ENGR 090 Senior Design Project

Design and Construction of a Small-Scale, Self-Replicating Machine Tool

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Abstract

A small-scale milling machine was designed, fabricated and constructed. This machine was intended to be "self-replicating," in the sense that a user equipped with an existing copy of the machine as well as basic hand and power tools could reproduce all parts used in the machine's construction not easily available from large commercial retailers (McMaster-Carr, MSC, etc.). This machine was designed to have a work volume of at least 6" x 6" x 6", be cost-competitive with existing small milling machines (approximately $500-$700), be capable of performing light milling in mild steel, and be eventually intended for CNC implementation.

Overall, the machine was only moderately successful. Serious chatter problems, believed to be caused by inadequate fastener tightening, made it impossible to quantitatively evaluate the machine's performance. Furthermore, the machine significantly exceeded its budget target, costing $1173.86 not including tooling costs or shipping costs. However, the machine does make significant steps towards the development of a usable self-replicating subtractive manufacturing device, and can serve as a valuable pedagogic tool for future Swarthmore students interested in precision machine design.

Preface

About the Author

Julian Leland is a senior at Swarthmore College, pursuing a B.S. in Engineering (mechanical concentration) with a minor in Public Policy. Within engineering, he is interested primarily in mechanical design, manufacturing, and design for the developing world.

Acknowledgements

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Introduction

In recent years, the development of small, affordable rapid fabrication tools such as 3D printers have made the concept of “desktop manufacturing” a buzzword, even outside of the hobbyist and maker communities. Increasingly, the low cost and relative simplicity of these machines is making them accessible to educators, small businesses and other users who previously would have never considered integrating a computer-controlled machine into their work. Additionally, the concept of self-reproduction – that the machine can be used to create all non-standard parts (e.g. parts not easily purchased from industrial suppliers) involved in its construction – is becoming more fundamental to this new breed of tools, with users taking advantage of the self-reproducing nature of some machines to increase the dissemination (and popularity) of these new tools by simply building more.

Currently, most rapid fabrication tools currently available fall into one of three classes:

1. Additive manufacturing tools, such as the MakerBot or RepRap 3D printers
2. Non-contact subtractive manufacturing tools, such as the PlasmaCAM CNC plasma cutter
3. Routers, such as the ShopBot Desktop, Zenbot, or Probotix Fireball V4.

One class of machine that has largely been overlooked is the traditional milling machine. These machines are subtractive manufacturing tools like routers, but are built much more heavily to allow the working of metals at a (reasonably) rapid rate while still maintaining a high degree of cut precision. Historically, milling machines have been large, heavy pieces of equipment – for example, the Bridgeports currently available in the Engineering Department machine shop. Small milling machines have always been available, and are currently produced by a number of manufacturers including Sherline, Taig, Harbor Freight and LittleMachineShop. However, these machines are uniformly expensive (upwards of $600), limited in their functionality, and often of variable quality. Finally, they are not capable of self-reproduction: most of the components in these machines are either too large for the machine to produce, or require advanced machining processes (grinding, scraping, heat treating) to be usable.

The goal of this E90 project was to work towards filling this gap, by designing and constructing a 3-axis machine tool, primarily intended for milling use, which would able to mill basic materials including mild steel, and was <u>capable of creating all non-standard parts involved in its construction</u>. The machine was intended to be small (≤ 250 lbs., work volume of roughly 6” x 6” x 6”), constructed exclusively from widely available components and self-produced components without requiring advanced machining processes (grinding, etc.) and comparable in cost to its competitors ($500-$700). These characteristics are developed further below, in Problem Definition.

Design

The design of this machine tool was divided into four discrete sections. First, the problem that this E90 attempts to address was defined. The machine's function and basic specifications were outlined, and basic performance goals were set. This data was then used to develop design parameters for the machine, including expected maximum cutting force and driving frequencies affecting the machine. Second, the machine's frame was designed. Primary concerns for this section of the design process included manufacturability, static and dynamic performance, and cost. Third, the machine's linear motion systems, including X and Y axis bearings and motion components, were specified and designed. Finally, the machine's spindle unit and motor were selected and implemented. Because of the necessary interactions between these systems, design of the three major sub-systems was pursued in parallel, rather than serially.

Problem Definition

Beginning in mid-November, a basic outline of the machine's function and specifications was developed. The criteria used were based off of Slocum's suggested design plan for a machine tool (Slocum 1992), although some modifications have been made.

Definition of Function and Specifications

First, a series of overall goals for the project were established. These goals were intended to serve as a "compass" for selecting between competing designs or priorities – for example, to select between designs that have greater cost versus greater accuracy. In order of importance (and inviolability), these goals were:

1. **Self-replicability**: the machine must be able to create all parts involved in its construction that are not easily commercially available. Additionally, the creation of the machine should not require the builder to invest significantly in tools that the average hobbyist could not be reasonably expected to have access to. For example, construction should not require a surface plate, or large precision calipers.
2. **Cost**: the machine's total cost should be kept low (around $500-$700, excluding CNC components)
3. **Precision**: the machine should exhibit accuracy and precision of positioning as described below.
4. **Work Volume**: the machine's total work volume should be no smaller than 6" x 6" x 6".

With these goals established, specific aspects of the machine's form and function were addressed.

- **Geometry and Frame Design:** The machine was required to be relatively small, weighing around 250 lbs and taking up a total volume of no more than 36" x 36" x 36". Its work volume was specified to be 6" x 6" x 6"; this was estimated by the author to be the maximum size of parts typically created in the Swarthmore machine shop. Because of this relatively small work volume, the machine was also required to allow limited-mobility machining of parts larger than 6" x 6" x 6" – for example, by allowing parts to protrude from the machine's frame during machining operations.

No specific frame geometry was specified for the machine. This was done primarily to permit the development of alternative frame designs. The traditional C-frame design used in most small milling machines is relatively compliant; all frame elements between the cutting tool and the workpiece are cantilevered, requiring comparatively massive frame elements for a given degree of stiffness. Since the limited work volume of this machine precludes the use of large frame components, the required frame stiffness would need to be achieved through a smaller, more enclosed frame design.

- **Machine Type and Kinematics:** The machine was intended for use as a vertical milling machine (as opposed to a turning center or grinding machine), using singly-supported tools. No specific translation system was specified for the machine, to allow non-Cartesian systems to be implemented. Instead, positioning accuracy tolerances were specified; these are detailed further below, in Determination of Performance Goals. The machine was required, however, to be able to perform 3-axis profiling operations.

- **Materials Selection:** The materials used in the construction of this machine were required to be, in order of importance: 1) inexpensive, 2) easily rough-machined (for example, sawed to size using hand tools) or otherwise easily worked, and 3) easily available, ideally from industrial distributors such as MSC or McMaster-Carr.

- **Production and Assembly:** In keeping with the requirement that reproduction of the machine be feasible for the average hobbyist, a number of constraints were placed on the machine's production and assembly. Allowed production methods were limited to what the machine itself could theoretically perform (high-precision milling and drilling), and what a competent user could produce using basic hand tools (hacksaws, power drills, files, etc.). Precision required in parts was required not to exceed that achievable by the machine: truly high-precision parts (linear ways, etc.) were to be sourced from major manufacturers, and no grinding or significant scraping was to be permitted. Total tooling costs were to be kept as low as possible, with a minimum number of distinct tooling setups used to create the entire machine. Finally, assembly was placed under the same restrictions as production, with no tools or processes that the average user would not be expected to have access to being allowed (for example, press-fitting or welding).

  Notably, no specific production time frame was stipulated for this machine. Because of the do-it-yourself nature of this project, it is assumed that the user has more time to spend than they do money. Consequently, labor-intensive production and assembly processes are preferred over more efficient but higher-cost alternatives, with the caveat that processes still should be accessible to the average user (e.g. no scraping).

- **Maintenance:** In keeping with the cost requirement outlined above, the machine's maintenance requirements were to be kept as low as possible. No components requiring frequent replacement (more than once per year, given average student use) were allowed for use, and components that would reduce periodic maintenance requirements were preferred. Finally, it was required that any components requiring maintenance or replacement be relatively easily available to the operator.

- **Cost:** The cost of the machine was stipulated to be between $500 and $700, excluding CNC components. A preliminary budget was prepared, giving target budgets for each system to allow this budget to be met: this budget may be seen in Appendix 1: Preliminary BOM/Budget.

Determination of Performance Goals

In addition to the basic definition of the machine's form and characteristics outlined above, a series of specific quantitative performance goals for the machine were also developed. Because CNC control of the machine was not stipulated as a final deliverable for the project, some of these goals were intended to be used purely for design purposes, and were not actually intended to be measured at the conclusion of the project.

- **Maximum Cutting Force Determination:** One of the primary metrics needed before design could begin in earnest was the maximum cutting force that the machine would reasonably be subject to. A spreadsheet intended for calculating cutting force and machine power requirements was developed, using a synthesis of similar derivations from a variety of sources. The spreadsheet was then used to calculate the maximum expected tangential cutting force at the tooltip and required machining power for a variety of materials and cut parameters.

