce repetitive motion needed during operation. It will also reduce the fatiguing load of holding a pneumatic hammer in place for the user, by transferring the loads of operation into the tool and table instead of the user.

In order to minimize down time if replacement parts are needed or new configurations are created, the use of parts requiring CNC machining will be minimized. Serviceability is a concern when designing shop tools, so the design will emphasize the use of reusable or easily replaceable components. The driver head must be constructed from a non-marring material and will likely be sourced as a standard nylon headed chisel from the automotive body repair industry. Using industry standardized punch tips will allow for easy sourcing of replaceable components. The base table will be approximately 36” by 36” in work surface area and will sit at a height that fits the 95th percentile man and woman using the most recent American anthropometric data.

5 Description of Final Design

5.1 Overview of Final Design

As seen below, the tool will be built on a wooden butcher block in order to better serve as a prototype that can be transported around shop areas for testing. The subsystems are broken down as follows; first the hub and hub plate, followed by the locking mechanism, and finally the sliding tool holder. Operation of the tool follows a similar flow path, first the disk is loaded onto the hub and secured. Then the locking mechanism is adjusted and secured, followed by the alignment and tightening of the sliding tool holder. The last step is for the operator to don protective eyewear and actuate the impact tool via the pneumatic hand valve mounted to the table. The plastic bin will catch the freed blades and allow the operator to remove them for inspection. Further operational details, design analysis, component selection, maintenance, and safety considerations are addressed in each sub system following this overview. Due to the prototype nature of the tool, most adjustable features do not have set lock points, but instead, allow for full range adjustability to accommodate manufacturing tolerances and fine adjustment during testing.

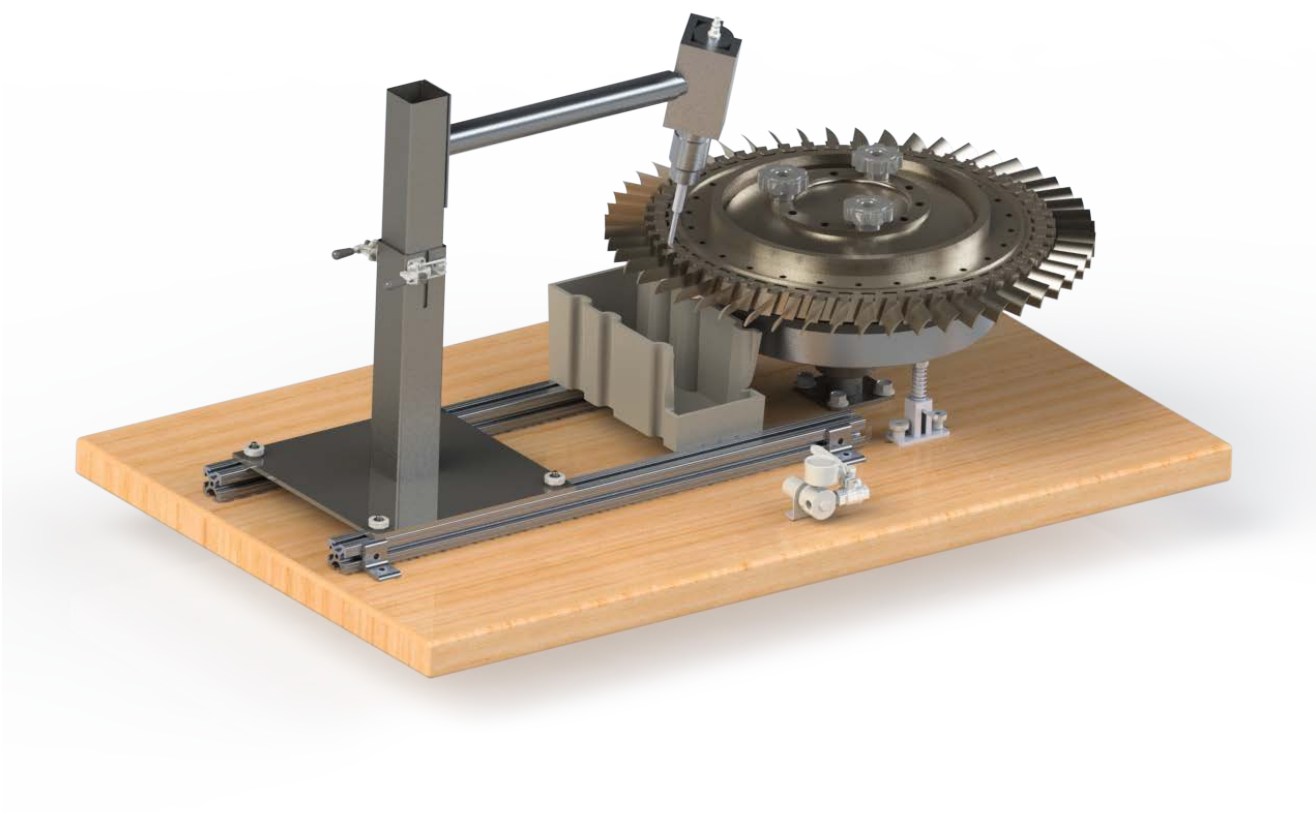

Figure 5.1.1. Isometric Render of the Final Design.

5.2 Rotating Hub Assembly

Figure 5.2.1. Rotating Hub Assembly.

The turbine blade removal tool hub assembly will provide the support structure for constraining and locating the rotor disk underneath the linear impact punch tool. The assembly consists of a five-lug two-bearing hub that will be mounted to a wood base structure. Fixed to the rotating hub will be a series of plates used to locate the rotor disk in a specific orientation. A spacer and precision machined hub-plate will be used to mount and clock the disk in precise, controlled, angular displacements corresponding to the angular change between blade root centers underneath the punch tool, rotated and locked into place manually by the operator. The rotor disk will be constrained axially to prevent movement of the rotor disk during impacting using fasteners that require no specialty tools for installation and removal. The hub assembly was designed with ease of use in mind by eliminating the need for the user to align the disk under the impact tool between removal of individual blades after initial tool setup with a specific rotor disk. A lock assembly to be discussed later in the report will stop rotation of the hub assembly during tool operation. This will prevent accidental disk strikes by the pneumatic impact tool by constraining the hub from rotating out of a desired impact location. Each component of the sub-assembly was analyzed to ensure sufficient robustness and strength during extreme and unlikely operations outside of normal design criteria. Detailed descriptions below exist for each component and engineering analysis of each item can be found in Appendix E3.

5.2.1 Butcher Block

Figure 5.2.1.1. Isometric Render of the Butcher Block

To allow for easy prototyping of the turbine blade removal tool, all subassemblies will be supported by a 48” x 30” x 2.25” maple wood butcher board to enhance mobility of the assembly while ensuring strong base support. A butcher block sourced from the culinary industry will be utilized for the base structure, providing a cost effective and versatile platform to fix all subsystems of the turbine blade removal tool. Maple was chosen as the desired wood type for the board due to its high hardness, low warpage and shrinkage rate, and machinability. Maple wood can be cut using simple power tools and resists splitting during precision cuts.

Figure 5.2.1.2. Bottom Render of the Butcher Block

To mount the hub assembly to the block, a 1.80 inch thru hole will be cut using a ShopBot CNC router at a centered location between the narrow sides 37.0in away from the right datum to allow for the centering of the hub spindle. Four thru holes created in a similar manner will surround this center hole to allow for bolting of the hub assembly to the board itself using grade 5 flange bolts and nuts tightened to a specified torque outlined in Appendix E6. On the bottom of the board, a 5.00in x 5.00in pocket that is 1.25in deep with .375in rounded corners will be created to recess the flange nuts below the bottom surface of the block, allowing for a flat bottom surface. Biasing the location of the hub assembly to one end along the length of the block will allow for a significant increase in travel distance for the punch tool assembly, enhancing the user's ability to mount and dismount rotor disks without tool interference. The hub lock assembly will mount to the butcher block in pockets with 10° of variation to allow for adjustability for different rotor disks. The angular distance between fir tree slots in the disk of interest is 6.96°, and a lack of angular relationship between the fir tree slots to other disk features requires the clocking and locking of the hub assembly to be variable by at least this amount to allow the user to setup the tool specific to the disk being disassembled. Carriage bolts installed through the bottom of the block in 5/16in slots recessed in a 5.00in x 2.00in x 1.25in pocket with .375in rounded corners will be used to mount the lock in the slots. Use of carriage bolts in a slot will allow the lock base to be adjusted to the location required to set the disk, punch, and lock in the correct timing and then fastened by thumb nuts above without risk of the fastener spinning below the board surface in its slot. Slots will be cut using a 5/16in center-point end milling cutter and pockets will be created from a .375in center-point end milling cutter on the ShopBot CNC router in the Cal Poly Machine Shop.

Deflection, bending stress, and stress rupture calculations for simply supported and cantilever loading of the base board using the calculated mass of all components to be mounted were performed to ensure the board will adhere to all operating requirements under adverse and extreme loading. Maximum deflection under extreme central loading conditions was calculated to be 0.033 inches, while maximum deflection under cantilever loading was 0.534 inches. The loading cases for the basis of the calculations would be extremely unlikely during tool use. Provided these cases existed, deflections remained minimal. To ensure a fracture failure would not occur during baseboard loading, rupture stresses under the previously outlined loading conditions were calculated. A maximum rupture stress of 1430 psi under cantilever point loading falls significantly below the rupture stress limit for Maple wood tabulated to be 15,800 psi [6].

It is not expected that the baseboard will require any routine maintenance and was designed with safety in mind. Maple wood resists intrusion by harsh chemicals and will not degrade as fast as other less dense wood types. Using a high density wood such as maple will allow for a heavy and stable tool base that will weigh approximately 93.9 lbs. This allows the overall tool center of gravity height to be reduced due to the quantity of heavy components mounted above the table during operation. Although large in size, the tool will still be portable by two persons when subassemblies are removed, allowing for easy transport for testing and assembly in non-shop locations.

5.2.2 Trailer Hub

Figure 5.2.2.1. Isometric Render of the Trailer Hub.

To provide stable rotor disk rotation for impacting tool alignment with the disk, selection of a commercially available hub-spindle assembly sourced from the industrial trailer industry was determined to be the best, most cost effective solution for constructing the base of the rotating hub assembly. Use of a hub designed for a light-duty industrial trailer provides a robust rotating structure designed for large operating loads under varying conditions over the course of decades of use. Loads exerted by the tool on the trailer hub selected for use are far less than the design loads that the hub is rated for. The preliminary tool design provided a basic model of a hub that could have been constructed out of raw stock steel and commercially available bearings, however, the cost and time required for production far exceeded that of sourcing a purchasable part.

The trailer hub selected is the 1750 lb five-lug hub and spindle produced by Tie Down Engineering and distributed by Northern Tool Inc. This hub features a 5in x 4.5in industry standard bolt pattern on its flange surface and is rated for 1750 lbs of load when mounted in a cantilever configuration with 7.75 inches of lever arm from its designated mounting point. The spindle is constructed of mild steel and is turned to provide a smooth surface for the two bearing races. The hub is supported by two back-to-back 1 1/16in tapered roller bearings that constrain the hub to one degree of freedom under pure rotation. The hub is constrained axially along the spindle length by an axial retention thrust washer and castle nut that fixes the flanged surface to the bearings and spindle. The flanged hub surface is made from low-carbon steel and is designed for wheel mating. The flange includes ½in- 20 x 1.25in wheel studs pressed into the flanged face. These studs will be replaced by ½in- 20 x 3.00in studs for use on the turbine blade removal tool. Although not a machined precision surface, the bolt flange is sufficiently flat for the blade removal tool application.

Reverse engineering of the trailer hub was completed by teardown, and a complete CAD model of the trailer hub was created. Bending deflection and stresses were calculated for the spindle assembly to ensure robustness under load. Modeling the spindle as a cantilever beam in bending, using the smallest spindle diameter for calculation, provided favorable results with a safety factor of 2.0. Maximum possible bending deflection was calculated to be .00079in under extreme and unlikely loading conditions. This confirmed that the device chosen was safe for the application and would provide trouble-free operation under load. Minimizing deflection in all components was critical to the overall tool design requirements, as small deflections compounded through many components could allow the rotating hub assembly to move relative to the punch tool, which could cause a mis-strike to occur, damaging the disk.

Calculations for the life expectancy and strength requirements of the bearings used in the hub assembly were completed to ensure the bearing loads were not in excess of the design loads for the bearings installed in the hub. Upon disassembling the tool, the bearings used were found to be SKF L44600 1 1/16" tapered roller bearings. Design specifications for these bearings were researched in published catalogs by SKF Bearing Inc [9]. The C_0 and C_{10} static and dynamic load values tested at 99% reliability for one million cycles were found in the catalog and were compared to calculated required C_0 and C_{10} values for the turbine blade removal tool loading cases in worst-case configuration [Appendix E3]. Calculations proved that the bearing loads provided by the turbine blade removal tool on the hub were lower than the maximum load ratings provided by SKF, therefore, the bearing support was deemed robust and safe.

The hub assembly structure is designed to be mounted in a 1.75in ID hollow tube steel axle, however, a conveniently placed 3.75in x 3.75in x 0.25in thick square mounting flange with ½in thru holes placed in a 2.75in square pattern allows for the hub to be fixed in stand-alone configuration without a support tube on certain trailers. This flat flange will be used to mount the hub assembly in a vertical configuration to the baseboard using four ½in-13 2.00in grade 5 partial-thread flange bolts and locking nuts.

The hub requires minimal routine servicing, only requiring periodic greasing through the provided zerk fitting at the top of the hub. The manufacturer specified 6oz of NGLI Molybdenum based grease be pumped into the hub every 6 months or 6,000 road miles. Due to the lack of significant rotation during use on the blade removal tool, only the 6 month interval for servicing is applicable. If needed, replacement parts are easily sourced, as all 1750 lb trailer axle hub assemblies with a 5" x 4.5" bolt circle pattern share similar spindle, hub, and bearing geometries as specified by SAE J2475_201407. Replacement bearings and grease retention seals can be found online or at local retailers.

The hub chosen is tested to safe operation up to 75 miles per hour using a 28-inch nominal diameter wheel and tire. Rotation at these speeds far exceeds the slow rotation that the device will see during operation on the turbine blade removal tool. The hub is also designed for significant off-center side loading during turning with a heavily loaded trailer. These situations are far in excess of any possible loading scenario that will exist during use of the turbine blade removal tool, therefore, the device was deemed safe for use and provides minimal safety hazard. Pinch points are non-existent on the hub itself and all hub weight and components affixed to the hub will be supported by the baseboard.

5.2.3 Hub Spacer

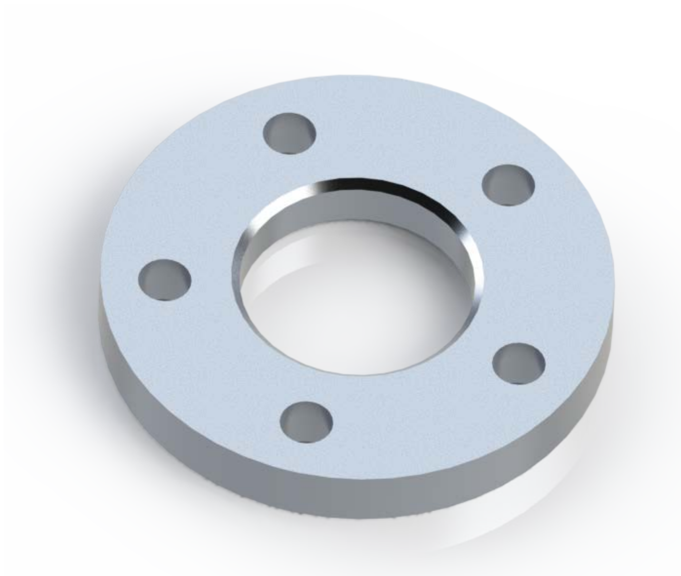

Figure 5.2.3.1. Isometric Render of the Hub Spacer

To provide spacing for the machined hub plate, a small .750in thick 6061-T6 spacer will be used to raise the hub plate above the trailer hub central bearing assembly. To allow the use of the trailer hub, the hub spacer in combination with the hub plate must vertically clear the 2.77in portion of the bearing support riser that extends above the bolt flange on the trailer hub. A small spacer is used to reduce the need for a 3.00in thick or greater hub plate that the disk will mount to. This reduces the amount of aluminum used for the creation of the hub assembly and allows for two smaller components to be used instead of one larger component without adversely affecting tool operation. Use of smaller components reduces cumbersome assemblies and component weight, allowing for single user setup and disassembly of the device without specialty tools or lifting equipment.

The hub spacer will be machined from 6061-T6 6in x 1in nominal round stock extruded aluminum using a Haas VF-2 CNC 3-axis mill. The spacer will have a contoured 2.75in diameter center through hole with 0.10in X 45° chamfers along the edges at the spacer flat surface. The hole will be created using stepped drilling, followed by contouring using a ½in 4-flute HSS end mill. Chamfering the round edges will allow for easy installation of the spacer onto the trailer hub, as the chamfer will help guide the spacer onto the center bearing assembly. Five equally spaced .625in diameter thru holes will be placed on the face of the spacer to allow for the passing of threaded studs. The top and bottom faces of the hub spacer will be face machined using a 3in 10-flute carbide face mill in the Haas VF-2 to ensure smooth surface finish, flatness, and parallel relationships for both faces. The outer face of the spacer will not be machined as it is unnecessary, however, edges will be deburred to ensure no sharp portions exist. 6061-T6 aluminum alloy was chosen due to its ease of machinability, corrosion resistance, easy raw-stock sourcing, and low cost per weight compared to other metals.

The load on the hub spacer exists from the mass of the stacked components above it, the impact from the tool during use, and the preload force from the lug nut compression against the precision hub plate that will hold the disk. No deflection existed in pure compression when calculated for the hub spacer, and no significant bending loads are expected as both faces of the hub spacer will have complete support by the trailer hub and hub plate assemblies. The hub spacer will require no maintenance and poses no significant safety threat to the user or tool. No alignment or adjustment of the spacer will be necessary for tool operation.

5.2.4 Hub Plate

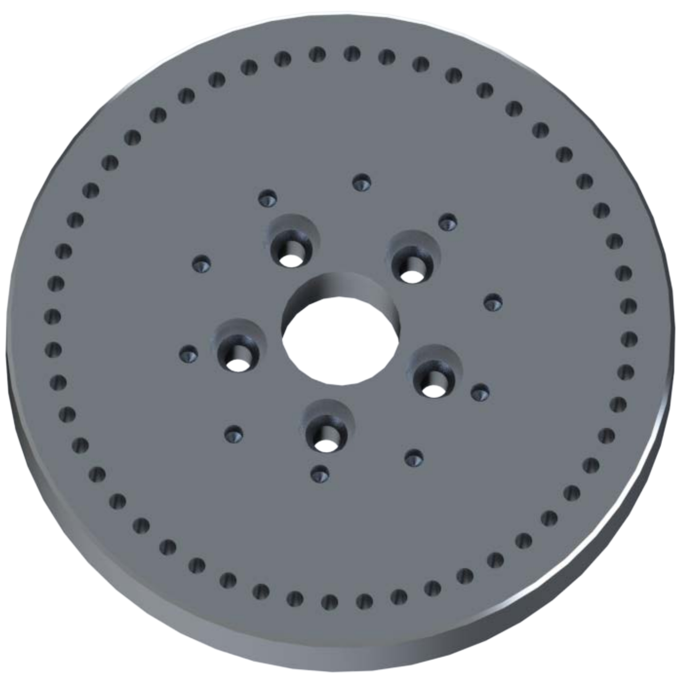

Figure 5.2.4.1. Isometric Render of the Hub Plate.

The core of the hub assembly and most critical feature of the turbine blade removal tool is the precision machined hub plate that will be used to support and locate the rotor disk assembly during blade removal. The hub plate is designed to sit on top of the hub spacer and mount to the assembly using ½in-20 60° lug nuts that thread onto the wheel studs protruding from the trailer hub. The plate will provide a flat machined surface where padding will exist to mount the disk while protecting the sensitive curvic power transfer teeth. Created from 14.5in x 2.0in nominal raw stock 6061-T6 aluminum, the entire piece is designed to be machined in a long single-mount operation in a Haas VF-3 CNC 3-axis mill. The design only requires the disk to be rotated 180° one time during processing, and all machining can be completed using the standard mill vice currently installed in the VF-3 mill.

6061-T6 aluminum alloy was chosen as the base material for the hub plate for similar reasons as the hub spacer. 6061-T6 is has low cost per weight, is strong and low density compared to steels, has significant resistance to corrosion and corrosive solvents and oils, and is easily machined with standard high speed steel tools. 6061-T6 can be sourced in large and oddly shaped geometries without greatly increasing cost and can be machined faster than steels. Since the hub will need to be lifted during disk exchange between rotor disk assemblies, weight is a concern to ensure a single user could safely lift and move the component. 6061-T6 allows for a low-weight hub plate while still providing strength and robustness needed for long term operation.

The hub plate will be face milled using a 3in 10-flute carbide face mill in the Haas VF-3 on both faces to a precise thickness of 1.950in with flatness and parallel tolerances between faces. The outer edges of the plate will be chamfered to remove the sharp corner edge, however, the outer face will not be contoured as it is not necessary. Similar to the hub spacer, the hub plate will have a contoured 2.60in diameter center thru hole with a 45°.10in chamfer along the edge that is placed over the center bearing support on the trailer hub. The diameter of the trailer hub bearing support is 2.57in and the clearance of fit between the hub plate center hole and bearing support is small. Chamfering the edge of the thru hole will allow the plate to self-center on the bearing support for ease of installation. The center hole will be created using stepped drilling, followed by contouring using a ½in 4-flute HSS end mill. Five equally spaced .600in diameter thru holes placed in industry standard 5in x 4.5in configuration will be machined on the face of the hub plate to allow for the passing of threaded studs from the trailer hub into the plate. The studs will extend 1.25in into the plate where they will mate with the corresponding lug nuts outlined previously. Above the .600in thru hole for each stud will be a 60° taper from .600in to .953in to allow for lug nut mating. This will be created using a 1.00in 60° HSS countersink end mill cutter. A counter-bore of 1.250in will be created extending from the top face of the plate to the top of the countersink to allow for the use of a 19mm standard deep socket for torqueing of the lug nuts.

At a farther radius from the center than the five lug nut holes will be 10 equally spaced ½in-13 tapped holes extending to a depth of .50in into the disk that correspond to the geometry of the specific tie bolt hole pattern for the rotor disk of interest. The angular and radial tolerances of these tapped holes are critical for the alignment of the plate and rotor disk assembly. Threaded rods will be installed into these holes which will help align and fix the rotor disk to the hub plate during use.

To prevent hub assembly rotation during operation, the hub plate will have 52 equally spaced holes near the outer edges of the plate. Each individual hole corresponds to the center location of a blade root on the disk of interest. These holes will be used with the locking mechanism discussed later in the report to clock and lock the hub assembly into place to prevent rotation during impacting operation. Holes will be drilled and then reamed using a high speed steel reamer to a precise .3750in diameter and will extend completely through the disk.

The hub plate was tested for bending deflection and bending stress under extreme situations that are unlikely to exist under any possible scenarios during tool use. Maximum bending deflection was calculated to be .0019 inches during static loading and .0024 inches during impacting. These deflections are unlikely to occur at all as the calculations used to provide these deflections assumed worst case cantilever loading with the smallest cross-section support web used for the hub plate. Transverse shear stress was negligible at only 80.93 psi, and maximum possible bending stress was small at 2707 psi at the innermost minimum cross section of the hub plate. All of these figures proved that the hub plate was sufficiently robust for the loading cases that will exist during disk mounting, dismounting, and blade removal, even under extremely unlikely circumstances such as dropping a rotor disk onto the hub plate from elevation or excessively high impact forces from the pneumatic tool. All calculations were completed with a safety factor of 2.0. Appendix E4 provides diagrams and calculation examples for analysis of the hub plate.

The hub plate will require no required maintenance as it is a standalone part without any features requiring cleaning or lubrication. It is recommended that users avoid gouging the faces of the part to ensure flatness of the faces remain throughout the life of the product. The hub plate itself poses little safety risk as there are no individual moving parts on the plate. Weighing only 28.6 pounds, the plate is safe to lift by a single operator under OSHA lifting regulations. With a maximum diameter of 14.50in, the hub plate itself is not cumbersome to move when needed.

5.2.5 Rubber Cushion Pad

Figure 5.2.5.1. Isometric Render of the Rubber Cushion Pad.

To ensure no damage to the curvic teeth occurs when placing the rotor disk onto the hub plate, the top surface of the hub plate will be lined with a ½in thick EPDM round rubber cushion surface. The rubber pad will have ten .625in thru holes stamped into it corresponding to the threaded holes for the nylon studs that will extend upward from the hub plate to locate the rotor disk assembly. 12in x 0.5in nominal square rubber sheet will be cut using a heated wire to an 11.5in nominal diameter circle, far wider than the diameter of the curvic teeth location from the center of the rotor disk assembly. The rubber pad will sit in between the hub plate and the disk itself and will be fixed to the hub plate by the extension of three nylon studs threaded into the hub plate extending vertically 4.50in.

EPDM rubber was chosen as the material for the cushion pad due to its extreme resistance to oil, gasoline, solvent, and oxidant intrusion. It is a medium-hardness rubber that will not compress significantly under large loads. Some damping will be provided to minimize vibration between the disk during impacting and the rest of the tool assembly, increasing the life of components and preventing the loosening of any fasteners on other components of the tool.

The expected load due to stacked component weight including a safety factor of 2.0 exerted on the pad is 428.2 lbs assuming a rotor disk weight of 200.0 lbs. This value is far less than the compression pressure rating per square inch for the EPDM rubber chosen. This load will be distributed across the curvic teeth which the disk will rest on during blade removal. Minimal maintenance will be required on this part, only requiring occasional replacement if permanent depressions form along the surface of the curvic teeth resting location. The part poses no significant safety hazard to the operator or tool during installation, tool operation, or removal.

5.2.6 Studs and Fasteners

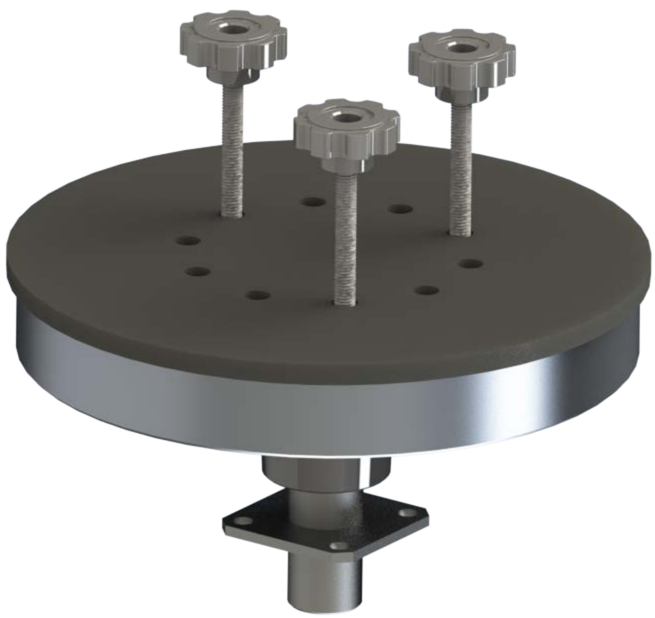

Figure 5.2.6.1. Isometric Render of the Studs and Fasteners.

To affix the disk to the hub plate without damage, three individual fully threaded ½in-13 x 5.00in Nylon 6,6 studs will be threaded into any three of the ten threaded holes in the hub plate corresponding to the tie bolt hole pattern in the disk of interest. The disk will be lowered over these studs using the current Solar Turbines disk lifting tool. The studs will be threaded into the plate in 3 of the 10 holes not being occupied by the rubber expanding fasteners in the lifting tool plate. Once the nylon studs are installed in the hub plate, the rotor disk and lifting tool can be lowered onto the rubber pad on top of the hub plate. The Solar Turbines rotor lifting tool can be removed after resting the disk on the rubber pad and the three nylon studs will extend approximately 1.25" above the disk surface.

Three nylon 6,6 ½in-13 knurled hand-thread fasteners will be threaded onto the three nylon studs to provide a downward force on the disk, fixing it to the full rotating hub assembly, constraining it axially. Due to the weight of the rotor disk assembly, it was determined that a large downward force was not required by the hand fasteners as the disk weight will help to prevent its own rotation and translation relative to the hub plate surface. A downward force of 1000 lbs distributed over the three fasteners can be achieved at a tightening torque of 10.38 ft-lb for each knurled fastener seen in Appendix E6.

Torque higher than 20 ft-lb is not recommended by the stud and fastener manufacturer due to the low deformation resistance of nylon threads. The clamp force at approximately half of the maximum fastener torque allows constraint of the disk axially on the platform without the need for specialty tools for setup.

Nylon 6,6 was chosen for the studs and fasteners due to its low hardness compared to the rotor disk material. This will prevent the studs from scratching or deforming the disk upon contact to eliminate the risk of adding any discontinuities or deformations to the critical components. The chosen fasteners are easily sourced, changed, discarded, and installed if replacement is necessary. Nylon 6,6 is resistant to oils, solvents, oxidants, and other shop materials which can degrade other plastics. It is a safe material while providing adequate strength compared to other softer polymer materials. Small safety hazards exist between the studs and fasteners, specifically a finger pinch point between the rotor disk surface and knurled fastener due to the large surface area under the fastener that can easily trap a user's finger if careful attention is not given during the tightening process. Users should only tighten one knurled fastener at a time in a three-pass torque configuration ending at approximately 10 ft-lb to ensure adequate and even torqueing of the fasteners.

Table 5.2.1. Rotating Hub Assembly Bill of Materials.

Subsystem	Part Number	Description	Supplier	Supplier Number	Package	Quantity
Base Table (100)	101	60"x30"x2.25" Maple Butcher Block	Butcher Block USA	HD3060	1	1
Hub Assembly (200)	201	Tie Down Engineering Hub/Spindle Assembly	Northern Tool	128008	1	1
Hub Assembly (200)	202	1/2-13 2" Flange Bolt Grade 5	McMaster-Carr	92979A479	10	1
Hub Assembly (200)	203	1/2-13 Flange Nut Grade 5	McMaster-Carr	92018A540	25	1
Hub Plate (300)	301	1/2"-13 Nylon Threaded Rod	McMaster-Carr	93665A684	5	3
Hub Plate (300)	302	1/2-13 Comfort-Grip Fluted-Rim Knob	McMaster-Carr	5532T37	1	3
Hub Plate (300)	303	12"x12"x1/2" Oil-Resistant Buna-N Rubber Sheet	McMaster-Carr	8635K168	1	1
Hub Plate (300)	304	1"x1"x0.65" 4130 Square Stock Steel	McMaster-Carr	6582K43	1	1
Hub Plate (300)	305	1/2-20 x 3.00" Wheel Studs	Summit Racing	8020	20	1
Hub Plate (300)	306	14.5"x1.0" 6061-T6 Extruded Alumium Plate	Speedy Metals	Rd 6061-T6 Aluminum, Extruded	1	1
Hub Plate (300)	307	6.5" x 1.0" 6061-T6 Extruded Aluminum Plate	Speedy Metals	Rd 6061-T6 Aluminum, Extruded	1	1
Hub Plate (300)	308	1" 60° Countersink Milling Tool	ICS Cutting Cools	CSKXL160-100	1	1

5.3 Locking Mechanism

Figure 5.3.1. Isometric Render of the Locking Mechanism

One of the critical considerations of the final design is how to lock the turbine disk in place and prevent rotation during blade removal. To do so, a locking mechanism was designed that will enable the user to rotate the disk to the desired blade for removal, lock the system in place, and perform the blade removal without rotational movement. The mechanism consists of a locking pin and hub plate mentioned previously that act as a clocking mechanism so the user can easily move from blade to blade with distinct accuracy. An issue faced during design is that there is no angular relationship between the broached fir tree slots and any other distinct disk geometry. This causes angular variability of as much as ~7^o between any two different disks. To account for this variation, rounded slots that stretch 10^o will be machined into the butcher block to allow the locking mechanism to be adjusted as needed. This will ensure that no matter how the disk is placed in the mechanism, it can be easily adjusted for proper blade removal. Additionally, the mechanism utilizes a spring to hold the pin in an upward and locked position within the hub plate. For ease of adjustment, a handle will protrude from the frame past the edge of the disk that allows the operator to release the pin for rotation of the hub assembly. The locking pin assembly will be fixed to the work table or butcher block for stability and is located directly underneath the corresponding holes of the hub plate as seen in Figure 5.3.1.

Once all locking mechanism parts are manufactured to the engineering specifications, assembly is quite simple. The pin is inserted into the base framing through the 0.500in. hole and rotated so that the ¼in tapped hole is normal to the slot in the frame. At this point, the ¼in handle can be hand threaded into the bottom of the pin. Next, the spring is placed over the shaft and retained using two jam nuts. The final step is to mount this assembly to the butcher block by placing two carriage bolts through the bottom slots in the butcher block aligned so the threads extend upward through the base plate. The hand nuts can then be hand tightened for final assembly at a desired angle relative to the disk.

To use this device, the user will hold the handle all-the-way down allowing the disk to rotate freely. The user will need to rotate the hub assembly until the pin is aligned with any hole in the hub plate. They will then release the handle, and the tapered tip of the pin will help guide the pin into the appropriate hole. The user will align the assembly such that the impact hammer is aligned with a fir tree root of a blade, and once completed, will hand tighten the thumb nuts locking the entire system in place. At this point, the blade is ready for removal and mechanism is clocked for the remaining blades.

This locking mechanism is quite safe and has very little risk to the operator. Sharp corners are avoided and there is no risk of cuts to the operator. The only safety concern is pinching that could occur to the operator’s finger during the process. The pinching force magnitude is under 15 lbf. and will cause no serious injury.

5.3.1 Locking Mechanism Base

Figure 5.3.1.1. Isometric Render of Locking Mechanism Base.

The base will serve as the intermediate connection between the locking mechanism and the butcher block. It consists of a ⅛ in. flat steel plate with rounded edges to eliminate sharp corners. The outside edges will be also filleted to a radius of 0.25 in. The plate has two 0.397in (Drill Size X) clearance holes that will be used for mounting to the 10º slots in the butcher block using ⅜in - 16 square carriage bolts. The slots in the butcher block will prevent the square carriage bolts from rotating which allows the mechanism to be easily adjusted without the need of specialty tools. The bottom of the butcher block has a 2in x 5in x 1.25 in. filleted rectangular cut at the location of the slots so the bolt heads do not protrude beyond the bottom surface, ensuring the butcher block remains flat on the work surface. 1in steel tubing with a wall thickness of 0.065 in. will be welded to this base as framing and the calculations for stress analysis can be seen in Appendix E7. This framing will have a 5/16in slot milled through the centerline which acts as a guide for the handle which has a diameter of ¼ in.

The base plate will come from ⅛ in. thick low carbon steel plating as stock. This material was selected due to its rigidity and machinability. It will provide rigid support for this mechanism, preventing movement and deflection during loading. The base plate will need to support the locking mechanism from the horizontal component of the blade removal tool which is equal to 61.8 lb, including a factor of safety of 2.0. No facing will be necessary as flatness is not a concern for this component. The contouring and 0.397 in. holes will be completed using a drill press with an X letter size drill, and the corners will be ground to avoid sharp edges. Once the base plate is complete, the steel tubing will be fillet welded on all edges to the base. The analysis can be seen in Appendix E8.

A small plate will be welded to the top of the steel tubing and will be used to guide the pin vertically into the locking plate. The plate will be ¼ in. thick with a 0.500 in. thru hole reamed in the center to guide the pin in vertical movement. The top plate will be manufactured from ¼in low carbon steel sheet stock. The 0.500 in. hole will be reamed ensuring a tight tolerance and was determined using a LC11 class fit. The hole/shaft calculations can be seen in Appendix E10.

5.3.2 Locking Pin

Figure 5.3.2.1. Isometric Render of Locking Pin

The locking pin has two different diameters to serve two purposes. The larger diameter will translate through the hole in the frame which will serve as a guide for the pin during vertical movement. A spring will be placed between the hub plate and the lock frame and will be retained and preloaded at the upper end by a pair of jam nuts. The top smaller pin diameter is used as the locking portion that will protrude into the hub plate locking holes for fixing the hub assembly. The overall length of the pin was determined through analysis of relating components to determine proper geometry. At full extension in the locked position, the release handle will touch the top of the slot in the lock base frame and the spring retaining jam nuts will rest on the bottom of the hub plate. This will prevent any further upward movement of the lock. In this position, the spring will have slight preload, ensuring the pin remains in a locked position. In the lowered, unlocked position, the pin will touch the base of the frame and will provide a clearance of 0.100 in. between the pin tip and hub plate allow for rotation of the disk to the desired blade for removal.

Pin manufacturing will begin with ½” low carbon steel rod stock. The stock will be turned and faced to the proper geometry. The larger diameter of the pin will have a diameter of 0.487 in. that will slide through the 0.500 in. reamed hole in the frame. These dimensions were determined through an LC11 class fit with a H13 hole to allow translational movement while maintaining positional accuracy. Calculations can be found in Appendix E10 . Located near the bottom edge of the 0.487 in. diameter portion of the pin is a ¼in - 20 tapped hole for the handle to thread into. The top of the 0.487 in. diameter portion of the pin will be threaded using a ½in - 20 die for the jam nuts to thread. The narrow top diameter of the pin will be turned to a diameter of 0.363 in. to match the 0.3750 reamed holes in the locking plate through LC11 class fit with a H13 hole. This smaller diameter of the pin will have a length of 1.0 inch and is chamfered at the tip 0.1in. x 45^{o} to aid in self-centering in the hub plate lock holes.

5.3.2.1 Locking Handle

As previously mentioned, a simple ¼" steel rod will serve as the handle to lower and raise the pin. This handle will be manufactured using ¼" low carbon steel rod and faced to the proper length. One end of the rod will be threaded using a ¼ in.- 20 die that will match the tapped hole on the pin.

5.3.3 Locking Spring

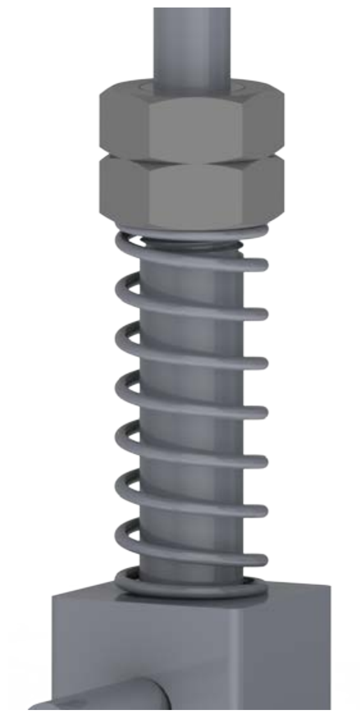

Figure 5.3.3.1. Isometric Render of the Spring.

The spring for the lock assembly was carefully chosen after specifying the lock geometry and performing calculations for needed compression and extension length. The spring will be placed around the 0.487 in. diameter pin and will require an inside diameter that is larger. Additionally, it will be constrained between the top plate of the frame and the two locking nuts and a preload is desired to keep the pin in the upward and locked position. Thus, an uncompressed length greater than that distance is required. On the contrary, at the fully lowered position when the pin is removed and touching the bottom plate, the spring needs to have a fully-compressed length that will allow for the full range of compression without coil binding. Another important consideration was the force required to lower the pin completely. It was determined a force between 10-15 lbs. would be preferable for a repeated operation. Knowing these constraints, a range of desired spring constants was calculated. Since this spring is to be mounted between the two relatively flat surfaces of the top plate and jam nuts, a closed and flat end spring was chosen for this application. A spring with part number 9657K375 was then selected from McMaster-Carr Inc. as it fulfilled all of the design requirements. All calculations and considerations can be found in Appendix E11 outlining the spring selection process. If a spring wears over time or is unsatisfactory to the operation, the spring can be easily replaced by removing the two jam nuts. Additionally, the jam nuts can be easily over or under tightened allowing the operator to adjust the force required.