  The maximum expected cutting force was found to be 138 lbf (614 N). This was found for a .2" x .375" cut at 2400 RPM and a feed of .001" per tooth, using a .375" end mill, in AISI 1018 CR steel. Because this is a fairly heavy cut for most machining operations, it was assumed that this represented the heaviest use the machine was likely to see, and would provide a good basis for design. A factor of safety of 1.5 was applied, bringing the maximum cutting force to 200 lbf (890 N), which was used throughout the design process as the maximum expected load.[1]

  A full derivation of the equations used in the spreadsheet may be found in

[1] It should be noted that commercial cutting force calculators indicate that this calculation severely overestimates the cutting force. Notably, the cutting force calculator provided online by the Kennametal Corporation indicates that the expected cutting force produced by the cut above is only 46 lbf (205 N) (Kennametal Corporation 2012).

Appendix 2: Machining Force and Power Derivation. A sample of the spreadsheet may be seen in Appendix 3: Machining Force and Power Calculation Worksheet, and the report pages from the different tests conducted to determine the maximum cutting force may be seen in Appendix 4: Material/Cut Combination Test Results.

- **Maximum and Minimum Translation Rates:** The minimum translation rate required from the machine was found during the calculation of the maximum cutting force; based off of speed and feed tables in Machinery's Handbook, it was found to be 5.19 in/min, or .086 in/sec. This rate was stipulated for design purposes rather than as an actual performance parameter, to ensure that the machine was able to execute smooth translations at this rate without jerking or "cogging" when CNC control is eventually implemented. Similarly, a maximum translation rate of 3 in/sec was also stipulated as a design parameter, with the intent that the machine should be able to fully transverse its work volume in under 2 seconds.

- **Axis Positioning Accuracy:** Axis positioning accuracy was specified extremely generally, so as to permit non-Cartesian translation systems to be used. Specifications were that the machine should be able to position a part to within ± .0005" anywhere within its work volume, in any cardinal direction; that the repeatability of positioning should be ± .0005"; and that the resolution of positioning should be at least .0005". Like the maximum and minimum translation rates, these specifications were primarily intended to be used as design goals rather than as actual measurable performance goals.

- **Cutting Accuracy:** The machine was specified to be able to perform a .125" x .1" full-width cut in mild steel without experiencing total error motion greater than .001". This cut was expected to produce a maximum cutting force of 86 lbf (383 N) by the cutting force spreadsheet developed above, although other cutting force calculators have predicted a significantly lower value. Unlike the positioning accuracy and translation rate specifications, this metric was intended both as a design parameter and a measurable performance goal, and will be measured at the conclusion of the project.

- **Spindle Capacity:** The maximum cutting force calculations developed above indicated a maximum required spindle power of 1.2 HP (895 W); other force and power calculators have indicated a much lower maximum power requirement of .44 HP (328 W)

Frame Design

With the basic form and specifications of the machine established, the design of the frame was addressed. As mentioned earlier, the design of the frame presents a particularly unique challenge for this project. Most commercial machine tools are comprised of frame components that are significantly larger than the machine, making self-replication for these tools infeasible. Additionally, much of the stiffness of commercial machine tools comes from the massiveness of those frame members, meaning that any frame design used would need to make up for that loss of stiffness through other means. Consequently, in addition to meeting the cost and material availability requirements stated above, the frame of this machine would need to be:

1. sufficiently modular that all precision machining required for frame components could be completed by the machine;
2. sufficiently compliant in assembly that any frame elements which could not be precision machined could be machined using hand tools and then placed in precise alignment during assembly.

Preliminary Frame Design

Preliminary frame design focused primarily on maximizing the stiffness of the frame. As mentioned above, most commercial machines use a C-frame design, like that used on the traditional "Bridgeport-style" vertical mill shown in Figure 0. This design maximizes operator accessibility at the expense of rigidity, since all frame components between the cutting tool and the workpiece are cantilevered. Because the machine developed by this project is intended for eventual use as a CNC machining center, the accessibility requirements of the machine were reduced, creating an opportunity for increasing the stiffness of the frame through the use of closed frame designs.

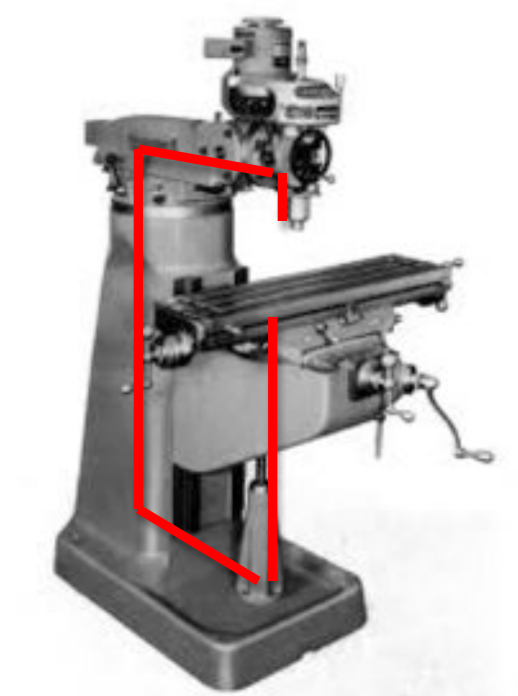

Figure 0 - C-Frame Milling Machine

A series of "mockup" frame designs were created in Dassault Systemes' Solidworks. To allow comparison of the innate stiffness characteristics of the different frame geometries, these frame designs all used a common frame member profile, defined arbitrarily as a 1" x 1" solid bar made of AISI 1020 CR steel. All frame designs were developed to allow a 12" x 12" x 12" work volume to fit within the frame of the machine, and a common spindle unit was used with all frames. Some frames were based off of existing frame designs, including the Lindsey Tetraform machine tool and the Ingersoll Hexapod concept tool developed at the National Institute of Standards and Technology; others were simply the author's creation. The frames were created from separate parts and then merged to create homogenous bodies, discounting the effect of joints on the ultimate stiffness of the frame

Each frame was simulated in Solidworks Simulation. Frames were defined as fixed at each base corner, and the spindle unit and tool were defined to be rigid. A 100-N load was applied to the tooltip; multiple tests were conducted with the load placed in different orientations, to determine whether any weak or strong axes existed within the frames. Images of the frames tested, as well as the results of those tests, may be seen in Appendix 5: Preliminary Frame Testing Results.

For each frame, the quantity 1/(∂V) was calculated; this quantity is the stiffness per unit volume of frame material, and was selected as an evaluation criteria to allow simultaneous optimization of static performance and cost. The following values were determined (higher values are better):

Frame Design	1/(∂V)
Trigonal Pyramidal	1.9382
Square Pyramidal	2.3744
Square (Corner Support)	1.2338
Square (Edge Support)	1.5851
Double Tetrahedral	15.5558

Table 1 - Stiffness/Unit Volume

As can be seen the above table, the double tetrahedral frame design – shown below in Figure 1 – performed significantly better than the other frame designs. Combined with the comparative ease of manufacturing of this frame design, it was selected for further development in the specific frame design phase.

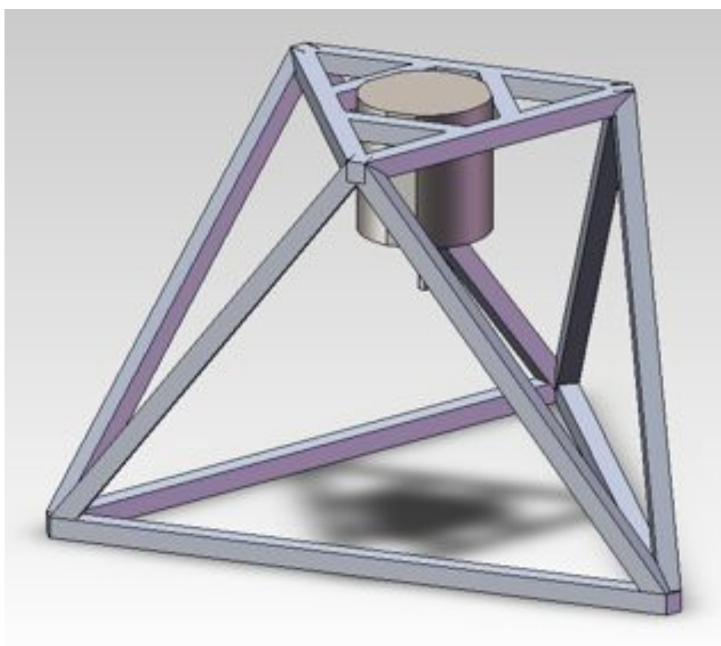

Figure 1 - Double Tetrahedral Frame Design

Specific Frame Design

With a general frame design selected, the frame was further developed.

Materials Selection

The first step in specific frame design was to select a base construction material for the frame. As discussed earlier, the primary criteria used for selecting frame materials were that they be 1) inexpensive, 2) easily rough-machined or otherwise easily worked, and 3) easily available. Given these criteria, two primary materials were initially investigated: hollow steel section framing, and aluminum extrusion framing (commonly sold under the 80/20 brand name).

Advantages of the aluminum extrusion included its lighter weight, increased ease of machining, and the higher dimensional precision of the extrusions. Additionally, the extrusion is designed to allow easy assembly and fixturing of parts, which would simplify the assembly process; the steel frame requires bolted connections, which are challenging to design and harder to assemble. However, the aluminum extrusions are not as stiff as the steel framing, and the natural damping properties of aluminum are lower than that of steel. Finally, aluminum extrusion is typically much more expensive than steel sections, and preliminary cost estimates showed that the steel frame would be significantly cheaper and easier to procure.

To choose between the materials, simplified Solidworks models of both the aluminum extrusion frame and the steel section frame were developed (Figure 2). The aluminum frame was simplified to use square beams with the same second moment of inertia as the aluminum extrusions; the steel frame used defined bolted connection joints, rather than incorporating actual bolted connections. In both frames, a mockup spindle was used; it was defined as rigid for the purposes of simulation. A series of simulations, including static deflection, dynamic behavior and thermal expansion, were conducted on each frame design.

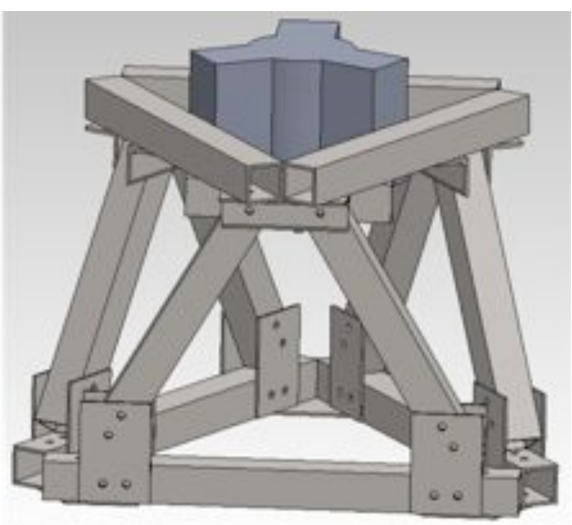

Figure 2 - Simplified Simulation Frames

Static Performance

A simulated load of 67 lbf (300 N) was applied at the tooltip. In both frames, areas at the corners of the bases of the frames were defined as fixed. Under these loading conditions, the aluminum frame experienced a maximum deflection of .0012”, while the steel frame experienced a maximum deflection of .0004”. As a check, a second test was run with the steel frame defined as universally bonded as opposed to having bolted connections. For this test, the maximum deflection experienced was .00006”. Because of the author’s unfamiliarity with the use of bolted connections in Solidworks, it is currently unclear which more accurately represents the machine’s performance. However, since the aluminum frame had been defined as universally bonded in the original test, it was accepted that the steel frame was demonstrating greater static performance.

Dynamic Performance

The dynamic performance of the frame designs was explored in two ways. First, the expected natural frequency of each frame was calculated using the simple approximation:

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

where k is stiffness and m is mass. The expected natural frequency for the steel frame was approximately 90 Hz, while the expected frequency for the aluminum frame was approximately 140 Hz. This was also checked in Solidworks, using a simple fundamental frequency simulation. The simulation reported higher natural frequencies, reporting a frequency of 370 Hz for the steel frame.

Unfortunately, neither of these natural frequencies are ideal for machine tools. Both lead to “danger speeds” – rotational speeds that are liable to cause excitation at the machine’s natural frequency – which are within the range of speeds that the machine could reasonably be expected to reach. Additionally, the disagreement between the on-paper approximation of the natural frequency and the FEA results brought into question the validity of both methods. Consequently, the dynamic performance data was discarded; the steel frame was chosen to have superior dynamic performance as a function of its weight and the greater natural damping characteristics of the steel.

Thermal Performance

Finally, the thermal performance of the designs was analyzed, using a Solidworks ambient temperature simulation. The steel frame again performed significantly better, deforming both at a slower rate and to a lesser degree than the aluminum frame. Interestingly, in both simulations, the majority of the thermal expansion occurred in the Z direction: the symmetry of the double tetrahedral frame design makes it relatively resilient to thermal deformation.

In light of these results, the steel frame design was deemed superior, and was selected for use.

Linear Motion System Design

With the design of the frame completed, the design of the linear motion systems for the X, Y and Z axes was turned to. Early on in the project, alternative linear motion systems incorporating novel mixed linear and rotary motion systems were investigated. However, it quickly became clear that the additional complexity required by these systems was not justified by the performance gains they would yield, and simple stacked linear motion systems were selected.

Determination of Performance Requirements

The first step in developing the linear motion systems was to determine the maximum performance that would be required of those systems. A spreadsheet was developed which approximates the reaction forces and moments on bearings produced by a load anywhere in 3-space, for 4-, 3- and 2-bearing carriages; this spreadsheet may be seen in Appendix 6: Bearing Load and Moment Calculation Worksheet. This spreadsheet is based on a derivation originally created by Slocum (Slocum 1992), which develops an approximation for the loads experienced by the bearings in a 4-bearing carriage; this derivation was expanded to the 3- and 2-bearing cases by the author. All three derivations are fully presented in Appendix 7: Bearing Load and Moment Calculation Derivations

Using this spreadsheet, bearing loads were calculated for 4-, 3-, and 2-bearing stages, for 200 lbf loads applied in the X, Y and Z directions. In all trials, the point of application of the force was defined to be at the center of the work volume, while the stage was defined at the location that would produce the greatest stresses on the bearings (typically, at the extreme end of its travel). From these tests, the 3-bearing stage configuration was determined to be the most cost-effective, yielding only slightly higher loads than a 4-bearing configuration and avoiding the high moments produced in a 2-bearing configuration, while still reducing the cost of the system significantly. Design loads were found to be ± 250 lbf in the X and Y directions, and ± 600 lbf in the Z direction.

Selection of Bearing System

With the maximum expected bearing loads determined, a bearing system was selected. Early on, it was decided that because of time constraints and manufacturing limitations, linear motion systems for the X and Y axes would be purchased off-the-shelf, rather than fabricated. A variety of bearing systems were investigated, including wheel-and-rail systems, sliding bearings, ball bearing sleeves with linear shafting, and recirculating ball-bearing block systems. No standalone linear motion system for the Z axis was purchased: this was addressed elsewhere, as explained below.

Based off of literature reviews and discussions with bearing system manufacturers, a recirculating ball-bearing block system was selected. These systems are the gold standard among both amateur and professional machine tool builders: they are extremely stiff, and impart very little running resistance. Additionally, they are capable of handling impact loads, and will continue to slide smoothly under intermittent loading – an important characteristic for machine tool bearings. Unfortunately, these bearings are also extremely expensive: purchasing the 6 bearing blocks required for this machine from a commercial outlet (for example, McMaster-Carr) would cost nearly $800

6 AGW-15CB bearings were purchased from Automation Overstock, an online automation component reseller. Because these bearings are legacy products, they were available for a fraction of their usual cost: all 6 bearing blocks, plus the accompanying rails, were purchased for $206.50.

Selection and Design of Linear Actuation System

In addition to the linear bearings, a linear actuation system was specified and designed. The primary driving factor in the design of the linear actuation system was the fact that the machine is eventually intended for CNC control, but would initially be manually operated. Consequently, the linear actuation system would need to be both usable by a human operator, but also compatible with eventual CNC control.

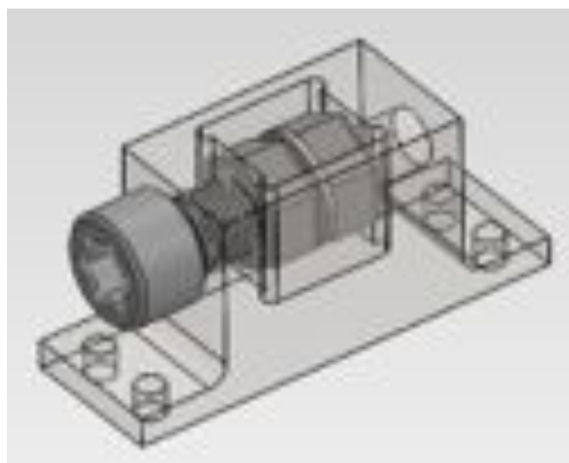

Figure 3 - Backlash-Reducing Nut

A system based around the 3/8" – 10 Acme screw thread standard was selected. Acme screws, which have been used for decades as power- and motion-transmission screws, are easily available, and a fraction of the cost of ball screw systems. The 10 TPI formfactor also allows for easy control by a human operator: one turn of the screw corresponds to a total motion of .1", which is a convenient measurement. The leadscrews were singly supported by two angular-contact ball bearings in a DF configuration, which allows for slight angular misalignment of the screws while still providing good axial stiffness. Additionally, a backlash-reducing nut system was designed, which uses a spring washer to provide preload between two Acme nuts (Figure 3). This system facilitates eventual CNC implementation: by eliminating backlash from the leadscrew system, it increases the machine's ability to determine its position by measuring the rotation of the screw rather than by measuring the actual motion of the stage. Additionally, it also allows the use of less expensive "standard-grade" Acme hardware, rather than the more costly "precision-grade" hardware. Both the precision and standard grade hardware have the same lead accuracy (roughly ±.004"-.006" per foot), and differ only in terms of the quality of their thread fit. Because the backlash-reducing nut system eliminates any looseness in the thread fit, the lower-grade hardware may be used without detriment.