Table 5.3.1. Locking Mechanism Bill of Materials.

Subsystem	Part Number	Description	Supplier	Supplier Number	Package	Quantity
Locking Mechanism (400)	401	1/2 - 20 Thin Hex Nut	McMaster-Carr	93839A825	25	1
Locking Mechanism (400)	402	Flat End Compression Spring	McMaster-Carr	9657K375	6	1
Locking Mechanism (400)	403	High Speed Steel Chucking Reamer - 0.3750	McMaster-Carr	2995A69	1	1
Locking Mechanism (400)	404	3/8"-16 x1.25" Square Carriage Bolt	McMaster-Carr	90185A626	25	1
Locking Mechanism (400)	405	3/8" - 16 Thumb Nut	McMaster-Carr	91833A113	1	3
Locking Mechanism (400)	406	High Speed Steel Chucking Reamer - 0.5000	McMaster-Carr	2995A74	1	1
Locking Mechanism (400)	407	Low Carbon Steel Rod - 1/4" Diam - 1Ft	McMaster-Carr	8920K115	1	1
Locking Mechanism (400)	408	Low Carbon Steel Rod - 1/2" Diam - 1Ft	McMaster-Carr	8920K155	1	2
Locking Mechanism (400)	409	Low Carbon Steel Sheet - 3"x3"x3/16"	McMaster-Carr	1388K662	1	2
Locking Mechanism (400)	410	Low Carbon Steel Sheet - 3"x3"x1/4"	McMaster-Carr	1388K102	1	1

5.4 Sliding Tool Holder and Air System

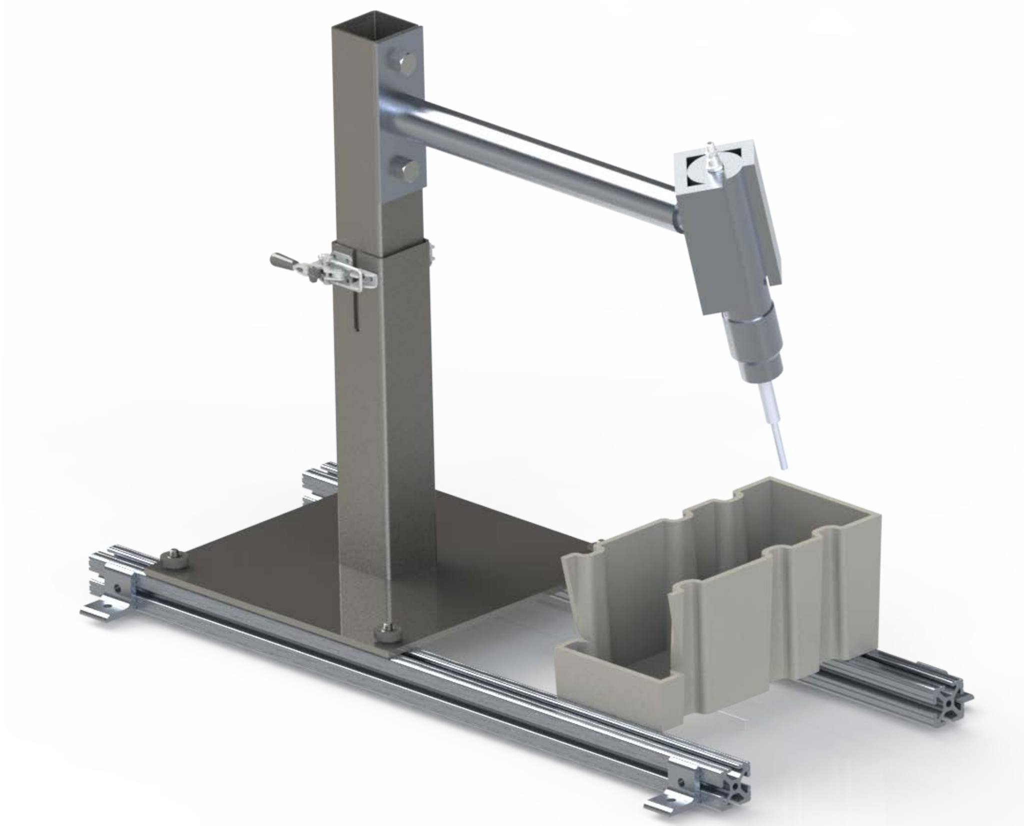

Figure 5.4.1. Isometric Render of the Sliding Tool Holder

To accommodate manufacturing tolerances, differences in disk diameter, and ease of loading or unloading a disk; the design chosen to hold the impact tool is an overhead arm with sliding adjustment on slotted aluminum rails for radial adjustment. Height adjustment is accomplished through the use of a telescoping pair of nested box steel tubes. By slotting the larger of the two tubes, it can be clamped to the smaller tube to fix the height of the impact tool relative to the disk. This height adjustment is similar to seat post adjustments on a bicycle and incorporates two toggle clamps for a total clamping force of 720lb. By incorporating the toggle clamps the user only has to release the two small levers in order to adjust the height of the tool. The aluminum rails for radial adjustment also allow infinite location possibilities to accommodate any disk diameter and will be marked visually or with physical stops once the positions required for each disk type have been established. The base plate will be clamped to the rails with four T-studs and hand operable nuts to ensure the user needs as few tools as possible to load or unload a disk. This is necessary because the overhead arm must be moved radially to work smoothly with the existing overhead crane system when installing and removing rotor disks. A plastic catch bin has been incorporated into the rail system to catch the blades as they are removed to prevent damage. To drive out the seized blades, an inline air hammer with non-marring tips has been selected to provide the rapid impact needed to remove the blades. This impact tool will be held securely by a commercial bicycle repair stand bolted to the vertical steel tube.

Figure 5.4.2. Isometric Render of the Tool Holder.

The operation of the Sliding Tool Holder is simple and requires no tools per the design requirements. Once the disk is mounted to the hub and locked into place, the base plate is moved forward on the aluminum rails until aligned with the blade. Future iterations would include a laser sight, but due to cost considerations this feature is not included on this iteration. Once the punch is aligned, the catch bin is mounted underneath the disk and impacting location. Following all alignments, the clamping thumb-nuts are tightened to hold the base place and punch in place. Ideally the height will not change from disk to disk, but in the event it does, the vertical tool holder clamps are released and the punch holder set to the correct height. Testing will determine the height for a given disk thickness and location markings will be placed on the riser tubes for easy future adjustment. With all portions of the tool in their locked position to prevent movement, the operator will put on the required PPE and set the air regulator to the desired pressure. When ready to begin impacting, they will operate the spring loaded air valve to commence impact tool operation. Once the blade has been freed it will either fall into the plastic catch bin or require a gentle tug from below to remove it.

5.4.1 Component Selection

The aluminum rails are commonly used for custom structural framing applications due to their easily utilized universal T-slots that accommodate many available fixtures and attachments such as the T-studs that will secure the base plate to the rails. With slots on all sides, the rails will also be secured to the table using L brackets bolted to the table. By fully supporting the aluminum rails on the table they will not be loaded in bending and only loaded in compression along their entire length. Calculation proved that stress and deflection were not a concern. Four L-brackets with #10 wood screws will be used to secure the rails to the table at all corners, more attachment points can be easily added or easily moved to accommodate changes needed during testing.

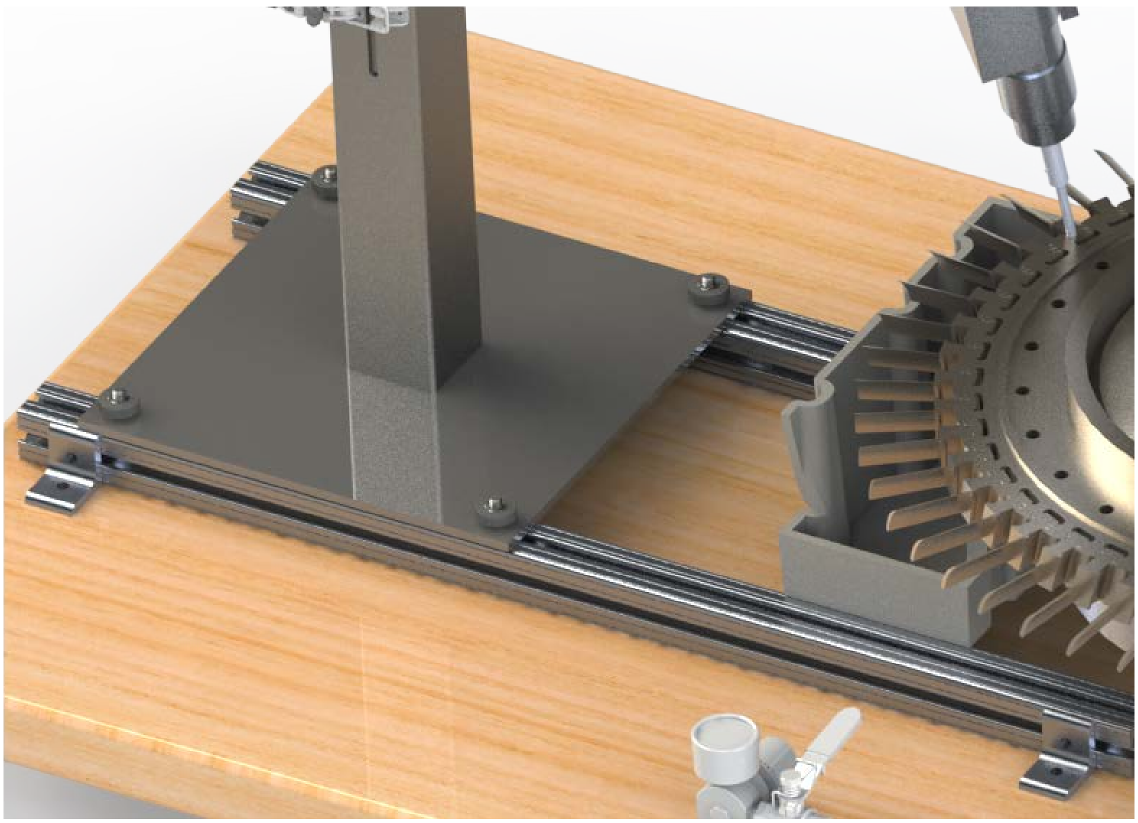

Figure 5.4.1.1. Isometric Render of the Sliding Rails.

The impact tool will be held in place using an off the shelf Park Tool PRS-4W-1 Repair Stand bicycle clamp. After receiving suggestions from Solar Turbines Engineers, further research into these style clamps was completed and they fit the design requirements for the prototype of the tool. These clamps have the ability to firmly secure the impact tool and allow for a slight angle offset to more closely match the broach angles of the fir tree slots. The PRS-4W-1 clamp also provides an easily adaptable wall mount flange to fix it to the rest of the tool holder. The secure clamping is provided by a compound action similar to Vise-Grips, and the angular adjustment is infinite with a clamping screw to secure it. By using a commercial bicycle repair clamp designed to hold large bikes in place during high load work such as breaking loose stuck fasteners, we can be certain that the clamp will be strong enough to withstand the loads from the impact tool. Testing completed at Solar Turbines in February 2017 determined that the required force to remove blades was small (27lb) repeated impacts, and the sliding tool holder was designed to withstand a maximum of 200lb from the impact tool (estimated 100lb maximum force with a Safety Factor of 2).

Figure 5.4.1.2. Photo of Park Tool PRS-4W1 Bike Holder.

The impact tool used is the Klutch 47927 due to its compact, inline profile, use of standard sized attachments, variable stroke, reasonable cost, and lower impact force compared to most air chisels on the market. The use of standard .401 shank attachments is important for decreasing the service cost and time for the tool by utilizing readily available auto body air hammer tips that come with non-marring nylon or brass ends. Both the Klutch tool and hammer attachments are as pictured.

Figure 5.4.1.3. Photo of the Klutch Air Hammer and Non-Marring Tips.

The catch bin is a commercially available plastic hardware bin that was selected for its size, ease of access, and cost. Mounted to the rails, the bin will not interfere with any of the other mechanisms so it will rarely require any repositioning. By selecting a plastic hardware bin, the blades will land on a durable but non-damaging surface in accordance with design requirements for preserving the blades for remanufacturing.

5.4.2 Safety

Safety is addressed on the sliding tool holder with several features to ensure the user does not accidentally place their hand under the impact tool, or inadvertently release the height adjustment clamps. The air system will not rely on the built in trigger of the impact tool as it will be locked, instead the user will operate a spring loaded ball valve mounted to the table. This spring loaded valve defaults to closed if the user removes their hand and allows a fine degree of throttling for the air supply to the impact tool. The air pressure will also incorporate a regulator for establishing a maximum air pressure that is sent to the impact tool, this ensures the user does not accidentally send the full 100 psi available directly to the impact tool. The adjustable alignment features on the rails and vertical tube use vibration resistant methods for securing their position to prevent accidental release.

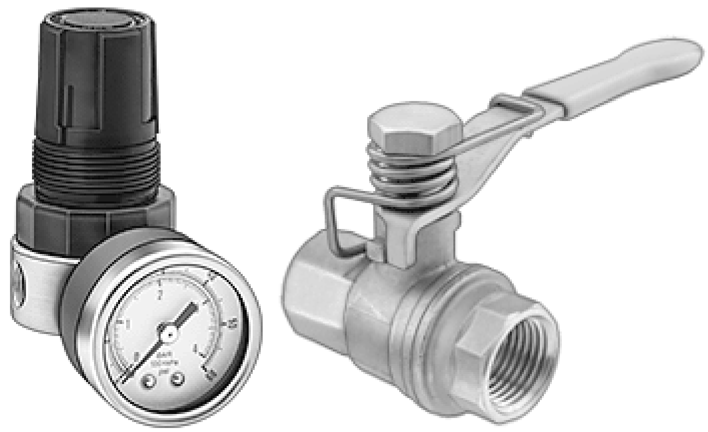

Figure 5.4.2.1. Photos of the Air Regulator and Spring-Loaded Valve.

It is important to note that during use, the tool will require the operator to wear a face shield or eye protection in the event that any debris are ejected from the disk. Hearing protection may be worn by the operator in the event that the noise from the air tool is louder than is comfortable.

5.4.3 Strength Validation

The chief concern in the design of this tool, is not stress, but deflection given the importance of accuracy. The area of where the deflection occurs is in the main upright where the deflection and deformation angle both translate into misalignment of the impact tool with the blade. Cantilever beam calculations were performed to evaluate the size of the tubing used while maintaining the accuracy of the punch tip during use. The calculations in Appendix E12 demonstrate that the choice of 2.25 inch square tube for the smaller tube and 2.5 inch square for the larger tube were appropriate given the tool tip deflection of .0015 inches and availability. The area where stress was the governing factor was at the weldment along the base of the main upright. Calculations were performed to select a ¼in. fillet weld size using an E70 series electrode and can be found in Appendix E13. Maximum stress in the weld was found to be 3.3ksi which is far below the rated 21ksi shear strength for E70 series electrode. The torsional load on the main upright created by the horizontal component of the impact tool load resulted in a misalignment of the punch tip of .0145in as shown in Appendix E13. A fatigue analysis was performed on the base plate weld to ensure that infinite lifetime was achieved, these calculations can be seen in Appendix E14 using the Gerber criteria. A fatigue factor of safety of 1.23 was calculated using the built in load factor of safety of 2.0 and the known conservative Gerber method found in Shigley's Mechanical Design 10th Ed.

The exact dimensions of the slit in the side of the larger tube to clamp on the smaller nested tube will require testing due to the nature of the loaded area. Because deflection calculations ignore the effects of small features and connections, these calculations would not provide reliable information for the deflection of the sides of the tube, nor any deflections in the locking clamp applying the load. It is important to note that the edges of the slit will be rounded to resist crack propagation from stress concentrations.

5.4.4 Assembly and Manufacturing

The components requiring manufacturing are the upright tubes, base plate, and attaching clamps. The base plate will have the hole pattern for the rails drilled before welding. The tubes are sized to be nested properly so they will only require being cut to length. The larger, lower tube will have a slit cut in the sides by vertical band saw to provide a starting point for tuning the clamping strength. The toggle clamps will be welded to the sides of the larger tube, bridging the slit to provide the clamping force. After the slit has been cut and tested for clamping with a finalized size, the large tube will be welded to the base plate as discussed above. The smaller upper tube will have the hole pattern drilled to mount the Park Tool clamp prior to assembly, dimensions of the Park Tool Clamp will be verified before the upper tube is drilled.

For assembly, the rails will be screwed to the table surface before mounting the base plate to the T-studs. Following the base plate mounting, the upper tube will be inserted and clamped into the base tube and the Park tool clamp mounted to the upper tube. The final step for the sliding tool holder is to fix the impact tool in the clamp and adjust the tool to the mounted disk. The air system will be mounted to the table after all other systems are mounted to ensure there is no interference with hoses and it is comfortable to operate.

5.4.5 Maintenance and Repair

The tool will require minimal maintenance as outlined in the PDR design requirements. The sliding tool holder will require only a cursory check for any cracks or deformation before each use. Daily oiling of the air system similar to any other air tool will be required if the shop air supplied is not already. In addition, debris should be cleaned from sliding tool surfaces if there is any significant build up from the shop environment.

Repairs if necessary will be simple due to the modular and basic nature of the sliding tool holder design. Again, this was a requirement from PDR to minimize any downtime during usage of the tool. No components of the sliding tool holder require CNC machining, while the most labor intensive repairs would require minor welding as outlined above in the manufacturing section. In order to facilitate faster repairs and minimized downtime, a full BOM with sources for each purchased component is supplied to prevent confusion while sourcing replacement parts, seen in Appendix C.

5.4.6 Cost Analysis

The Sliding Tool Holder contains as many purchased parts as possible to limit custom manufacturing that is both expensive and impractical for adoption of the tool in Solar Turbines facilities. A full cost breakdown for each purchased component is available in the BOM (Appendix C) and no CNC machining is required for the sliding tool holder subassembly. A shipping estimate is included in the overall BOM as most components and materials are sourced from McMaster Inc. without concrete shipping quotes. McMaster Inc. was chosen as the main supplier for most parts due to their prompt shipping and accurate technical details to limit manufacturing delays once parts are ordered.

Table 5.4.1. Sliding Tool Holder Bill of Materials.

Subsystem	Part Number	Description	Supplier	Supplier Number	Package	Quantity
Sliding Tool Holder (500)	501	Klutch Air Needle Scaler	Northern Tool	47927	1	1
Sliding Tool Holder (500)	502	Double-Locking Pull-Action Toggle Clamp	McMaster-Carr	51335A71	1	2
Sliding Tool Holder (500)	503	T-Slotted Framing, Single Rail, Silver, 1-1/2" High x 1-	McMaster-Carr	47065T102	1	2
Sliding Tool Holder (500)	504	Corner Bracket for 1-1/2" High Single Rail, Silver	McMaster-Carr	47065T845	1	4
Sliding Tool Holder (500)	505	Drop-in Fastener with Stud, for 1-1/2" High Single Rail	McMaster-Carr	47065T234	1	4
Sliding Tool Holder (500)	506	Zinc-Plated Steel, Number 10 Size, 3/4" Long	McMaster-Carr	91070A245	100	1
Sliding Tool Holder (500)	507	Solid Tube Framing, Steel, 2-1/2" Square	McMaster-Carr	4931T146	1	1
Sliding Tool Holder (500)	508	Solid Tube Framing, Steel, 2-1/4" Square	McMaster-Carr	4931T145	1	1
Sliding Tool Holder (500)	509	Plastic Bin Box	McMaster-Carr	4666T63	1	1
Sliding Tool Holder (500)	510	Extended Straight Bracket for 1-1/2" High Rail, Silver	McMaster-Carr	47065T261	1	1
Sliding Tool Holder (500)	511	Reinforced Plastic Knurled-Head Thumb Nut	McMaster-Carr	93886A150	10	1
Sliding Tool Holder (500)	512	12"x12" plate for base of tool holder	Speedy Metals	1/4" 1045HR Steel Plate,	1	1
Sliding Tool Holder (500)	513	Park Tool Deluxe Wall Mount Repair Stand - PRS-4W	Amazon	Select Silver color	1	1
Air System (600)	601	Air Regulator	McMaster-Carr	6763K13	1	1
Air System (600)	602	Air Regulator Bracket	McMaster-Carr	6763K21	1	1
Air System (600)	603	Straight connector	McMaster-Carr	5485K22	1	1
Air System (600)	604	Brass On/Off Valve with Spring Close Lever Handle	McMaster-Carr	4088T8	1	1

6 Manufacturing Report

6.1 Sliding Tool Holder

Manufacturing for the Sliding Tool Holder began with the base plate which was ordered as a 12in. x 12in. square mild steel plate, and holes were drilled on a drill press to "O" size to allow for easy fitment of the 5/16in. mounting studs (.003in. clearance). The sliding rails were ordered in 4ft. lengths to allow for significant adjustment in their final mounted length and were cut down to a length of approximately 43in to fit on the new table. Regarding the vertical tubes, the lengths were cut using an abrasive cutoff wheel instead of a band saw due to availability. The holes were matched to the Park Tool clamp due to its deviation from the specified hole pattern (See Figure 6.1.1). The clamp mounting pattern deviated by as much as 1/4in from the specified dimensions requiring the match drill process.

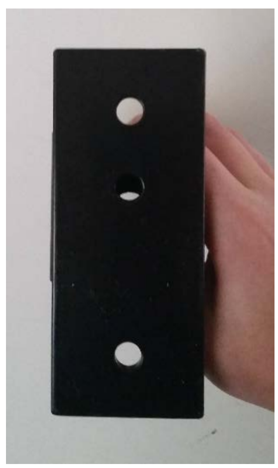

Figure 6.1.1. Mounting Flange of the Park Tool Clamp.

With the lengths for the tubes set, and holes drilled, the lower tube was located on the base plate and fixtured with magnets for welding, seen in Figure 6.1.2. Note, this image shows one magnet in place, however three magnets were used while the open side was welded.

Figure 6.1.2. Vertical Lower Tube ready for welding.

The welding was performed with a Millermatic 251 GMAW welder as shown below in Figure 6.1.3.

Figure 6.1.3. Performing stitch welds

The welds were kept short, and completed on opposite sides, with time allowed for cooling to minimize deflection. There was minor heat deflection of the tube in one direction from welding seen in Figure 6.1.4.

Figure 6.1.4. Perpendicularity measurements on the vertical tube.

The welds were ground smooth with a flap wheel for better paint adhesion shown in figure 5. The clamps were welded using the same Miller 251 to the lower vertical tube and slots were cut to size allowing for the clamping action.

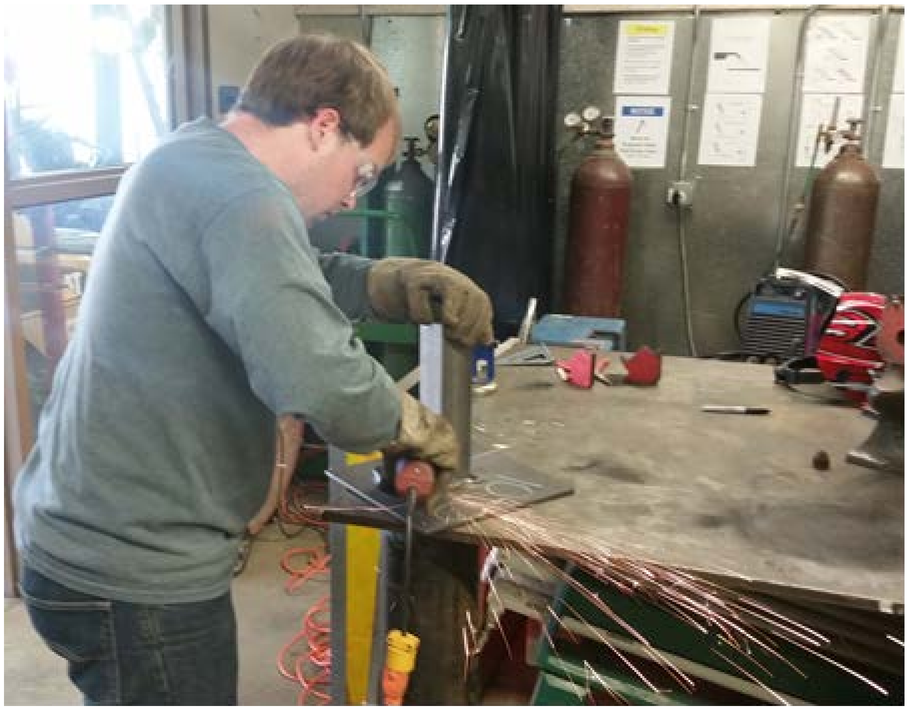

Figure 6.1.5. Finish Grinding of Welds.

The slots to allow the clamping action in the vertical tubes were cut after the toggle clamps were welded in place and adjusted for draw length. This was done so that the clamping could be tested and the geometry of the slots fine tuned. It was found that adding an additional cut from the centered slot to the outer face of the lower tube, as shown below**,** was needed to eliminate twisting and lower the necessary force from the user. This is because it dramatically lowers the geometric stiffness at the point of bending. To finish the vertical tubes, the slots were rounded and ground smooth with a rotary tool to limit stress concentrations and prevent cracking.

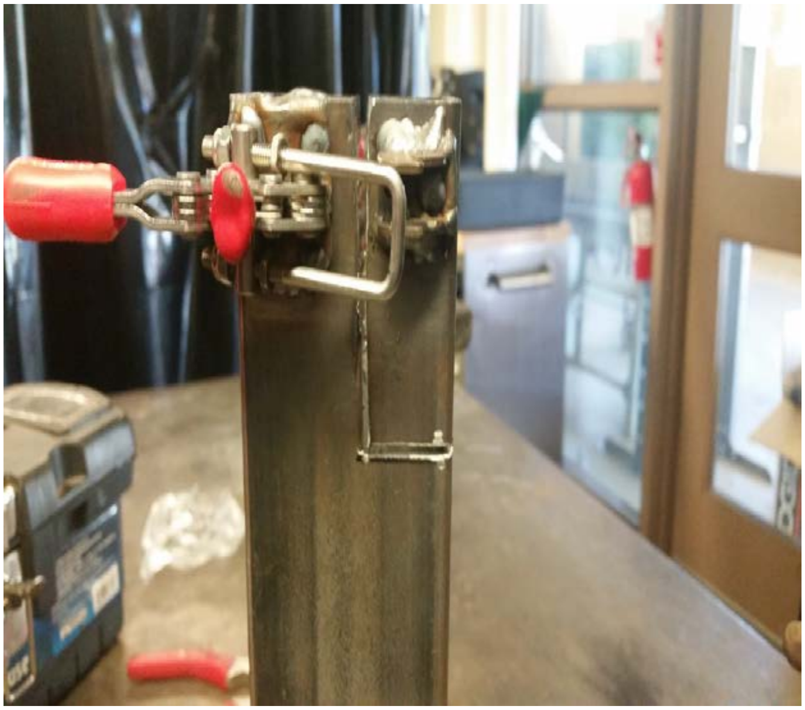

Figure 6.1.6. Modified Slits in the Lower Clamping Tube.

For assembly, the rails were bolted to the table surface before mounting the base plate to the T-studs. Following the base plate mounting, the upper tubes were inserted and clamped into the base tube and the Park tool clamp mounted to the upper tube. The final step for the sliding tool holder was to fix the impact tool in the clamp and adjust the tool to fit the 3D printed sector model. The air system was mounted to the table after all other systems were mounted to ensure there was no interference with hoses and it was comfortable to operate. A self-coiling air hose was chosen for the connection between the user controls and the impact tool so that the hose would not tangle or hang loose in the various positions of the sliding tool holder.

6.2 Locking Mechanism

Manufacturing of the locking mechanism began with the base plate. The base plate stock thickness was increased to ¼" from 3/16" as designed. The extra stock thickness allowed for full depth contouring of the base. The CAM used to generate the code for milling was created in HSMWorks, and machining was performed on a Haas Mini Mill. A ½" 4-flute end mill was first used to contour the stock and create the pocket. Next, the 0.397" holes were peck drilled using a size X drill. The part was then removed and the remaining stock was faced using a ¾" 4-flute end mill on a manual mill.

Figure 6.2.1. Base Plate after Op. 1 of machining.

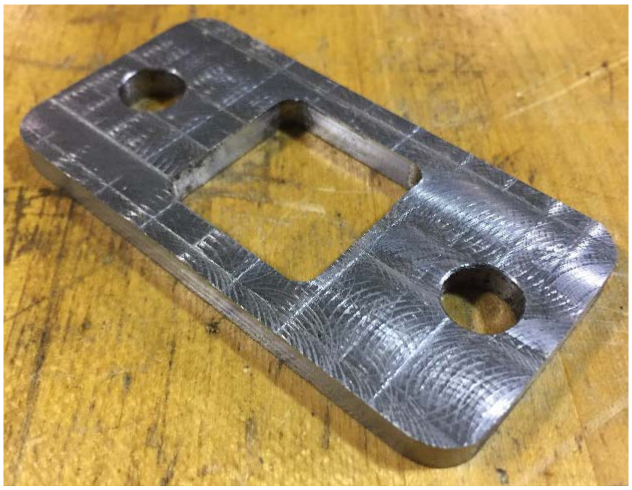

Figure 6.2.2. Finished Locking Mechanism Base Plate.

Next, the small guiding plate was manufactured using similar methodology to the base plate. The stock thickness remained unchanged; however, the part thickness was increased to ⅜" from ¼" to allow additional material for clamping. The CAM used to generate the code for machining was also created in HSMWorks. Machining began by contouring the stock using a ½" 4-flute end mill. Then a 15/32" drill was used to create the thru-hole two sizes under desired, prior to reaming with a 0.500" HSS reamer. The part was then removed and faced using a manual mill in similar fashion to the base.

The locking pin was manufactured third. First, the half inch steel stock was cut to length on a horizontal band saw. Then turning and facing was completed on the HAAS TL-1 Lathe using conversational control for inputs. The shaft main diameter was set to be a nominal diameter of 0.487" and was measured to be 0.4875". To create the ½"-20 threads on the end of the pin, the locking pin was placed into the chuck of a manual lathe and locked. Using a tailstock for centering, the threads were cut by hand with a die. While considering how to drill the hole for the ¼"-20 threads for handle installation, a flat was determined to be the best option to machine into the shaft before drilling. The drilling was performed with a ¼" drill on a manual drill press and then tapped with ¼" - 20 threads by hand.

The locking pin handle was first cut to length using a horizontal band saw. The part was placed into a manual lathe, using the tailstock for centering, and a ¼"-20 die was used to cut the threads by hand.

Figure 6.2.2. Completed Locking Pin and Handle.

The second-to-last part to be manufactured was the square tube framing. The overall length was adjusted to accommodate the aforementioned changes in height of the base and guiding plate. The stock was cut to length using a horizontal band saw. The part was then clamped in the vise of a manual mill and a 9/32" 4-flute end mill was used to create the slot for travel of the locking pin.

After speaking with a professor in the Manufacturing Department of Cal Poly, a new part was recommended for the locking mechanism. A delrin insert was designed to fit within the square tubing to guide and support the locking pin throughout travel as well as during blade removal. The insert required contouring to match the contour of the square tubing, deep drilling to allow the locking pin to travel through, and a slot for the locking handle to pass during lock travel. The CAM to generate the code to machine the insert was created using HSMWorks. The first step was to cut the stock for the insert which was completed on a horizontal band saw. Next, a Haas Mini Mill was used for machining and started by using a ¾" 3-flute end mill to contour the delrin to the internal geometry of the square tube. A ½" drill was then used to drill the thru hole for the shaft, and then the part was removed. A horizontal band saw was used to cut the remaining stock. Lastly, the slot for the lock handle was created using the same 9/32" 4-flute end mill on a manual mill.

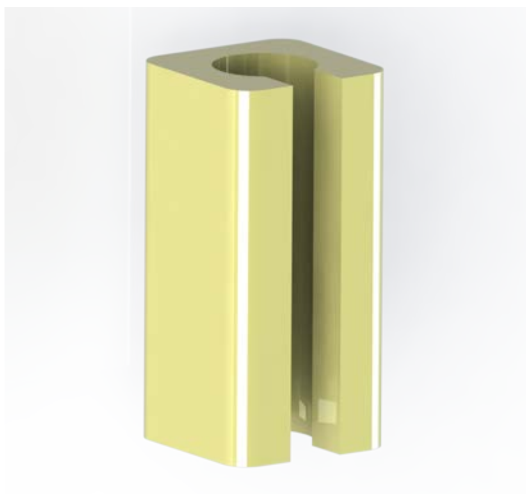

Figure 6.2.3. Isometric Render of the Delrin Insert.

Upon completion of manufacturing of each individual component, assembly of the locking mechanism took place. First, the guiding plate was MIG welded to the square tubing and then the delrin insert was pressed into the tubing from the opposite end. The base plate was then MIG welded to the bottom end of the square tubing, and then the pin and handle was inserted.

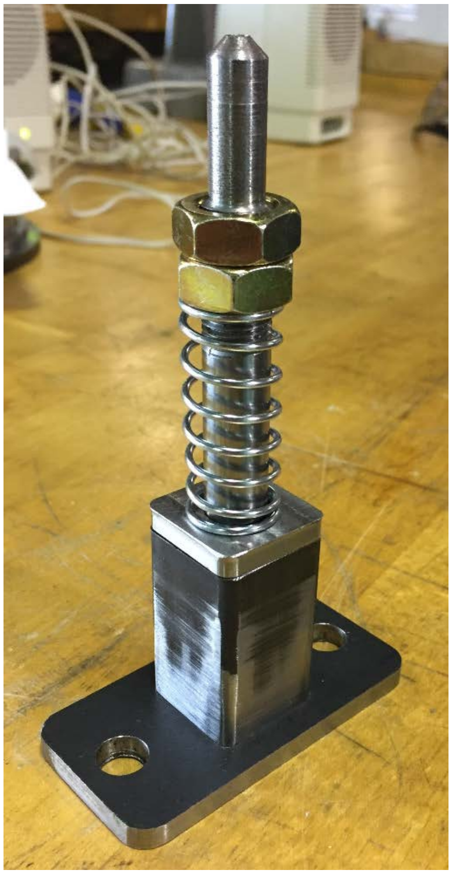

Figure 6.2.4. Completed Locking Mechanism.

6.3 Rotating Hub Assembly

The off-the shelf trailer hub selected for the base of the rotating hub assembly only required slight modification for use in the Turbine Blade Removal Tool. The ½”-20 studs supplied with the trailer hub that extend outward from the flange used to secure the stacked components to the rotating base were too short for use on the tool. Longer studs with matching interference fit taper and knurl were installed in place of the supplied studs. Effective length of the replacement studs is 2.85”, extending usable thread area by 1.20” longitudinally. A simple arbor press was used to press the factory studs out and install the replacement studs.

Figure 6.3.1. Trailer Hub in as-ordered condition.

The hub spacer used to increase clearance between the trailer hub bearing support and the disk mounting location was designed to be a simple-to-manufacture part. Requiring simple 2-D milling operations allowed for its creation at the Cal Poly Machine Shop. Starting from 6.5" nominal diameter, 1.125" nominal thickness 6061-T6 stock, the component was faced on each flat surface, followed by simple contour operations to create the necessary holes in the proper locations. To constrain the stock in the mill a set of soft-jaws, also created from excess 6061 material, were cut to match the contour of the rough stock. This allowed for more secure mounting of the stock in the mill to prevent the part from dislodging under cutting loads. This also allowed for precision flatness post-machining as the soft-jaws were cut by the same machine that completed the finishing operations on the hub spacer, ensuring flatness tolerances were met.

A Haas VF-2 was used to complete all machining operations. Once the stock was installed in the machine, a 3" 4-flute HSS face mill was used to machine .250" off of the first face in five .050" depth passes. Once completed, a ¾" 2-flute HSS end mill was used to contour cut the center hole that allows the trailer hub bearing support to pass through, as well as the five holes that allow the threaded studs to pass through from the trailer hub flange face. The hub spacer was then removed from the machine, rotated 180°, and faced on the back side. Three .050" passes removed .150" of material. All machined surfaces maintained a finish of 64 Ra or better.

Using a precision work table and a dial indicator, flatness of the machined hub spacer was tested to ensure .010" flatness was maintained across both faces. Maximum deviation across one face was .007", demonstrating the the design tolerance was met. Parallelism was also measured, maximum deviation was found to be .005" across the length of the part at various sections. Post-manufacture testing proved the part was machined successfully, meeting all design criteria.

Figure 6.3.2. Hub Spacer on top of the Trailer Hub (longer studs not shown).

The most complex part of the rotating hub assembly is the precision hub plate. With significant complex geometry, tight tolerancing, and precise relationships between features, significant machining was required to complete this part. Due to its sheer size, at over 15" in diameter, the group was unable to construct this part in San Luis Obispo, as no equipment had the ability to machine a part this size. Solar Turbines' Manufacturing Engineering department volunteered to construct the part on the group's behalf. Drawings and CAD models were sent to the manufacturing engineering group at Solar Turbines in San Diego, CA. A meeting was held with the project sponsor to modify the design to conform to Solar's requirements while also enhancing the machinability of the part. Solar used in-house tooling used for gas-turbine rotor disk production to make a one-off hub plate for the prototype Turbine Blade Removal Tool and shipped it to the group in October 2017. Although the group did not make the CAM for this part; facing, turning, drilling, reaming, and threading operations were used in its construction.

Upon receiving the part from Solar's manufacturing department, the item was scrutinized to ensure it conformed to the final design. One error was found in that the 52 holes used in conjunction with the locking mechanism were chamfered on the incorrect face, an error deemed non-critical and acceptable by the group. Flatness across the part was maintained at a maximum of .005" variance, and parallelism at .003" variance using metrology tools in the Cal Poly Machine Shop. The hub plate was assembled with the hub spacer and trailer hub, demonstrating proper operation as designed with all other components.

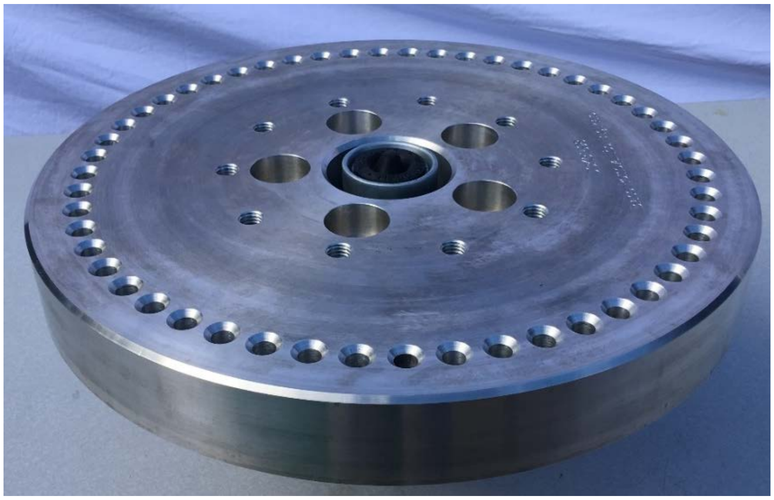

Figure 6.3.3. Completed Hub Plate as received from Solar Turbines.

The rubber pad used to cushion the turbine rotor disk assembly, that sits on top of the hub plate was received from the supplier as nominal 14” x 14” x ½” sheet stock. To create the round pad, a stencil was created from the CAD model of the rubber pad. This stencil was attached to the rubber sheet to aid in guiding a manual cut of the rubber using a vertical band saw. The ten thru holes seen in the CAD model were created using a drill press with heated drill bits. The stencil marked the location of each hole to ensure drilling accuracy. Although not a precision part, significant efforts were made to ensure the rubber pad met design standards, however, no concrete dimensions were required for this part, and nominal dimensions were all that were required in the design. Post-manufacture analysis proved the rubber pad was created successfully and will suffice for prototype testing.