Spindle & Power Systems Design

The last major design task addressed was the selection of the spindle and drive motor.

Spindle Unit Selection

Early on in the project, a variety of options for spindle units were investigated, including off-the-shelf commercial spindle units, repurposed spindles from other applications such as jewelry manufacture or wood routing, and custom-built spindles. Because of the high loads placed on spindles, and the tight tolerances involved in their manufacture, repurposed and custom-built spindles were shelved, and off-the-shelf spindles were investigated.

Ultimately, a spindle unit sold by LittleMachineShop.com was selected. This spindle unit incorporates both an R8 spindle and a 350 W DC motor with controller, which allows continuously variable speed between 0 and 2500 RPM. It can accept up to a 1/2” drill or a 5/8” endmill. Additionally, this unit is also sold with a matching dovetail column, which incorporates a rack and pinion drive. This column was also purchased, and was used to provide Z-axis motion.

Construction

The construction of this machine comprised three separate phases: specific component design, fabrication, and assembly.

Design

Specific components used in the frame, linear motion systems and spindle unit were designed and modeled in Solidworks. A total of 39 unique components were designed in the course of this project; these components can be seen, along with assembly drawings, in Appendix 8: Machine Part & Assembly Drawings.

Although there are too many parts to describe the design process of each part individually, a number of common techniques and processes were used in designing all of the parts involved in this project:

- **Use of Slotted Holes**: As discussed earlier, one of the greatest challenges faced in this project was the issue of ensuring stress-free, precise assembly without requiring precision machining beyond the capacity of the machine. One of the primary ways that this was addressed was through the use of slotted connections in the frame and in certain parts of the linear motion assemblies. Slotted holes were designed in accordance with the AISC specification for long-slotted holes (AISC 2010). As dictated by the specification, washers were used to cover slotted holes, except in the case of rail attachment blocks, where the use of washers would be impractical.

- **Oversized Holes in Bolted Connections**: In cases where strain-tightened bolted connections were used, expected imprecision in fabrication could be addressed simply through oversizing of the holes. Oversized holes may exceed the diameter of the bolt by 1/8” for a 1/2” bolt, which gives more than sufficient flexibility for most assembly situations.

- **Design for Adjustment**: Many of the parts of this machine were deliberately designed to allow adjustment in specific directions, in order to accommodate imperfections in fabrication. A good example of this is the design of the main frame members. Because all frame members are much larger than the work envelope of the machine, many of their features are dimensioned from opposite ends of the member. Since it is assumed that the length of the member can only be accurately determined to .01”, the primary direction that dimensional errors are expected to occur in is along this axis. In order to accommodate this, all parts of the frame are designed to accommodate this. The frame is designed to be able to dilate and contract, through the use of combined standard holes and slotted holes.

- **Common Parts and Stock Sizes**: Whenever possible, parts were designed to be used in multiple locations, to reduce the total number of unique parts and simplify the fabrication process. Additionally, a concerted effort was made to keep the total number of stock material sizes used as small as possible.

- **Parts Requiring High-Precision Manufacturing:** The following parts are all considered to be high-precision parts. Especial care was taken during their manufacturing to ensure that they were produced to the precise specifications detailed in the part drawings:
  - TopShortBearingRailClamp V2 – Left – FinalDesign
  - TopShortBearingRailClamp V2 – FinalDesign
  - ShortBearingRailClampV2 – FinalDesign
  - BearingBlockMountPlate – Final Design

Fabrication

Fabrication was conducted over a period of two months, from March to April 2012. Fabrication took place both in the Swarthmore Engineering machine shop, as well as at the home of the author. Although careful records of fabrication time were not kept, it is estimated that fabrication took between 200 and 250 hours.

Tools used during fabrication included:

- milling machines (both Bridgeports and a small Benchmaster mill)
- bandsaws (horizontal and vertical)
- belt sander
- wire wheel
- bench grinder
- lathe (for the lead nut push screws. This operation could have easily be performed on a milling machine, or even in a standard drill press.)

In the interests of efficiency, the primary fabrication constraint placed on the assembly process – that the work volume used for precision machining operations be no greater than 6” x 6” x 6” – was ignored, with the extended travel of the machine tools during fabrication being taken full advantage of. Additionally, the capacity of the tools used was significantly higher than that of the tools that would presumably be available to the average user: full-sized milling machines, bandsaws and sanders were used.

A relatively small set of tooling was required to produce the parts. Aside from standard endmills and drills, the following tools were required:

- M4 tap set and matching tap drill (3.3 mm)
- M3 tap set and matching tap drill (2.5 mm)
- 10-32 tap set
- ¼-28 tap set
- 3/8-24 tap set

- ½-20 tap set
- 15/64th endmill
- Edge finder

Assembly

Assembly was completed relatively quickly. Asides from the basic hand tools (box wrenches, socket wrenches, hex wrenches, hammers) used during assembly, precision tools used included:

- precision level. A high-quality machinist's level was used, which is capable of measuring differences of .0005" difference over 12", or .0024 degrees.
- machinist's square.
- 6" dial calipers
- shim stock. A package of shim stock incorporating sizes between .001" and .05" was used both for shimming and for measuring gaps.

The full assembly process was documented as it was conducted: notes from this process may be seen in Appendix 9: Assembly Notes. While the machine was satisfactorily assembled, there is still significant room for improvement in the assembly process: future builders should treat the attached notes as rough guidelines at best.

Assembly Techniques

A number of specific techniques were discovered during the assembly process, which greatly simplified the various assembly tasks. Some of these techniques are listed here, to facilitate the efforts of future builders.

- **Use of bearing block mount plate as reference for X and Y axes:** The bearing block mount plate, which is among the highest-precision parts in the entire assembly, was used frequently as a reference for positioning the other linear motion components. This is a highly recommended technique, as the bearing block mount plate is also a relatively non-compliant component, and is also critical to the proper functioning of the translation system. Future assembly efforts will use the plate as the sole datum for the installation of the rest of the linear motion systems.

- **Levering at base of side triangles:** Although the machine's frame is theoretically designed to be sufficiently compliant to allow for errors in manufacturing, binding between fasteners and parts of the machine makes it difficult to shift frame components. Beyond simply forcing parts (a good hammer is critical for assembling this machine), one particularly effective technique used when leveling the top frame of the machine was to place a steel bar in the gap between the base frame members and the side frame members, and lever upwards with the bar.

- **Tightening inner bolts first:** Although there is theoretically sufficient clearance to allow a wrench to access both bolts when working with a double-bolted connection (for example, the connections between the base and side members), it is much easier to install the inner bolt first and tighten it snugly to hold frame members in place. Then, the outer bolt can be installed to further secure the joint, and both bolts can be strain-tightened simultaneously.

- **Use of long screws when installing into captive nuts:** Many of the connections involved in the linear bearing systems involve captive nuts that are threaded into the frame members before assembly. The proper method for installing these connections is as follows:
    a. Before assembly, insert all captive nuts into frame members. Thread bolts into nuts to hold in place.
    b. Before beginning assembly, procure screws in all needed sizes (1/4-28 and 3/8-24) that are significantly longer than the screws that will actually be used during installation.
    c. Use these longer screws to facilitate attachment of parts to the captive nuts without allowing the captive nuts to fall into the frame members. The frame rapidly becomes extremely heavy as it is assembled, and re-aligning the nuts will prove significantly more difficult.
- **Use of top table as surface plate:** The top table is specified as a Blanchard-ground plate. Although its surface finish is fairly rough, it is a relatively flat surface, and may be used as a stand-in for a surface plate if needed.

Performance Evaluation

Because of the late date of the machine's final assembly, and because of further alignment work that was required as a condition of the IEEE grant used in this project, a full quantitative evaluation of the machine's characteristics and performance was not conducted. However, the machine was tested briefly on a small set of materials: the results are described below.

Cutting Performance

Three cutting tests were performed with the machine. In the first test, a piece of 304 stainless sheet measuring .125" thick was placed in the machine's vise. A small hole was center-drilled, followed by a 25/64" hole. The machine exhibited some vibration during this test, but satisfactorily drilled the hole.

In the second cutting test, a 3/8" endmill was used to square a chunk of osage orange wood, a particularly hard and fibrous wood. The machine exhibited little vibration during this process, even during relatively heavy cuts (.375" x .2"). Additionally, the surface finish produced by this cutting process indicates that the table and spindle unit are relatively square, although this has not been confirmed through direct measurement.

In the third and final cutting test, the 3/8" endmill was used in face- and end-milling operations on a short section of 6061-T6 aluminum bar. At very light cut levels when end milling, the machine's performance was satisfactory, leaving a relatively fine surface finish. However, when the cut size was increased, the machine began to chatter violently. The machine was observed to exhibit a particularly unusual chattering pattern, wherein it would cut without trouble momentarily, and then would abruptly "jump", as though the cutting tool had been deflected sideways. This chatter significantly impacted the surface finish of the part, as can be seen in Appendix 10: Machining Test Results.