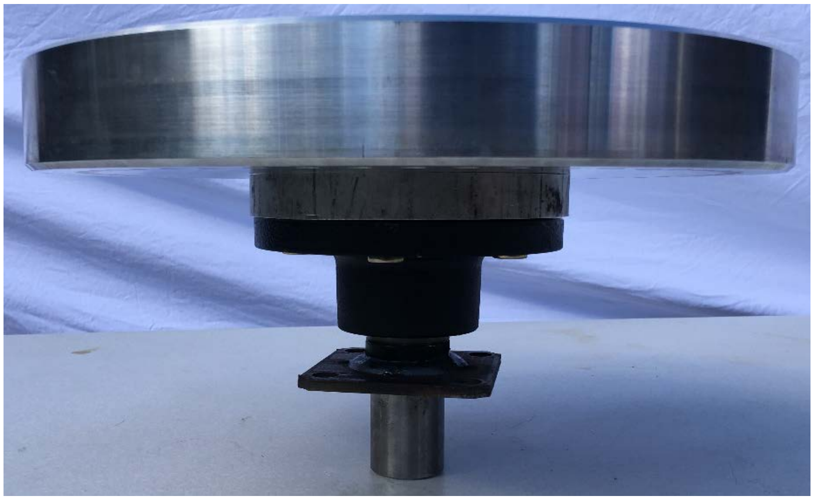

Figure 6.3.4. Rotating Hub Assembly.

7 Post-Assembly Testing

After completing the turbine blade removal tool prototype, significant testing was completed to ensure product durability, accuracy, and conformance to design standards. The adjustable tool holder subassembly was tested for adjustability in many configurations to determine if the prototype could be used across many product lines, where the rotor disk assemblies vary in size and geometric form. Successful testing demonstrated that height, broach angle, and radial length adjustability performed as expected. Fellow students unaffiliated with the group developing the Turbine Blade Removal Tool were asked to adjust the tool holder in various fashions to determine if the device was easy to use, comfortable, and manipulatable by a single user. Positive feedback was received with only a few comments regarding improvements that could be made, discussed later in the future recommendations section.

The hub assembly was tested with the locking mechanism as a unit. The hub was first tested for rotation and runout. Minimal runout occurred, less than .010 in rotation once mounted to the table. Rotation was easy to initiate with a light push to the hub plate. The lock was mounted to the table and adjusted to fit perfectly in radial alignment with the 52 locking holes in the hub plate. The 3-D printed sector model was mounted to the hub assembly, and was placed under the impact tool. Upon rotating the hub plate to each of the 52 locking locations, it was found that each individual fir tree broach maintained alignment with the impact tool head as rotation proceeded.

The locking mechanism was tested for pin security and ease of release and engagement. After creating a wear surface between the pin and the delrin bushing inside the lock, the locking mechanism became easy to release with approximately 15 pounds of downward force on the lock handle. When released, the installed retention spring quickly caused the pin head to extend upward into the hub plate, securing the disk in place.

All assemblies performed as expected, and the group determined that the initial prototype needed no major modifications prior to unveiling at the senior project expo. All components were painted after testing to prevent corrosion and enhance visual appeal, and the project was prepared for display. Upon final assembly a safety check was performed, to ensure no sharp edges or pinch points existed without visual warning. Review with the Cal Poly Mechanical Engineering department representatives led to successful sign off of the project as a safe device for demonstration.

Table 7.1. Testing Summary.

Spec. #	Parameter Description	Requirement or Target	Tolerance	Risk	Compliance	Status
1	Rotor Damage	None	Go / No-Go	H	A, T, I	Test
2	Blade Damage	Minimal	Go / No-Go	H	A, T, I	Test
3	Human Driven Force	50 lbf	Max	M	A, T	Pass
4	Tool Cost	$4,000	Max	M	A	Pass
5	Blade Removal Rate	140 blades/hr	Min	M	A, T	Test
6	Tool Assembly Weight	4000 lbf	Max	L	A, I	Pass
7	User Approval	80%	Go / No-Go	L	T	Test
8	Service Interval	1 Disk Overhaul	Min	L	T	Pass
9	Tool Lifetime	100 Disk Overhauls	Min	L	A, T	Test
10	Service Time	1 Hour	Max	L	T	Pass
11	Shop Resources	2	Max	M	A	Pass
12	Training Time	1 Hour	Max	M	A	Test
13	Manufacturing Time	8 Hours	Max	M	A, T	Fail

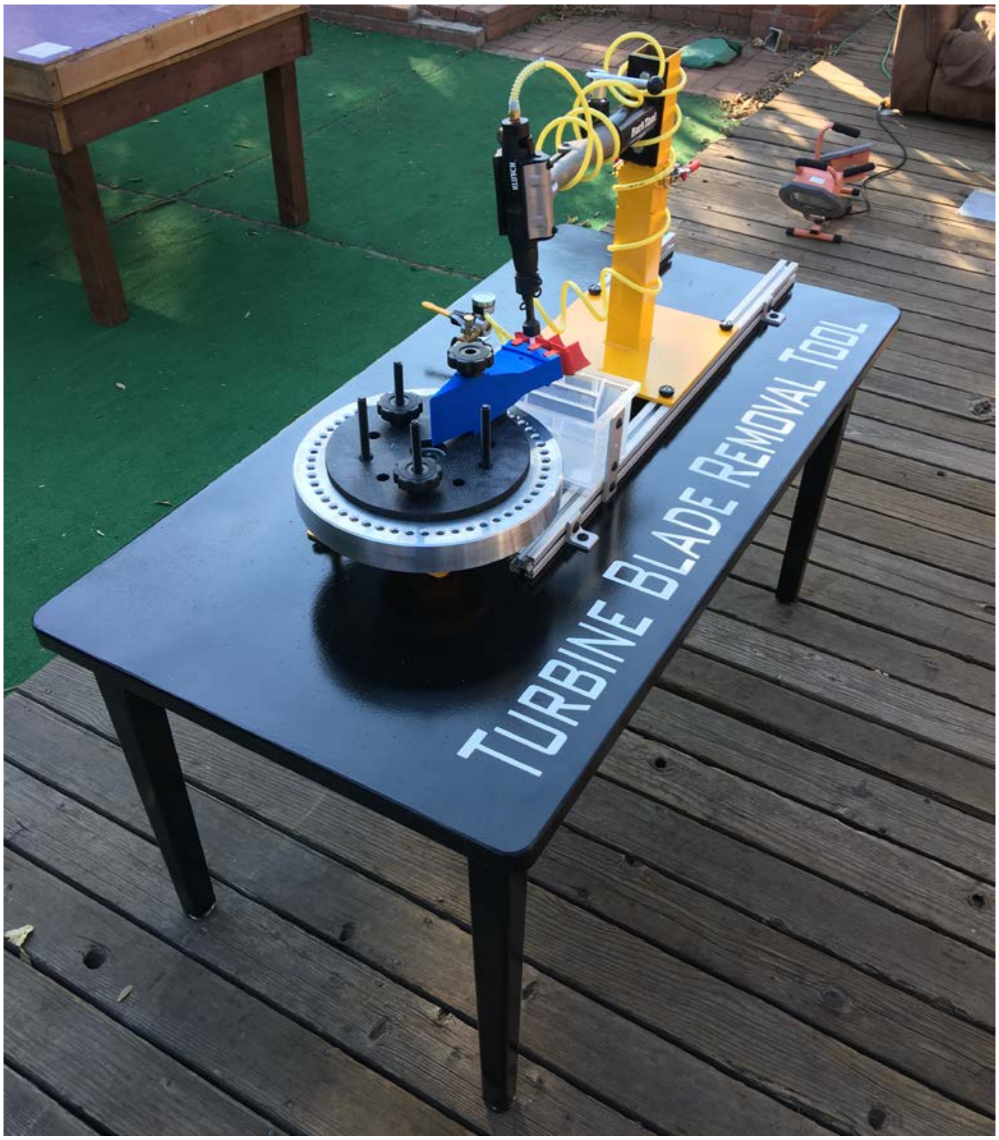

Figure 7.1. Finished project after testing.

8 Safety

The constructed prototype encompasses many features that will ensure safety to the user. Using a table-based setup allows all components to be supported by the floor, removing the need to have the device supported by a technician. The remote punch system allows users to remove their hands from the setup during blade impacting, preventing operator injury. Using a pneumatic setup only requires the operator to control a single valve to initiate tool operation, removing the need for human force input. Minimizing the need for high voltage electrical ensures operators are protected from injury due to shock, and pressurized air will be a maximum of 100 psi as per Solar's shop air standards. No high-speed rotating equipment exists on the tool, and pinch points have been kept to a minimum. Any hazardous or sharp locations on the tool have been clearly marked to ensure clear visual warning to operators and passerby.

9 Future Recommendations

The first suggested change for a future iteration of the Turbine Blade Removal Tool would be to create a more mobile tool, given the end goal of presenting this project at the Exposition and the availability of a table at no cost from Cal Poly Surplus, the table mounting was the best choice for this iteration. However, a mobile work surface, such as a standalone base that could be placed on different tables, or a wheeled device, would enable the tool to be moved to different disassembly bays or facilities as needed.

A variation on the above mentioned recommendation is to have the air controls and lock mounted to the sliding rails that support the tool holder. This would allow the entire tool to be moved separate from the table, eliminating any attachments besides the rails and hub assembly to the mounting surface. This would greatly improve mobility of the tool and create the possibility of moving the tool by hand with two operators. An additional benefit to assembling the tool in this configuration would be a dramatic reduction in the required storage space in Solar's facilities, enabling Solar to keep more of these tools on hand for rapid disassembly, potentially even in the field.

A suggestion made by many people who used our prototype tool was that the lock handle should be extended to prevent the user's arm from coming in contact with the hub plate and rotor disk assembly, once mounted. This improvement would significantly improve the user experience, and would allow for quicker lock operation between impacts. The current locking mechanism uses a short linear handle that extends from the lock base, which may be found awkward during use, especially for tall users. A revision to this piece would be simple for a second iteration and would remove the most common complaint from people polled to test the tool.

When discussing the need to construct a hub plate for each disk type and blade count to be used on the Turbine Blade Removal tool, it was suggested that labels for each plate be etched onto the outer edge of each hub plate. This would allow for faster tool assembly between disks, as users would be able to easily identify the proper hub plate needed for a specific disk. Since the hub plates are large, it would be cumbersome for a user to have to search for a specific disk if proper labeling did not exist when the plates are stored in a storage bay. Etching the edges with a part number and description would be a simple operation during plate machining.

The lever arm distance that the Park Tool bike clamp provides for the impact tool is significant. A bike clamp was used as the tool holder to reduce prototype cost and to simplify construction due to time constraints. Ideally, this lever arm would be reduced to a minimum to reduce tool vibrations and reduce deflection at the impact point. Since the tool holder assembly is radially adjustable for length, the lever arm that holds the impact tool does not need to be long, just longer than the greatest distance between blade root and tip for the largest disk desired for disassembly.

Figure 9.1. Final Project and Group Photo.

10 References

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11 Appendices

11.1 Appendix A: Quality Function Deployment Matrix and Pugh Matrix

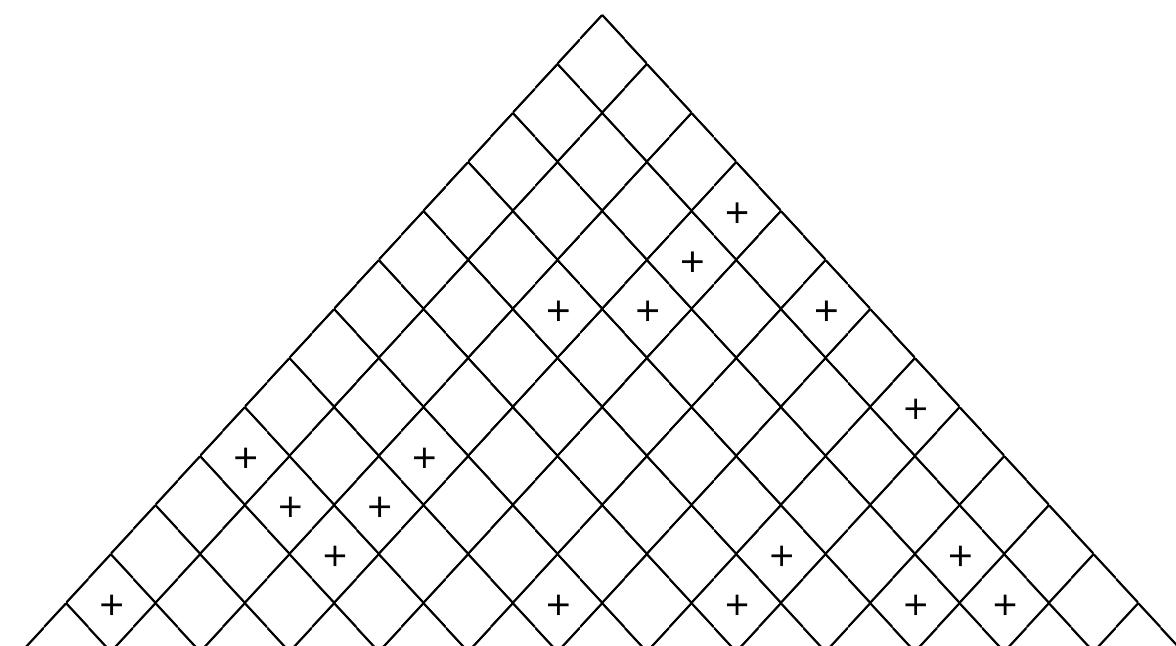
See Next Page(s)

Appendix A: Quality Function Deployment (QFD) matrix.

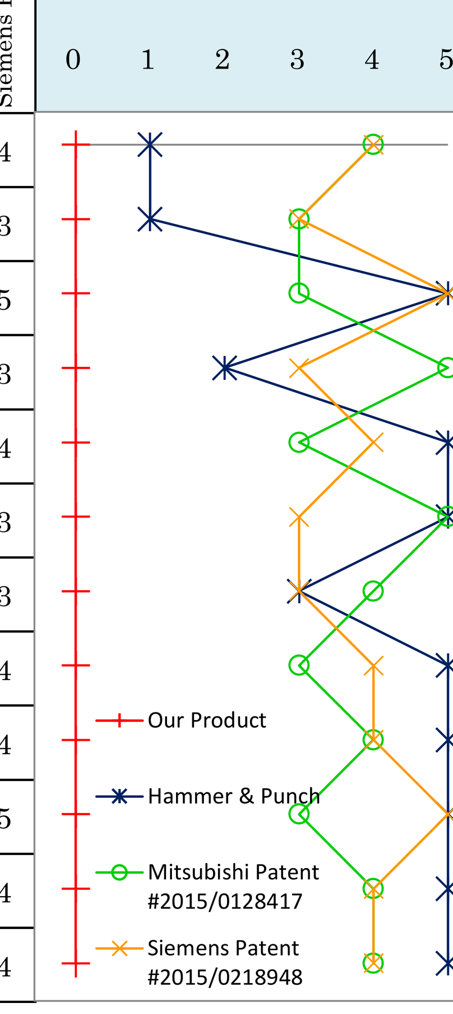
Correlations	
Positive	+
Negative	−
No Correlation	

Relationships	
Strong	●
Moderate	○
Weak	▽

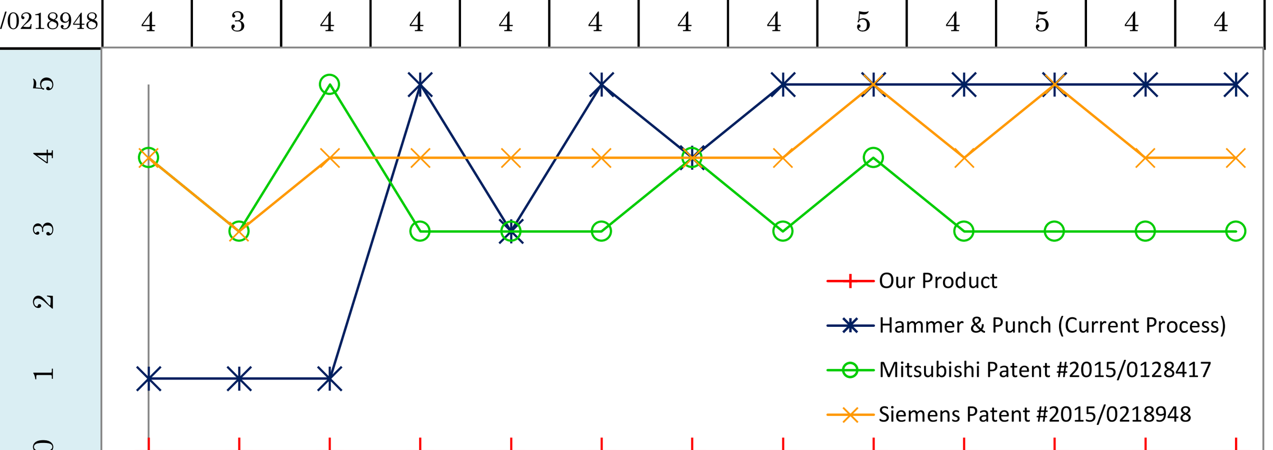
Direction of Improvement	
Maximize	▲
Target	◇
Minimize	▼



									Column #	1	2	3	4	5	6	7	8	9	10	11	12	13				
	WHO: Customers								Direction of Improvement	◇	◇	◇	▼	▲	◇	◇	◇	◇	▼	▼	▼	▼	NOW: Current Product Assesment - Customer Requirements			
Row #	Weight Chart	Relative Weight	Solar Turbines Mechanical Engineers	Solar Turbines Mechanics, Overhaul, Assembly	Solar Turbines FSR	Tool Engineers and Manufacturing	**Total Score**	**Overall Rank**	HOW: Engineering Specifications / WHAT: Customer Requirements (explicit & implicit)	Visual Inspection of Disk Damage	Visual Inspection of Blade Damage	Meets 2017 OSHA Tool Guidelines	Manufacturing Cost	Blades Removed Per Hour	Portable by Single Person	Ergonomic by user survey	Number of uses before service	Number of uses before replacement	Employee Use Training Time	Number of Shop Resources Required	Repair and Service Time	Manufacturing Tooling and Time	Our Product (Not Yet Designed)	Hammer & Punch	Mitsubishi Patent #2015/0128417	Siemens Patent #2015/0218948
1		15.38%	12	12	12	12	48	1	Disk Damage Prevention	●				○									0	1	4	4
2		5.45%	9	3	4	1	17	12	Blade Damage Prevention		●			●									0	1	3	3
3		8.65%	4	11	8	4	27	5	Easy to Use			○		▽	○	●			●	○			0	5	3	5
4		8.33%	8	10	2	6	26	7	Ergonomics of Tool Use / OSHA Approval			●		▽	○	●			▽	▽			0	2	5	3
5		7.69%	1	7	9	7	24	8	Easily Transportable			○			●	○			○				0	5	3	4
6		8.65%	10	4	10	3	27	4	Multiple Disk Configuration	▽	▽		○	○		▽	○	○	●	○	●	●	0	5	5	3
7		9.29%	11	5	11	2	29	2	Rapid Blade Removal	▽	●		●	●			○	○	●	○	▽	○	0	3	4	3
8		6.41%	7	1	1	11	20	9	Low Manufacturing Cost			▽	●		▽	▽	○	○		○	○	●	0	5	3	4
9		8.33%	3	9	6	8	26	6	Low Maintenance				○	▽			●	●	○	○	●	○	0	5	4	4
10		6.41%	2	6	7	5	20	9	Requires Minimal Shop Resources				○			▽			●	●	▽		0	5	3	5
11		8.97%	6	8	5	9	28	3	High Reliability				●	○			●	●		●	●	○	0	5	4	4
12		6.41%	5	2	3	10	20	9	Long Tool Life				●	▽			○	●			●	●	0	5	4	4



	1	2	3	4	5	6	7	8	9	10	11	12	13
HOW MUCH: Target	Go / No-Go	Go / No-Go	<100 lbf of human driven force	<$1000	>35 blades per hour	<50 lbf section weight	>80% of users approve tool	No service required during one disk overhaul	>100 disks	<1 hr	<=2 external mechanical drivers	<1 hr	Tool manufacturing completed in < 1 shift
Max Relationship	9	9	9	9	9	9	9	9	9	9	9	9	9
Technical Importance Rating	156.41	141.35	130.45	350	263.46	126.6	197.44	248.08	286.54	353.53	270.83	326.28	273.08
Relative Weight	5%	5%	4%	11%	8%	4%	6%	8%	9%	11%	9%	10%	9%
Weight Chart													
Current Product Assesment - Engineering Specifications: Our Product (Not Yet Designed)	0	0	0	0	0	0	0	0	0	0	0	0	0
Hammer & Punch (Current Process)	1	1	1	5	3	5	4	5	5	5	5	5	5
Mitsubishi Patent #2015/0128417	4	3	5	3	3	3	4	3	4	3	3	3	3
Siemens Patent #2015/0218948	4	3	4	4	4	4	4	4	5	4	5	4	4



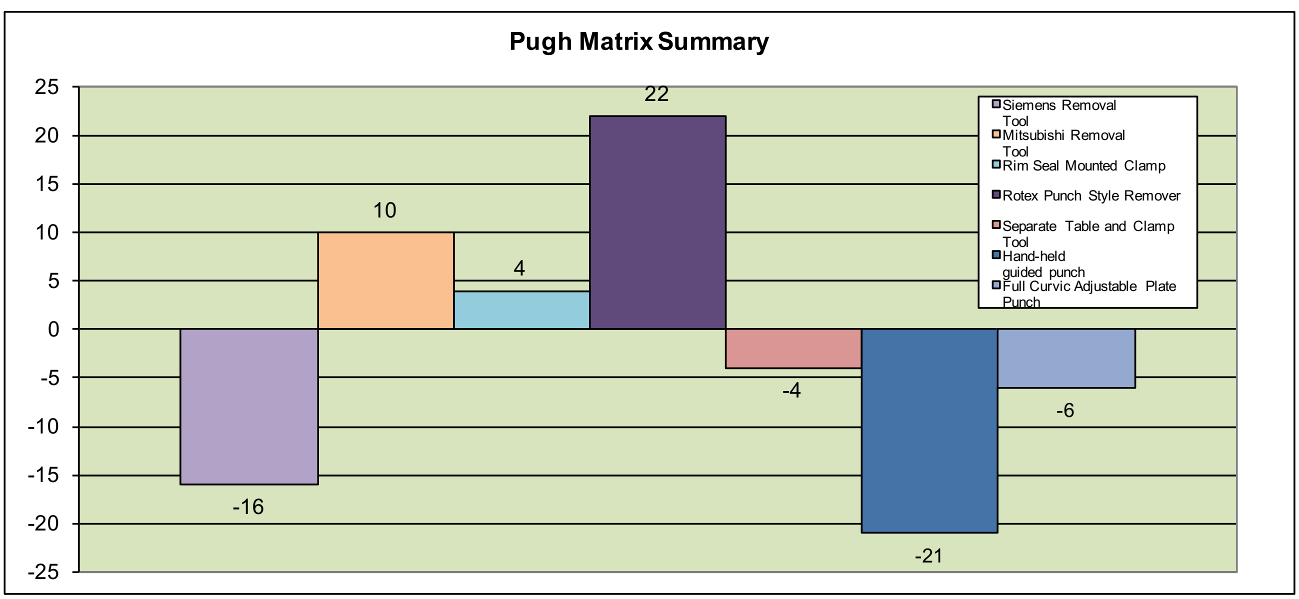
Column #	1	2	3	4	5	6	7	8	9	10	11	12	13

Appendix A: Pugh Matrix

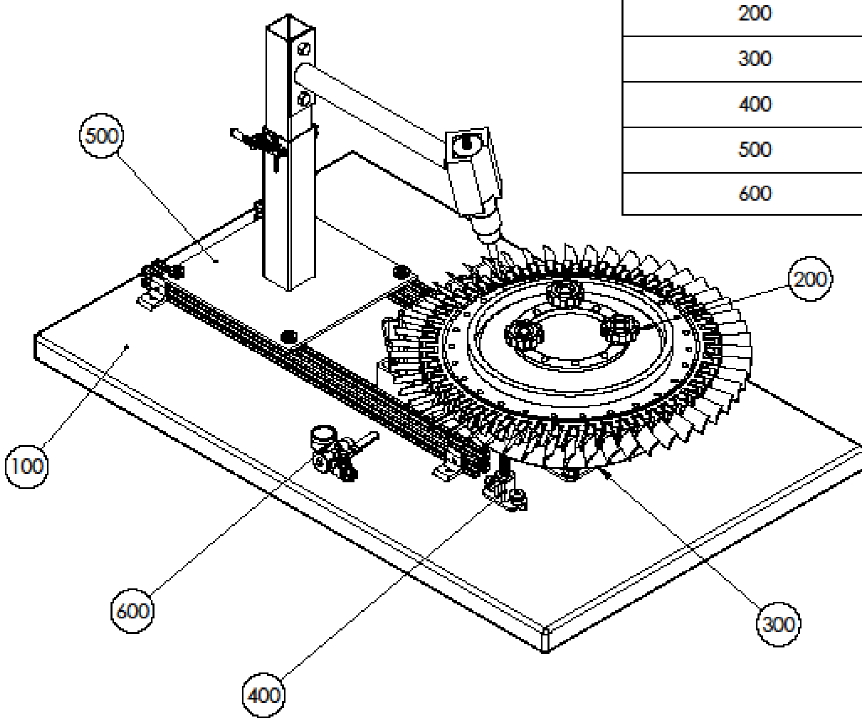
Pugh Matrix

Concept Selection Legend	
Better	+
Same	S
Worse	-

Key Criteria	*Importance Rating*	Current Solar Removal Process	Siemens Removal Tool	Mitsubishi Removal Tool	Rim Seal Mounted Clamp	Rotex Punch Style Remover	Separate Table and Clamp Tool	Hand-held guided punch	Full Curvic Adjustable Plate Punch
			Solution Alternatives						
Effectiveness in Use	**10**		+	+	+	+	+	S	+
Ease of Use	**5**		-	+	-	+	-	-	+
Manufacturing Cost	**5**		-	-	-	-	-	S	-
Durability	**6**		+	-	+	+	+	S	-
Adjustability	**6**		-	+	+	+	+	-	-
Space and Shop Requirements	**3**		S	S	S	-	-	S	-
Product Line Compatability	**2**		-	-	S	+	+	-	-
Efficiency	**8**		-	+	-	+	-	-	+
Maintenance	**6**		-	-	S	-	-	S	-
Portability	**1**		S	S	S	-	-	S	-
Sum of Positives			2	4	3	6	4	4	4
Sum of Negatives			6	4	3	4	6	6	6
Sum of Sames			2	2	4	0	0	0	0
Weighted Sum of Positives			16	29	22	37	24	0	23
Weighted Sum of Negatives			32	19	18	15	28	21	29
TOTALS			**-16**	**10**	**4**	**22**	**-4**	**-21**	**-6**



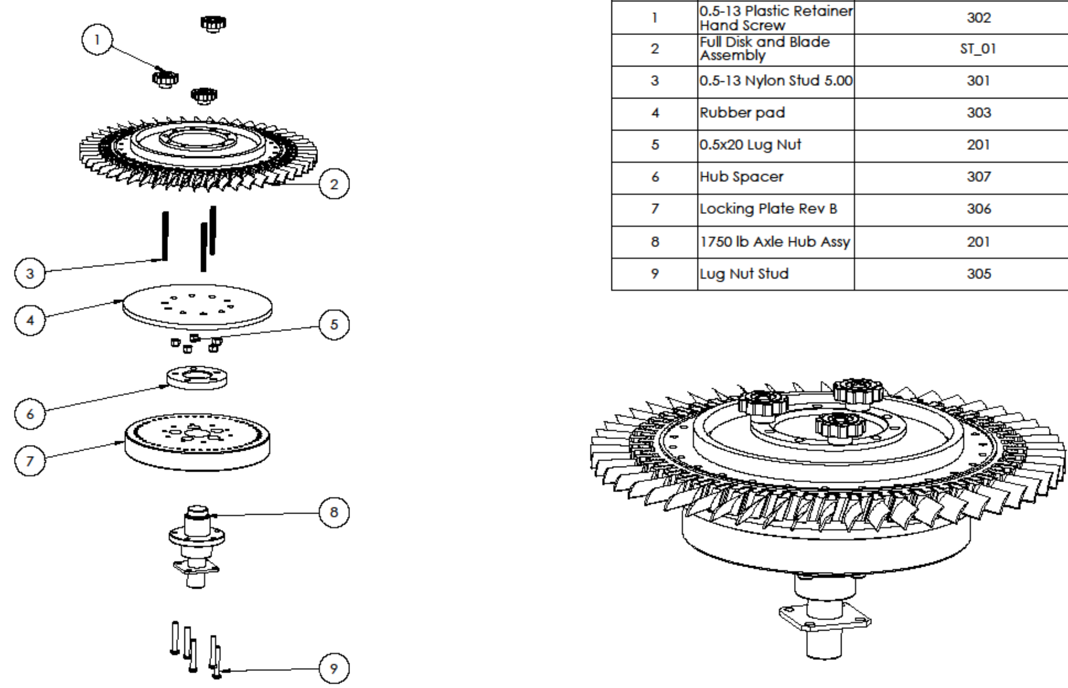
11.2 Appendix B: Drawing Packet (Assemblies with BOM, Detailed Part Drawings, Process/Instrumentation, Electrical Drawings)



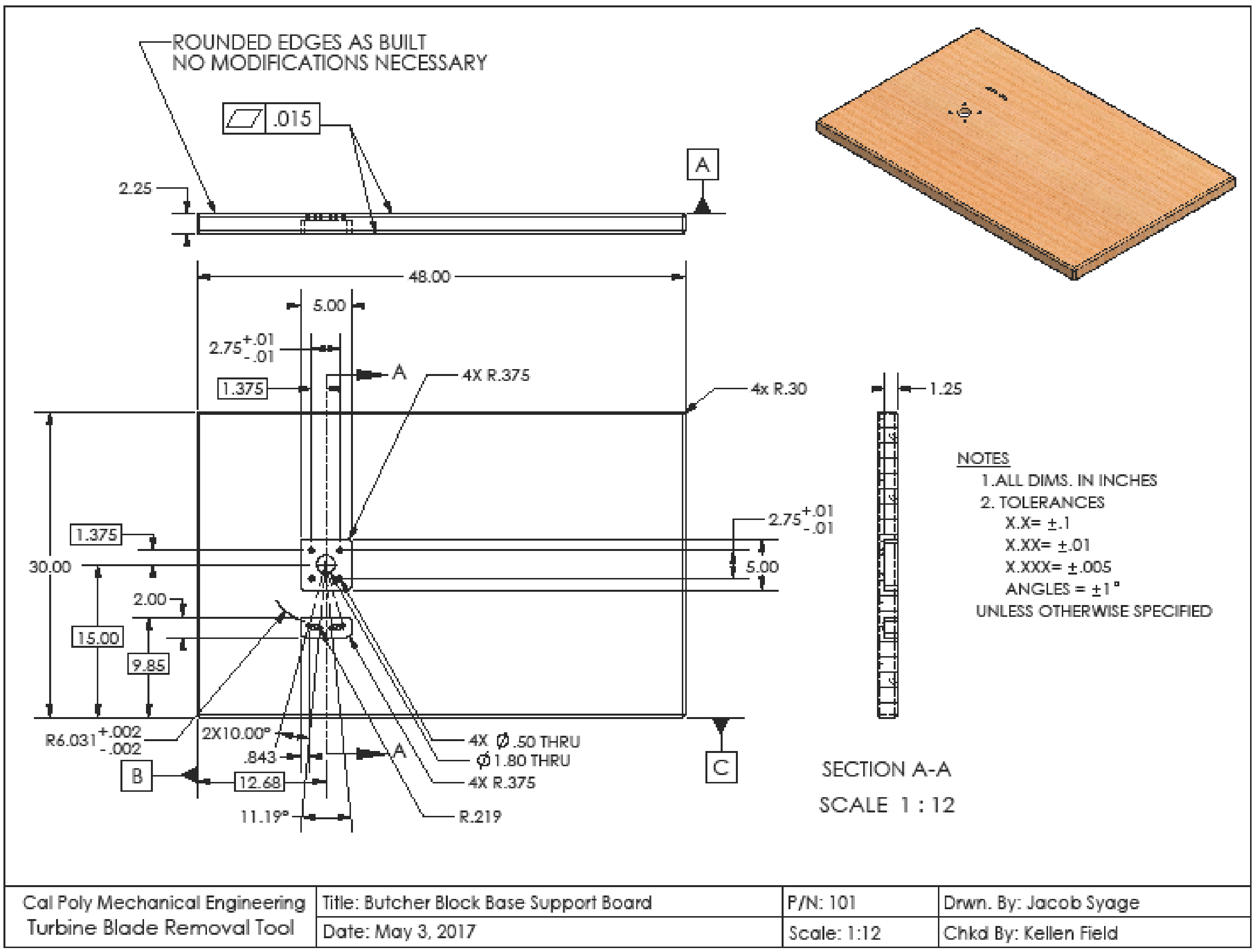
ASSEMBLY NUMBER	DESCRIPTION
100	BUTCHER BLOCK BASE
200	HUB ASSEMBLY UNDER DISK
300	HUB PLATE UNDER DISK
400	LOCKING MECHANISM
500	SLIDING TOOL HOLDER
600	AIR SYSTEM

Cal Poly Mechanical Engineering SOLIDWORKS Educational Product. For Instructional Use Only		Title: FULL ASSEMBLY		Drwn. By: K. FIELD
ME 429 - SPRING 2017	ASSEMBLY #: 100	Date: 5/3/2017	Scale: 1:8	Chkd. By: J. SYAGE

ITEM NO.	ITEM NAME	PROJECT PART NUMBER	QTY.
1	0.5-13 Plastic Retainer Hand Screw	302	3
2	Full Disk and Blade Assembly	ST_01	1
3	0.5-13 Nylon Stud 5.00	301	3
4	Rubber pad	303	1
5	0.5x20 Lug Nut	201	5
6	Hub Spacer	307	1
7	Locking Plate Rev B	306	1
8	1750 lb Axle Hub Assy	201	1
9	Lug Nut Stud	305	5

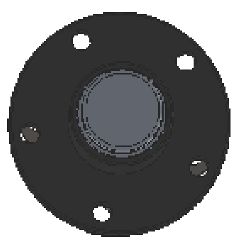


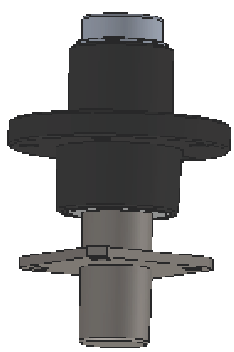
Cal Poly Mechanical Engineering Turbine Blade Removal Tool	Title: Full Hub Assembly	P/N: 200/300	Drwn. By: Jacob Syage
	Date: May 8, 2017	Scale: 1:8	Chkd By: Kellen Field

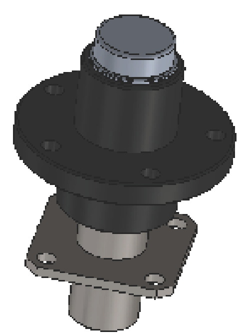


ROUNDED EDGES AS BUILT
NO MODIFICATIONS NECESSARY
.015
2.25
A
48.00
5.00
2.75 +.01 -.01
1.375
4X R.375
4x R.30
1.25
NOTES
1.ALL DIMS. IN INCHES
2. TOLERANCES
X.X= ±.1
X.XX= ±.01
X.XXX= ±.005
ANGLES = ±1°
UNLESS OTHERWISE SPECIFIED
1.375
30.00
2.00
15.00
9.85
5.00
R6.031 +.002 -.002
B
2X10.00°
.843
12.68
11.19°
4X Ø .50 THRU
Ø 1.80 THRU
4X R.375
R.219
C
SECTION A-A
SCALE 1 : 12
Cal Poly Mechanical Engineering
Turbine Blade Removal Tool
Title: Butcher Block Base Support Board
Date: May 3, 2017
P/N: 101
Scale: 1:12
Drwn. By: Jacob Syage
Chkd By: Kellen Field

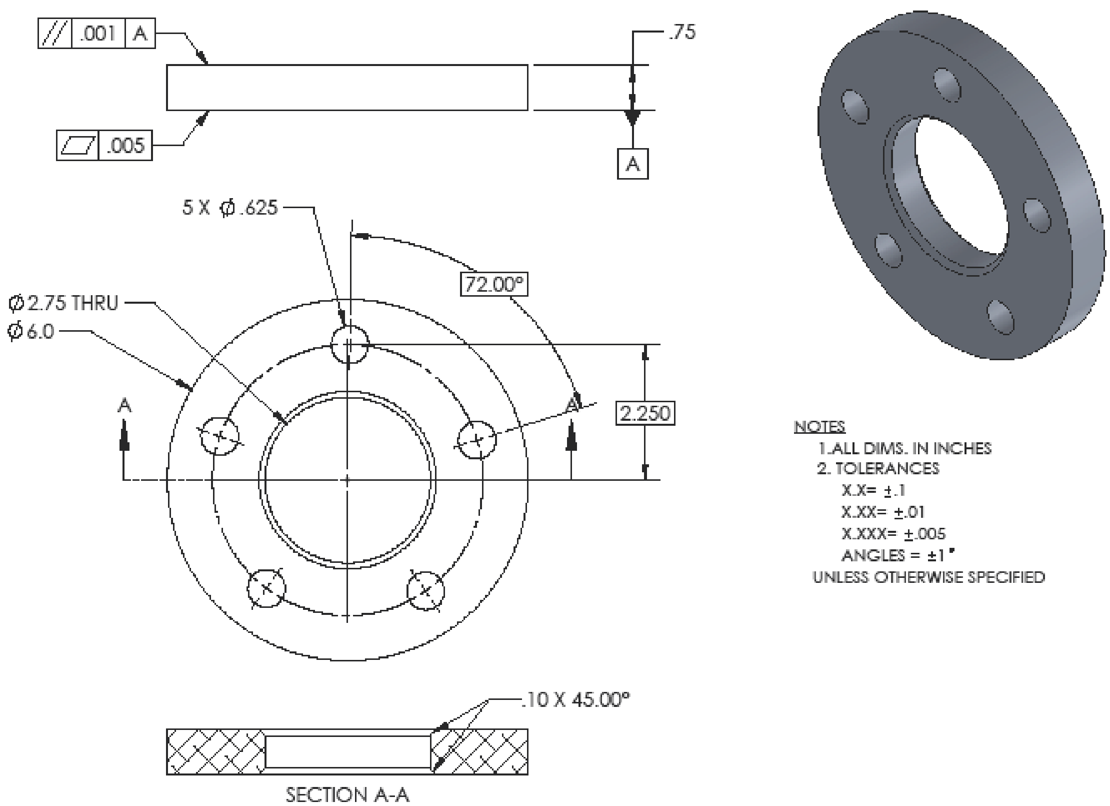
COMPONENET	MANUFACTURER	MANUFACTURER PART NUMBER	PROJECT PART NUMER	QUANTITY
1750LB TRAILER HUB ASSEMBLY	TIE DOWN ENGINEERING	128008	201	1



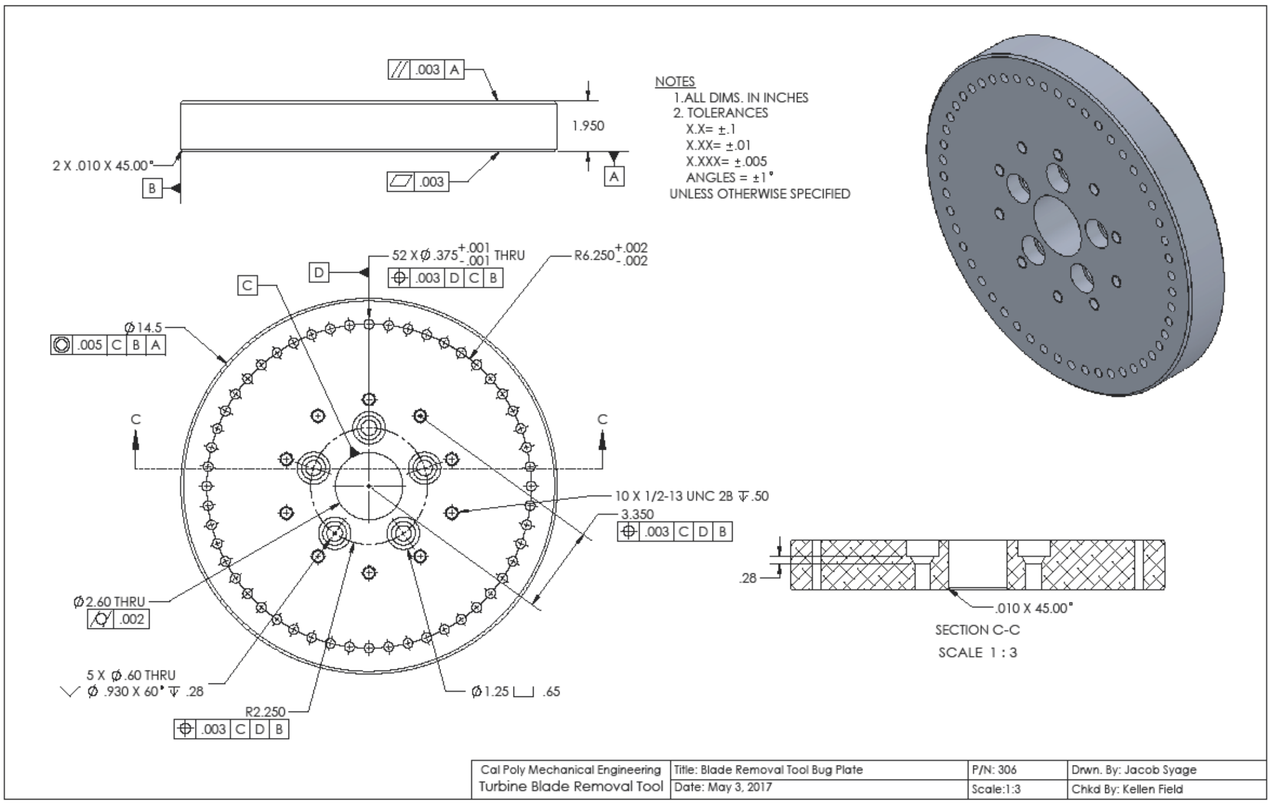




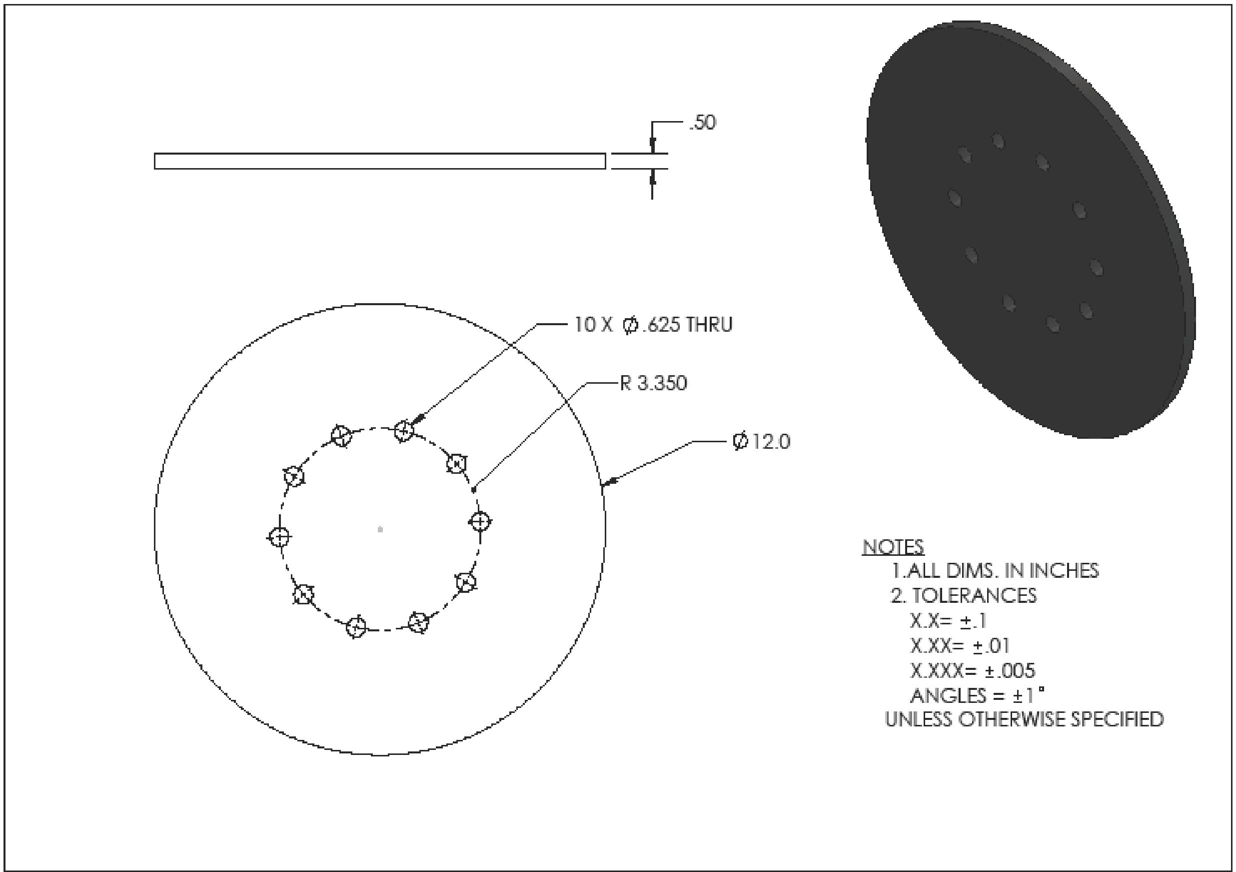
Cal Poly Mechanical Engineering Turbine Blade Removal Tool	Title: Tie Down Engineering 1750 Trailer Hub Assembly	P/N 201	Drwn. By: Jacob Syage
	Date: May 3, 2017	Scale: 1:3	Chkd By: Kellen Field



Cal Poly Mechanical Engineering Turbine Blade Removal Tool	Title: Hub Plate Spacer Block	P/N: 307	Drwn. By: Jacob Syage
	Date: May 3, 2017	Scale: 1:2	Chkd By: Kellen Field



.003 A
1.950
2 X .010 X 45.00°
B
.003
A
NOTES
1.ALL DIMS. IN INCHES
2. TOLERANCES
X.X= ±.1
X.XX= ±.01
X.XXX= ±.005
ANGLES = ±1°
UNLESS OTHERWISE SPECIFIED
52 X Ø .375 +.001 -.001 THRU
.003 D C B
R6.250 +.002 -.002
C
D
Ø14.5
.005 C B A
C
C
10 X 1/2-13 UNC 2B ⊽ .50
3.350
.003 C D B
.28
.010 X 45.00°
SECTION C-C
SCALE 1 : 3
Ø2.60 THRU
.002
5 X Ø .60 THRU
Ø .930 X 60° ⊽ .28
R2.250
.003 C D B
Ø1.25 ⊔ .65
Cal Poly Mechanical Engineering
Turbine Blade Removal Tool
Title: Blade Removal Tool Bug Plate
Date: May 3, 2017
P/N: 306
Scale:1:3
Drwn. By: Jacob Syage
Chkd By: Kellen Field

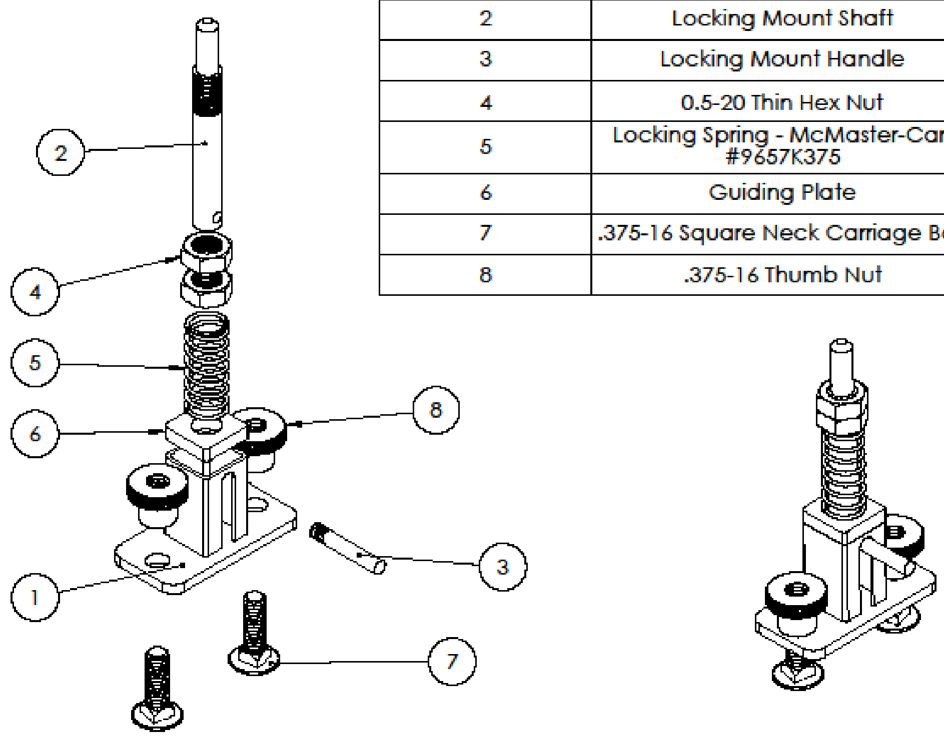


Cal Poly Mechanical Engineering Turbine Blade Removal Tool	Title: EPDM Rubber Cushion Pad	P/N: 303	Drwn. By: Jacob Syage
	Date: May 3, 2017	Scale: 1:4	Chkd By: Kellen Field

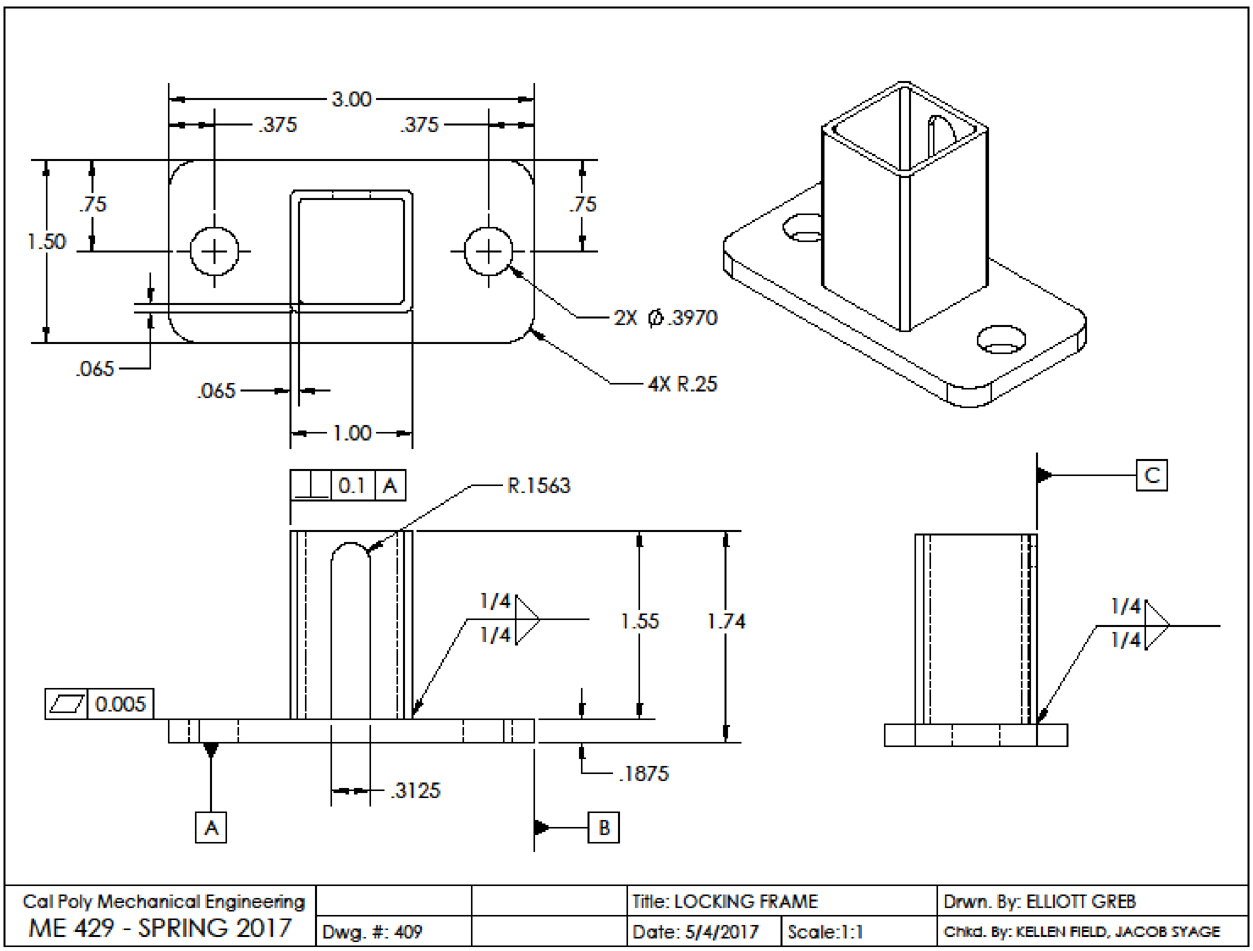
CAL POLY
SAN LUIS OBISPO

PROJECT:	Medium Duty Gas-Turbine Engine Blade Removal Tool				PART:	Removal Tool
TITLE:	Final Design Report				SECTION:	ME 430
ORIGINATOR:	J. Syage, E. Greb, K. Field		DATE:	11/30/2017	REVISION:	A
Reviewer:	J. Hernandez				PAGE: 79 of	178

ITEM NO.	DESCRIPTION	QTY.
1	Locking Mount	1
2	Locking Mount Shaft	1
3	Locking Mount Handle	1
4	0.5-20 Thin Hex Nut	2
5	Locking Spring - McMaster-Carr #9657K375	1
6	Guiding Plate	1
7	.375-16 Square Neck Carriage Bolt	2
8	.375-16 Thumb Nut	2

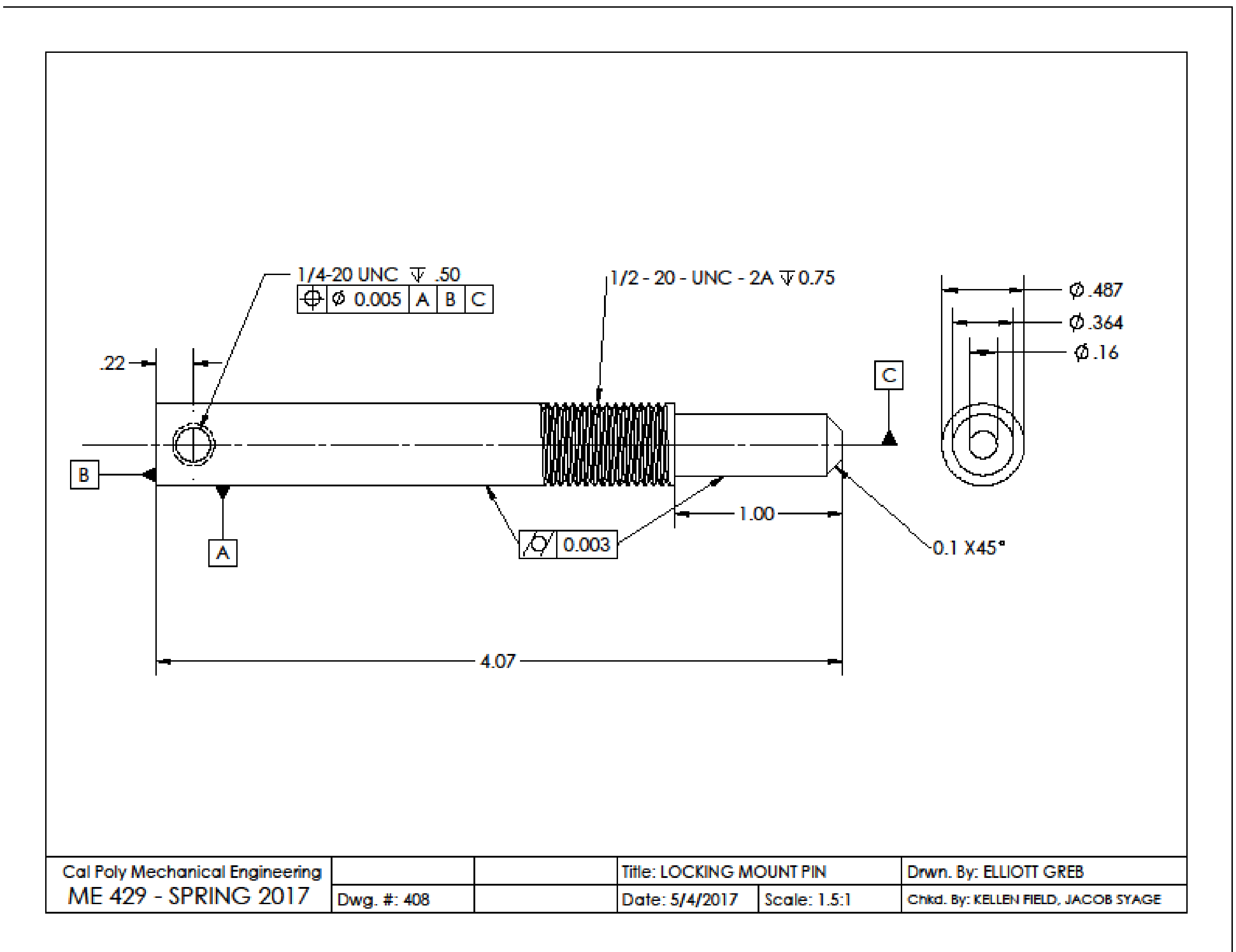


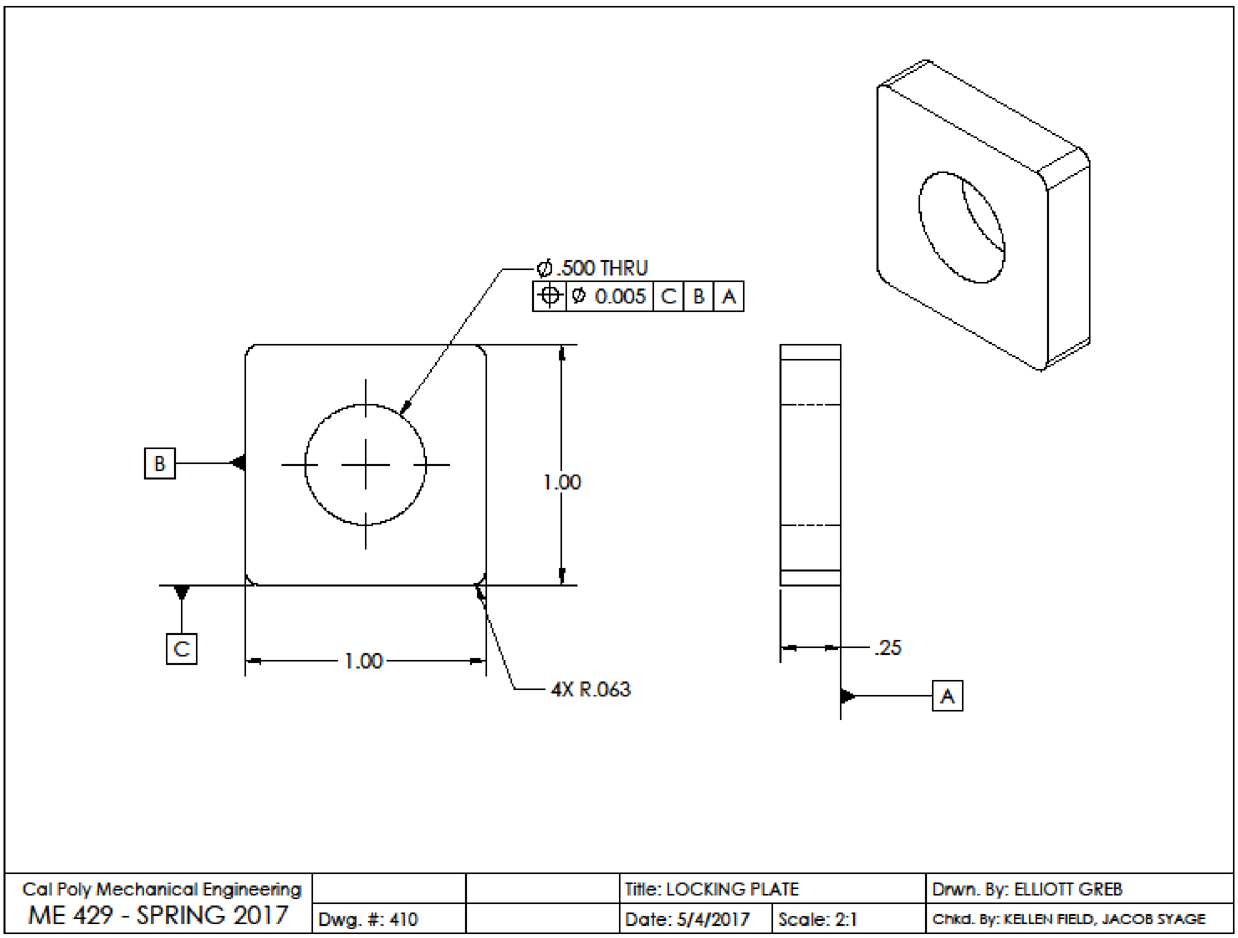
Cal Poly Mechanical Engineering ME 429 - SPRING 2017	Dwg. #: 400		Title: LOCKING MECHANISM ASSEMBLY	Drwn. By: ELLIOTT GREB
			Date: 5/4/2017 / Scale: 1:4	Chkd. By: KELLEN FIELD, JACOB SYAGE



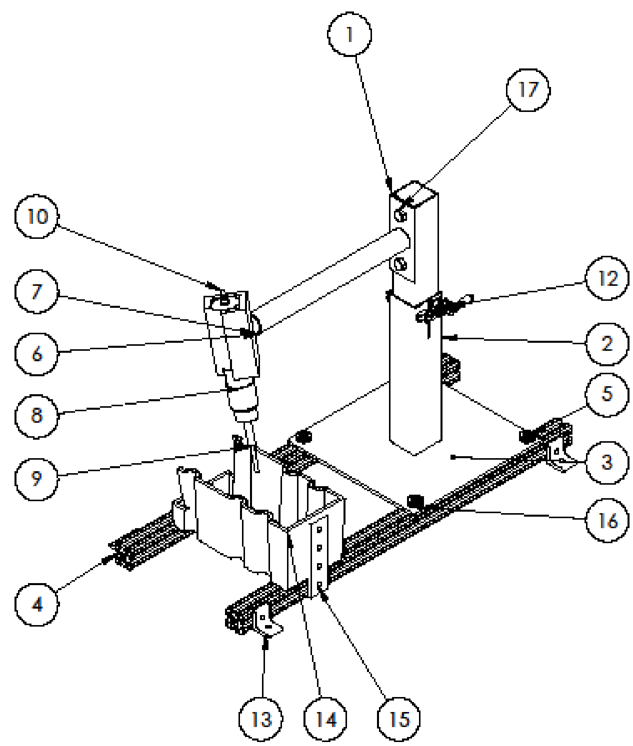
3.00
.375
.375
.75
1.50
.75
2X ⌀.3970
4X R.25
.065
.065
1.00
0.1 A
R.1563
1/4
1/4
1.55
1.74
0.005
.3125
.1875
A
B
C
1/4
1/4
Cal Poly Mechanical Engineering
ME 429 - SPRING 2017
Dwg. #: 409
Title: LOCKING FRAME
Date: 5/4/2017
Scale:1:1
Drwn. By: ELLIOTT GREB
Chkd. By: KELLEN FIELD, JACOB SYAGE

PROJECT:	Medium Duty Gas-Turbine Engine Blade Removal Tool				**PART:**	Removal Tool
TITLE:	Final Design Report				**SECTION:**	ME 430
ORIGINATOR:	J. Syage, E. Greb, K. Field		**DATE:**	11/30/2017	**REVISION:**	A
Reviewer:	J. Hernandez				**PAGE: 81 of**	178



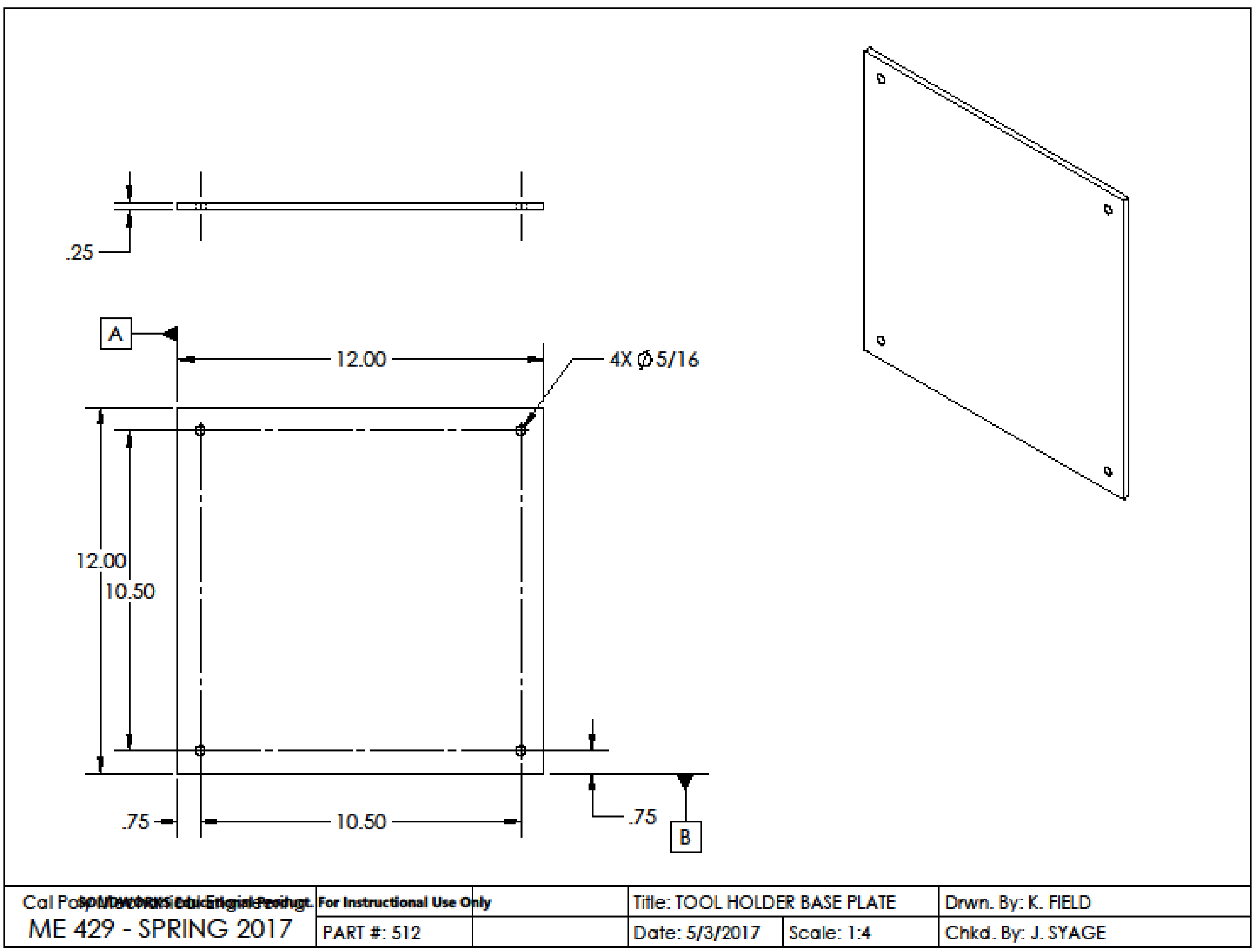


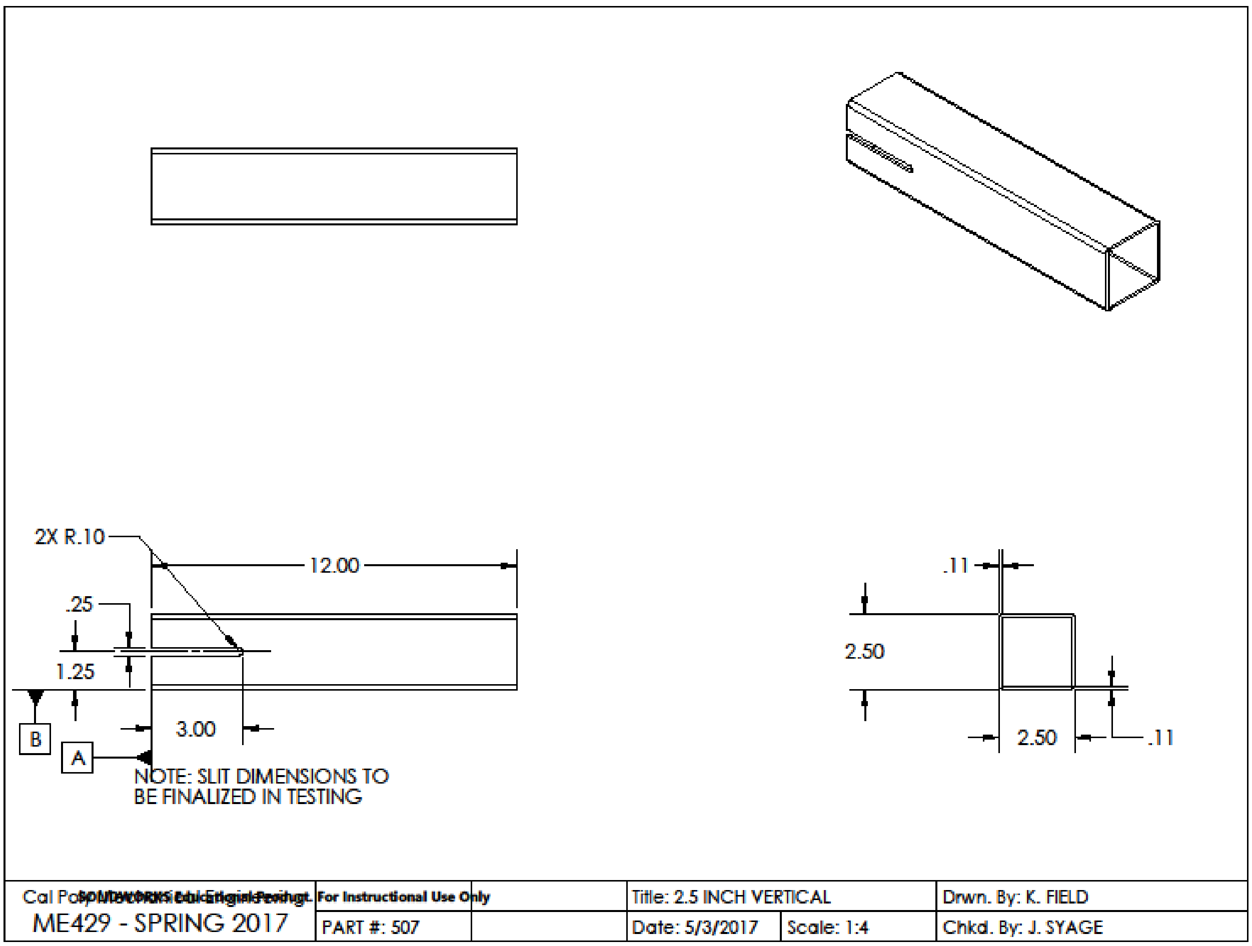
Ø .500 THRU
⌖ Ø 0.005 C B A
B
1.00
C
1.00
4X R.063
.25
A
Cal Poly Mechanical Engineering
ME 429 - SPRING 2017
Dwg. #: 410
Title: LOCKING PLATE
Date: 5/4/2017
Scale: 2:1
Drwn. By: ELLIOTT GREB
Chkd. By: KELLEN FIELD, JACOB SYAGE

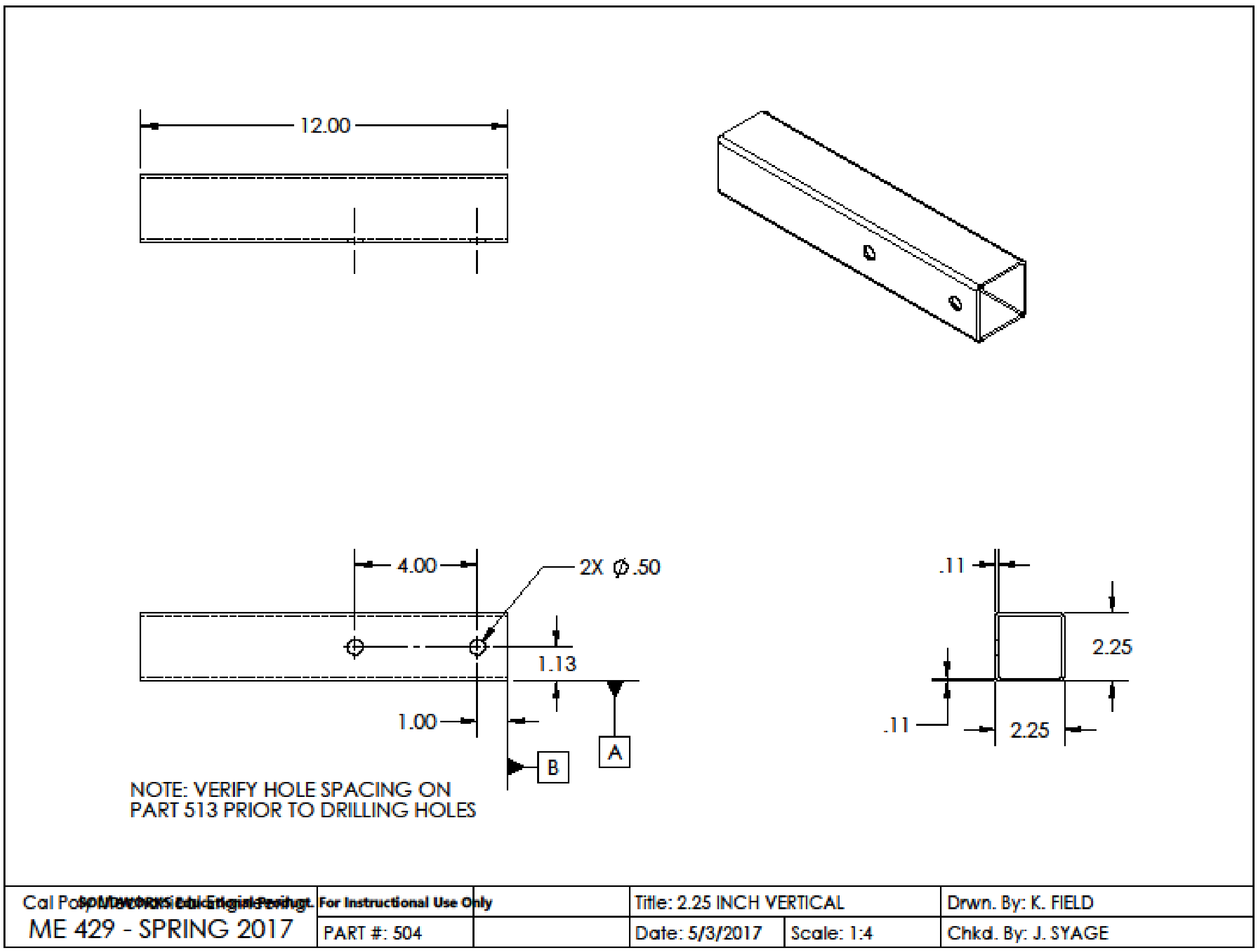


ITEM NO.	DESCRIPTION	QTY.
1	2.25inch Square Tubing	1
2	2.5inch Square Tubing	1
3	Base Plate	1
4	Aluminum T-Slot Rail	2
5	T Nut with 5-16 Stud	4
6	Fixture Arm	1
7	Fixture Claw	1
8	Air Hammer	1
9	Air Hammer Punch	1
10	_25in Male Air Fitting	1
11	1/2-13 LOCKING NUT	2
12	360lb Draw Clamp	2
13	Rail Mounting Bracket	4
14	Blade Bin	1
15	47065T261_ALUMINUM T-SLOTTED FRAMING EXTRUSION	1
16	93886A150_ROUND ACETAL THUMB NUT W SHLDR	4
17	1/2-13X1 HEX BOLT	2

Cal Poly SOLIDWORKS Educational Product. For Instructional Use Only		Title: SLIDING TOOL HOLDER		Drwn. By: K. FIELD
ME 429 - SPRING 2017	ASSEMBLY #: 500	Date: 5/3/2017	Scale: 1:8	Chkd. By: J. SYAGE


.25
A
12.00
4X ⌀5/16
12.00
10.50
.75
10.50
.75
B
SOLIDWORKS Educational Product. For Instructional Use Only
ME 429 - SPRING 2017
PART #: 512
Title: TOOL HOLDER BASE PLATE
Date: 5/3/2017
Scale: 1:4
Drwn. By: K. FIELD
Chkd. By: J. SYAGE


2X R.10
12.00
.25
1.25
B
A
3.00
NOTE: SLIT DIMENSIONS TO
BE FINALIZED IN TESTING
.11
2.50
2.50
.11
Cal Poly Mechanical Engineering Dept.
SOLIDWORKS Educational Product. For Instructional Use Only
ME429 - SPRING 2017
Title: 2.5 INCH VERTICAL
Drwn. By: K. FIELD
PART #: 507
Date: 5/3/2017
Scale: 1:4
Chkd. By: J. SYAGE



12.00
4.00
2X ⌀.50
1.13
1.00
B
A
.11
2.25
.11
2.25
NOTE: VERIFY HOLE SPACING ON
PART 513 PRIOR TO DRILLING HOLES
SOLIDWORKS Educational Product. For Instructional Use Only
ME 429 - SPRING 2017
PART #: 504
Title: 2.25 INCH VERTICAL
Date: 5/3/2017
Scale: 1:4
Drwn. By: K. FIELD
Chkd. By: J. SYAGE

11.3 Appendix C: List of Vendors, Contact Information, and Pricing

See Next Page

Subsystem	Part Number	Description	Supplier	Supplier Number	Item Cost	Package	Quantity	Sub-Total
Base Table (100)	101	60"x30"x2.25" Maple Butcher Block	Butcher Block USA	HD3060	$107.48	1	1	$107.48
Hub Assembly (200)	201	Tie Down Engineering Hub/Spindle Assembly	Northern Tool	128008	$57.99	1	1	$57.99
Hub Assembly (200)	202	1/2-13 2" Flange Bolt Grade 5	McMaster-Carr	92979A479	$308.00	1	1	$308.00
Hub Assembly (200)	203	1/2-13 Flange Nut Grade 5	McMaster-Carr	92018A540	$10.05	10	1	$10.05
Hub Plate (300)	301	1/2"-13 Nylon Threaded Rod	McMaster-Carr	93665A684	$5.93	25	1	$5.93
Hub Plate (300)	302	1/2-13 Comfort-Grip Fluted-Rim Knob	McMaster-Carr	5532T37	$7.32	5	3	$21.96
Hub Plate (300)	303	12"x12"x1/2" Oil-Resistant Buna-N Rubber Sheet	McMaster-Carr	8635K168	$3.28	1	3	$9.84
Hub Plate (300)	304	1"x1"x0.65" 4130 Square Stock Steel	McMaster-Carr	6582K43	$31.94	1	1	$31.94
Hub Plate (300)	305	1/2-20 x 3.00" Wheel Studs	Summit Racing	8020	$20.88	1	1	$20.88
Hub Plate (300)	306	14.5"x1.0" 6061-T6 Extruded Alumium Plate	Speedy Metals	Rd 6061-T6 Aluminum, Extruded	$17.25	1	1	$17.25
Hub Plate (300)	307	6.5" x 1.0" 6061-T6 Extruded Aluminum Plate	Speedy Metals	Rd 6061-T6 Aluminum, Extruded	$59.40	1	1	$59.40
Hub Plate (300)	308	1" 60° Countersink Milling Tool	ICS Cutting Cools	CSKXL160-100	$22.50	20	1	$22.50
Locking Mechanism (400)	401	1/2 - 20 Thin Hex Nut	McMaster-Carr	93839A825	$93.78	1	1	$93.78
Locking Mechanism (400)	402	Flat End Compression Spring	McMaster-Carr	9657K375	$11.53	25	1	$11.53
Locking Mechanism (400)	403	High Speed Steel Chucking Reamer - 0.3750	McMaster-Carr	2995A69	$6.70	6	1	$6.70
Locking Mechanism (400)	404	3/8"-16 x1.25" Square Carriage Bolt	McMaster-Carr	90185A626	$21.44	1	1	$21.44
Locking Mechanism (400)	405	3/8" - 16 Thumb Nut	McMaster-Carr	91833A113	$7.07	25	1	$7.07
Locking Mechanism (400)	406	High Speed Steel Chucking Reamer - 0.5000	McMaster-Carr	2995A74	$9.21	1	3	$27.63
Locking Mechanism (400)	407	Low Carbon Steel Rod - 1/4" Diam - 1Ft	McMaster-Carr	8920K115	$28.05	1	1	$28.05
Locking Mechanism (400)	408	Low Carbon Steel Rod - 1/2" Diam - 1Ft	McMaster-Carr	8920K155	$1.03	1	1	$1.03
Locking Mechanism (400)	409	Low Carbon Steel Sheet - 3"x3"x3/16"	McMaster-Carr	1388K662	$3.46	1	2	$6.92
Locking Mechanism (400)	410	Low Carbon Steel Sheet - 3"x3"x1/4"	McMaster-Carr	1388K102	$9.22	1	2	$18.44
Sliding Tool Holder (500)	501	Klutch Air Needle Scaler	Northern Tool	47927	$11.28	1	1	$11.28
Sliding Tool Holder (500)	502	Double-Locking Pull-Action Toggle Clamp	McMaster-Carr	51335A71	$49.93	1	2	$99.86
Sliding Tool Holder (500)	503	T-Slotted Framing, Single Rail, Silver, 1-1/2" High x 1-1/2" Wide, Hollow	McMaster-Carr	47065T102	$21.78	1	2	$43.56
Sliding Tool Holder (500)	504	Corner Bracket for 1-1/2" High Single Rail, Silver	McMaster-Carr	47065T845	$4.61	1	4	$18.44
Sliding Tool Holder (500)	505	Drop-in Fastener with Stud, for 1-1/2" High Single Rail	McMaster-Carr	47065T234	$1.58	1	4	$6.32
Sliding Tool Holder (500)	506	Zinc-Plated Steel, Number 10 Size, 3/4" Long	McMaster-Carr	91070A245	$9.04	100	1	$9.04
Sliding Tool Holder (500)	507	Solid Tube Framing, Steel, 2-1/2" Square	McMaster-Carr	4931T146	30.32	1	1	30.32
Sliding Tool Holder (500)	508	Solid Tube Framing, Steel, 2-1/4" Square	McMaster-Carr	4931T145	27.36	1	1	27.36
Sliding Tool Holder (500)	509	Plastic Bin Box	McMaster-Carr	4666T63	$5.44	1	1	$5.44
Sliding Tool Holder (500)	510	Extended Straight Bracket for 1-1/2" High Rail, Silver	McMaster-Carr	47065T261	$7.40	1	1	$7.40
Sliding Tool Holder (500)	511	Reinforced Plastic Knurled-Head Thumb Nut	McMaster-Carr	93886A150	$10.76	10	1	$10.76
Sliding Tool Holder (500)	512	12"x12" plate for base of tool holder	Speedy Metals	1/4" 1045HR Steel Plate,	$22.21	1	1	$22.21
Sliding Tool Holder (500)	513	Park Tool Deluxe Wall Mount Repair Stand - PRS-4W	Amazon	Select Silver color	$191.66	1	1	$191.66
Air System (600)	601	Air Regulator	McMaster-Carr	6763K13	$44.46	1	1	$44.46
Air System (600)	602	Air Regulator Bracket	McMaster-Carr	6763K21	$8.32	1	1	$8.32
Air System (600)	603	Straight connector	McMaster-Carr	5485K22	$2.30	1	1	$2.30
Air System (600)	604	Brass On/Off Valve with Spring Close Lever Handle	McMaster-Carr	4088T8	$31.26	1	1	$31.26
						Sub-Total		$1,465.80
						Est. Total with Tax		$1,597.72
						Total with Tax and Shi		$1,890.88

11.4 Appendix D: Vendor Supplied Component Specifications and Data Sheets

11.4.1 Appendix D.1: Klutch Pneumatic Needle Scaler/Hammer Tool

Klutch 2-in-1 Air Needle Scaler and Hammer Kit

Item# 47927 ★★★☆☆ 1 Review | 1 Answered Question

Only $99.99

1 **Add to Cart**

- Use as air needle scaler or hammer
- Lightweight composite housing; tool weighs only 2.8 lbs.
- Low vibration design
- 4,600 BPM
- 12 needles

See full description

Backordered Online — will ship in 30 or more Business Days

View Shipping + Delivery Estimates

Check Store Availability

Hover over image to zoom

[+] What do you think of our product images?