Unfortunately, at this point it was observed that some of the machine's screws which had not been fully tightened were being loosened dangerously by the chattering: consequently, further testing was abandoned. However, although the degree of the machine's chattering is extremely high, it is not believed to be due to inherent design flaws in the machine. Rather, because of the magnitude of the chatter, it is believed that a critical connection was not adequately tightened, which led to the "jump"-type chatter described above. This caused already-loose bolts and screws to further loosen themselves, which contributed further to the chatter. Further tests will need to be conducted after the final leveling is completed to determine whether this is the case.

Machine Characteristics

Beyond simple cutting performance, the machine's performance can be analyzed through a number of other metrics.

- **Cost:** Not including tooling costs or shipping and handling costs, the total cost of raw materials used in the machine is \$1173.86. Shipping costs totaled to \$201.98, and tooling costs totaled to $193.80. It is hard to say whether the cost of raw materials quoted here is actually representative of the true cost to a future builder: although many items were purchased through industrial suppliers like MSC and McMaster-Carr, and could be purchased at a much lower cost elsewhere, other parts were purchased with tax-exempt status or educational discounts, which may not be available to future builders.

- **Self-Replication:** It is currently unclear if the machine is capable of self-replication. In the current design of the machine, there are only two parts – the top table, and the bearing block mount plate – which would be particularly difficult, although not impossible, for the machine to reproduce. However, since the machine has yet to demonstrate its ability to mill mild steel with any degree of accuracy, it cannot be said whether or not it would be capable of performing this task.

- **Work Volume:** Because of the use of the off-the-shelf Z-axis column, the work volume turned out to be significantly larger than expected in the Z direction. The final work volume size is approximately 5.35" by 4.85" by 14"**.**

- **Usability:** Unfortunately, in its current state, the machine is difficult and even dangerous to use. This is largely due to the awkward positioning of the leadscrew drive handles within the work frame of the machine. Obviously, for a CNC-equipped machine this would not be an issue; however, if manual operation is desired, an alternative system needs to be found.

Conclusion

The machine was presented to the Swarthmore Engineering Department on May 2nd, 2012. Slides from the presentation may be seen in Appendix 11: Final Presentation Slides.

Overall, this E90 is regarded as marginally successful. The machine's performance was not able to be quantitatively analyzed, and the few qualitative tests that were run indicated significant issues with the machine's rigidity: it is currently being determined whether these are due to errors in the machine's assembly, or whether they are due to more profound issues with the design of the machine. Because of these limitations, it cannot be conclusively determined whether or not the machine would actually be capable of self-replication. Finally, the machine does not quite match its original work envelope goals (although it does exceed them in the Z direction), and significantly misses its cost goal.

However, despite these problems, this E90 has still proved to be a worthwhile project. In addition to being a good exercise in mechanical design, design for manufacture and creative addressing of constraints, this project has also made significant steps towards the development of a self-replicating milling machine – a true subtractive machining analogue to the RepRap.

Although this iteration of the machine has yet to demonstrate its usability, it is hoped that by releasing the part drawings and other associated documentation to the public through a Creative Commons Attribution-Share Alike license, other users may be able to contribute to the development of this machine and help it attain its full potential

Avenues for improvement of the machine's design have been divided into two categories: near-term improvements, which could conceivably be carried out by future Swarthmore students on this iteration of the machine, and long-term improvements and design changes, which should be carried out by other builders.

Near-Term Improvements

- **Develop better fastener-tightening protocol:** Bolts and screws loosening under cutting conditions is an extremely dangerous phenomenon, and one which appears to be a serious concern with this machine. A better method of ensuring fasteners do not loosen under impact loading is seriously needed. Possible options include use of Loctite or other thread-locking adhesives, strain-tightening of all fasteners, or redesign of connections.

- **Improve Z-axis:** Currently, the machine's Z- axis is not counterbalanced. Under cutting conditions, the pinion gear engagement system may loosen, allowing the pinion gear to spin freely, and possibly leading to the spindle unit slowly creeping down the Z-axis, or potentially falling if the Z-axis gib is loosened too quickly. Some type of counterweight or other support mechanism for the Z-axis should be designed and implemented to prevent this.

- **Realign X-Y Carriage:** The X-Y carriage needs to be realigned to reduce binding. When this is done, careful notes should be taken on the alignment process used, to help improve the current process.

- **Redesign lead nut assembly:** The lead nut assemblies used are unfortunately currently prone to binding when adequately tightened. In the short term, the steel lead nuts used should be replaced with bronze nuts to reduce binding, and the lead nut housings should be re-machined to provide a tighter fit around the nuts. In the longer term, the system should be redesigned to ensure that the backlash removal system is working as intended.

- **Rebuild spindle unit:** The spindle unit was purchased at a discount because it had been previously dropped; additionally, it was dropped one further time by the author. The unit almost certainly needs its bearings replaced, and may require a full rebuild.

Long Term Improvements/Design Changes

- **Replace internally terminating bolts with through-bolts:** Future designs should replace as many internally terminating bolts with through-bolts, like those used on the side frame members, which are both easier to assemble and tension.

- **Redesign to facilitate assembly:** The assembly process is relatively straightforward, except for the alignment process. Potentially, this process could be simplified by redesigning some components of the linear motion systems to use a smaller number of independent parts, and instead using single parts which can assure alignment through machining rather than through complex alignment processes. Other options include implementing more adjusting hardware – for instance, push screws.

- **Redesign to facilitate easier self-replication**: As mentioned above, there are some components currently used in the machine that could not be trivially reproduced given the machine's constraints; these parts should be redesigned to allow the machine to reproduce them. Additionally, the complexity of the current design means that the fabrication process is extremely lengthy. Future designs of this machine could potentially incorporate a far simpler frame, such as a heavy gantry-type frame, to reduce machining time and component count. Additionally, other assemblies such as the bearing rail mount blocks could be simplified to reduce part counts.

- **Redesign to reduce cost:** Currently, the machine is not cost-competitive with existing alternatives. This is partially due to economic constraints: milling machines are a well-understood technology, and can be manufactured easily when industrial manufacturing equipment is available (specifically, rapid casting, shaping and grinding facilities). However, it is also partially due to the use of expensive off-the-shelf components for some of the machine's parts. By designing the spindle unit and linear motion systems, rather than simply purchasing them off-the-shelf, the cost of this machine could potentially be reduced significantly. Additionally, this would be more in keeping with the design philosophy of the machine. An off-the-shelf spindle unit is a highly specialized part which is only available from some manufacturers; ideally, this would be a part that the machine could reproduce itself.

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Appendices

Appendix 1: Preliminary BOM/Budget

The following BOM/budget was prepared as part of the E90 Project Proposal delivered in November 2011. It outlines the estimated costs of the different machine subsystems.

Item Name	Predicted Cost	Comments
Spindle	$50 - $150	The spindle is one of the few machine-tool specific components that will need to be purchased rather than built, due to the grinding required to finish the inside of the spindle taper. Commercial spindle units from Sherline, Taig and Foredom are being investigated: additionally, the possibility of purchasing just the spindle taper, and building the bearing housing and spindle mounting, is being investigated.
Framing Material	$50 - $100	Framing materials being investigated include aluminum extrusions such as 80/20, engineered steel sections such as Unistrut, and common steel rod profiles. Cost, stiffness, and damping ability are primary characteristics of interest.
Motor and Drive Components	$75 - $150	The power requirements of this machine have yet to be determined. Potential sources for motors include commercially available motors intended for powering machine tools, and surplus motors that are easily procured (for example, washing machine motors).
Translation Components	$50 - $150	Linear translation is likely to be accomplished using lead screws or ball screws, due to accuracy and "holding power" requirements.
Linear Bearings	$100 - $200	Linear bearing systems will primarily be evaluated on the basis of cost and stiffness. Possible candidates include commercially-available pre-built linear ways, and rail-and-bushing systems.
Raw Material	$50 - $150	Raw metal stock for this project will likely prove to be a fairly significant portion of the total budget, since many of the components machined will have large external profiles. Care should be taken to ensure that all parts are designed to minimize stock size requirements (and thus decrease the total cost of stock).
Total	$375 - $900	

Appendix 2: Machining Force and Power Derivation

The cutting force multiplier derived here is used in the machining force and power calculation spreadsheet to calculate the expected cutting force.

It is assumed that the user has access to the following data (some data available from Machinery's Handbook):

- K_P: the power constant of the material being cut (Machinery's Handbook)
- The diameter and number of teeth of the milling cutter being used
- r: the desired radial width of the cut
- d: the desired axial depth of the cut
- f_t: the specified feed per tooth for the desired cut parameters (Machinery's Handbook)
- V: the specified cutting speed for the desired cut parameters (Machinery's Handbook)
- W: the tool wear factor specified for the desired cut parameters (Machinery's Handbook)
- E: the efficiency of power transmission for the given machine

The cutting power required to perform the specified cut using a single-pointed tool is defined by Machinery's Handbook (Industrial Press 2008, 1056) to be

$$P_C = K_P C Q W$$

where P_C is cutting power, C is the feed factor, and Q is the material removal rate.

From this, cutting force is calculated as:

$$F_C = \frac{P_C \cdot 550}{(V/60)}$$

assuming that Pc is in HP, and V is in ft/min.