11.4.2 Appendix D.2: 5 Lug Hub/Spindle End Unit

Tie Down Engineering 5-Lug Hub/Spindle End Unit for Build Your Own Trailer Axle System — 1750-Lb. Capacity Per Hub, Model# 80117

Item# 128008 ★★★★★ 6 Reviews | 22 Answered Questions

Hover over image to zoom

Only $69.99

1 **Add to Cart**

- 5-lug hub/spindle end unit
- Fits 2in. x 2in. tube (Item# 124997) and spindle/flange set (Item# 124994)
- Sold individually, 2 end units required per axle
- Includes brake flange

In Stock Online

View Shipping + Delivery Estimates

Not Available in Stores

11.4.3 Appendix D.3: Park Tool Deluxe Wall Mount Repair Stand – PRS-4W

Roll over image to zoom in

Park Tool

Park Tool Deluxe Wall Mount Repair Stand - PRS-4W

22 customer reviews | 4 answered questions

Price: $191.66 & **FREE Shipping**. Details

Only 10 left in stock - order soon.

Want it tomorrow, May 3? Order within **3 hrs 32 mins** and choose **One-Day Shipping** at checkout. Details

Sold by Bicycle Addiction and Fulfilled by Amazon in easy-to-open packaging. Gift-wrap available.

Color: **silver**

Size:

100-3C

- Designed to be easily mounted to any post or wall stud
- Shop quality stand using the same clamp and locking system as the PRS-2, PRS-3 and PRS-4
- Powder coat finish
- Uses 100-3C clamp

11.4.4 Appendix D.4: Compressed Air Regulator

Compact Compressed Air Regulator

Relieving, Aluminum Housing, 1/4 NPT

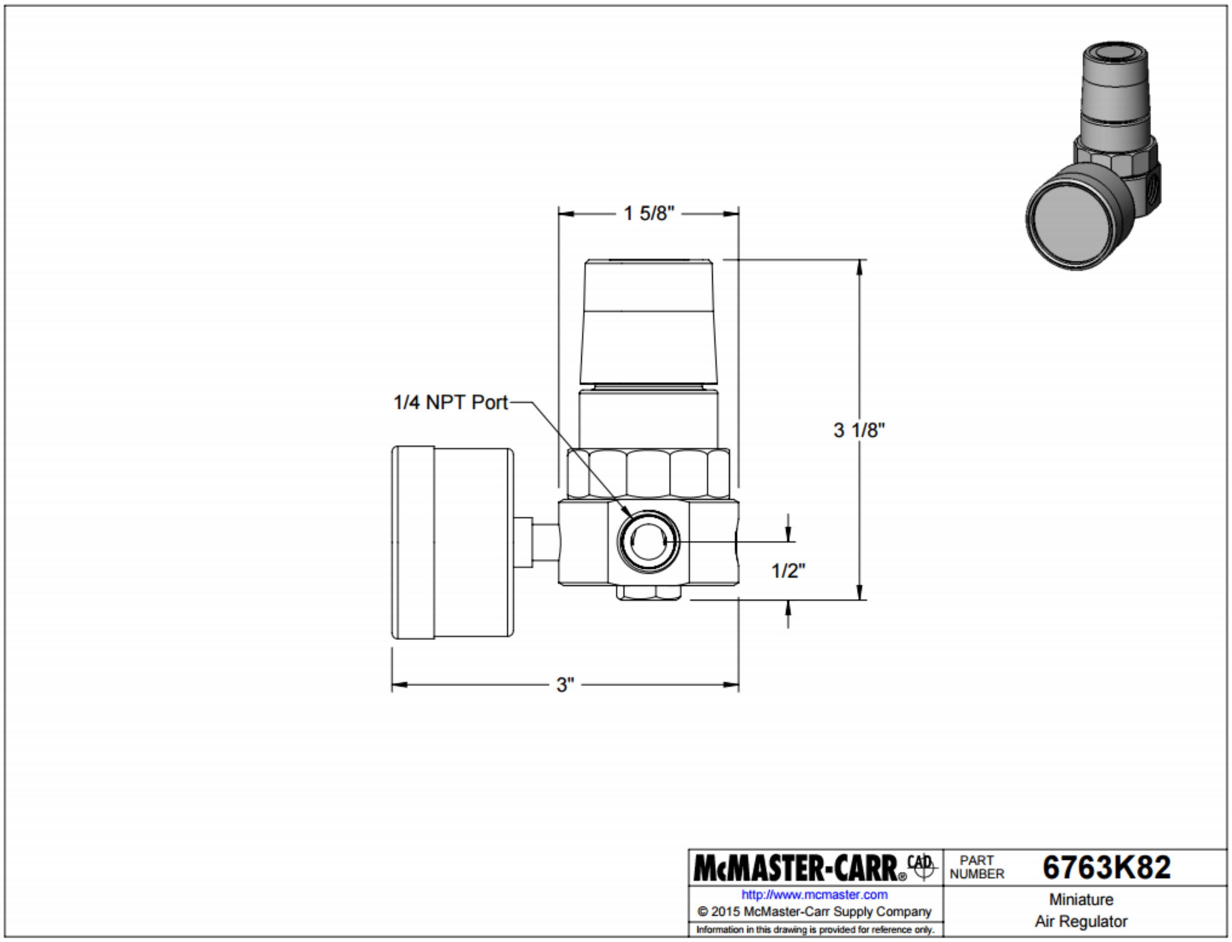
Pressure Regulating Range, psi
- 0-25
- 0-60
- 0-125

Each

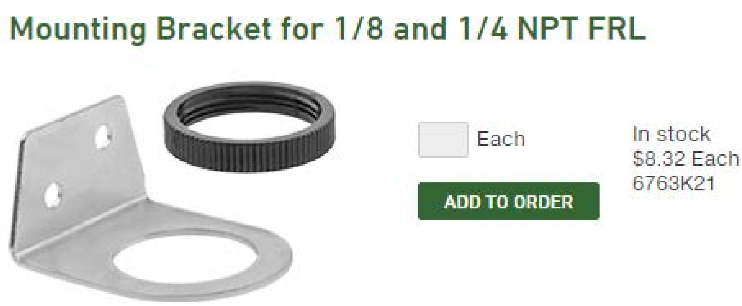
ADD TO ORDER

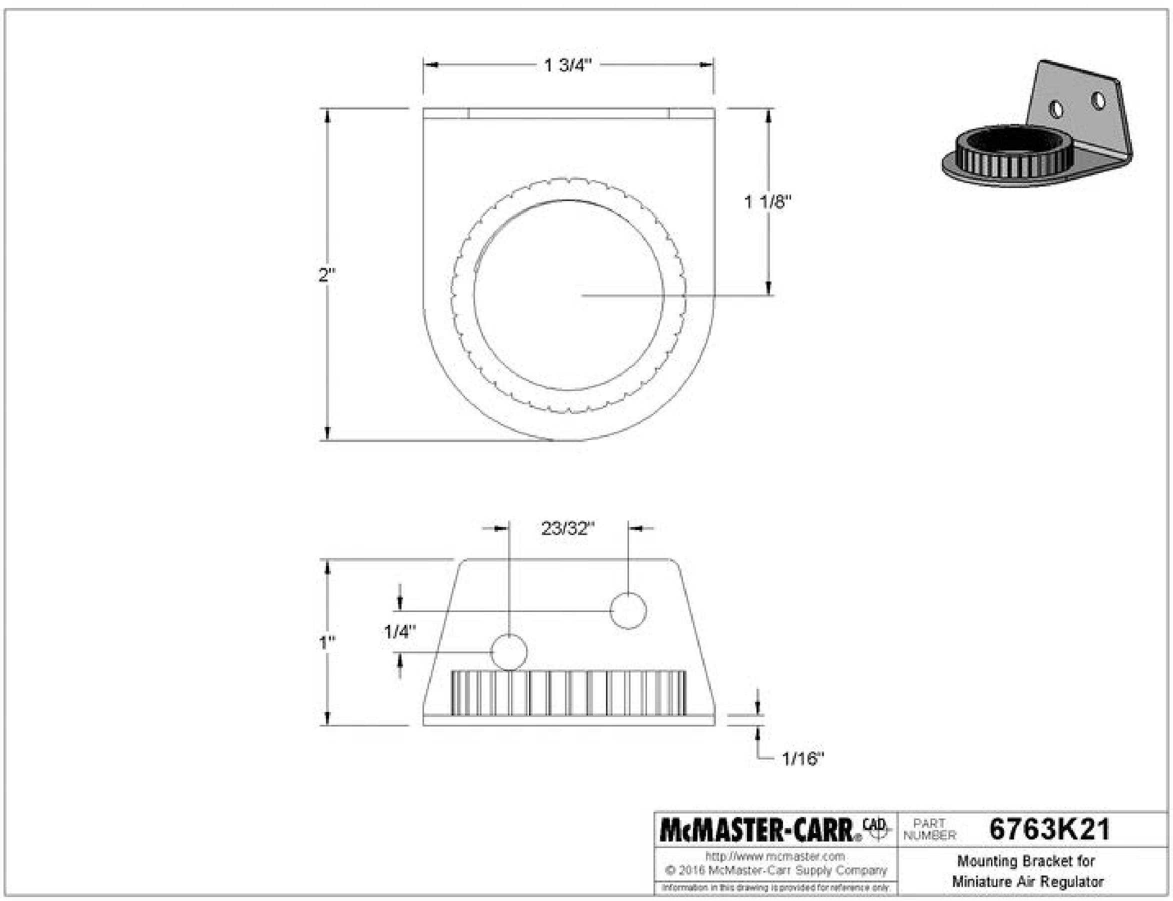
$42.57 Each
6763K82

Component	Regulator
Regulator Type	Relieving
Pipe Size	
Inlet	1/4
Outlet	1/4
Connection Type	Pipe
Pipe Connection Type	Threaded
Connections	NPT Female
Airflow	10 scfm @ 100 psi
Maximum Pressure	300 psi
Maximum Temperature	125° F
Accuracy	±1 psi
Housing Material	Aluminum
Overall	
Height	3 1/8"
Width	1 5/8"
Gauge Included	Yes
Gauge Style	Built Into Body
Adjustability	Adjustable
Adjustment Type	Knob
Related Product	Mounting Brackets


1 5/8"
1/4 NPT Port
3 1/8"
1/2"
3"
McMASTER-CARR
CAD
http://www.mcmaster.com
© 2015 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
PART NUMBER
6763K82
Miniature
Air Regulator

11.4.5 Appendix D.5: Mounting Bracket for Compressed Air Regulator





11.4.6 Appendix D.6: Brass On/Off Valve with Spring Close Lever Handle

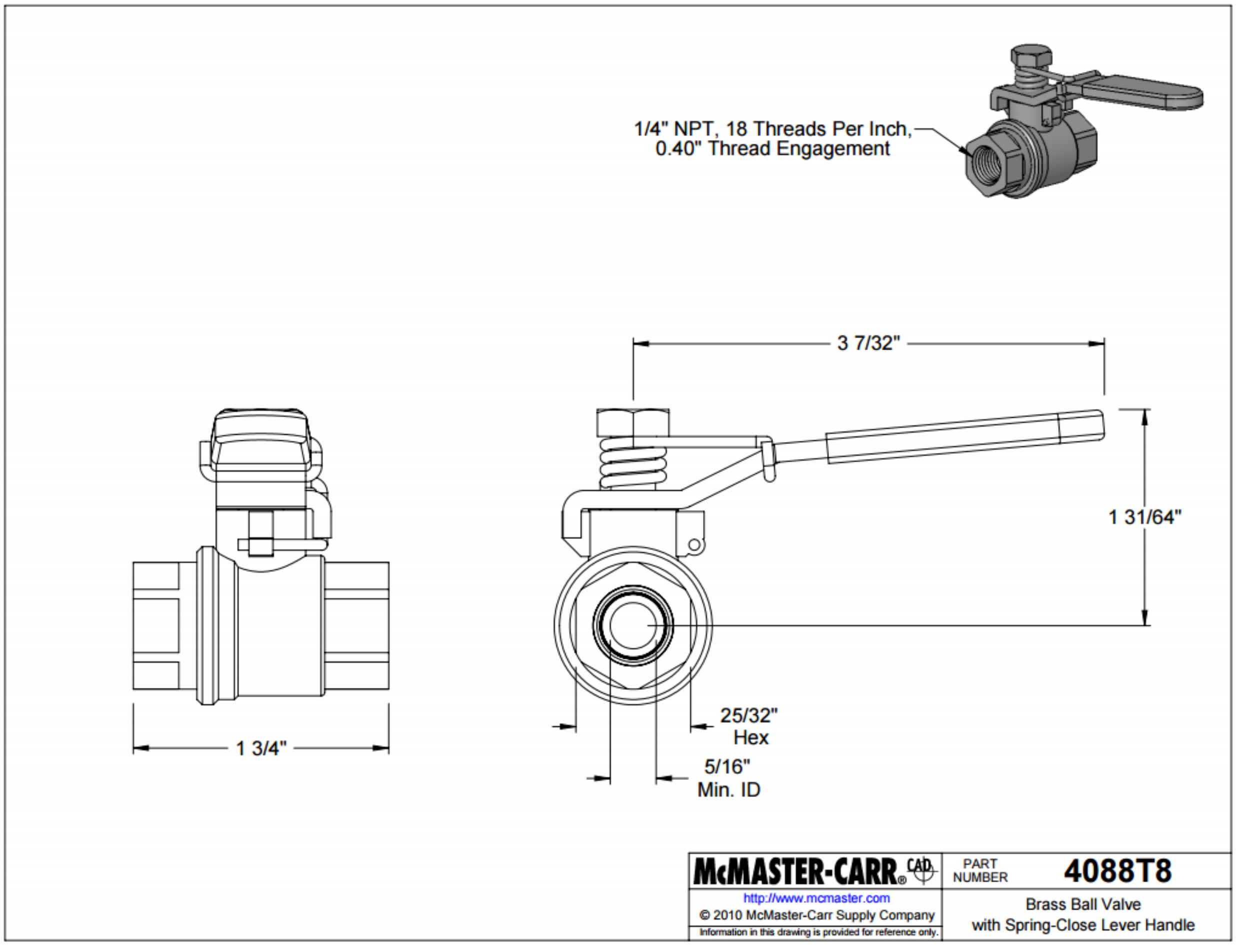
Brass On/Off Valve with Spring Close Lever Handle
1/4 NPT Female

Each

ADD TO ORDER

In stock
$31.26 Each
4088T8

Valve Function	On/Off
For Use With	Air, Inert Gas, Oil, Steam, Water
Activation	Manual
Connection Type	Pipe × Pipe
Connection	Threaded NPT Female × Threaded NPT Female
Pipe Size	1/4 × 1/4
Shape	Straight
End-to-End Length	1 3/4"
Maximum Pressure @ Temperature	600 psi @ 100° F
Maximum Steam Pressure @ Temperature	150 psi @ 366° F
Temperature Range	-50° to 400° F
Vacuum Rating	29.9 in. of Hg
Port Type	Full
Material	
Seal	Fluoroelastomer
Seat	PTFE
Ball	Brass
Body	Brass
Valve Operation	Handle
Handle Type	Spring Close Lever
Valve Type	Ball
Flow Coefficient (Cv)	6.5
Specifications Met	C-UL Listed, CSA Certified, CSA-US Certified, FM Approved, UL Listed


1/4" NPT, 18 Threads Per Inch,
0.40" Thread Engagement
3 7/32"
1 31/64"
1 3/4"
25/32"
Hex
5/16"
Min. ID
McMASTER-CARR
CAD
PART NUMBER
4088T8
http://www.mcmaster.com
© 2010 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Brass Ball Valve
with Spring-Close Lever Handle

11.4.7 Appendix D.7: Medium Hardness Maple Butcher Block

Butcher Block USA

Our Products >> Maple Block Top 30 x 60 x 2-1/4

Heavy-duty 2-1/4" thick Hard Maple workbench top resists warping and stands up to years of industrial workbench use.

- made from edge-grain hard rock Northern maple
- 7/8" wide wood strips
- solid wood strips - no finger joints
- choice of food-safe mineral oil or unfinished
- heavy duty 2-1/4" thickness
- square 1/8" eased edge
- simply the best workbench top you can buy

Genuine maple product made in the USA.

Certified Forest Stewardship Council (FSC) butcher block tops available to meet your LEED and sustainable forestry requirements.

Item #: HD3060

Maple Block Top 30 x 60 x 2-1/4

Price:
~~$308.00~~

Adjusted Price:
$320.00

This item ships freight.

* Marked fields are required.

Finish: *
Mineral Oil (+$12.00)

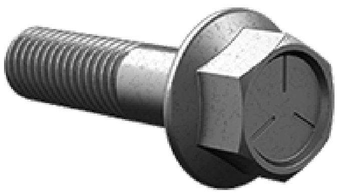
Qty: *
1

ADD TO CART

11.4.8 *Appendix D.8: Grade 5 ½”-13 x 2.00” Hex Head Flange Bolt*

Grade 5 Steel Flanged Hex Head Screws

Medium-Strength, 1/2"-13 Thread Size, 2" Long

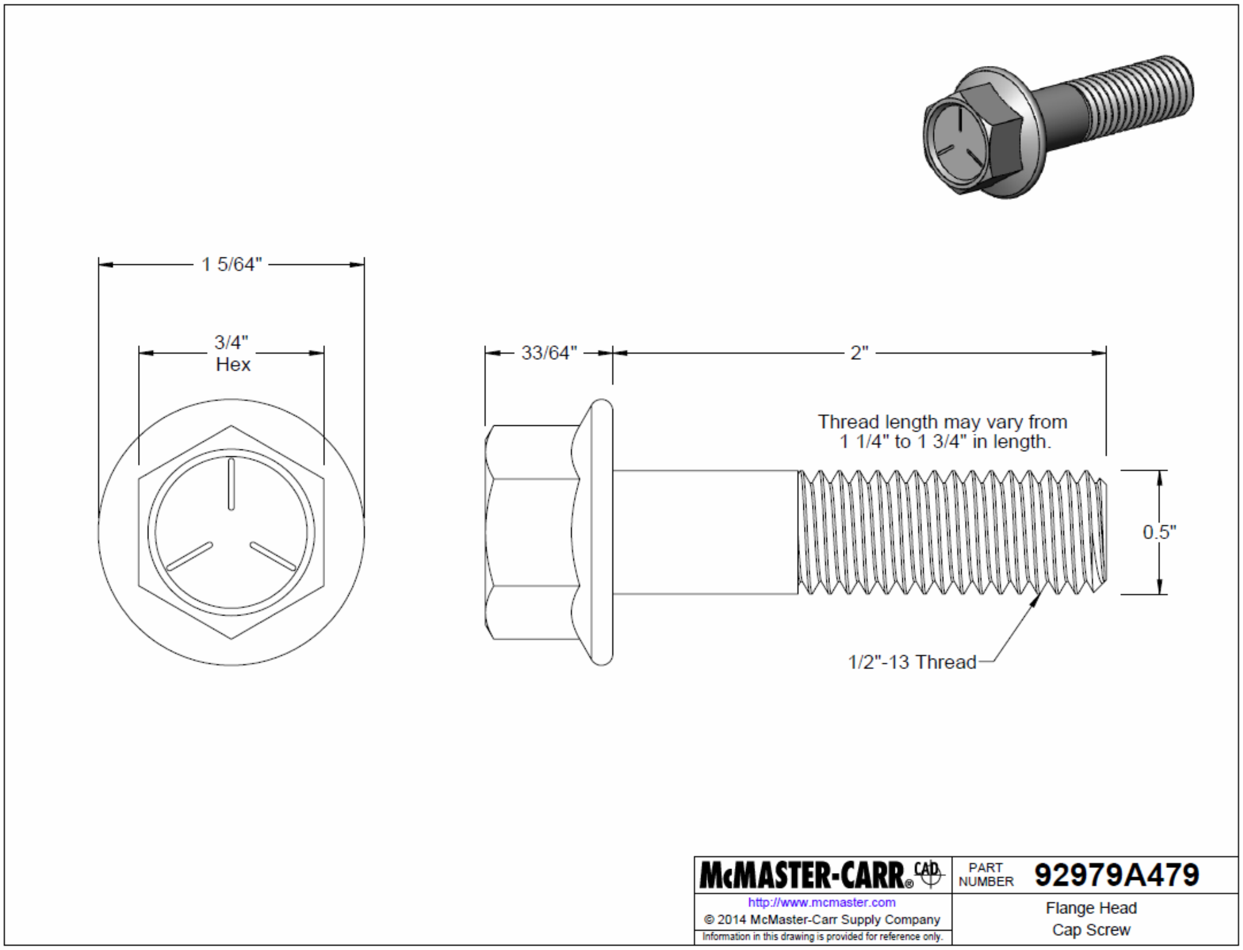


Packs of 10

ADD TO ORDER

In stock
$10.05 per pack of 10
92979A479

Thread Size	1/2"-13
Length	2"
Threading	Partially Threaded
Minimum Thread Length	1 1/4"
Head Width	3/4"
Head Height	33/64"
Flange Diameter	1 5/64"
Material	Zinc-Plated Steel
Hardness	Rockwell C22
Tensile Strength	110,000 psi
Fastener Strength Grade/Class	Grade 5
Screw Size Decimal Equivalent	0.500"
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 2A
Thread Direction	Right Hand
Head Type	Hex
Hex Head Profile	Standard
Drive Style	External Hex
Screw Features	Flanged
Specifications Met	IFI 111, SAE J429
System of Measurement	Inch
RoHS	Compliant


1 5/64"
3/4"
Hex
33/64"
2"
Thread length may vary from
1 1/4" to 1 3/4" in length.
0.5"
1/2"-13 Thread
McMASTER-CARR
CAD
PART NUMBER
92979A479
http://www.mcmaster.com
© 2014 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Flange Head
Cap Screw

11.4.9 Appendix D.9: ½”-13 High-Strength Steel Nylock Nut

High-Strength Steel Nylon-Insert Flange Locknut

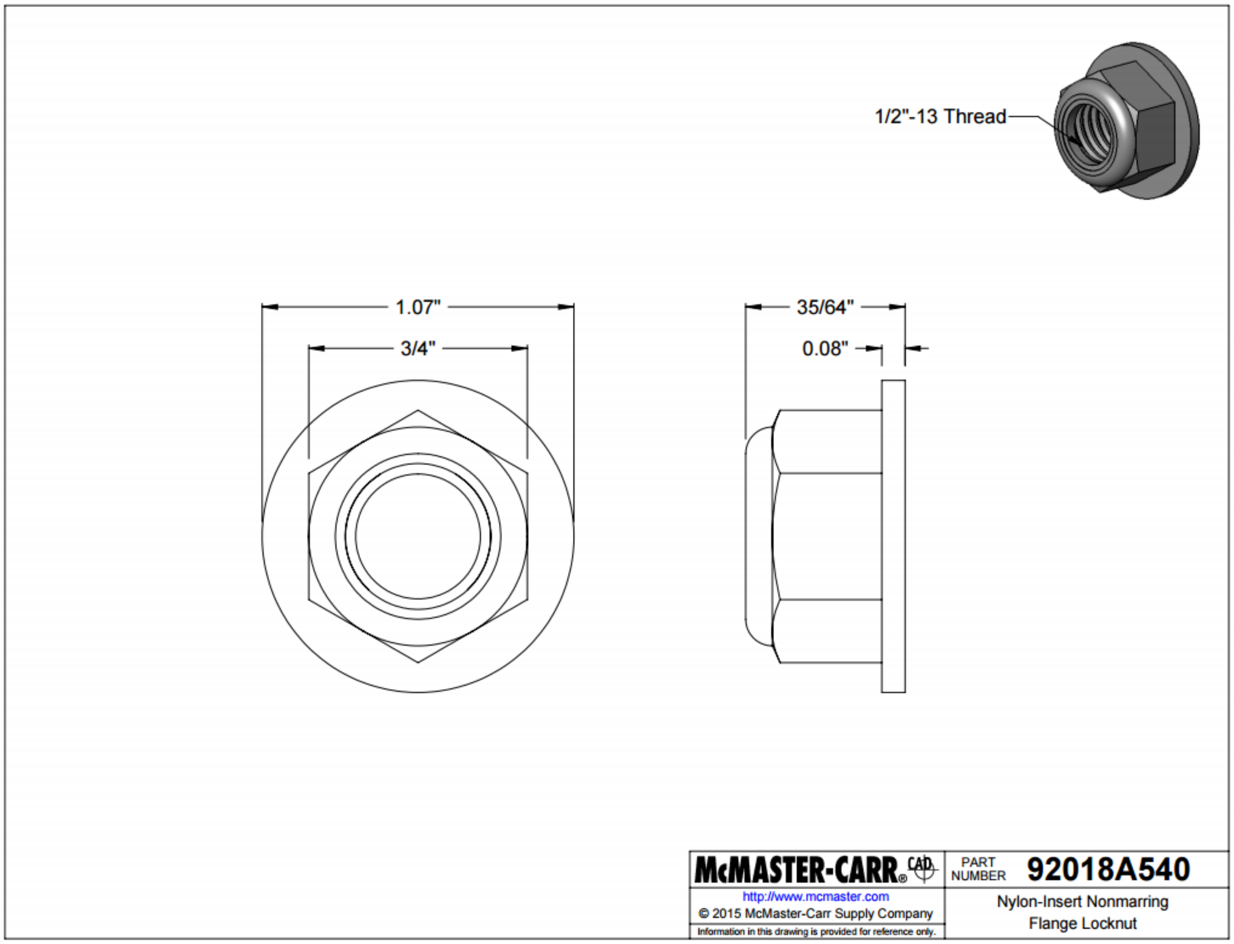
Grade G, Zinc Yellow-Chromate Plated, 1/2"-13 Thread Size

Packs of 25

ADD TO ORDER

In stock
$5.93 per pack of 25
92018A540

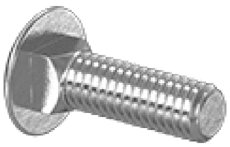
Material	Zinc Yellow-Chromate Plated Steel
Fastener Strength Grade/Class	Grade G
Thread Size	1/2"-13
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 2B
Thread Direction	Right Hand
Width	3/4"
Height	35/64"
Flange	
Diameter	1.07"
Thickness	0.08"
Insert Maximum Temperature	250° F
Drive Style	External Hex
Nut Type	Flange, Locknut
Hex Nut Profile	Standard
Locking Type	Nylon Insert
System of Measurement	Inch
RoHS	Not Compliant


1/2"-13 Thread
1.07"
3/4"
35/64"
0.08"
McMASTER-CARR
CAD
PART NUMBER
92018A540
http://www.mcmaster.com
© 2015 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Nylon-Insert Nonmarring
Flange Locknut

11.4.10 Appendix D.10: ⅜”-16 x 1.25” Grade 5 Carriage Bolt

Grade 5 Steel Square-Neck Carriage Bolt

Medium-Strength, 3/8"-16 Thread Size, 1-1/4" Long

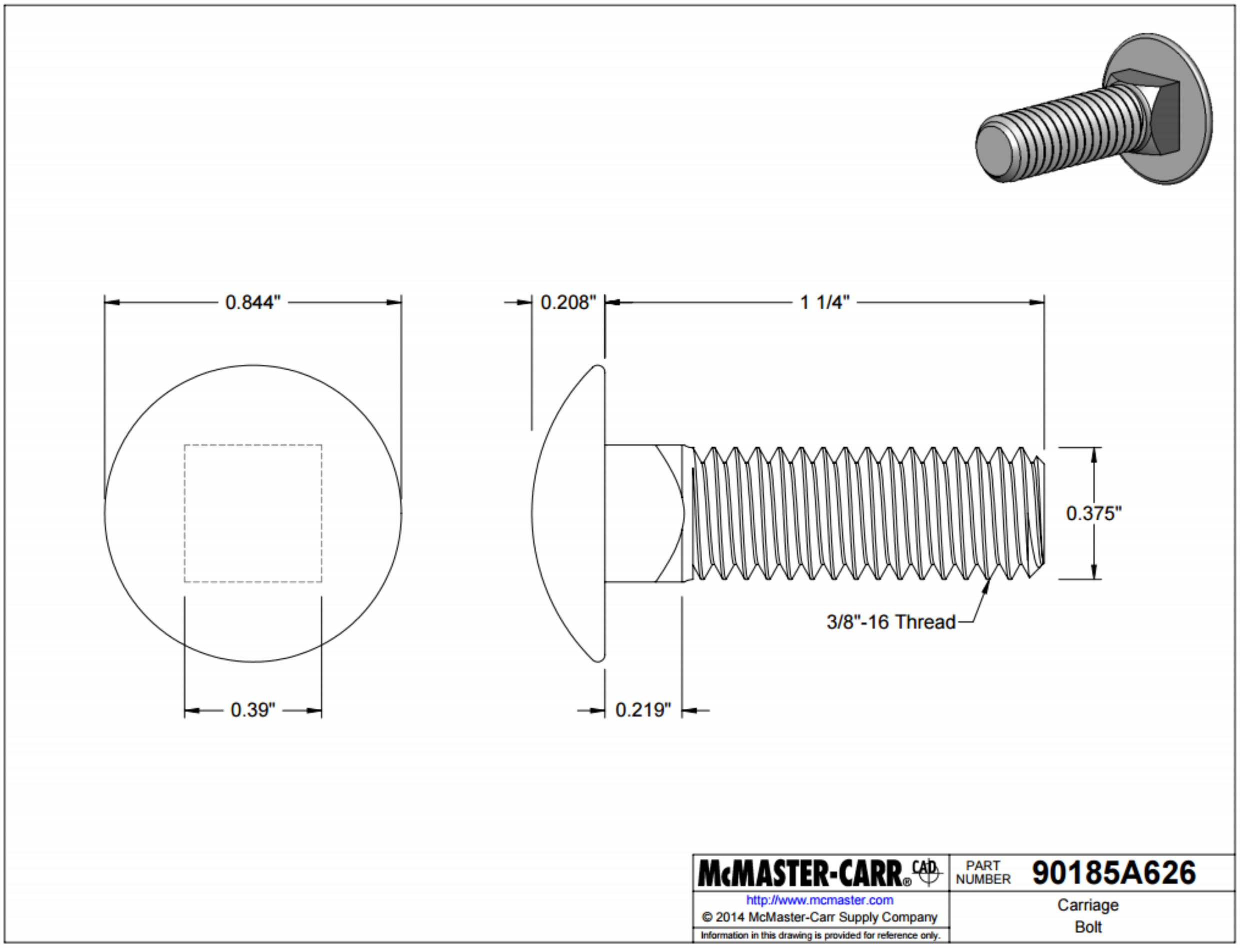


Packs of 25

ADD TO ORDER

In stock
$7.07 per pack of 25
90185A626

Thread Size	3/8"-16
Length	1 1/4"
Threading	Fully Threaded
Head Diameter	0.844"
Head Height	0.208"
Neck Width	0.39"
Neck Length	0.22"
Material	Zinc-Plated Steel
Fastener Strength Grade/Class	Grade 5
Hardness	Rockwell C25
Tensile Strength	120,000 psi
Screw Size Decimal Equivalent	0.375"
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 2A
Thread Direction	Right Hand
Head Type	Rounded
Rounded Head Profile	Wide
Rounded Head Style	Carriage
Neck Type	Square
Specifications Met	ANSI B18.5, SAE J429
System of Measurement	Inch
RoHS	Compliant

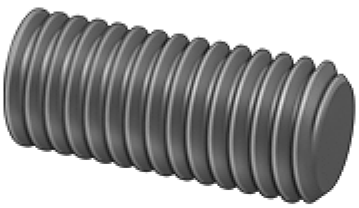


0.844"
0.208"
1 1/4"
0.375"
3/8"-16 Thread
0.39"
0.219"
McMASTER-CARR
PART NUMBER
90185A626
http://www.mcmaster.com
© 2014 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Carriage Bolt

11.4.11 Appendix D.11: ½"-13 x 5.00" Nylon Threaded Rod

Nylon Threaded Rod

1/2"-13 Thread Size, 5" Long



Color
- ● Black
- ○ White

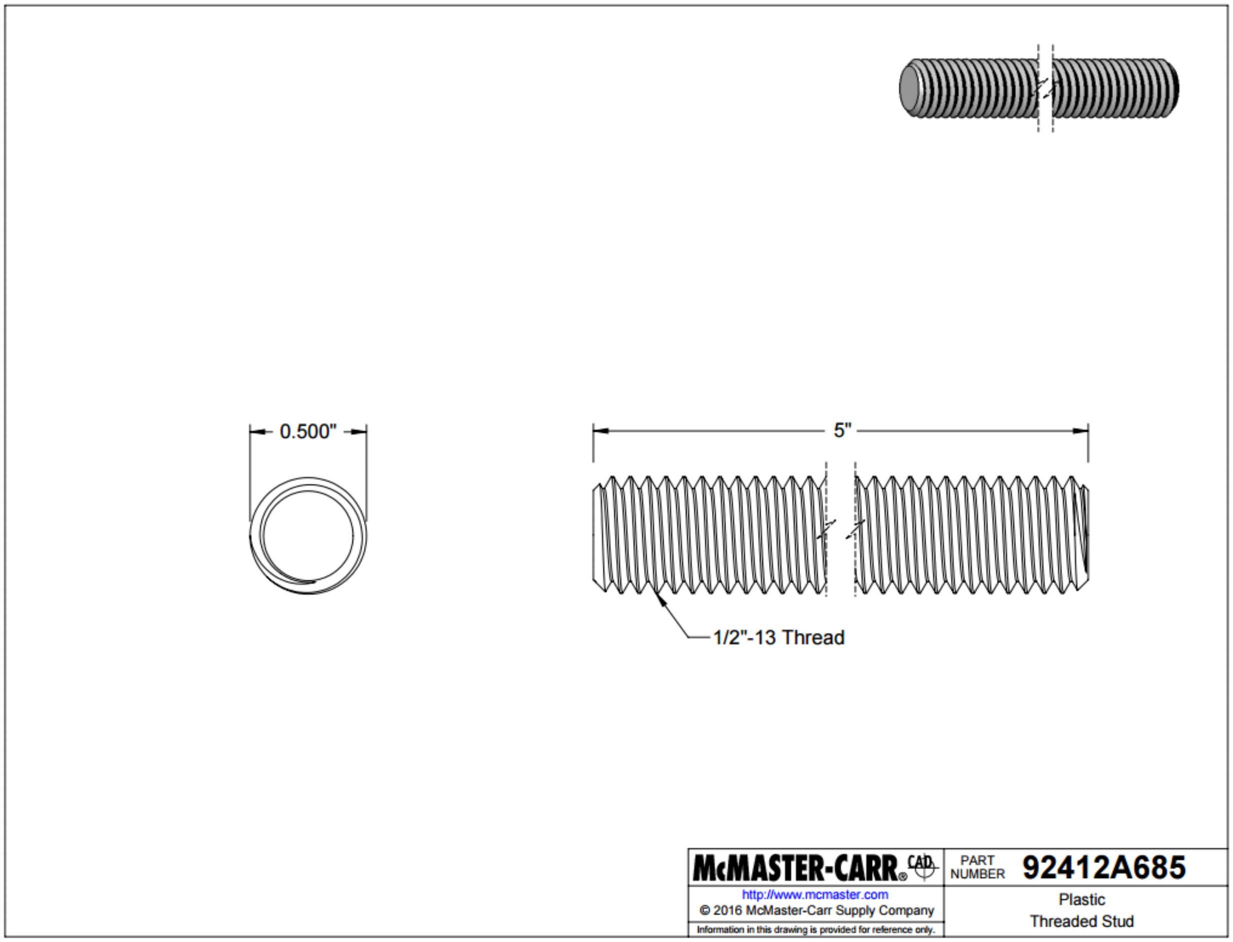
Packs of 5

ADD TO ORDER

$7.32 per pack of 5
93665A684

Material	Nylon 6/6 Plastic
Thread Size	1/2"-13
Length	5"
Tensile Strength	Not Rated
Hardness	Not Rated
Minimum Temperature	Not Rated
Maximum Temperature	185° F
Thread	
Direction	Right Hand
Type	UNC
Spacing	Coarse
Fit (External)	Class 2A
Threading	Fully Threaded
System of Measurement	Inch
RoHS	Compliant

Made from nylon 6/6, these threaded rods resist oil, grease, and solvents (except mineral acids). They're nonconductive, making them good for use around sensitive electrical components.