Although this derivation is accurate for a single pointed tool, it does not accurately represent the force situation that exists when a cutting tool with multiple cutting blades is used.

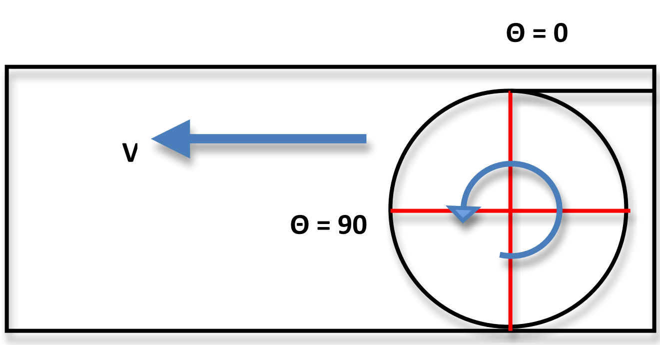

Figure 4 - Cutting Diagram

It is assumed that for a multiple-bladed tool, the cutting force on a single blade varies as a function of the angular position of the tool, from 0 at Θ = 0º to F_C at Θ = 90º, and then back to 0 at Θ = 180º. At any point, this cutting force can be decomposed into tangential and radial components (relative to the cutting tool). These components can be described in terms of F_C as:

$$F_r = F_{C_{Max}} \cdot \sin(\theta) \text{ and } F_t = F_{C_{Max}} \cdot \cos(\theta)$$

and the combined cutting force on any tooth may be described as:

$$\vec{F}_C = F_{C_{Max}}[\sin(\theta)\cos(\theta)\hat{i} + \sin^2(\theta)\hat{j}]$$

At any given time, the greatest number of teeth contributing force to a cutting operation is two. For two teeth, separated by an angle Ψ, the force vectors may be combined to give a maximum cutting force vector defined as:

$$\vec{F}_C = F_{C_{Max}} \cdot [(\cos(\theta)\sin(\theta)\hat{i} + \sin^2(\theta)\hat{j}) + (\cos(\theta+\psi)\sin(\theta+\psi)\hat{i} + \sin^2(\theta+\psi)\hat{j})]$$

By assuming that the maximum radial cutting force is no greater than one-half the tangential cutting force (Tata McGraw-Hill 1982, 653), and decomposing forces into those acting in the direction of feed (thrust forces) and those acting perpendicular to the direction of feed (perpendicular forces), the forces acting on a 2-flute endmill can be expressed as:

$$\vec{F}_X = F_{C_{Max}}[\sin(\theta)\cos(\theta) + .5 \cdot \sin^2(\theta)]$$

$$\vec{F}_Y = F_{C_{Max}}[-.5 \cdot \sin(\theta)\cos(\theta) + \sin^2(\theta)]$$

The combined X and Y forces produce a maximum resultant force vector at Θ = 90º, where the magnitude F = 1.11 F_{CMax}. It should be noted that this is the force acting on a single tooth, which is the greatest number of teeth that can contribute to cutting at any one time in a 2-fluted endmill.

For a 4-fluted endmill, the maximum cutting condition occurs at Θ = 45º, when two of the blades are cutting. By analyzing each tooth separately as above and combining, the magnitude of the resultant force vector can be shown to also be equal to 1.11 F_{CMax}. Consequently, this value is used in the spreadsheet to determine the magnitude of the maximum cutting force.

Appendix 3: Machining Force and Power Calculation Worksheet

The following pages are images of the sheets in the machining force and power calculation worksheet developed for this E90. A digital copy of this spreadsheet is available in the attached DVD.

Spreadsheet for Estimation of Machining Forces and Power Requirements - End Milling Operations

Date Modified:

Project:	E90 Mill - Cutting Force Calculations
Iteration:	E90 Proposal Draft Iteration
Date:	11/18/2011

Sheet Author:	
Name:	Julian Leland
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Project:	E90 Mill - Cutting Force Calculations
Iteration:	E90 Proposal Draft Iteration
Date:	11/18/2011

Metric	**Value**	**Units**	**Description**
Material To Cut			
Material Name	AISI 1030		Name of material being machined
K_P	0.78	HP/(in^3/min)	Power contstant. Available in *Machinery's Handbook* - pg. 1054
		kW/(cm^3/sec)	
Cutting Tool			
Material Name	HSS		
Tool Diameter	0.25	in	
		mm	
Number of Teeth	4		
Cut Parameters			
r	0.25	in	Desired radial width of cut.
		mm	
d	0.1	in	Desired axial depth of cut
		mm	
f_t	0.001	in/tooth	Feed per tooth. Available in *Machinery's Handbook*. May require multiple steps to calculate
		mm/tooth	
V	85	ft/min	Required cutting speed. Available in *Machinery's Handbook* - specified as "s".
		m/min	
W	1.2	-	Tool wear factor. Available in Machinery's Handbook.
f_m	5.194817343	in/min	Linear feed rate of cut
		cm/sec	
N	1298.704336	rpm (inch unit	
		rpm (metric ui	Spindle speed
Q	0.129870434	in^3/min	Material Removal Rate
		cm^3/sec	

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Project:	E90 Mill - Cutting Force Calculations
Iteration:	E90 Proposal Draft Iteration
Date:	11/18/2011

Metric	Value	Units	Description
Cutting Forces Present In Tool			
F_C	76.73918955	lbf	Tangential cutting force
		N	
F_{MAX}	85.1805004	lbf	Maximum force vector magnitude (2- or 4-flute endmill).
	0	N	See full report for derivation.

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Project:	E90 Mill - Cutting Force Calculations
Iteration:	E90 Proposal Draft Iteration
Date:	11/18/2011

Metric	**Value**	**Units**	**Description**
Power Requirements of Tool			
E	0.75	-	Efficiency of power transmission (.1 = 10%)
C	1.626058084	(feed in in.)	Feed factor for calculating power constant
		(feed in mm.)	
P_C	0.197661549	HP	Power at cutting tool
	0	kW	
P_M	0.263548732	HP	Power required from motor
	0	kW	

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Project:	E90 Mill - Cutting Force Calculations
Iteration:	E90 Proposal Draft Iteration
Date:	11/18/2011

Final Report

Material To Cut			
Material Name	AISI 1030		Name of material being machined
KP	0.78	HP/(in3/min)	Power Constant
	0	kW/(cm3/sec)	
Cutting Tool			
Material Name	HSS		
Tool Diameter	0.25	in	
	0	mm	
Number of Teeth	4		
Cut Parameters			
r	0.25	in	Desired radial width of cut.
	0	mm	
d	0.1	in	Desired axial depth of cut
	0	mm	
ft	0.001	in/tooth	Feed per tooth
	0	mm/tooth	
V	85	ft/min	Required cutting speed (at cutting edge)
	0	m/min	
W	1.2	-	Tool wear factor
fm	5.1948173	in/min	Linear feed rate of cut
	0	cm/sec	
N	1298.7043	rpm (inch units)	Spindle speed
	0	rpm (metric units)	
Q	0.1298704	in3/min	Material Removal Rate
	0	cm3/sec	
Power Requirements of Tool			
E	0.75	-	Efficiency of power transmission (.1 = 10%)
C	1.6260581	(feed in in.)	Feed factor for calculating power constant
	0	(feed in mm.)	
PC	0.1976615	HP	Power at cutting tool
	0	kW	
PM	0.2635487	HP	Power required from motor
	0	kW	
Cutting Forces Present In Tool			
FC	76.73919	lbf	Tangential cutting force
	0	N	
FMAX	85.1805	lbf	Maximum force vector magnitude (2- or 4-flute endmill). See full report for derivation.
	0	N	

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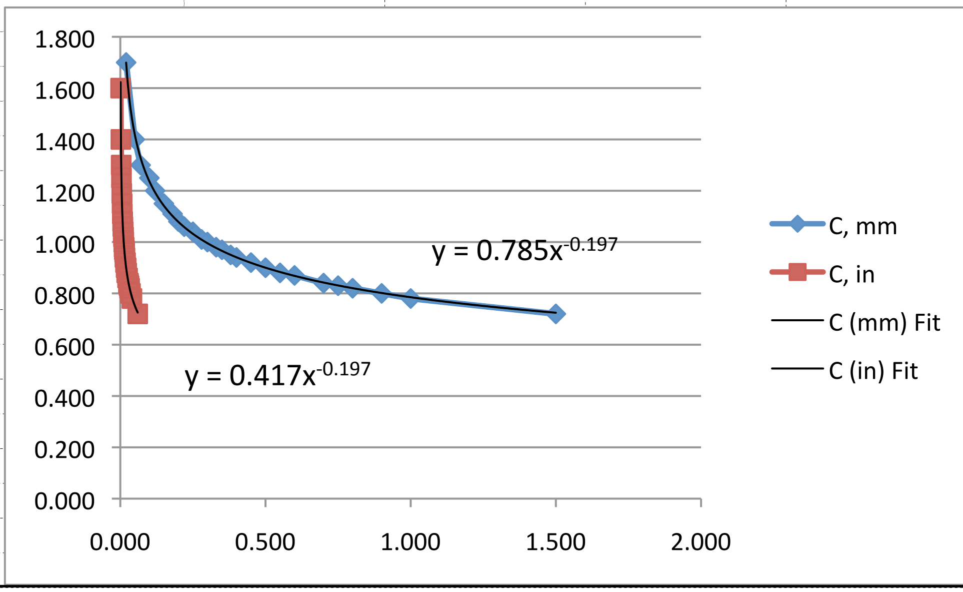
Project:	E90 Mill - Cutting Force Calculations
Iteration:	E90 Proposal Draft Iteration
Date:	39403