0.500"
5"
1/2"-13 Thread
McMASTER-CARR
CAD
http://www.mcmaster.com
© 2016 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
PART NUMBER
92412A685
Plastic
Threaded Stud

11.4.12 Appendix D.12: Comfort-Grip ½”-13 Knurled Knob

Comfort-Grip Fluted-Rim Knob

1/2"-13 Threaded Through Insert, 2-1/2" Diameter

Each

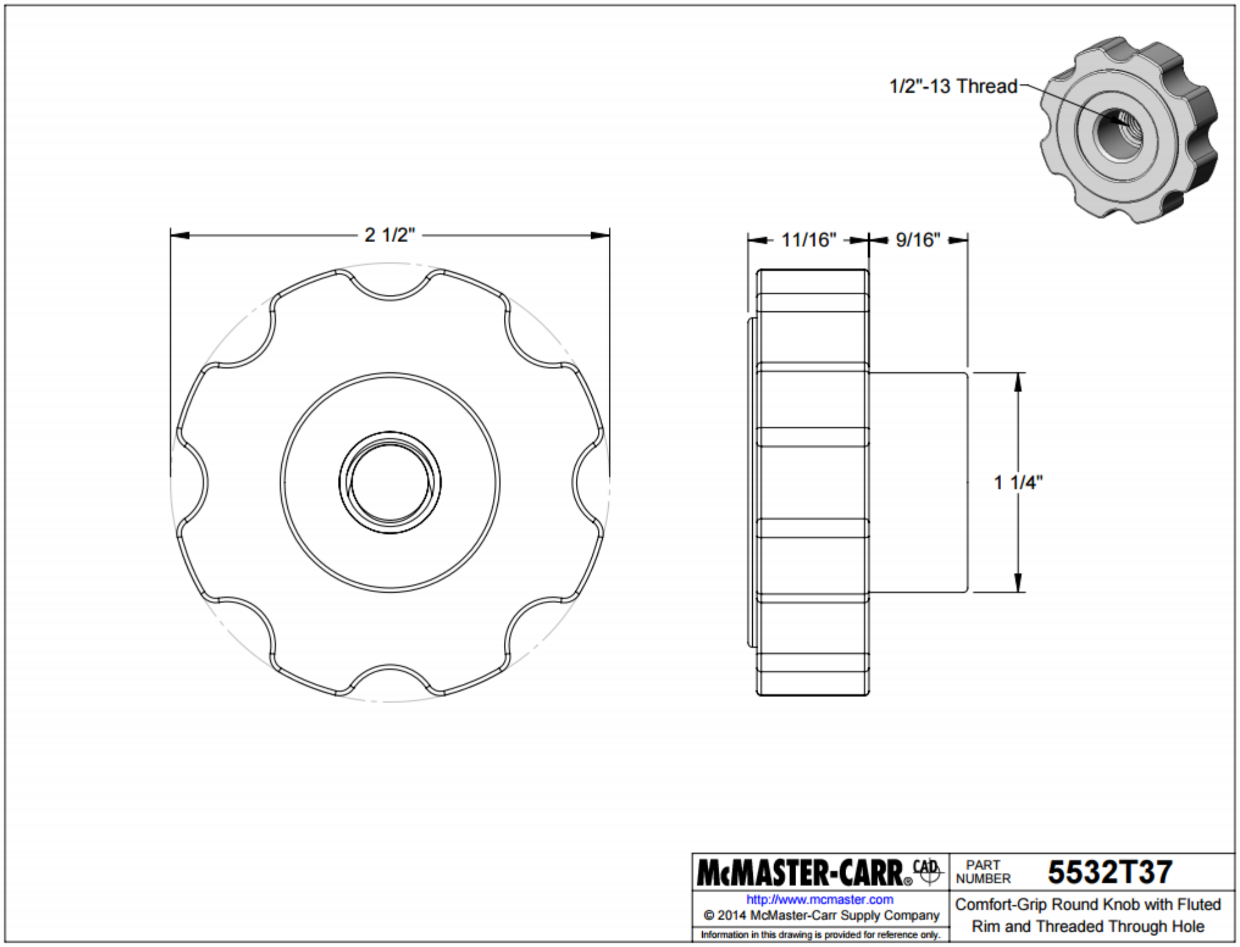
ADD TO ORDER

In stock
1-9 Each $3.28
10 or more $2.92
5532T37

Thread	
Size	1/2"-13
Type	UNC
Depth	7/16"
Direction	Right Hand
Head	
Diameter	2 1/2"
Height	11/16"
Shape	Round
Texture	Ribbed
Hub	
Diameter	1 1/4"
Height	9/16"
Material	Rubber-Coated Polypropylene Plastic
Color	Black
Insert Material	Steel
Temperature Range	-15° to 200° F
Hole Shape	Round
Mounting Style	Threaded Through Hole
Knob Type	Plain
RoHS	Compliant

An economical alternative to metal, plastic resists oil, grease, and solvents. Knobs have a rubber coating that provides a comfortable gripping surface.

PROJECT:	Medium Duty Gas-Turbine Engine Blade Removal Tool			**PART:**	Removal Tool
TITLE:	Final Design Report			**SECTION:**	ME 430
ORIGINATOR:	J. Syage, E. Greb, K. Field	**DATE:**	11/30/2017	**REVISION:**	A
Reviewer:	J. Hernandez			**PAGE: 107 of**	178



11.4.13 *Appendix D.13: ⅜"-16 Stainless Steel Knurled Thumb Nut*

18-8 Stainless Steel Knurled-Head Thumb Nut

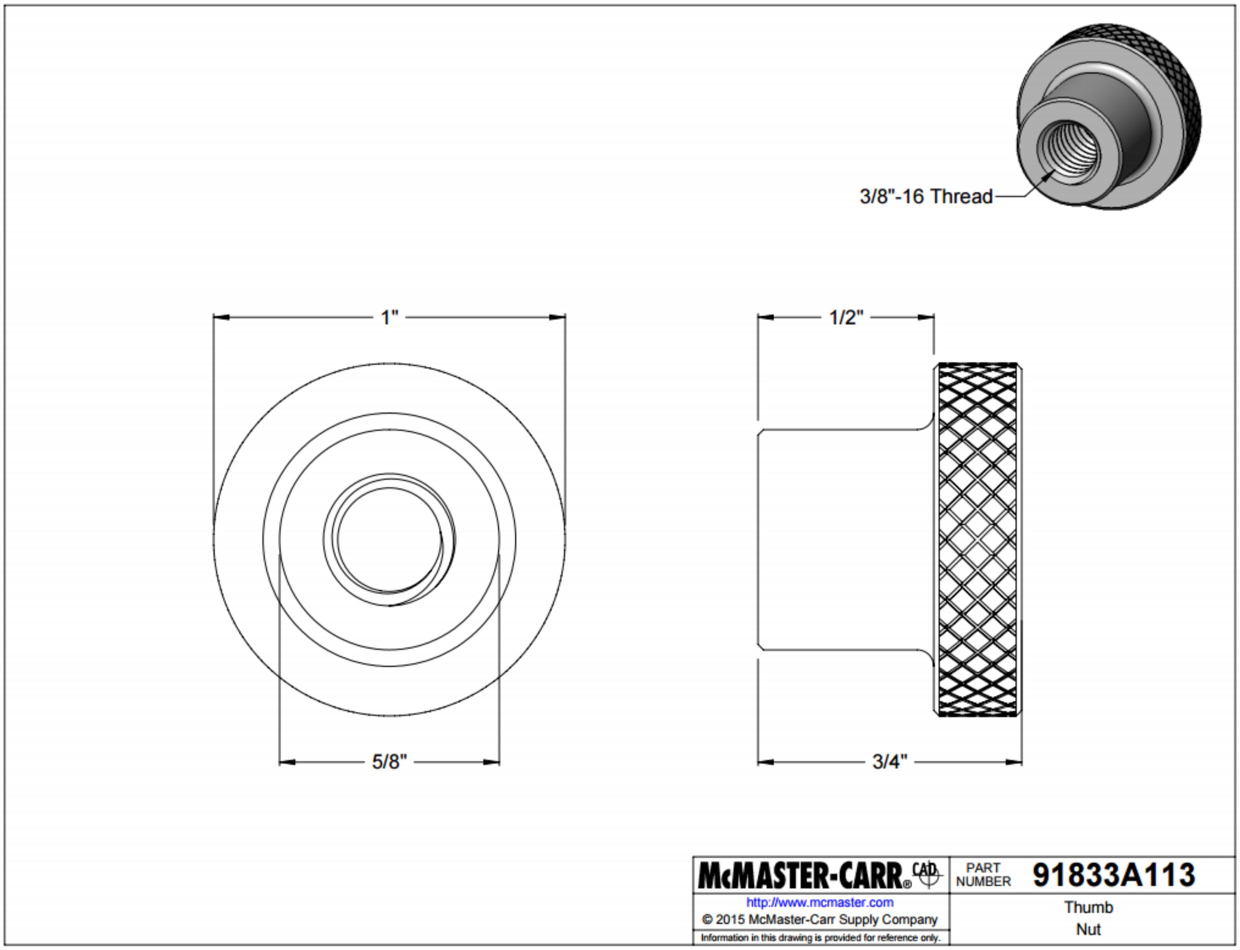
3/8"-16 Thread Size

Packs of 1

ADD TO ORDER

In stock
$9.21 per pack of 1
91833A113

Material	18-8 Stainless Steel
Thread Size	3/8"-16
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 2B
Thread Direction	Right Hand
Collar	
Diameter	5/8"
Height	1/2"
Head Diameter	1"
Head Texture	Knurled
Height	3/4"
Threading	Fully Threaded
Nut Type	Thumb
Thumb Nut Head Shape	Round
Thumb Nut Profile	Standard
System of Measurement	Inch
RoHS	Compliant



3/8"-16 Thread
1"
5/8"
1/2"
3/4"
McMASTER-CARR
PART NUMBER
91833A113
http://www.mcmaster.com
© 2015 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Thumb Nut

11.4.14 Appendix D.14: 5/16”-18 Plastic Thumb Nuts

McMASTER-CARR

thumb nut

Reinforced Plastic Knurled-Head Thumb Nut

5/16"-18 Thread Size

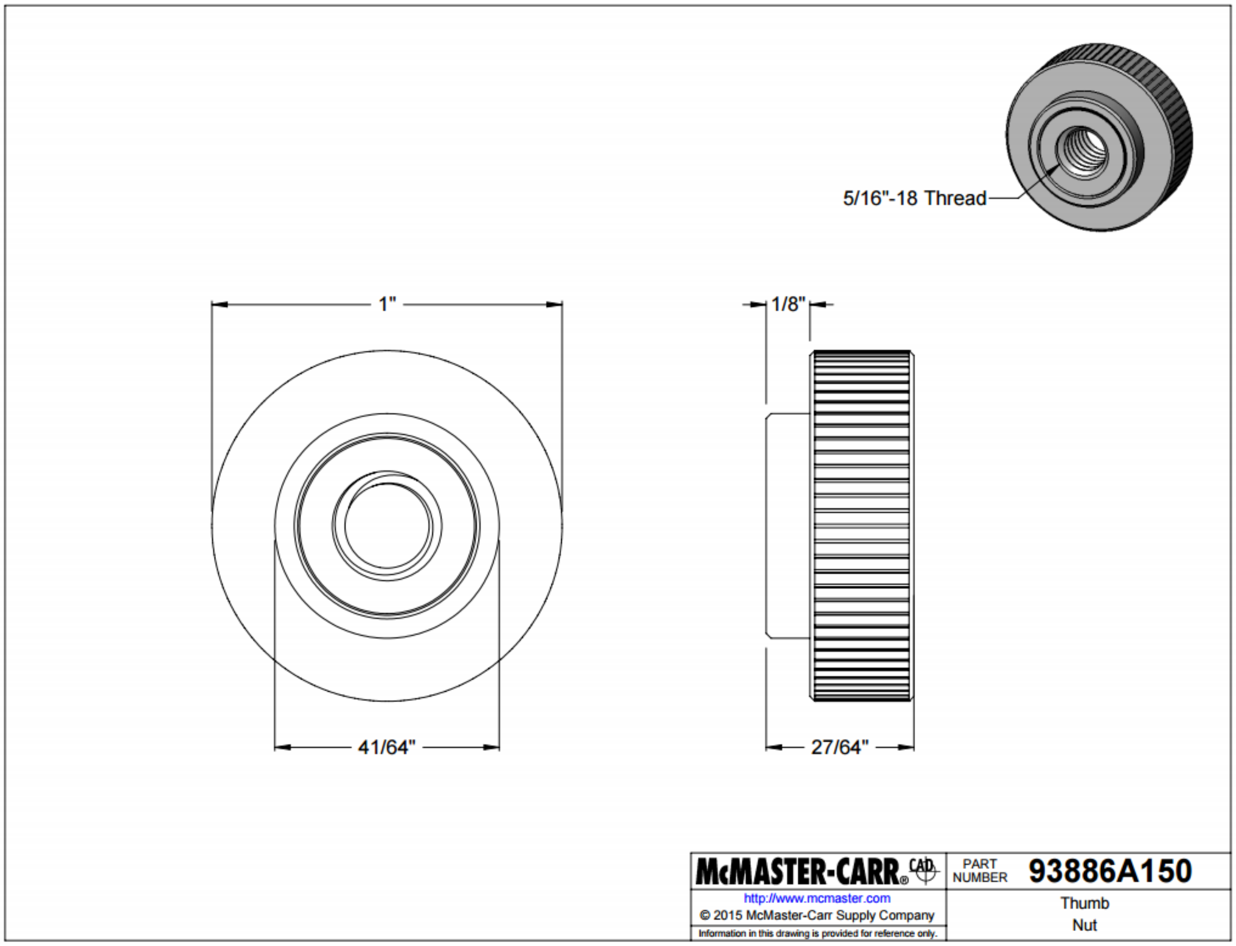
Packs of 10

ADD TO ORDER

In stock
$10.76 per pack of 10
93886A150

Material	Acetal Plastic
Thread Size	5/16"-18
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 2B
Thread Direction	Right Hand
Threading	Fully Threaded
Collar	
Diameter	41/64"
Height	1/8"
Head Diameter	1"
Head Texture	Knurled
Height	27/64"
Temperature Range	32° to 180° F
Color	Black
Nut Type	Thumb
Thumb Nut Head Shape	Round
Thumb Nut Profile	Standard
System of Measurement	Inch
RoHS	Compliant

An aluminum insert with strong threads withstands more torque during installation. They're chemical resistant, nonconductive, and lightweight. The head is knurled for slip-resistant controlled adjustment and the collar raises the head to make it easy to grip.


5/16"-18 Thread
1"
41/64"
1/8"
27/64"
McMASTER-CARR
http://www.mcmaster.com
© 2015 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
PART NUMBER
93886A150
Thumb
Nut

11.4.15 Appendix D.15: ½"-20 x 3.00" Full Thread Moser Engineering Wheel Studs

Moser Engineering Wheel Studs 8020

Click to Enlarge Image

$22.50

1 **Add To Cart** + Wish List + Compare

- Fast Shipping
- Tech Advice
- Low Price Guarantee
- Easy Returns

★★★★★ (6)

Review This Product

Wheel Studs, Press-In, 1/2-20 in. x 3.0 in., Right Hand Thread, .685 in. Knurl Set of 10

Estimated Ship Date: Today

Would you rather pick it up? Select Location

Check Application

Overview | Applications | Suggested Parts | Reviews | Show All

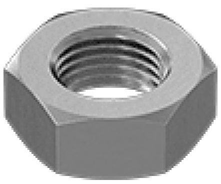
Brand:	Moser Engineering
Manufacturer's Part Number:	8020
Part Type:	Wheel Studs
Product Line:	Moser Engineering Wheel Studs
Summit Racing Part Number:	**MSR-8020**
Wheel Stud Style:	Press-in
Wheel Stud Thread Size:	1/2-20 in. RH
Underhead Length (in):	3.000 in.
Knurl Diameter (in.):	0.685 in.
Quick-Start Nose:	No
Quantity:	Sold as a set of 10.
In-Store Pickup:	Choose In-store pick-up (OH, GA, NV) on our web site.

Moser Engineering offers wheel studs in both press-in and screw-in styles to fit most popular applications.

11.4.16 Appendix D.16: ½-20 Thin Steel Hex Nut

Zinc Yellow-Chromate Plated Steel Thin Hex Nut

Grade 8, High-Strength, 1/2"-20 Thread Size



Packs of 25

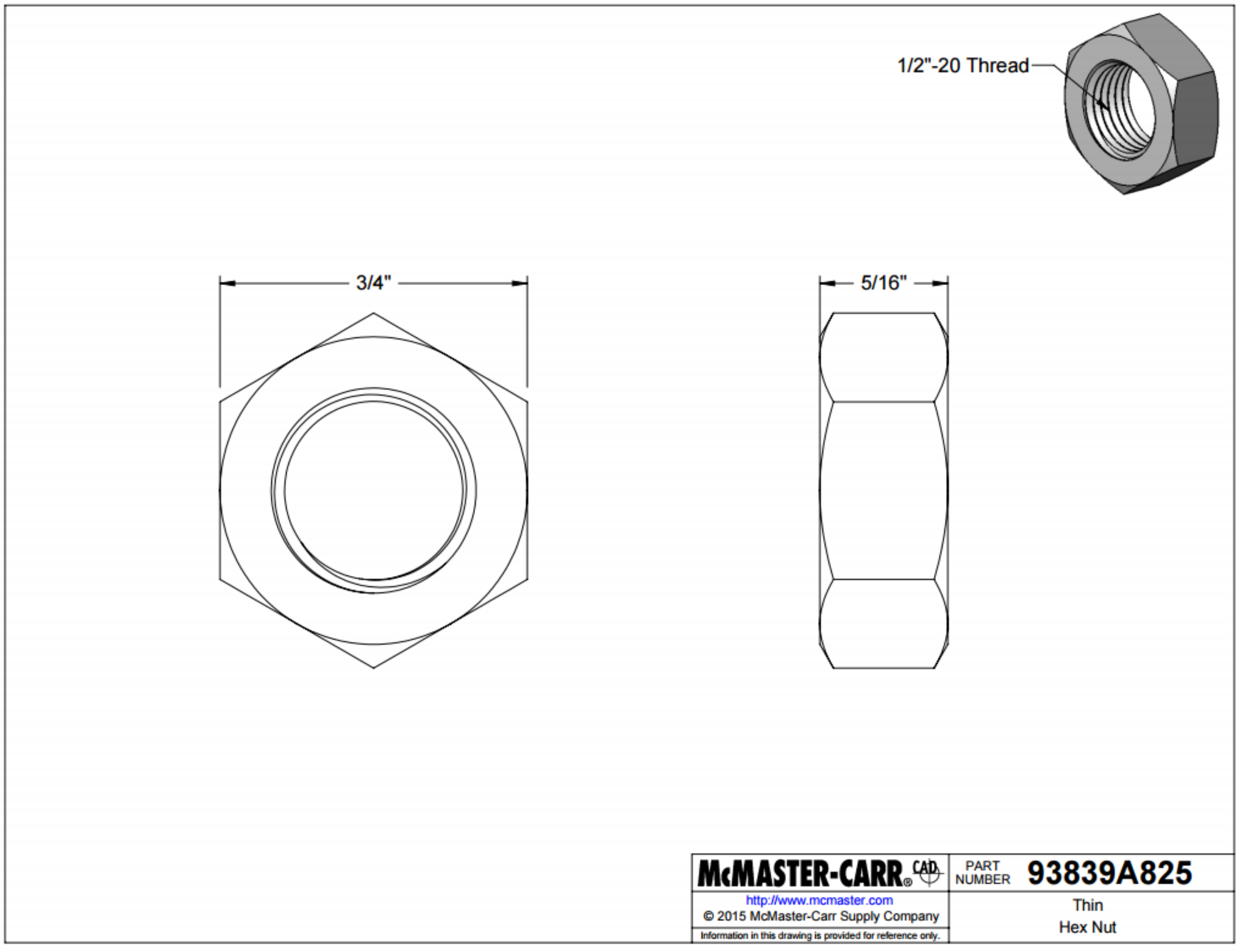
ADD TO ORDER

In stock
$11.53 per pack of 25
93839A825

Material	Zinc Yellow-Chromate Plated Steel
Fastener Strength Grade/Class	Grade 8
Thread Size	1/2"-20
Thread Type	UNF
Thread Spacing	Fine
Thread Fit	Class 2B
Thread Direction	Right Hand
Width	3/4"
Height	5/16"
Drive Style	External Hex
Nut Type	Hex
Hex Nut Profile	Thin
System of Measurement	Inch
RoHS	Not Compliant

Also known as jam nuts, these are about half the height of standard hex nuts. Use in low-clearance applications or jam one against a standard hex nut to hold it in place. They're about 25% stronger than medium-strength steel thin hex nuts.

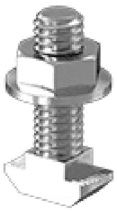
Zinc yellow-chromate plated steel nuts are corrosion resistant in wet environments.


1/2"-20 Thread
3/4"
5/16"
McMASTER-CARR CAD
PART NUMBER 93839A825
http://www.mcmaster.com
© 2015 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Thin
Hex Nut

11.4.17 *Appendix D.17: 1.50" T-Slotted Framing Stud*

T-Slotted Framing

Drop-in Fastener with Stud, for 1-1/2" High Single Rail



Each

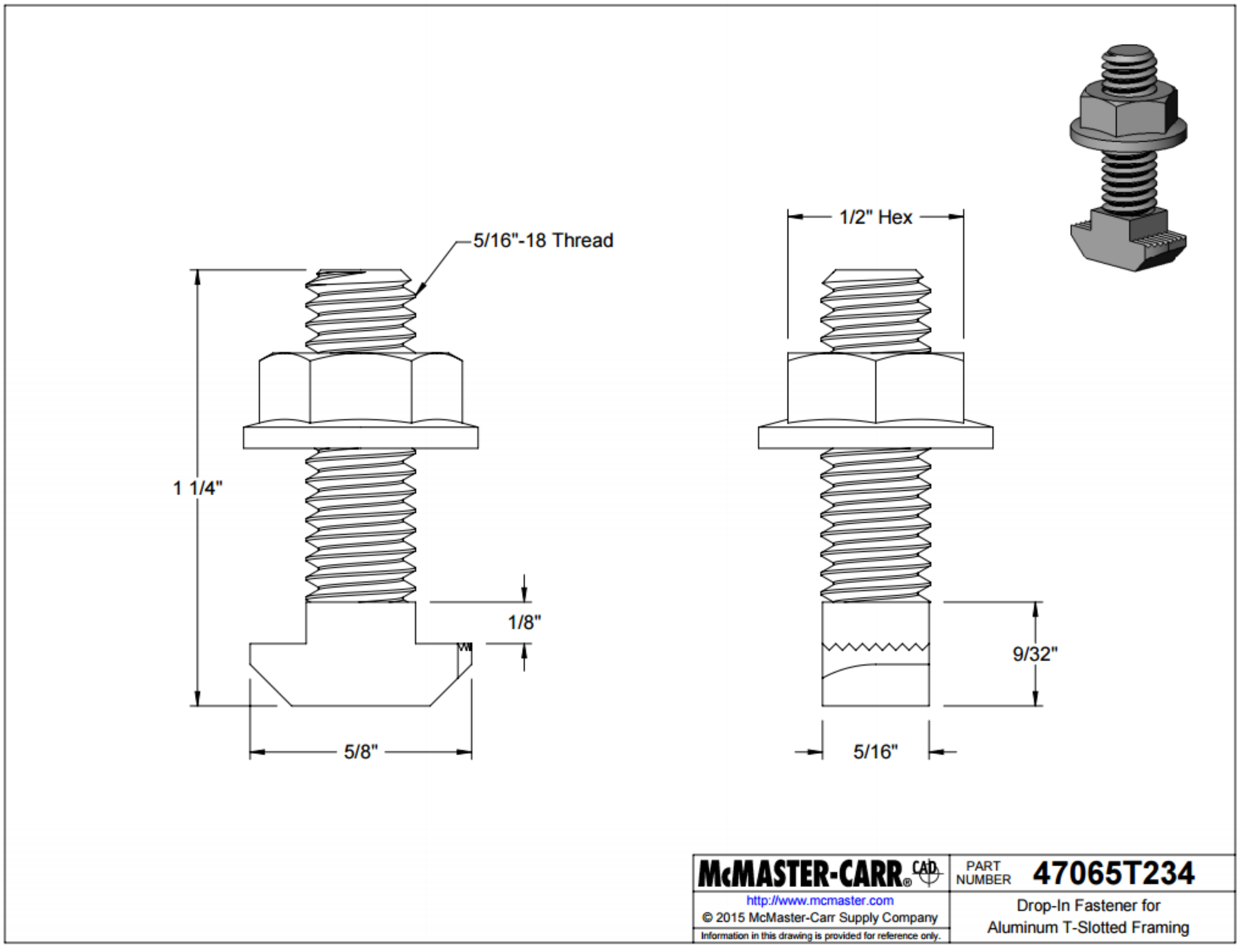
ADD TO ORDER

In stock
$1.58 Each
47065T234

For Rail Height	
Single	1 1/2"
Double/Quad	3"
Material	Zinc-Plated Steel
Installation Type	Drop In
Fastener Drive Style	External Hex
Thread Size	5/16"-18
Fastener Type	With Stud
Framing Type	T Slot
T-Slot Framing Component	Fastener
RoHS	Compliant

Make machine guards, carts, workstations, and more with our most versatile framing. It has continuous T-slots for attachment points, making structures easy to configure. Use fasteners to connect components.

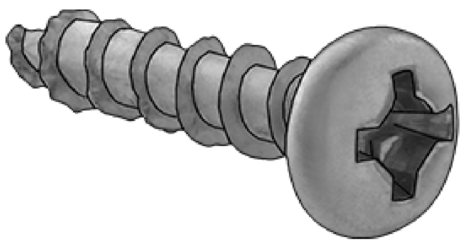
Drop-in fasteners can be placed directly into the T-slot to attach components—no need to disassemble your structure. They consist of a screw and nut. Fasteners with stud twist into place for quick attachment of inside braces and other accessories. They can be used for grounding anodized rails. Tighten with a socket wrench.


5/16"-18 Thread
1/2" Hex
1 1/4"
1/8"
9/32"
5/8"
5/16"
McMASTER-CARR
PART NUMBER
47065T234
http://www.mcmaster.com
© 2015 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Drop-In Fastener for
Aluminum T-Slotted Framing

11.4.18 Appendix D.18: ¾" Rounded Head Screws

Rounded Head Screws for Plywood and Osb

Zinc-Plated Steel, Number 10 Size, 3/4" Long



Packs of 100

ADD TO ORDER

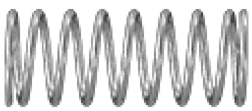
In stock
$9.04 per pack of 100
91070A245

Material	Zinc-Plated Steel
Screw Size	No. 10
Screw Size Decimal Equivalent	0.190"
Length	3/4"
Head	
Diameter	0.37"
Height	0.13"
Drive	
Size	No. 2/No. 2
Style	Square/Phillips
Drill Bit Size	3/32"
Drill Bit Size Decimal Equivalent	0.094"
Approximate Threads per Inch	13
Thread Direction	Right Hand
Threading	Fully Threaded
Tapping Method	Thread Forming
Head Type	Rounded
Rounded Head Profile	Standard
Rounded Head Style	Pan
Tip Type	Pointed
Shank Cross Section	Round
System of Measurement	Inch
For Use In	Plywood, Oriented Strand Board
RoHS	Compliant

11.4.19 *Appendix D.19: Zinc-Plated Closed and Flat Compression Spring*

Compression Spring

Zinc-Plated Wire, Closed and Flat End, 1.75" Long, 0.594" ID



Packs of 6

ADD TO ORDER

In stock
$6.70 per pack of 6
9657K375

Spring Type	Compression
Material	Zinc-Plated Music-Wire Steel
End Type	Closed and Flat
Overall Length	1.75"
OD	0.72"
ID	0.594"
Wire Diameter	0.063"
Wire Shape	Round
Compressed Length	0.54"
Maximum Load	15.50 lbs.
Rate	13.00 lbs./in.
RoHS	Compliant

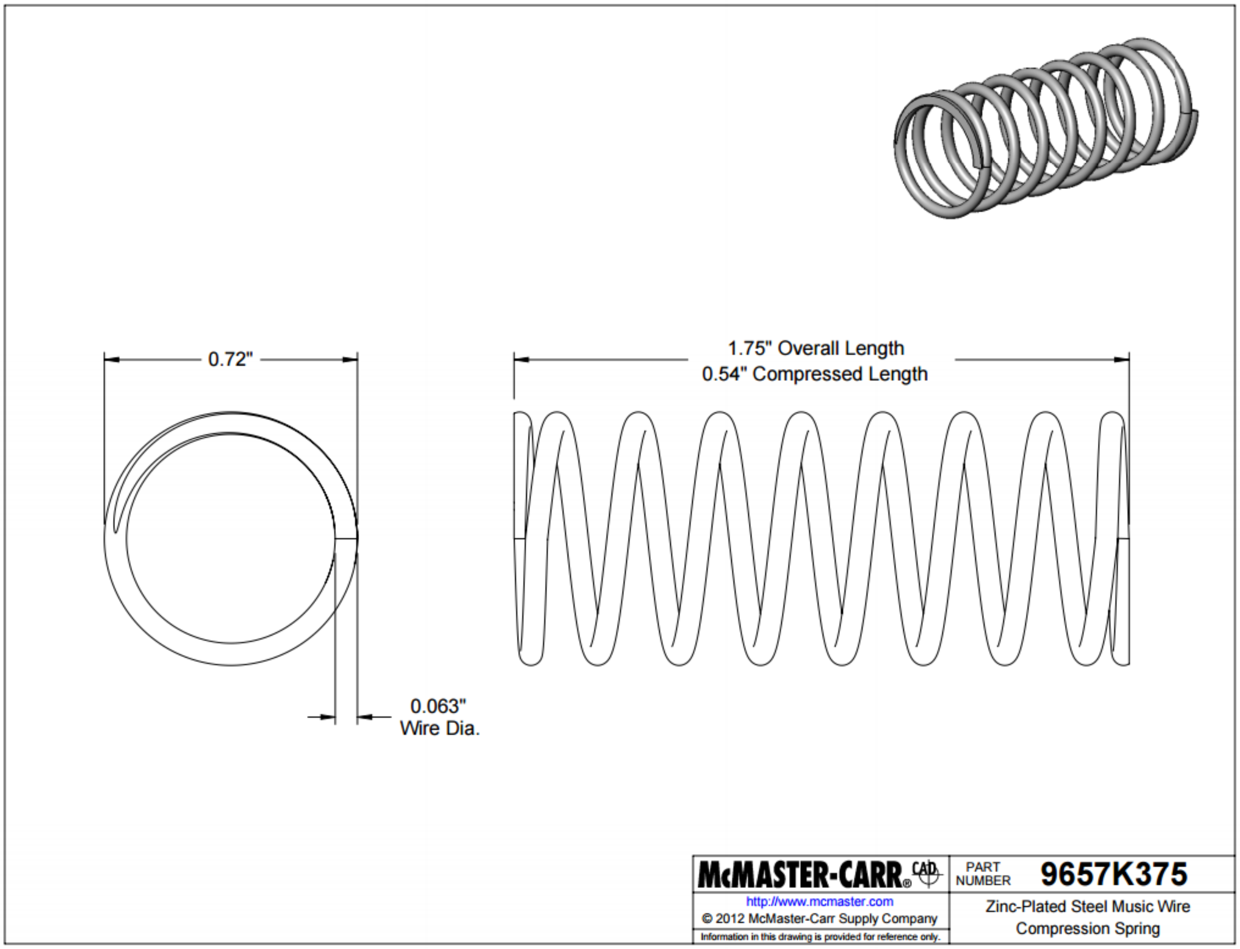
Music-wire steel springs are strong.

Zinc-plated springs provide moderate corrosion resistance.

Rate—As you push a compression spring, it gets harder to push. The higher the rate, the harder it is to compress the spring.

On springs with closed and flat ends, the last coil is ground flush so the springs stand straighter and are easier to stack.

Don't see the size you need? Additional sizes are available.

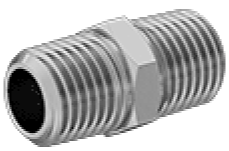


0.72"
1.75" Overall Length
0.54" Compressed Length
0.063"
Wire Dia.
McMASTER-CARR CAD
http://www.mcmaster.com
© 2012 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
PART NUMBER
9657K375
Zinc-Plated Steel Music Wire
Compression Spring

11.4.20 Appendix D.20: ¼" Straight Connector High Pressure Gas Pipe Fitting

High-Pressure Brass Pipe Fitting

Straight Connector with Hex Body, 1/4 NPT Male



Each

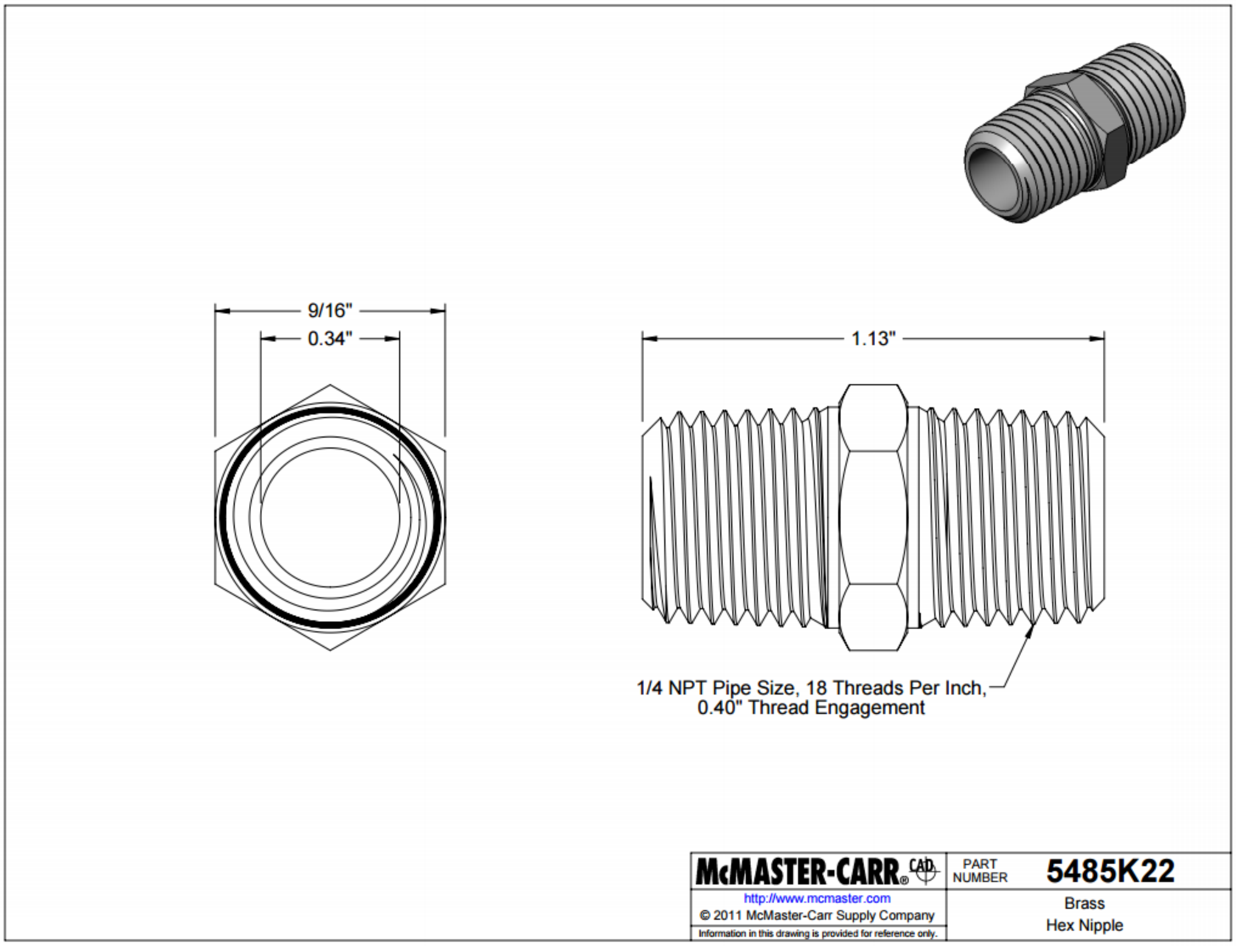
ADD TO ORDER

In stock
$2.30 Each
5485K22

Shape	Straight
Type	Connector
Body Shape	Hex
Connection Type	Pipe
Pipe Connection Type	Threaded
Connection	NPT Male
Pipe Size	1/4
Length	1.13"
For Use With	Air, Natural Gas, Oil, Water
Material	Brass
Fabrication	Machined from Bar
Maximum Pressure	1,000 psi @ 72° F
RoHS	Not Compliant

These fittings have the strength to handle high-pressure applications. They are brass for good corrosion resistance.

PROJECT:	Medium Duty Gas-Turbine Engine Blade Removal Tool				PART:	Removal Tool
TITLE:	Final Design Report				SECTION:	ME 430
ORIGINATOR:	J. Syage, E. Greb, K. Field		DATE:	11/30/2017	REVISION:	A
Reviewer:	J. Hernandez				PAGE: 121 of	178



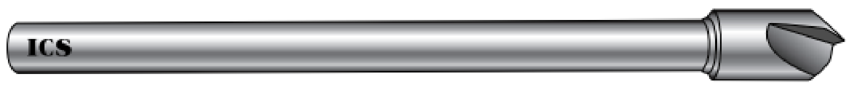
11.4.21 Appendix D.21: 1.00" 60° Long Reach Countersink End Mill CSKXL160-100

HIGH SPEED STEEL

LONG-REACH SINGLE FLUTE COUNTERSINKS

ICS USA | SPECIAL PURPOSE | RIGHT HAND CUT

EXTRA LENGTH STRAIGHT SHANK | FAST CUT 60°, 82°, 90° & 100° POINT ANGLES | ECCENTRIC RELIEF



High Speed Steel Long-Reach (Extended Length) Single Flute Countersinks are used for difficult to reach tool and die work. The single flute is designed for chamfering and deburring holes too small for multi-flute countersinks.