Interpolation Data

Feed Factor Calculations (C - Machinery's Handbook, pg. 1057)
Equations are of form y = ax^b

Feed [in]	C		Feed [mm]	C
0.001	1.600		0.020	1.700
0.002	1.400		0.050	1.400
0.003	1.300		0.070	1.300
0.004	1.250		0.100	1.250
0.005	1.190		0.120	1.200
0.006	1.150		0.150	1.150
0.007	1.110		0.180	1.110
0.008	1.080		0.200	1.080
0.009	1.060		0.220	1.060
0.010	1.040		0.250	1.040
0.011	1.020		0.280	1.010
0.012	1.000		0.300	1.000
0.013	0.980		0.330	0.980
0.014	0.970		0.350	0.970
0.015	0.960		0.380	0.950
0.016	0.940		0.400	0.940
0.018	0.920		0.450	0.920
0.020	0.900		0.500	0.900
0.022	0.880		0.550	0.880
0.025	0.860		0.600	0.870
0.028	0.840		0.700	0.840
0.030	0.830		0.750	0.830
0.032	0.820		0.800	0.820
0.035	0.800		0.900	0.800
0.040	0.780		1.000	0.780
0.060	0.720		1.500	0.720
a	0.417			0.785
b	-0.197			-0.197



1.800
1.600
1.400
1.200
1.000
0.800
0.600
0.400
0.200
0.000
0.000
0.500
1.000
1.500
2.000
$y = 0.785x^{-0.197}$
$y = 0.417x^{-0.197}$
C, mm
C, in
C (mm) Fit
C (in) Fit

Appendix 4: Material/Cut Combination Test Results

The following pages are report pages generated by the milling force and power spreadsheet while expected cutting force values were being generated for a variety of materials and cut parameters as part of the preliminary design process.

Spreadsheet for Estimation of Machining Forces and Power Requirements - End Milling Operations

Date Modified:

Project:	E90 Mill - Cutting Force Calculations
Iteration:	E90 Proposal - Final
Date:	11/21/2011

Sheet Author:	
Name:	Julian Leland
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Phone:	301-661-8391

Project:	E90 Mill - Cutting Force Calculations
Iteration:	E90 Proposal - Final
Date:	11/21/2011

Final Report

Material To Cut			
Material Name	304 Stainless		Name of material being machined
KP	0.6	HP/(in3/min)	Power Constant
	0	kW/(cm3/sec)	
Cutting Tool			
Material Name	HSS		
Tool Diameter	0.25	in	
	0	mm	
Number of Tee	4		
Cut Parameters			
r	0.25	in	Desired radial width of cut.
	0	mm	
d	0.1	in	Desired axial depth of cut
	0	mm	
ft	0.001	in/tooth	Feed per tooth
	0	mm/tooth	
V	102	ft/min	Required cutting speed (at cutting edge)
	0	m/min	
W	1.1	-	Tool wear factor
fm	6.233780811	in/min	Linear feed rate of cut
	0	cm/sec	
N	1558.445203	rpm (inch unit	Spindle speed
	0	rpm (metric units)	
Q	0.15584452	in3/min	Material Removal Rate
	0	cm3/sec	
Power Requirements of Tool			
E	0.75	-	Efficiency of power transmission (.1 = 10%)
C	1.626058084	(feed in in.)	Feed factor for calculating power constant
	0	(feed in mm.)	
PC	0.16725208	HP	Power at cutting tool
	0	kW	
PM	0.223002773	HP	Power required from motor
	0	kW	
Cutting Forces Present In Tool			
FC	54.11096699	lbf	Tangential cutting force
	0	N	
FMAX	60.06317336	lbf	Maximum force vector magnitude (2- or 4-flute endmill). See full report for derivation.
	0	N	

Project:	E90 Mill - Cutting Force Calculations	
Iteration:	E90 Proposal - Final	
Date:		11/21/2011

Final Report

Material To Cut			
Material Name	304 Stainless		Name of material being machined
KP	0.6	HP/(in3/min)	Power Constant
	0	kW/(cm3/sec)	
Cutting Tool			
Material Name	HSS		
Tool Diameter	0.25	in	
	0	mm	
Number of Tee	4		
Cut Parameters			
r	0.01	in	Desired radial width of cut.
	0	mm	
d	0.5	in	Desired axial depth of cut
	0	mm	
ft	0.014	in/tooth	Feed per tooth
	0	mm/tooth	
V	117.8	ft/min	Required cutting speed (at cutting edge)
	0	m/min	
W	1.1	-	Tool wear factor
fm	100.7916795	in/min	Linear feed rate of cut
	0	cm/sec	
N	1799.85142	rpm (inch unit	Spindle speed
	0	rpm (metric units)	
Q	0.503958398	in3/min	Material Removal Rate
	0	cm3/sec	
Power Requirements of Tool			
E	0.75	-	Efficiency of power transmission (.1 = 10%)
C	0.966827137	(feed in in.)	Feed factor for calculating power constant
	0	(feed in mm.)	
PC	0.321578832	HP	Maximum power at cutting tool
	0	kW	
PM	0.428771776	HP	Maximum power required from motor
	0	kW	
Cutting Forces Present In Tool			
FC	90.085751	lbf	Tangential cutting force
	0	N	
FMAX	99.99518361	lbf	Maximum force vector magnitude (2- or 4-flute endmill). See full report for derivation.
	0	N	

Project:	E90 Mill - Cutting Force Calculations		
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Material To Cut			
Material Name	AISI 1018 CR		Name of material being machined
KP	0.69	HP/(in3/min)	Power Constant
	0	kW/(cm3/sec)	
Cutting Tool			
Material Name	HSS		
Tool Diameter	0.375	in	
	0	mm	
Number of Tee	4		
Cut Parameters			
r	0.375	in	Desired radial width of cut.
	0	mm	
d	0.2	in	Desired axial depth of cut
	0	mm	
ft	0.001	in/tooth	Feed per tooth
	0	mm/tooth	
V	238	ft/min	Required cutting speed (at cutting edge)
	0	m/min	
W	1.1	-	Tool wear factor
fm	9.696992373	in/min	Linear feed rate of cut
	0	cm/sec	
N	2424.248093	rpm (inch unit	Spindle speed
	0	rpm (metric units)	
Q	0.727274428	in3/min	Material Removal Rate
	0	cm3/sec	
Power Requirements of Tool			
E	0.75	-	Efficiency of power transmission (.1 = 10%)
C	1.626058084	(feed in in.)	Feed factor for calculating power constant
	0	(feed in mm.)	
PC	0.897586162	HP	Maximum power at cutting tool
	0	kW	
PM	1.196781549	HP	Maximum power required from motor
	0	kW	
Cutting Forces Present In Tool			
FC	124.4552241	lbf	Tangential cutting force
	0	N	
FMAX	138.1452987	lbf	flute endmill). See full report for derivation.
	0	N	Assumes full-width cut.

Project:	E90 Mill - Cutting Force Calculations		
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Final Report

Material To Cut

Material Name	AISI 1018 CR		Name of material being machined
KP	0.69	HP/(in3/min)	Power Constant
	0	kW/(cm3/sec)	
Cutting Tool			
Material Name	HSS		
Tool Diameter	0.375	in	
	0	mm	
Number of Tee	4		
Cut Parameters			
r	0.02	in	Desired radial width of cut.
	0	mm	
d	0.5	in	Desired axial depth of cut
	0	mm	
ft	0.01	in/tooth	Feed per tooth
	0	mm/tooth	
V	186.2	ft/min	Required cutting speed (at cutting edge)
	0	m/min	
W	1.1	-	Tool wear factor
fm	75.86470503	in/min	Linear feed rate of cut
	0	cm/sec	
N	1896.617626	rpm (inch unit	Spindle speed
	0	rpm (metric units)	
Q	0.75864705	in3/min	Material Removal Rate
	0	cm3/sec	
Power Requirements of Tool			
E	0.75	-	Efficiency of power transmission (.1 = 10%)
C	1.033084998	(feed in in.)	Feed factor for calculating power constant
	0	(feed in mm.)	
PC	0.594863887	HP	Maximum power at cutting tool
	0	kW	
PM	0.793151849	HP	Maximum power required from motor
	0	kW	
Cutting Forces Present In Tool			
FC	105.4270047	lbf	Tangential cutting force
	0	N	
FMAX	117.0239752	lbf	flute endmill). See full report for derivation.
	0	N	Assumes full-width cut.