SIZE	SHANK DIAMETER	OVERALL LENGTH	ITEM NO. 60°	ITEM NO. 82°	ITEM NO. 90°	ITEM NO. 100°	PRICE
1/4	1/4	6"	CSKXL160-016	CSKXL182-016	CSKXL190-016	CSKXL100-016	$18.72
5/16	1/4	6"	CSKXL160-020	CSKXL182-020	CSKXL190-020	CSKXL100-020	19.26
3/8	1/4	6"	CSKXL160-024	CSKXL182-024	CSKXL190-024	CSKXL100-024	22.00
1/2	3/8	6"	CSKXL160-032	CSKXL182-032	CSKXL190-032	CSKXL100-032	30.00
5/8	3/8	6"	CSKXL160-040	CSKXL182-040	CSKXL190-040	CSKXL100-040	35.50
3/4	1/2	6"	CSKXL160-048	CSKXL182-048	CSKXL190-048	CSKXL100-048	39.90
1	1/2	6"	CSKXL160-100	CSKXL182-100	CSKXL190-100	CSKXL100-100	59.00

Note: Special length countersinks are available. Inquire for pricing.

11.4.22 Appendix D.22: .3750" HSS Chucking Reamer

High-Speed Steel Chucking Reamer

3/8" Diameter, 1-3/4" Flute Length

3 Each

In stock
$21.44 Each
2995A69

ADD TO ORDER

Material	High-Speed Steel
Flute Style	Straight
Reamer Diameter	0.3750"
Length	
Flute	1 3/4"
Overall	7"
Number of Flutes	6
Cutting Diameter Tolerance	0" to 0.0002"
Diameter Rating	Standard
Shank Type	Round
Reamer Type	Chucking (Machine)
For Use On	Aluminum, Brass, Bronze, Iron, Plastic, Stainless Steel, Steel

The shank diameter is slightly smaller than the reamer diameter.

Straight-flute reamers are for general purpose cutting.

11.4.23 Appendix D.23: .5000" HSS Chucking Reamer

High-Speed Steel Chucking Reamer

1/2" Diameter, 2" Flute Length

Each

ADD TO ORDER

In stock
$28.05 Each
2995A74

Material	High-Speed Steel
Flute Style	Straight
Reamer Diameter	0.5000"
Length	
Flute	2"
Overall	8"
Number of Flutes	6
Cutting Diameter Tolerance	0" to 0.0002"
Diameter Rating	Standard
Shank Type	Round
Reamer Type	Chucking (Machine)
For Use On	Aluminum, Brass, Bronze, Iron, Plastic, Stainless Steel, Steel

The shank diameter is slightly smaller than the reamer diameter.

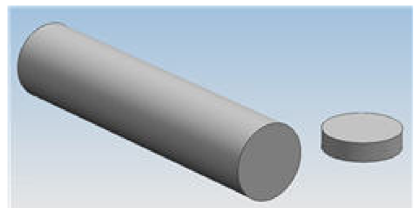
Straight-flute reamers are for general purpose cutting.

11.4.24 Appendix D.24: 6.50” Nominal 6061-T6 Extruded Aluminum Plate

6-1/2" {A} Rd 6061-T6511 Aluminum, Extruded

E-mail this product to a friend

Click here for material description, specification sheets and typical uses.



View larger image

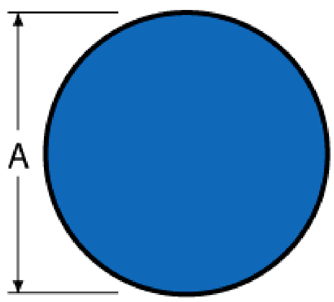
Rd 6061-T6511 Aluminum, Extruded

Dimensions:
A: 6-1/2"

Material: Aluminum

Grade: 6061

Shape: Round

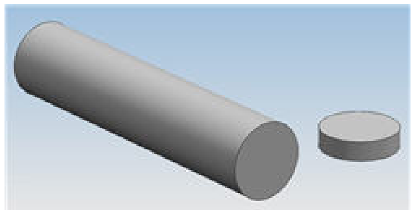


11.4.25 Appendix D.25: 14.50" Nominal 6061-T6 Extruded Aluminum Plate

14-1/2" {A} Rd 6061-T6 Aluminum, Extruded

E-mail this product to a friend

Click here for material description, specification sheets and typical uses.



View larger image

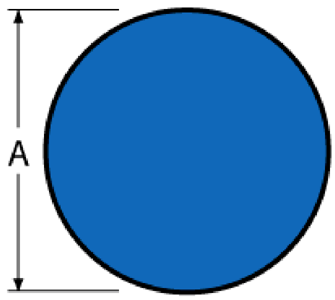
Rd 6061-T6 Aluminum, Extruded

Dimensions:
A: 14-1/2"

Material: Aluminum

Grade: 6061

Shape: Round

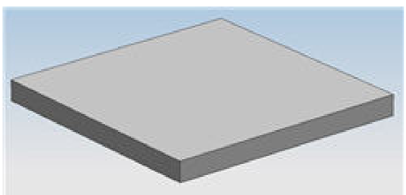


11.4.26 Appendix D.26: 12" x12" x1/4" 1045 Steel Plate

1/4" 1045 Hot Rolled, Steel Plate

E-mail this product to a friend

Click here for material description, specification sheets and typical uses.



View larger image

Speedy metals can custom cut plate materials to size. Call or e-mail us for more info

Notice **!!!** Plates have a flame cut edge.

1045 Hot Rolled, Steel Plate

Dimensions:
1/4"

Material: Steel

Grade: 1045

Shape: Plate

Finish: Hot Rolled

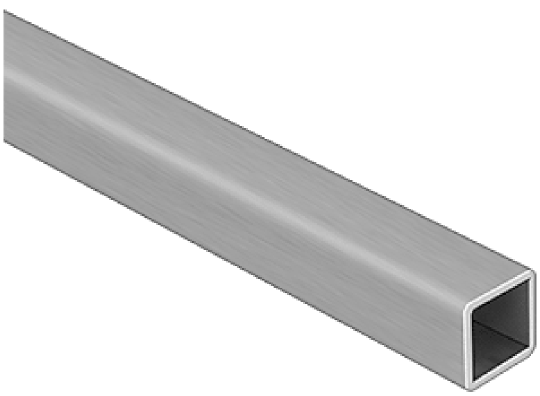
Length	Cutting Tolerance	Weight	Price	Order
12"x12" Plate	Plus or Minus 1/4"	10.8000 lbs	Price: $22.21	Quantity: 1 Add to Cart Add to Wish List

11.4.27 Appendix D.27: 2-½" Square Steel Tubing

McMASTER-CARR

Find

Solid Tube Framing, Steel, 2-1/2" Square



Length, ft.	
4	Each
6	ADD TO ORDER
8	
12	4931T146

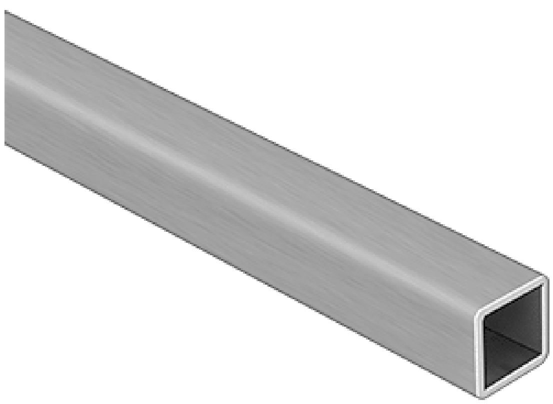
Material	Steel
Shape	Square
Size	2 1/2" sq.
Thickness	0.105"
RoHS	Compliant

11.4.28 Appendix D.28: 2-¼" Square Steel Tubing

MCMASTER-CARR®

Find

Solid Tube Framing, Steel, 2-1/4" Square



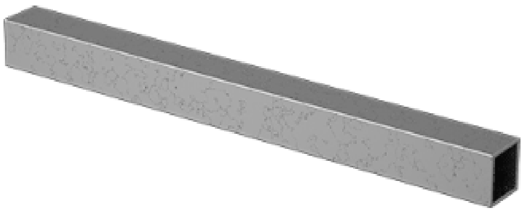
Length, ft.	Each
4	
6	ADD TO ORDER
8	
12	4931T145

Material	Steel
Shape	Square
Size	2 1/4" sq.
Thickness	0.105"
RoHS	Compliant

11.4.29 Appendix D.29: 1" x1" x .065" 4130 Rectangular Steel Tube

Easy-to-Weld 4130 Alloy Steel

Rectangular Tube, .065" Thick Wall, 1" x 1"

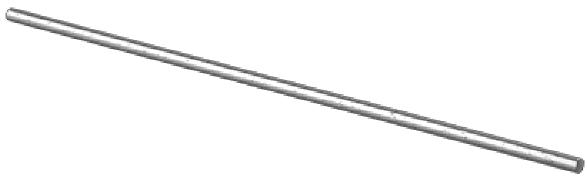


Length	Each
1/2 ft.	
1 ft.	ADD TO ORDER
2 ft.	
3 ft.	6582K43
6 ft.	

Grade	4130
Shape	Rectangular Tube
Finish	Unpolished
Wall Thickness	0.065"
Wall Thickness Tolerance	±0.0065"
Height	1"
Height Tolerance	±0.020"
Width	1"
Width Tolerance	±0.020"
Straightness Tolerance	0.075" per 3 ft.
Yield Strength	75,000 psi
Hardness	Hard (Rockwell C19)
Specifications Met	AMS-T-6736 and MIL-T-6736B
Construction	Cold Drawn
Material Condition	Annealed
Material Composition	
Carbon	0.27-0.34%
Manganese	0.35-0.60%
Silicon	0.15-0.40%
Phosphorus	0.011-0.035%
Sulfur	0.002-0.04%
Chromium	0.80-1.15%
Molybdenum	0.15-0.25%
Vanadium	0-0.035%
Aluminum	0.039%
Copper	0-0.25%
Hydrogen	0-2 ppm Max.
Nickel	0-0.25%
Niobium	0.05% Max.
Titanium	0.03% Max.
Iron	94.53-98.23%
Nominal Density	0.283 lbs./cu. in.
Electrical Resistivity	22.3 microhm-cm @ 68°F
Thermal Coefficient of Expansion per °F	7.6×10^{-6} (68° to 752° F)
Elongation Range	25-28%
Length Tolerance	±1"
RoHS	Compliant

11.4.30 Appendix D.30: ¼" Low Carbon Steel Rod

Low-Carbon Steel Rod
1/4" Diameter



Length	Each
1 ft.	
3 ft.	ADD TO ORDER
6 ft.	8920K115

Grade	1018
Shape	Rod
Finish	Unpolished
Diameter	1/4"
Diameter Tolerance	-0.002"
Yield Strength	54,000 psi
Hardness	Medium (Rockwell B70)
Specification Met	ASTM A108
Construction	Cold Drawn
Material Composition	
Carbon	0.13-0.20%
Manganese	0.30-0.90%
Silicon	0.15-0.30%
Phosphorus	0.04% Max.
Sulfur	0.50% Max.
Iron	98.06-99.42%
Nominal Density	0.283 lbs./cu. in.
Electrical Resistivity	15.9 microhm-cm @ 32° F
Thermal Conductivity	29.4 Btu/sq. ft./ft./hr./°F @ 212° F
Coefficient of Thermal Expansion (Text)	6.7-7.5 × 10^{-6}
Elongation Range	10-36%
RoHS	Not Compliant

11.4.31 Appendix D.31: ½" Low Carbon Steel Rod

Low-Carbon Steel Rod

1/2" Diameter

Length	Each
1 ft.	
3 ft.	
6 ft.	

ADD TO ORDER

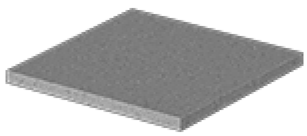
8920K155

Grade	1018
Shape	Rod
Finish	Unpolished
Diameter	1/2"
Diameter Tolerance	-0.002"
Yield Strength	54,000 psi
Hardness	Medium (Rockwell B70)
Specification Met	ASTM A108
Construction	Cold Drawn
Material Composition	
Carbon	0.13-0.20%
Manganese	0.30-0.90%
Silicon	0.15-0.30%
Phosphorus	0.04% Max.
Sulfur	0.50% Max.
Iron	98.06-99.42%
Nominal Density	0.283 lbs./cu. in.
Electrical Resistivity	15.9 microhm-cm @ 32° F
Thermal Conductivity	29.4 Btu/sq. ft./ft./hr./°F @ 212° F
Coefficient of Thermal Expansion (Text)	$6.7\text{-}7.5 \times 10^{-6}$
Elongation Range	10-36%
RoHS	Not Compliant

11.4.32 Appendix D.32: 3/16” Low Carbon Steel Plate

Low-Carbon Steel Sheet

3/16" Thick, 3" x 3", Ground Finish



Each

ADD TO ORDER

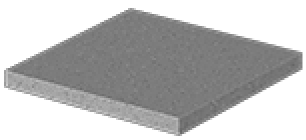
In stock
$9.22 Each
1388K662

Shape	Sheet
Finish	Ground
Thickness	3/16"
Thickness Tolerance	±0.001"
Width	3"
Width Tolerance	+1/2", −1/16"
Length	3"
Length Tolerance	+1/2", −1/16"
Yield Strength	36,000 psi
Hardness	Not Rated
RoHS	Compliant

11.4.33 Appendix D.33: ¼" Low Carbon Steel Plate

Low-Carbon Steel Sheet

1/4" Thick, 3" x 3", Ground Finish



Each

ADD TO ORDER

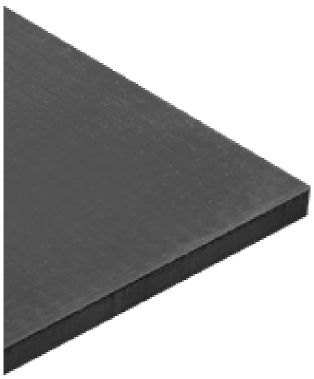
In stock
$11.28 Each
1388K102

Shape	Sheet
Finish	Ground
Thickness	1/4"
Thickness Tolerance	±0.003"
Width	3"
Width Tolerance	±1/8"
Length	3"
Length Tolerance	±1/8"
Yield Strength	36,000 psi
Hardness	Not Rated
RoHS	Compliant

11.4.34 Appendix D.34: 12” x 12” x 1/2” Oil Resistant Rubber Sheet

Oil-Resistant Buna-N Rubber Sheet

12" x 12", 1/2" Thick



Durometer
40A (Medium Soft)
50A (Medium)

Each

ADD TO ORDER

$31.94 Each
8635K168

Construction	Solid
Cross Section Shape	Rectangle
Material	Buna-N
Texture	Smooth
Thickness	1/2"
Thickness Tolerance	-0.047" to +0.047"
Width	12"
Width Tolerance	-0.250" to +0.250"
Length	12"
Length Tolerance	-0.25" to +0.25"
Backing Type	Plain
For Use Outdoors	No
Temperature Range	-20° to 170° F
Tensile Strength	800 psi
Color	Black
Specifications Met	ASTM D2000 BF
Durometer Tolerance	-5 to +5
RoHS	Compliant

Buna-N rubber is also known as nitrile, NBR, and acrylonitrile butadiene. It resists fuel oil, hydraulic oil, ethylene glycol, and grease.

11.4.35 Appendix D.35: Plastic Catch Bin

McMASTER-CARR

Find

Plastic Bin Box

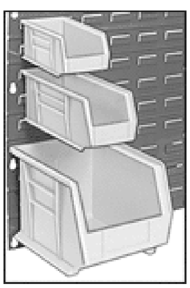
5-1/2" Wide, 5" High, 10-7/8" Deep

Color
- Blue
- Red
- Semi-Clear
- Yellow

Each

ADD TO ORDER

1-11 Each $5.44
12 or more $4.90
4666T63



Width	5 1/2"
Depth	10 7/8"
Height	5"
Capacity	30 lbs.
Nestable	No
Stackable	Yes
Hangable	Yes
Material	Polypropylene Plastic
Features	Label Holder
Related Products	Polystyrene Plastic Dividers Polystyrene Plastic Lids

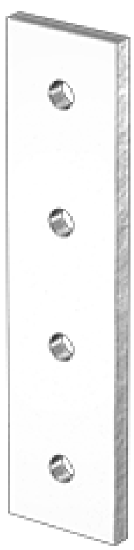
11.4.36 *Appendix D.36: Extended T-slotted Framing Bracket*

McMASTER-CARR

Find

T-Slotted Framing

Extended Straight Bracket for 1-1/2" High Rail, Silver

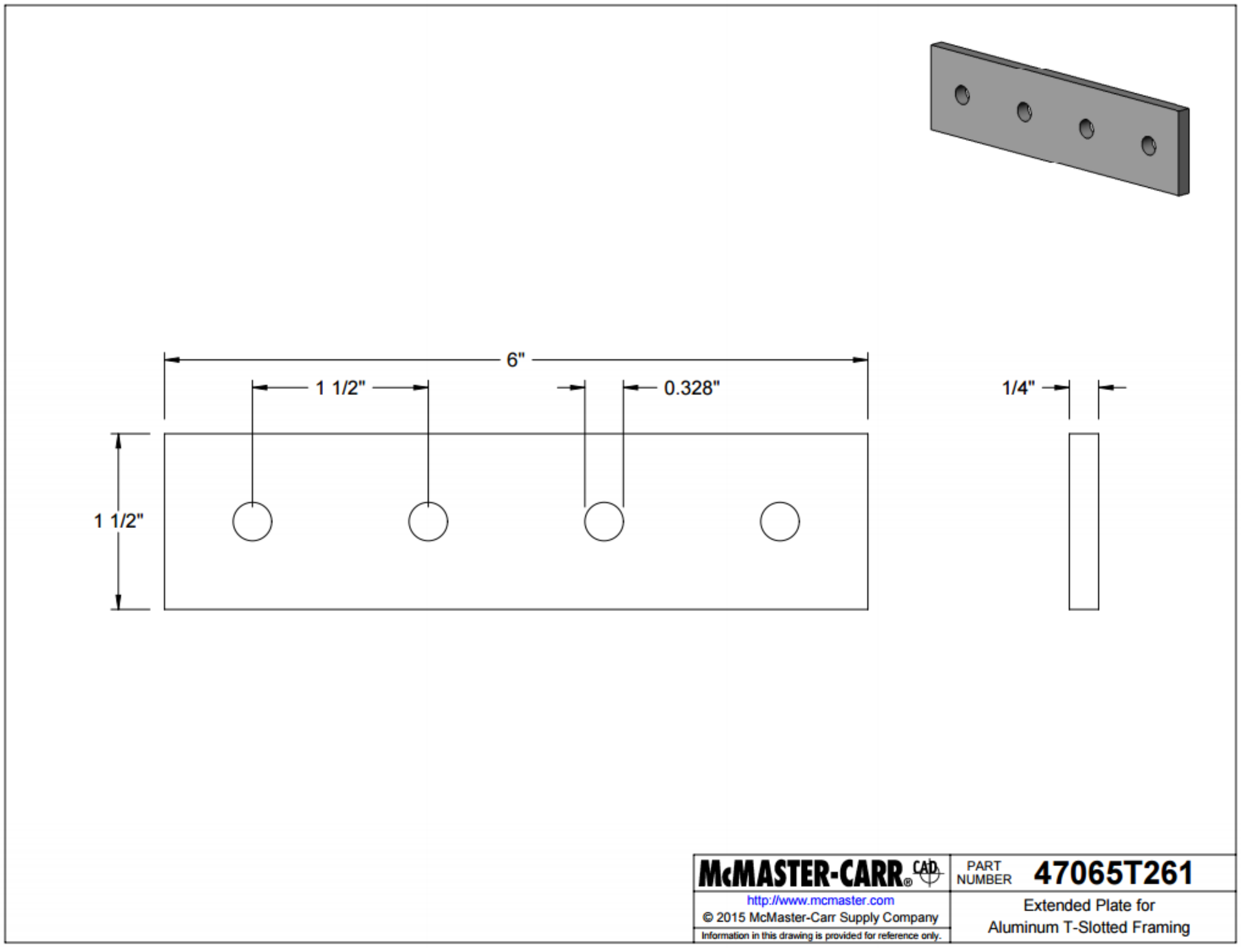


Each

ADD TO ORDER

In stock
$7.40 Each
47065T261

For Rail Height	1 1/2"
Color	Silver
Material	Anodized Aluminum
Length	6"
Mounting Fasteners Included	Yes
For Rail Style	Single
Bracket Type	Extended Straight
Framing Type	T Slot
T-Slot Framing Component	Surface Bracket
RoHS	Compliant



6"
1 1/2"
0.328"
1/4"
1 1/2"
McMASTER-CARR CAD
PART NUMBER
47065T261
http://www.mcmaster.com
© 2015 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Extended Plate for
Aluminum T-Slotted Framing

11.4.37 Appendix D.37: 1.50” x 1.50” T-Slotted Framing Rail

T-Slotted Framing

Single Rail, Silver, 1-1/2" High x 1-1/2" Wide, Hollow

Length, ft.	
1	5
2	6
3	8
4	10

Each

ADD TO ORDER

47065T102

Rail	
Height	1 1/2"
Width	1 1/2"
Rail Construction	Hollow
Color	Silver
Finish	Anodized
T-Slot Width	0.32"
Center Hole Diameter	0.26"
Rail Type	Standard
Rail Style	Single
Framing Type	T Slot
T-Slot Framing Component	Rail

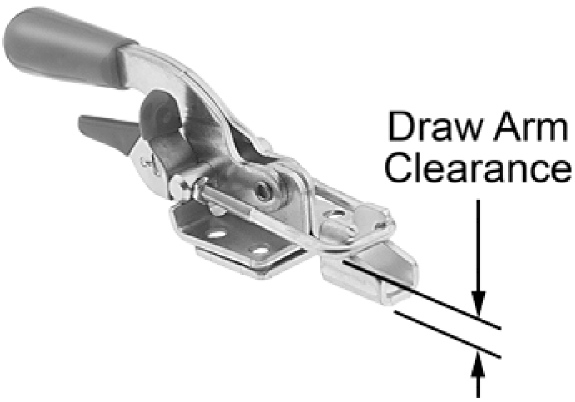
Make machine guards, carts, workstations, and more with our most versatile framing. It has continuous T-slots for attachment points, making structures easy to configure. Use fasteners to connect components.

Hollow rails are lighter and more economical than solid rails.

11.4.38 Appendix D.38: Double-Locking Pull-Action Toggle Clamp

Double-Locking Pull-Action Toggle Clamp

U-Bolt, 360 lb. Maximum Holding Capacity, Stainless Steel



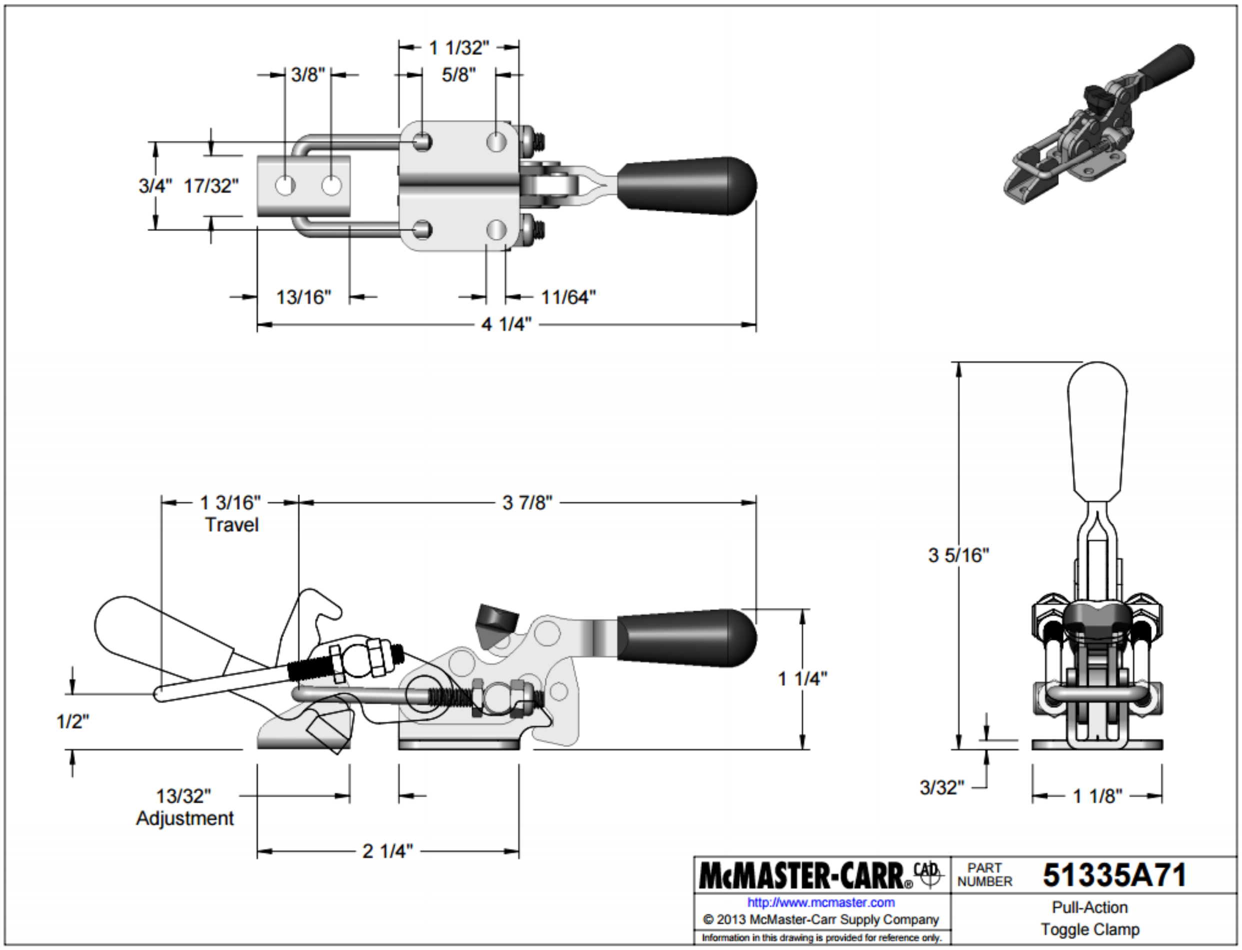
Each

ADD TO ORDER

In stock
$49.93 Each
51335A71

Clamping Action	Pull Only
Holding Capacity	360 lbs.
Draw Arm	
Style	U
Travel	1 1/4"
Adjustability	Adjustable
Adjustment Length	13/32"
Clearance	1/2"
When Clamped	
Overall Height	1 1/4"
Overall Length	3 7/8"
Body Material	Stainless Steel
Handle	
Style	Lever
Material	Plastic
Grip Style	Cushion
Includes	Latch Plate
Mount Type	Screw On
Mounting Fasteners Included	No
Mounting Holes	
Number of	6
Diameter	11/64"
RoHS	Compliant

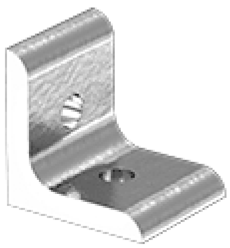
Made of stainless steel for use in harsh environments, use these clamps to secure lids and doors on drums and containers. Pull the handle back to engage the draw arm. A locking lever secures the clamp to prevent unintended openings in high-vibration applications.


1 1/32"
3/8"
5/8"
3/4"
17/32"
13/16"
11/64"
4 1/4"
1 3/16"
Travel
3 7/8"
3 5/16"
1 1/4"
1/2"
13/32"
Adjustment
2 1/4"
3/32"
1 1/8"
McMASTER-CARR
PART NUMBER
51335A71
http://www.mcmaster.com
© 2013 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Pull-Action
Toggle Clamp

11.4.39 Appendix D.39: 1.50” x 1.50” T-Slotted Framing Bracket

T-Slotted Framing

Corner Bracket for 1-1/2" High Single Rail, Silver



Each

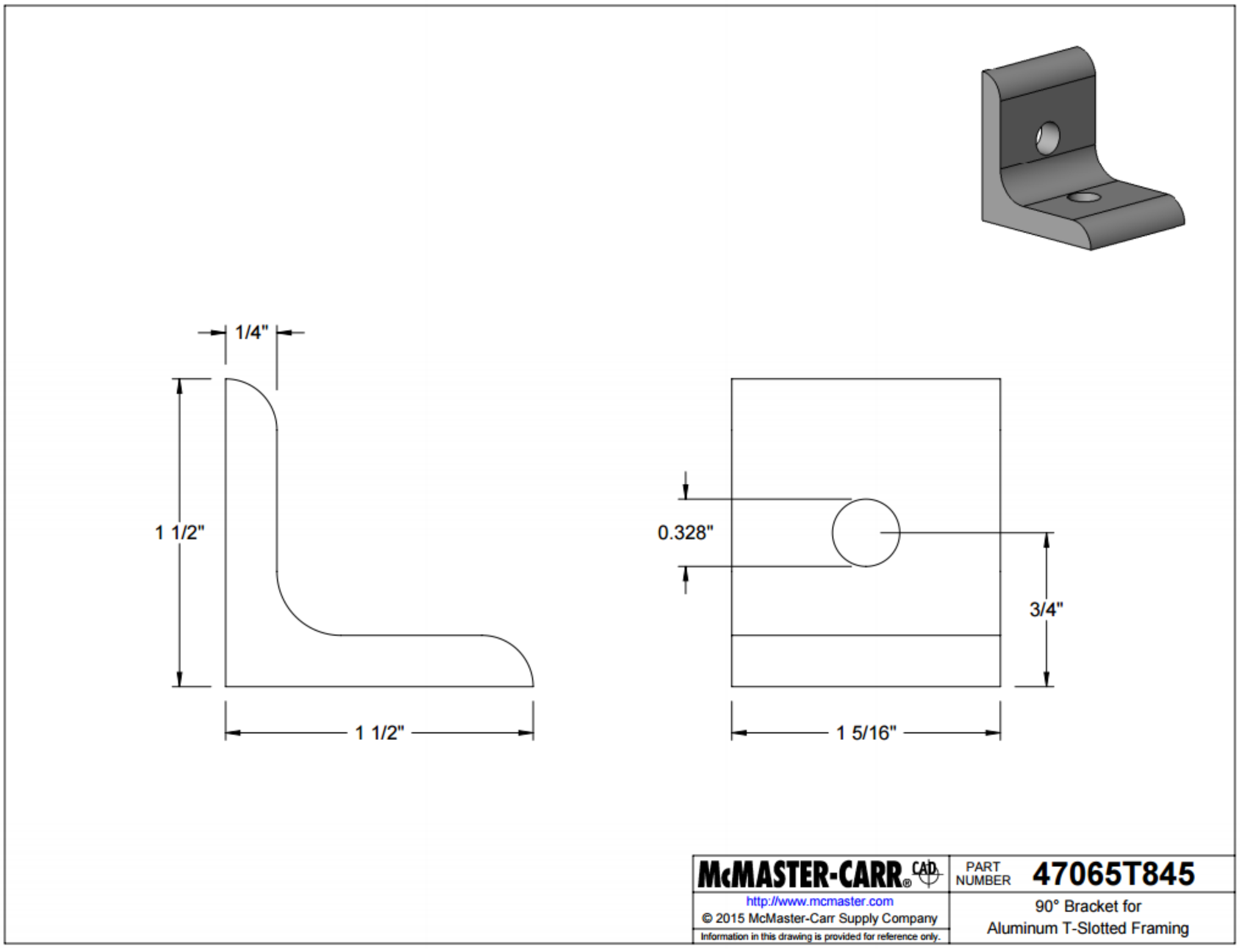
ADD TO ORDER

In stock
$6.41 Each
47065T845

For Rail Height	1 1/2"
Color	Silver
Material	Anodized Aluminum
Length	1 1/2"
Mounting Fasteners Included	Yes
For Rail Style	Single
Bracket Type	Corner
Framing Type	T Slot
T-Slot Framing Component	Corner Bracket
RoHS	Compliant

Make machine guards, carts, workstations, and more with our most versatile framing. It has continuous T-slots for attachment points, making structures easy to configure. Use fasteners to connect components.

Corner, extended corner, straight surface, extended straight surface, corner surface, and tee surface brackets connect rails without machining.

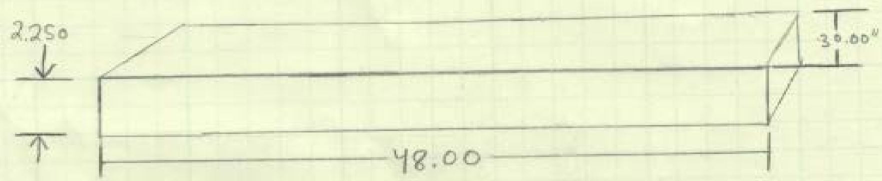


1/4"
1 1/2"
1 1/2"
0.328"
3/4"
1 5/16"
McMASTER-CARR CAD
http://www.mcmaster.com
© 2015 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
PART NUMBER 47065T845
90° Bracket for
Aluminum T-Slotted Framing

11.5 Appendix E: Detailed Supporting Analysis

11.5.1 Appendix E.1: Analysis of Table Structure

MAPLE TABLE WEIGHT CAPACITY –



ASSUMING COMPONENT WEIGT OF 377.25 LBS AND A SAFETY FACTOR OF 2.0 IN SHEAR AND BENDING, ANALYSIS FOR STATIC DEFLECTION IN WORST CASE CANTILEVER CONFIGURATION IS AS FOLLOWS.

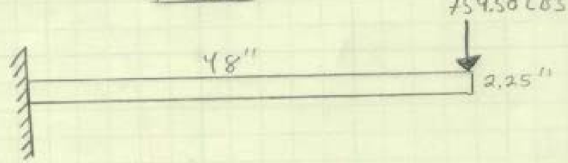
WEIGHT OF COMPONENTS: 377.25 LBS

SAFETY FACTOR: 2.0

EQUIVALENT LOAD: 754.50 LBS

ASSUMING POINT LOADING AT LONG END

FBD



DOWNWARD DEFLECTION AT END –

$E = 1{,}830{,}000 \ lb/in^2$

$I = \frac{30in\,(2.5in)^3}{12} = 39.0625 in^4$

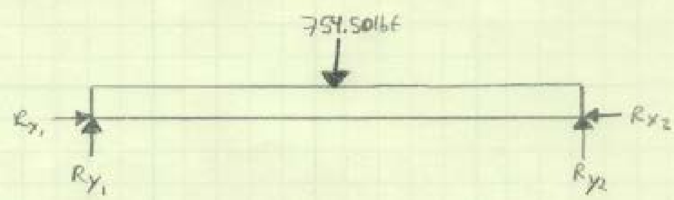
$$\delta_y = \frac{Px^3}{3EI}$$

$$\delta_y = \frac{(754.50lbf)(48in)^3}{3\left(1{,}830{,}000\frac{lbf}{in^2}\right)(39.0625in^4)}$$

$$\boxed{\delta_y = 0.3891in}$$

- NEGLIGIBLE DEFLECTION UNDER WORST-CASE CANTILEVER WITH SAFETY FACTOR

- IF LOADED IN CENTER LOADING



$$\delta_y = \frac{PL^3}{48EI}$$

$E = 1{,}830{,}000 \ lbf/in^2$

$I = 39.0625 in^4$

$$\delta_y = \frac{(754.50 lbf)(48in)^3}{48\left(1{,}830{,}000\frac{lbf}{in^2}\right)(39.0625in^4)}$$

$$\boxed{\delta_y = 0.0243in}$$

- NEGLIGIBLE DEFLECTION UNDER WORST-CASE CENTRAL LOADING CASE WITH SAFETY FACTOR.

• STRESS IN CANTILEVER AND THREE-POINT BENDING

$I = 39.0625 \, in^4$

• SECTION MODULUS = Z $\quad Z = \frac{bh^2}{6} = 25.3125 \, in^3$

CANTILEVER CONFIGURATION –

$$\sigma_{rupture} = \frac{PL}{Z}$$

$$\sigma_{rupture} = \frac{754.50 \, lbf \, (48 \, in)}{25.3125 \, in^2}$$

$$\boxed{\sigma_{rupture} = 1430.75 \, psi}$$

CENTER LOADING CONFIGURATION –

$$\sigma_{rupture} = \frac{PL}{4Z}$$

$$\sigma_{rupture} = \frac{754.50 \, lbf \, (48 \, in)}{4(25.3125 \, in^2)}$$

$$\boxed{\sigma_{rupture} = 357.68 \, psi}$$

• ALL WORST CASE LOADING SCENARIOS ARE CALCULATED TO BE LESS THAN THE RUPTURE AND BENDING STRESS MATERIAL PROPERTIES FOR NORTH AMERICAN MEDIUM HARDNESS MAPLE WOOD. THE DESIGN IS SUFFICIENTLY SAFE.

Table E.1.1: Maple Butcher Block Dimensions.

Table Dimensions		
Length	48.00	in
Width	30.00	in
Thickness	2.25	in
Surface Area	1440.00	in^2
Cross Section Area	67.50	in^2
Volume	3240.00	in^3
I_{xx} Inertia	28.48	in^4
Section Modulus	25.31	in^3

Table E.1.2: Maple Wood Properties.

Maple Wood Properties		
Density	44.00	lbf/ft^3
Specific Gravity	0.60	--
Rupture Modulus	15800.00	lbf/in^2 (psi)
Elastic Modulus	1830000.00	lbf/in^2 (psi)
Crush Strength	7830.00	lbf/in^2 (psi)
Maximum Loading	377.25	lbf
Safety Factor	2.00	

Table E.1.3: Butcher Block Bending Deflection.

Bending Deflection		
Cantilever	0.534	in
Simply Supported Central Loading	0.033	in

Table E.1.4: Butcher Block Stresses

Rupture Stress		
Cantilever	1430.76	lbf/in^2 (psi)
Simply Supported Central Loading	357.69	lbf/in^2 (psi)

11.5.2 Appendix E.2: Analysis of Trailer Hub Assembly

TRAILER HUB ASSY
ANALYSIS

BEARING CAPACITY 2 X 5709K83 TAPERED ROLLER

AXIAL THRUST LOAD CAPACITY – 1040 lbf
RADIAL STATIC LOAD CAPACITY – 1620 lbf

TOTAL LOAD CAPACITY – AXIAL – 2080 lbf
TOTAL LOAD CAPACITY – RADIAL – 3240 lbf

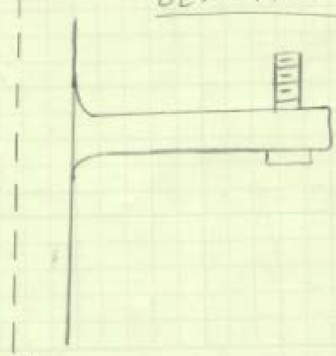
TOTAL AXIAL LOAD FROM ALL COMPONENTS –
377.25 lbf

SAFETY FACTOR – 2.0

EQUIVALENT WEIGHT – 755.50 lbf

THE HUB, BEARINGS, AND SPINDLE ARE
SUFFICIENTLY DESIGNED FOR THE LOADING CASE

BENDING OF HUB –



IMPACT LOAD – 100 lbf
SAFETY FACTOR – 2.0
EQUIVALENT LOA – 200 lbf

SECTION

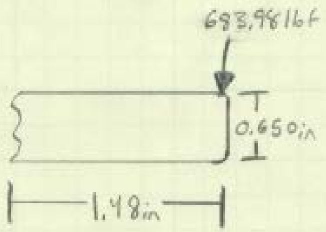
PROJECT:	Medium Duty Gas-Turbine Engine Blade Removal Tool				**PART:**	Removal Tool
TITLE:	Final Design Report				**SECTION:**	ME 430
ORIGINATOR:	J. Syage, E. Greb, K. Field		**DATE:**	11/30/2017	**REVISION:**	A
Reviewer:	J. Hernandez				**PAGE: 149 of**	178

WEIGHT OF DISK + HUB PLATE + CUSHION + FASTENERS – 241.99 lbf

EQUIVALENT STATIC LOAD WITH SAFETY FACTOR – 483.98 lbf

TOTAL LOAD – 683.98 lbf WORST CASE

ASSUMING SPINDLE IS RIGID AND BOLT FLANGE IS IN BENDING, CANTILEVER CONFIGURATION.



$E = 27{,}557{,}000 \text{ psi}$

$$\delta_y = \frac{PL^3}{3EI}$$

$$I = \frac{(1.48 \text{ in})(0.650 \text{ in})^3}{12} = 0.0338 \text{ in}^4$$

$$\delta_y = \frac{(683.98 \text{ lbf})(1.48 \text{ in})^3}{3\left(27{,}557{,}000 \frac{\text{lbf}}{\text{in}^2}\right)(0.0388 \text{ in}^4)}$$

$$\boxed{\delta_y = 6.914 \times 10^{-4} \text{ in}}$$

THE HUB DOES NOT DEFLECT SIGNIFICANTLY UNDER BENDING ALONG THE FACE.

Table E.2.1: 4340 Steel Properties.

4340 Steel Properties		
Density	490.75	lbf/ft^3
Tensile Strength	108000.00	lbf/in^2 (psi)
Yield Strength	68200.00	lbf/in^2 (psi)
Elastic Modulus	27557000.00	lbf/in^2 (psi)
Shear modulus	11600000.00	lbf/in^2 (psi)
Poisson's Ratio	0.28	--
Maximum Loading	341.99	lbf
Safety Factor	2.00	

Table E.2.2: Trailer Hub Face Dimensions.

Hub Face Dimensions		
Length	1.48	in
Thickness	0.65	in
I_{xx} Inertia	0.03	in^4

Table E.2.3: Trailer Hub Bending Deflection

Bending Deflection		
Cantilever	0.00079	in

11.5.3 Appendix E.3: Analysis of Trailer Hub Bearing Life

TRAILER HUB BEARING REQUIREMENTS

• NOTE: ALL LOADS ARE DESIGNED FOR WORST POSSIBLE LOADING SCENARIO.

LOADS:

AXIAL: 100 lbf

RADIAL: 272.89 LBF

SAFETY FACTOR: 2.0

APPLICATION FACTOR: 2.25 FOR MODERATE IMPACT NON-CONTINUOUS TOOLING ACCORDING TO SHIGLEY 7th EDITION DESIGN STANDARDS

• BEARING ANALYZED: SKF L44600

ASSUMED PARAMETERS:

- 8.00 RPM
- 7500 HOUR USAGE LIFE
- RELIABILITY 99%

BEARING ANALYSIS –

$F_a = 2(272.89\ lbf) = 545.78\ lbf$

$F_r = 2(100\ lbf) = 200\ lbf$

$C_o = 6744\ lbf$ → SKF L44600

$F_a/C_o = 0.08$

$F_a/VF_r = \frac{545.78\ lbf}{1.2(200\ lbf)} = 2.27$

$X_2 = 0.56$
$Y_2 = 1.50$ } INTERPOLATED SHIGLEY 8.2

$F_{equivalent} = 0.56(1.2)(200\ lbf) + 1.50(545.78\ lbf)$

$F_{equivalent} = 953.07\ lbf$

$$X_D = \frac{60(7500\,hrs)(8.00\,RPM)}{1{,}000{,}000\ cycles}$$

$$X_D = 3.60$$

C_{10} DYNAMIC REQUIREMENT PER BEARING –

$$C_{10} = (2.25 \cdot 953.07\,lbf)\left[\frac{3.60}{(0.02 + 4.439(1-.99))^{1/1.483}}\right]^{\frac{1}{10/3}}$$

$C_{10} = 7720.14\ lbf$ DISTRIBUTED ACROSS TWO IDENTICAL 1 1/16" TAPERED ROLLER BEARINGS.

· NOTE: IT IS ASSUMED THE LOAD SPLIT BETWEEN BEARINGS IS 50% / 50%

C_{10} REQUIRED PER BEARING = **3860.07 lbf**

Table E.3.1: Trailer Hub Manufacturer's Specs.

SKF		
Bearing	L 44600	
Speed Limit	15000	rpm
Co Static Load	6744	lbf
C10 Dynamic Load	5848	lbf
Tested Cycles	1000000	--

Table E.3.2: Trailer Hub Bearing Loads

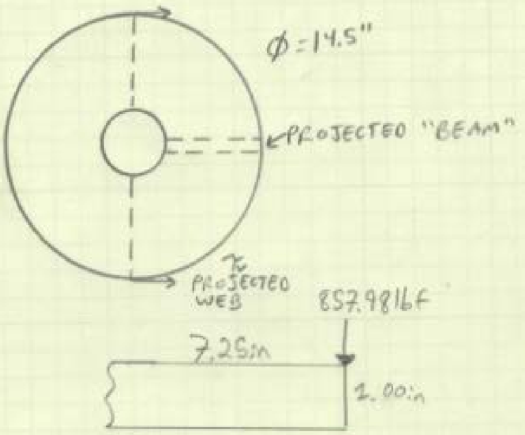
Loads		
Radial Load	100.00	lbf
Axial Load	272.89	lbf
Safety Factor	2.00	--
Application Factor	2.25	--
Roller Bearing Constant	43011.00	--
Ball Bearing Constant	3.00	--
Assumed Operation RPM	8.00	RPM
Desired Life	7500.00	Hours
Bearing Reliability	0.99	%
V Value for Outer Ring Rotation	1.20	--

Table E.3.3: Trailer Hub Equivalent Radial Loading

Equivalent Radial Loading		
Fa	545.78	lbf
Fr	200.00	lbf
Fa/Co	0.08	--
Fa/(V*Fr)	2.27	--
X2	0.56	--
Y2	1.50	--
Fe	953.07	lbf
xD	3.60	--
C10	**7720.14**	**lbf**
C10 Per Bearing	**3860.07**	**lbf**

11.5.4 Appendix E.4: Analysis of Hub Plate

HUB PLATE BENDING ALONG EDGES DUE TO TOOL IMPACTING



Static Load w/SF

657.98lbf

MAX STATIC + IMPACT LOAD:

857.98lbf w/SF

$I = 0.6041\ in^4$

$E = 10{,}000{,}000\ psi$

$$\delta_{max} = \frac{PL^3}{3EI}$$

$$\delta_{max} = \frac{(857.98lbf)(7.25in)^3}{3(10{,}000{,}000psi)(0.6041in^4)}$$

$$\boxed{\delta_{max} = 0.0180in}$$

$$\delta_{static} = \frac{(657.98lbf)(7.25in)^3}{3(10{,}000{,}000psi)(0.6041in^4)}$$

$$\boxed{\delta_{static} = 0.1383in}$$

$\Delta\delta$ during impact $= \delta_{max} - \delta_{static} = 0.0041in$

THE HUB PLATE WILL DEFLECT APPROXIMATELY .0041 in DURING EACH 100 lbf IMPACT BY THE TOOL. THIS IS NEGLIGIBLE AND SAFE.

BENDING STRESS –

TRANSVERSE SHEAR – $\frac{4V}{3A}$

$A_{web} = 0.5(1.0in)(14.5in)$

$A = 7.25in^2$

$$\tau_v = \frac{4(857.98lbf)}{3(7.25in^2)}$$

$$\boxed{\tau_v = 157.78psi}$$

THE DESIGN IS SUFFICIENT IN TRANSVERSE SHEAR LOAD CONDITIONS

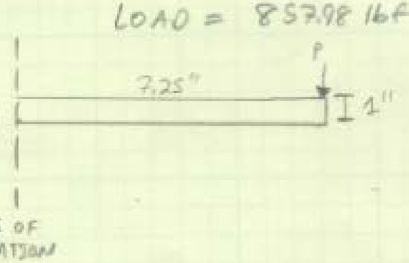
BENDING YIELD STRESS –

LOAD = 857.98 lbf

$$\sigma_b = \frac{Mc}{I}$$

$$\sigma_b = \frac{(857.98lbf)(7.25in)(.5in)}{\frac{7.25in(1in)^3}{12}}$$

$$\boxed{\sigma_b = 10{,}295.76psi}$$



THE BENDING STRESS WORST CASE IS CALCULATED TO BE SIGNIFICANTLY LESS THAN THE MATERIAL'S YIELD STRESS. THE HUB IS SAFE AND MEETS DESIGN CRITERIA.

Table E.4.1: Hub Plate Material Properties.

6061-T6 Properties		
Density	168.48	lbf/ft^3
Tensile Strength	45000.00	lbf/in^2 (psi)
Yield Strength	40000.00	lbf/in^2 (psi)
Elastic Modulus	10000000.00	lbf/in^2 (psi)
Shear modulus	3770000.00	lbf/in^2 (psi)
Poisson's Ratio	0.33	--
Static Loading	328.99	lbf
Dynamic Loading	100.00	lbf
Safety Factor	2.00	

Table E.4.2: Hub Plate Dimensions.

Hub Plate Dimensions		
Radius	7.25	in
Thickness	1.95	in
Mass	27.40	lbf
Area Inertia Modeled as Cantilever Beam	4.48	in^4
Rotational Inertia	720.11	$lbf*in^2$

Table E.4.3: Hub Plate Bending Deflection

Bending Deflection		
Cantilever Static	0.0019	in
Cantilever Dynamic	0.0024	in
Differential Deflection	0.0006	in

Table E.4.4: Hub Plate Shear Fracture

Shear Fracture		
Transverse Shear Stress	80.92	lbf/in^2 (psi)
Bending Stress	2707.63	lbf/in^2 (psi)

11.5.5 Appendix E.5: Analysis of Rotor Disk

ROTOR DISK DEFLECTION
AND APPLIED STRESS

- THE DEFLECTION OF THE ROTOR DISK ASSEMBLY UNDER IMPACTING MUST BE KEPT MINIMUM
- ASSUMPTIONS-
  - WASPALOY ASSUMED TO COMPRISE 100% OF DISK MATERIAL
  - BLADES AND IMPACT LOCATION ASSUMED TO BE PART OF DISK RIGID BODY.

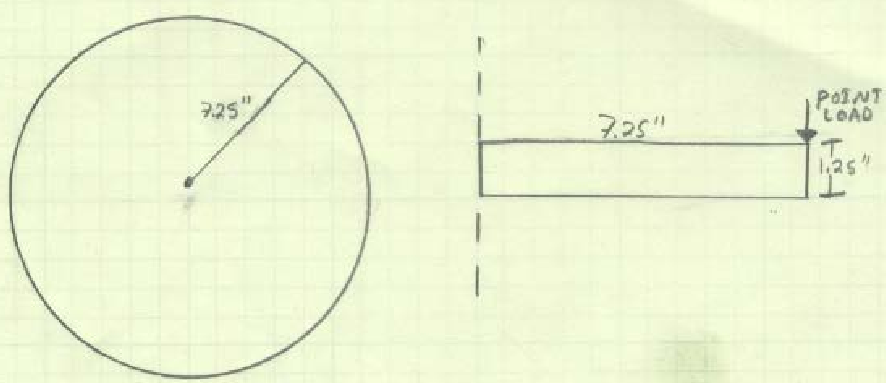
WASPALOY PROPERTIES-

DENSITY- 508.03 lbf/ft^3
TENSILE - 193,000 psi
YIELD - 132,000 psi
POISSONS - 0.30

LOADS-

STATIC - 0.00 lbf
DYNAMIC - 100.00 lbf
SAFETY FACTOR - 2.0

- DISK MODELED AS A UNIFORM SEMI-THIN DISK



BENDING DEFLECTION

DYNAMIC CASE – CANTILEVER BEAM BENDING ALONG A RECTANGULAR WEB CROSS SECTION.

$$\delta = \frac{P \cdot SF \cdot L^3}{3EI}$$

$$\delta = \frac{(100\,lbf)(2.0)(9.55\,in)^3}{3(30\times10^6\,psi)(1.55\,in^4)}$$

$$\boxed{\delta = 0.0012\,in}$$

AT WORST CASE IN AN UNREALISTIC EXTREME LOAD CASE THE DISK DEFLECTS 0.012 in. THIS IS FAR GREATER THAN THE ACTUAL DEFLECTION DUE TO THICKER DISK PROFILE AND DISTRIBUTION OF LOAD.

BENDING & TRANSVERSE SHEAR STRESSES

$$\tau_v = \frac{4V}{3A}$$

$$\tau_v = \frac{4(200\,lbf)}{3(9.55\,in \cdot 1.25\,in)}$$

$$\boxed{\tau_v = 22.34\,psi} \rightarrow \text{NEGLIGIBLE}$$

$$\sigma_b = \frac{P \cdot SF \cdot r \cdot e}{I}$$

$$\sigma_b = \frac{100\,lbf(2.0)(9.55\,in)(1.25\,in)}{1.55\,in^4}$$

$$\boxed{\sigma_b = 1536\,psi} \rightarrow \text{SMALL COMPARED TO MATERIAL YIELD.}$$

BENDING DEFLECTION

DYNAMIC CASE – CANTILEVER BEAM BENDING ALONG A RECTANGULAR WEB CROSS SECTION.

$$\delta = \frac{P \cdot SF \cdot L^3}{3EI}$$

$$\delta = \frac{(100\,lbf)(2.0)(9.55\,in)^3}{3(30\times10^6\,psi)(1.55\,in^4)}$$

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BENDING & TRANSVERSE SHEAR STRESSES

$$\tau_v = \frac{4V}{3A}$$

$$\tau_v = \frac{4(200\,lbf)}{3(9.55\,in \cdot 1.25\,in)}$$

$$\boxed{\tau_v = 22.34\,psi} \rightarrow \text{NEGLIGIBLE}$$

$$\sigma_b = \frac{P \cdot SF \cdot r \cdot c}{I}$$

$$\sigma_b = \frac{100\,lbf(2.0)(9.55\,in)(1.25\,in)}{1.55\,in^4}$$

$$\boxed{\sigma_b = 1536\,psi} \rightarrow$$ SMALL COMPARED TO MATERIAL YIELD.

Table E.5.1: Rotor Disk Material Properties (Waspaloy)

Waspaloy Properties		
Density	508.03	lbf/ft^3
Tensile Strength	193000.00	lbf/in² (psi)
Yield Strength	132000.00	lbf/in² (psi)
Elastic Modulus	30000000.00	lbf/in² (psi)
Poisson's Ratio	0.30	--
Static Loading	0.00	lbf
Dynamic Loading	100.00	lbf
Safety Factor	2.00	

Table E.5.2: Rotor Disk Dimensions

Rotor Disk Dimensions		
Radius	9.55	in
Thickness	1.25	in
Mass	200.00	lbf
Area Inertia Modeled as Cantilever Beam	1.55	in^4
Rotational Inertia	9120.25	lbf*in²

Table E.5.3: Rotor Disk Bending Deflection

Bending Deflection		
Cantilever Static	0.0000	in
Cantilever Dynamic	0.0012	in
Differential Deflection	0.0012	in

Table E.5.5: Rotor Disk Shear Fracture

Shear Fracture		
Transverse Shear Stress	22.34	lbf/in² (psi)
Bending Stress	1536.00	lbf/in² (psi)

11.5.6 Appendix E.6: Analysis of Trailer Hub Assembly Fastener Preloads

FASTENER PRELOAD –

FASTENERS –

1) 1/2" - 13 x 2.00 in FLANGE BOLT GRADE 5
2) 1/2" - 13 FLANGE NUT GRADE 5
3) 1/2" - 20 60° TAPERED LUG NUT GRADE 5
4) 1/2" - 13 x 5.00 in NYLON STUD

SINGLE EXAMPLE SHOWN, EQUATION MODIFIED FOR ALL FASTENERS TO CALCULATE PRELOAD FORCE AND CORRESPONDING TORQUE.

1/2" - 13 x 2.00" BOLT WITH 1/2" - 13 NYLOCK NUTS

NOMINAL DIAMETER – 0.50 in

THREADS PER INCH – 13

$A_d \rightarrow \frac{\pi(.5)^4}{4} \rightarrow 0.05\,in^2$

$A_t \rightarrow$ book $\rightarrow 0.14\,in^2$

$D_r \rightarrow \sqrt{\frac{4A_t}{\pi}} \rightarrow 0.43\,in$

$E \rightarrow 29.7\times10^6\,psi$

$l_t \rightarrow 2.00\,in$

$l_d \rightarrow 0.00\,in$

$K_b \rightarrow \frac{A_d A_t E}{A_d l_t + A_t l_d} \rightarrow 2{,}107{,}215\,psi$

$K_m \rightarrow$ MAPLE WOOD $\rightarrow 1.83\times10^6\,psi$

$D_m \rightarrow \frac{D_{nom} + D_r}{2} \rightarrow 0.463\,in$

$\theta_{Lead} \rightarrow \tan^{-1}\left(\frac{1}{\pi D_m TPI}\right) \rightarrow 0.053\,rad \rightarrow 3.03\,deg$

ASSUMED

CLAMP FORCE – 5000 lbf

BOLTS – 4

LOAD PER BOLT – 1250 lbf

PRELOAD TORQUE REQUIRED PER BOLT –

- $$T_{req} = \frac{F_i d_m}{2}\left(\frac{\tan(\lambda) + f\sec(\alpha)}{1 - f\tan(\lambda)\sec(\alpha)}\right) + \frac{F_i f_c d_c}{2}$$

- ASSUMED f_c COEFFICIENT OF THREAD FRICTION OF 0.20 FOR NEW, UNLUBRICATED THREADS

$$T_{req} = \frac{1250\,lbf(.463\,in)}{2}\left(\frac{\tan(3.03°) + 0.20\sec(30°)}{1 - 0.20\tan(3.03°)\sec(30°)}\right) + \left(\frac{1250\,lbf(0.20)(0.50)}{2}\right)$$

$$\boxed{T_{req} = 466.98\ in\cdot lbf = 38.92\ ft\cdot lb}$$

THIS FALLS IN A RANGE RECOMENDED FOR 1/2"-13 BOLTS ACCORDING TO FASTENAL INC AND ASTM A307 FOR GRADE 2 AND 5 FASTENERS

Table E.6.1: Analysis of Fasteners

Fastener 1/2-13 x 2.00" Bolt			**Fastener 1/2-20 Lug Nut**		
Nominal Diameter	0.50	in	Nominal Diameter	0.50	in
Threads Per Inch	13.00	--	Threads Per Inch	20.00	--
Ad	0.05	in^2	Ad	0.05	in^2
At	0.14	in^2	At	0.14	in^2
Dr	0.43	in	Dr	0.43	in
E	29700000.00	lbf/in^2 (psi)	E	29700000.00	lbf/in^2 (psi)
Lt	2.00	in	Lt	0.65	in
Ld	0.00	in	Ld	0.00	in
Fastener Stiffness	2107215.00	lbf/in^2 (psi)	Fastener Stiffness	6483738.46	lbf/in^2 (psi)
Clamp force	5000.00	lbf	Clamp force	15000.00	lbf
Number of Bolts	4.00	--	Number of Bolts	1.00	--
Load Per Bolt	1250.00	lbf	Load Per Bolt	15000.00	lbf
Member Stiffness	1830000.00	lbf/in^2 (psi)	Member Stiffness	29700000.00	lbf/in^2 (psi)
Dm	0.463	in	Dm	0.463	in
Lead Angle	0.053	Radians	Lead Angle	0.034	Radians
Lead Angle	3.030	Degrees	Lead Angle	1.971	Degrees
Bolt Preload Torque	**466.98**	**in*lbf**	**Bolt Preload Torque**	**637.41**	**in*lbf**
Bolt Preload Torque	**38.92**	**ft*lb**	**Bolt Preload Torque**	**53.12**	**ft*lb**

Table E.6.2: Analysis of Fasteners Cont.

Fastener 1/2-13 Nylon Stud		
Nominal Diameter	0.50	in
Threads Per Inch	13.00	--
Ad	0.05	in^2
At	0.14	in^2
Dr	0.43	in
E	425000.00	lbf/in^2 (psi)
Lt	5.00	in
Ld	0.00	in
Fastener Stiffness	12061.50	lbf/in^2 (psi)
Clamp force	1000.00	lbf
Number of Bolts	3.00	--
Load Per Bolt	333.33	lbf
Member Stiffness	29700000.00	lbf/in^2 (psi)
Dm	0.463	in
Lead Angle	0.053	Radians
Lead Angle	3.030	Degrees
Bolt Preload Torque	**124.53**	**in*lbf**
Bolt Preload Torque	**10.38**	**ft*lb**

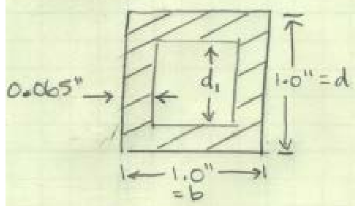
11.5.7 Appendix E.7: Analysis of Locking Mechanism Steel Tube Frame

CANTILEVER HOLLOW BEAM ANALYSIS

- ANALYZING THE LOCKING MECHANISM FRAME AT WORST CASE SCENARIO

KNOWNS:

CROSS SECTION:



LENGTH = 2.0 IN
LOAD = 61.8 lbf
$E = 27557000$ psi $= 2.7557 \times 10^7$ psi

FIND: DEFLECTION, δ
CANTILEVER BEAM STRESS, σ

SOLUTION:

- FIND $I_{xx} = I_{yy} = I$

$d_1 = 1.0 - 2(.065)$ IN

$d_1 = 0.87$ IN

$\rightarrow I = \frac{bd^3}{12} - \frac{b_1 d_1^3}{12}$

$I = \frac{(1.0)(1.0)^3 \text{ IN}^4}{12} - \frac{(0.87)(0.87)^3}{12} \text{ IN}^3$

$I = 0.0356 \text{ IN}^4$

- FIND δ

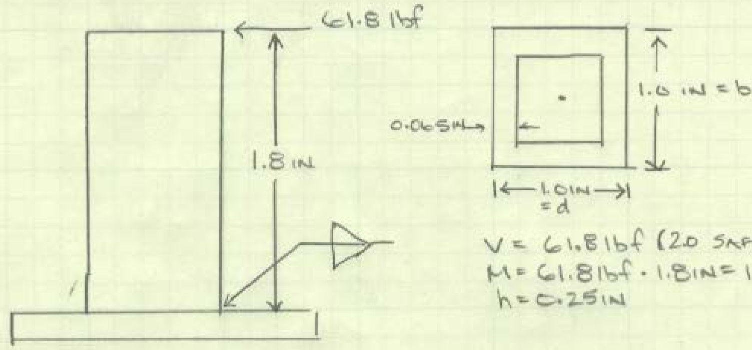
$\delta = \frac{PL^3}{3EI}$

$\delta = \frac{(61.8 \text{ lbf})(2.0 \text{ IN})^3}{3(2.7557 \times 10^7 \text{ lbf/IN}^2)(0.0356 \text{ IN}^4)}$

$\boxed{\delta = 1.68 \times 10^{-4} \text{ IN} = 0.000168 \text{ IN}}$

11.5.8 Appendix E.8: Analysis of Locking Mechanism Steel Tube Weld Stress

LOCKING MECHANISM – WELD STRESS (WORST CASE)



V = 61.8 lbf (20 SAFETY FACTOR)
M = 61.8 lbf · 1.8 IN = 111.24 lbf-in
h = 0.25 IN

FIND:
- FILLET WELD SHEAR, τ
- COMPARE TO τ_{ALLOW}

SOLUTION:

- $\tau' = \frac{V}{A}$ (PRIMARY SHEAR)

$\rightarrow A = 1.414h(b+d)$

$= 1.414(0.25\,in)(1.0 + 1.0\,in)$

$A = 0.707\,in^2$

$\tau' = \frac{61.8\,lbf}{.707\,in^2}$

$\tau' = 87.4\,psi$

- $\tau'' = \frac{1.414M}{bdh}$ (SECONDARY SHEAR)

$\tau'' = \frac{1.414(111.24\,lbf\text{-}in)}{(1.0\,in)(1.0\,in)(0.25\,in)}$

$\tau'' = 629.2\,psi$

$\tau = (\tau'^2 + \tau''^2)^{1/2}$

$\tau = 635.2\,psi$

$\tau_{ALLOW} = 21\,ksi > 635.2\,psi$ ✓

11.5.9 Appendix E.9: Analysis of Locking Pin

LOCKING PIN - CANTILEVER BEAM ANALYSIS

KNOWNS: 3/8" 4140 STEEL PIN
$E = 2.7557 \times 10^7$ psi
LOAD = 61.8 lbf
LENGTH = 3.37 IN

FIND: DEFLECTION, δ
CANTILEVER BEAM STRESS

SOLUTION:

$$I_{xx} = I_{yy} = I$$

$$I = \frac{1}{4}\pi r^4$$

$$= \frac{1}{4}\pi\left(\frac{.375}{2}\right)^4$$

$$I = 9.707 \times 10^{-4}\ \text{IN}^4$$

- FIND δ

$$\delta = \frac{PL^3}{3EI}$$

$$= \frac{(61.8\ \text{lbf})(3.37\ \text{IN})^3}{3(2.7557 \times 10^7\ \text{lbf/in}^2)(9.707 \times 10^{-4}\ \text{IN}^4)}$$

$$\boxed{\delta = 0.02947\ \text{IN}}$$

- FIND $\sigma_{b,MAX}$

$$\sigma_{b,MAX} = \frac{Mc}{I}$$

$$= \frac{(61.8\ \text{lbf} \cdot 3.37\ \text{IN})\left(\frac{.375}{2}\ \text{IN}\right)}{9.707 \times 10^{-4}\ \text{IN}^4}$$

$$\sigma_{b,MAX} = 40228\ \text{psi}$$

Table E.9.1: Locking Pin Material Properties

4140 Steel Properties		
Density	0.284	lbf/in³
Tensile Strength	95000	lbf/in² (psi)
Yield Strength	60200	lbf/in² (psi)
Elastic Modulus	27557000	lbf/in² (psi)
Shear Modulus	11600000	lbf/in² (psi)
Poisson's Ratio	0.3	--
Static Loading	0	lbf
Dynamic Loading	100	lbf
Broach Angle	18	Degrees
	0.314159265	Radians
Safety Factor	2.00	
Pin Loading	61.80339887	lbf

Table E.9.2: Locking Pin Dimensions in Worst Case Scenario

Locking Pin Dimensions (Worst Case), 3/8" Shaft, 3.37" Length		
Diameter	0.375	in
Length	3.37	in
Volume	0.372205	in³
Mass	0.105706	lbf
Area Inertia Modeled as Round Cantilever Beam, I_X	0.000971	in^4

Table E.9.3: Locking Pin Bending Deflection and Stress Analysis

Bending Deflection and Stress (3/8" Shaft)		
Cantilever Static	0.0000000	in
Cantilever Dynamic	0.0294750	in
Differential Deflection	0.0294750	in
Bending Stress	40229.86388	lbf/in² (psi)

11.5.10 Appendix E.10: Analysis of Locking Mechanism 0.375" Hole/Shaft Fit

LOCKING MECHANISM HOLE/SHAFT CALCULATIONS:

- HOLE/SHAFT FOR LOCKING PIN INTO LOCKING PLATE:
  - HOLE WILL BE REAMED TO $0.3750 \, {}^{+0.0002}_{-0.0000}$ IN
  - USING ANSI STANDARD FITS FOR A LC11 CLASS FIT

	HOLE (H13)	SHAFT
MAX	+0.009	−0.007
MIN	0	−0.016

  - APPLIED TO OUR 0.3750" REAMED HOLE

	HOLE (H13)	SHAFT
MAX	0.384	0.368
MIN	0.375	0.359

→ HOLE: 0.3795 ± 0.0045 IN
SHAFT: 0.3635 ± 0.0045 IN

  - FIND NOMINAL CLEARANCE KNOWING HOLE IS 0.3750 IN.

$C = 0.3750 - 0.3635$ IN

$C = 0.0115$ IN

- HOLE/SHAFT FOR LOCKING PIN THROUGH 1/4" STEEL PLATE
  - HOLE WILL BE REAMED TO $0.500 \, {}^{+0.0002}_{-0.0000}$ IN
  - USING ANSI STANDARD FITS FOR A LC11 CLASS FIT

	HOLE (H13)	SHAFT
MAX	+0.010	−0.008
MIN	0	−0.018

  - APPLIED TO OUR 0.5000" REAMED HOLE

	HOLE (H13)	SHAFT
MAX	0.510	0.492
MIN	0.500	0.482

$C = 0.5000 - 0.4870$

$C = 0.0130$ IN

HOLE: 0.505 ± 0.005 IN
SHAFT: 0.487 ± 0.005 IN

11.5.11 Appendix E11: Analysis and Selection of Locking Mechanism Spring

LOCKING MECHANISM - SPRING ANALYSIS

KNOWNS:

- FULLY EXTENDED SPRING LENGTH = 1.75 IN
- SHAFT DIAMETER = 0.487 IN
- SPRING COMPRESSION DISTANCE = 1.10 IN

ASSUMPTIONS:

- SPRING FORCE ≅ 10-15 lbf

FIND:

- DESIRED SPRING CONSTANT, K
- SPEC. A SPRING FROM MCMASTER-CARR

SOLUTION:

- FIND A ROUGH DESIRED SPRING CONSTANT

$$\text{FORCE} = K \Delta X$$

$$10/15 \text{ lbf} = K(1.10 \text{ IN})$$

$$K \cong 9.09 - 13.63 \text{ lbf/IN} \quad \text{(DESIRED)}$$

- WE WANT A SLIGHT PRELOAD TO RETAIN THE PIN UP
→ SO, UNCOMPRESSED LENGTH ≥ 1.75 IN

$$\ell_{UNCOMP.} \geq 1.75 \text{ IN}$$

- SPRING FULLY COMPRESSED LENGTH NEEDS TO BE LESS THAN THE DIFFERENCE BETWEEN THE FULLY EXTENDED LENGTH AND THE COMPRESSION DISTANCE

$$\ell_{COMP} < 1.75 - 1.10 \text{ IN}$$

$$\ell_{COMP.} < 0.65 \text{ IN}$$

- SPRING INNER DIAMETER NEEDS TO BE BIGGER THAN 0.487 IN (SHAFT/PIN) AND OUTER DIAMETER SMALLER THAN 0.75 IN (LOCKING NUT DIAMETER)

$$\phi_{ID} > 0.487 \text{ IN}$$

$$\phi_{OD} < 0.75 \text{ IN}$$

FROM THESE REQUIREMENTS, A SPRING WAS SELECTED.

- MCMASTER-CARR PART #: 9657K375

SPEC. COMPARISON:

- $K = 13.00 \text{ lbf/in}$ ✓
- $\ell_{uncomp.} = 1.75 \text{ in} \geq 1.75 \text{ in}$ ✓
- $\phi_{ID} = 0.594 \text{ in} > 0.50 \text{ in}$ ✓
- $\phi_{OD} = 0.72 \text{ in} < 0.75 \text{ in}$ ✓
- $\ell_{comp.} = 0.54 \text{ in} < 0.65 \text{ in}$ ✓

NOW TO FIND THE ACTUAL FORCE REQUIRED:

$$F = K \Delta x$$

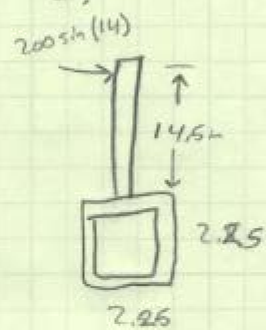
$$F = 13.00 \text{ lbf/in} (1.10 \text{ in})$$

$$\boxed{F \cong 14.3 \text{ lbf}}$$

11.5.12 *Appendix E.12: Analysis of Tool Holder Vertical Riser Beam Analysis*

DISPLACEMENT FROM TORSION
SHIGLEY's 10th ED.

TORQUE = 200lbf sin(14) 14.5 in
= 701 in-lb



$G = 11.5 \times 10^6$ psi

$A_m = 4.601 \text{ in}^2$

$L_m = 8.58$ in

$t = .105$ in

$h = 17$ in

$$\theta = \frac{h T L_m}{4 G A_m^2 t}$$

= .001 radians

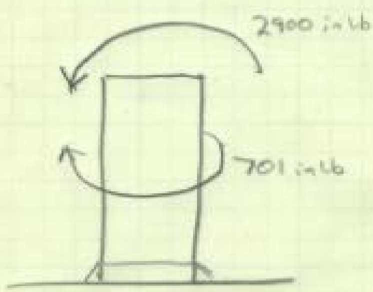
TIP DISPLACEMENT

$X_{TIP} = 14.5(.001)$

= .0145 in

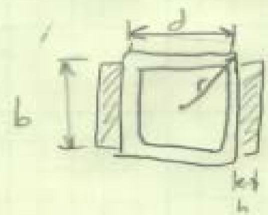
11.5.13 *Appendix E.13: Analysis of Tool Holder Vertical Rise Weld*

WELD STRESS, SHIGLEY'S 10th Ed.



PRIMARY SHEAR NEGLECTED FOR TORSION LOAD. $V = 200\sin(14)$

$V = 48$ lb



$b = 2.25$ in

$d = 2.25$ in

$h = .25$ in

$M_1 = 2900$ in lb

$M_2 = 701$ in lb

$J = .6711$ in^4

$r = \sqrt{2.25^2 + 2.25^2} = 3.182$ in

$$\tau_{bend} = \frac{1.414\,M}{b\,d\,h} = 3.23 \text{ ksi}$$

$$\tau_{TORSION} = \frac{M\,r}{J} = 1.6 \text{ ksi}$$

$$\tau_{TOTAL} = 4.9 \text{ ksi}$$

$$\tau_{allowable} = .30\,S_{ut}$$

$S_{ut} = 70$ ksi (E70 ELECTRODE)

$\tau_{ALLOW} = 21$ ksi

$4.9 \text{ ksi} < 21 \text{ ksi}$

WELD FATIGUE, SHIGLEY'S 10TH ED

ASSUME BASE IS 1018 HR

$S_{ut} = 58$ ksi $\quad S_y = 32$ ksi

E 70 ELECTRODE

$S_{ut} = 70$ ksi $\quad S_y = 57$ ksi $\quad$ TABLE 9-3

END OF PARALLEL FILLET WELD

$K_{fs} = 2.7$ $\quad$ TABLE 9-5

$K_a = 39.9\, S_{ut}^{-.995}$ $\quad$ AS FORGED SURFACE

$= .702$

THROAT AREA $\quad A = 1.414(.125)(2.25+2.25) =$

$K_b = .879\,\text{throat}^{-.107} = .891$

$K_c = .59$, TORSION

K_d = Temp = 1

K_e = reliability (90%) = .897

$K_f = 1$

$S_{se} = (.702)(.891)(.59)(1)(.897)(1)(.5)(58\text{ ksi})$

$S_{se} = 16.27$ ksi

$\tau_a' = 2.7(\tau_{tor}) = 26.5$ ksi

$\tau_a' > S_{se}$

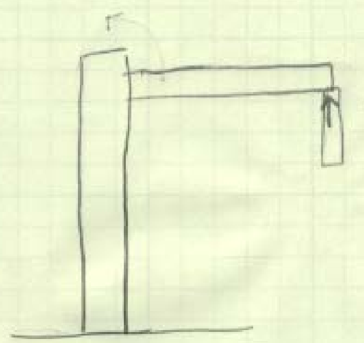
USE 1/4 FILLET WELDS, $\tau_a' = 13.23$ ksi

13.23 ksi < 16.27 ksi

11.5.14 Appendix E.14: Analysis of Tool Holder Vertical Riser Deflection

DEFLECTION OF UPRIGHT

SHIGLEY'S 10th ED.



ASSUME

CONSTANT 2.25" SECTION

(SAFE ASSUMPTION)

VARIABLES

SIDE LENGTH = 2.25"

WALL THICKNESS = .105"

HEIGHT =

$E = 30 \times 10^6$ psi

MOMENT ARM = 14.5"

TOOL LENGTH = 7"

$I = .08656\ in^4$

DEFLECTION FROM BENDING

$$y = \frac{ML^2}{2EI}$$

$= .16"$

$$\theta = \frac{ML}{EI}$$

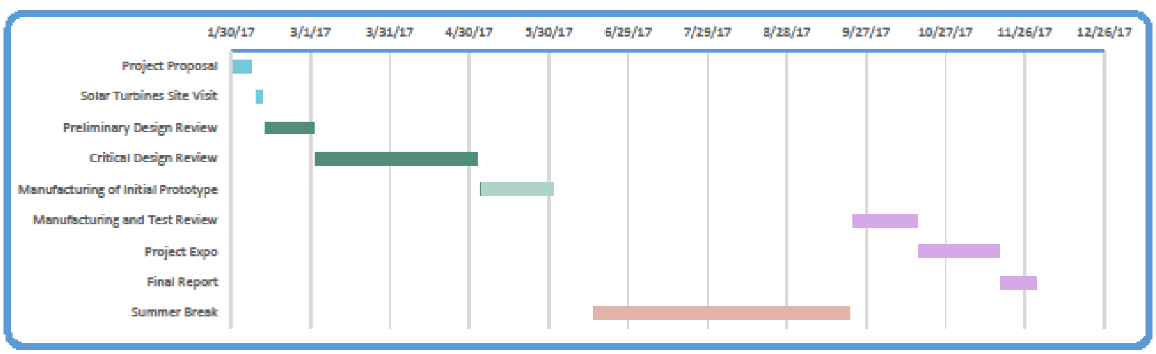
TIP DISPLACEMENT FROM BENDING

$$X_{TIP} = 7 \sin\left(\frac{ML}{EI}\right)$$

$= .0016"$

11.6 Appendix F: Gantt Chart

Today's Date	**5/3/2017**					
Task Name	**Start Date**	**End Date**	**Duration (Days)**	**Days Complete**	**Days Remaining**	**Percent Complete**
Project Proposal	1/30/2017	2/7/2017	8	8.00	0.00	100%
Solar Turbines Site Visit	2/9/2017	2/11/2017	2	2.00	0.00	100%
Preliminary Design Review	2/12/2017	3/2/2017	18	18.00	0.00	100%
Critical Design Review	3/3/2017	5/3/2017	61	61.00	0.00	100%
Manufacturing of Initial Prototype	5/4/2017	6/1/2017	28	1.00	27.00	4%
Manufacturing and Test Review	9/22/2017	10/16/2017	24	0.00	24.00	0%
Project Expo	10/17/2017	11/16/2017	30	0.00	30.00	0%
Final Report	11/17/2017	11/30/2017	13	0.00	13.00	0%
Summer Break	6/16/2017	9/21/2017	97	0.00	97.00	0%



11.7 Appendix G: Failure Mode and Effects Analysis (FMEA)

Item / Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s) / Mechanism(s) of Failure	Occurence	Recommended Action(s)
Mounted Hub Asembly	Disk is not secured	Disk falls off table	9	Bolts not tight enough	5	Stability calculation of table w/ disk. Train Technicians for proper setup procedures.
				Hub fails	4	
				Table leg breaks	2	
				Assembly tips over	1	
Rotating Hub Lock	Disk rotates during punching process	Disk is damaged	9	Weak locking mechanism	3	Ensure high safety factor on lock assembly. Train Technicians for proper alignment and maintenance. Use laser to help guide alignment.
		Blade Root is Damaged	7	User error during alignment	6	
		Punch tip is damaged	2			
	Disk does not rotate	Work cannot continue	2	Seizure/corrosion	2	
Slider Mechanism	Does not lock in place	Disk is damaged	9	Weak locking mechanism	3	Ensure user is aware of proper locking procedures. Use a robust locking mechanism with high safety factor. Develop a maintnance and inspection interval for slider lock.
		Blade Root is Damaged	7	User error during alignment	6	
		Punch tip is damaged	2			
	Unintentionally lock	Disk cannot be removed	2	Failure of sliders	3	
Hub Plate Retainer	Becomes Loose Relative to Disk	Disk Alignment Fails	7	Poor installation of plate retainer	4	Develop a torque spec/mark for proper installation tightness. Ensure a radial pattern of tightening is completed to enhance uniformity in clamping.
		Disk Falls From Hub Plate	8		1	
Tool Table	Falls or Collapses	Disk Damaged	9	Table support weight exceeded	6	Ensure a table is chosen with large weight-bearing safety factor. Keep high mass items low relative to the table surface. Bolt work table to the shop floor during use.
		Blades Damaged	7			
		Tooling Damaged	5	Center of gravity of equipment mounted too hight	3	
		User Injury	10			
Air System	Strikes operator hand	Injury	10	Improper training, lack of maintenance	6	Label clearly striking area. Require face shield during operation. Anti whip weights on air line.
	Flying debris					
	Line or valve burst					