Project:	E90 Mill - Cutting Force Calculations		
Iteration:	E90 Proposal - Final		
Date:	11/21/2011		

Final Report

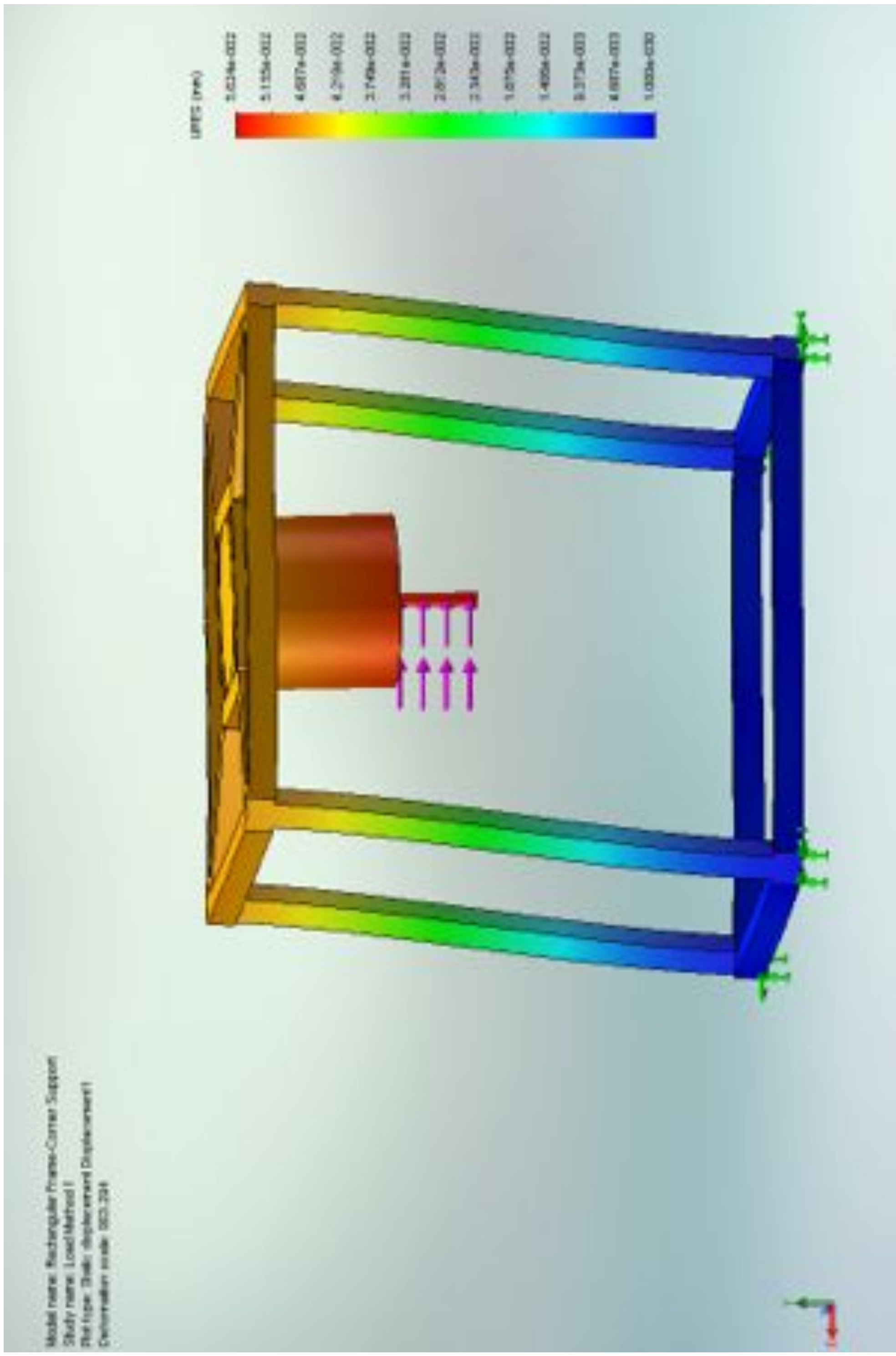
Material To Cut

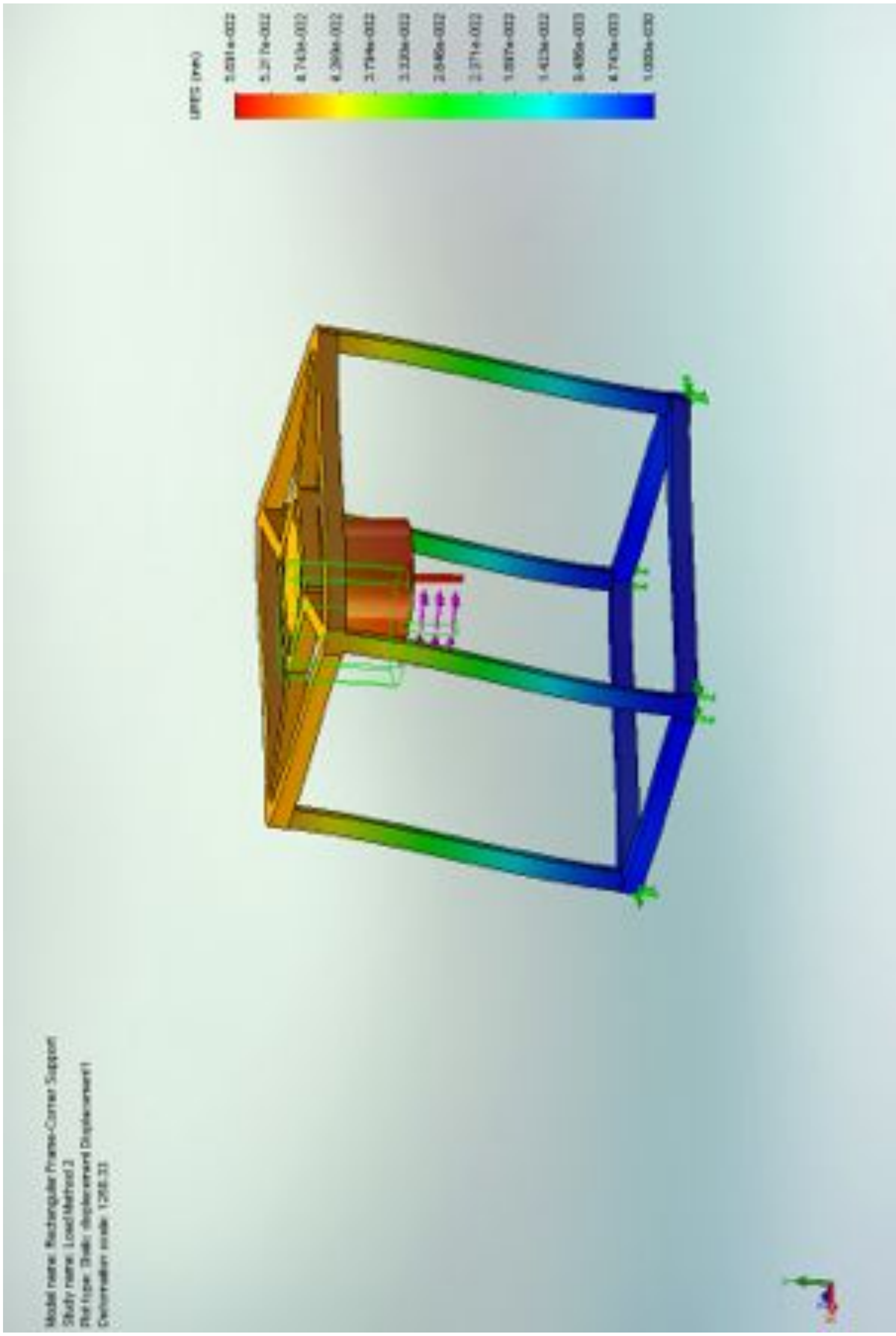
Material Name	6061 Aluminum		Name of material being machined
KP	0.33	HP/(in3/min)	Power Constant
	0	kW/(cm3/sec)	
Cutting Tool			
Material Name	HSS		
Tool Diameter	0.5	in	
	0	mm	
Number of Tee	2		
Cut Parameters			
r	0.5	in	Desired radial width of cut.
	0	mm	
d	0.25	in	Desired axial depth of cut
	0	mm	
ft	0.003	in/tooth	Feed per tooth
	0	mm/tooth	
V	1000	ft/min	Required cutting speed (at cutting edge)
	0	m/min	
W	1.1	-	Tool wear factor
fm	45.83662361	in/min	Linear feed rate of cut
	0	cm/sec	
N	7639.437268	rpm (inch unit	Spindle speed
	0	rpm (metric units)	
Q	5.729577951	in3/min	Material Removal Rate
	0	cm3/sec	
Power Requirements of Tool			
E	0.75	-	Efficiency of power transmission (.1 = 10%)
C	1.309613574	(feed in in.)	Feed factor for calculating power constant
	0	(feed in mm.)	
PC	2.7237825	HP	Maximum power at cutting tool
	0	kW	
PM	3.63171	HP	Maximum power required from motor
	0	kW	
Cutting Forces Present In Tool			
FC	89.8848225	lbf	Tangential cutting force
	0	N	
FMAX	99.77215297	lbf	flute endmill). See full report for derivation.
	0	N	Assumes full-width cut.

Appendix 5: Preliminary Frame Testing Results

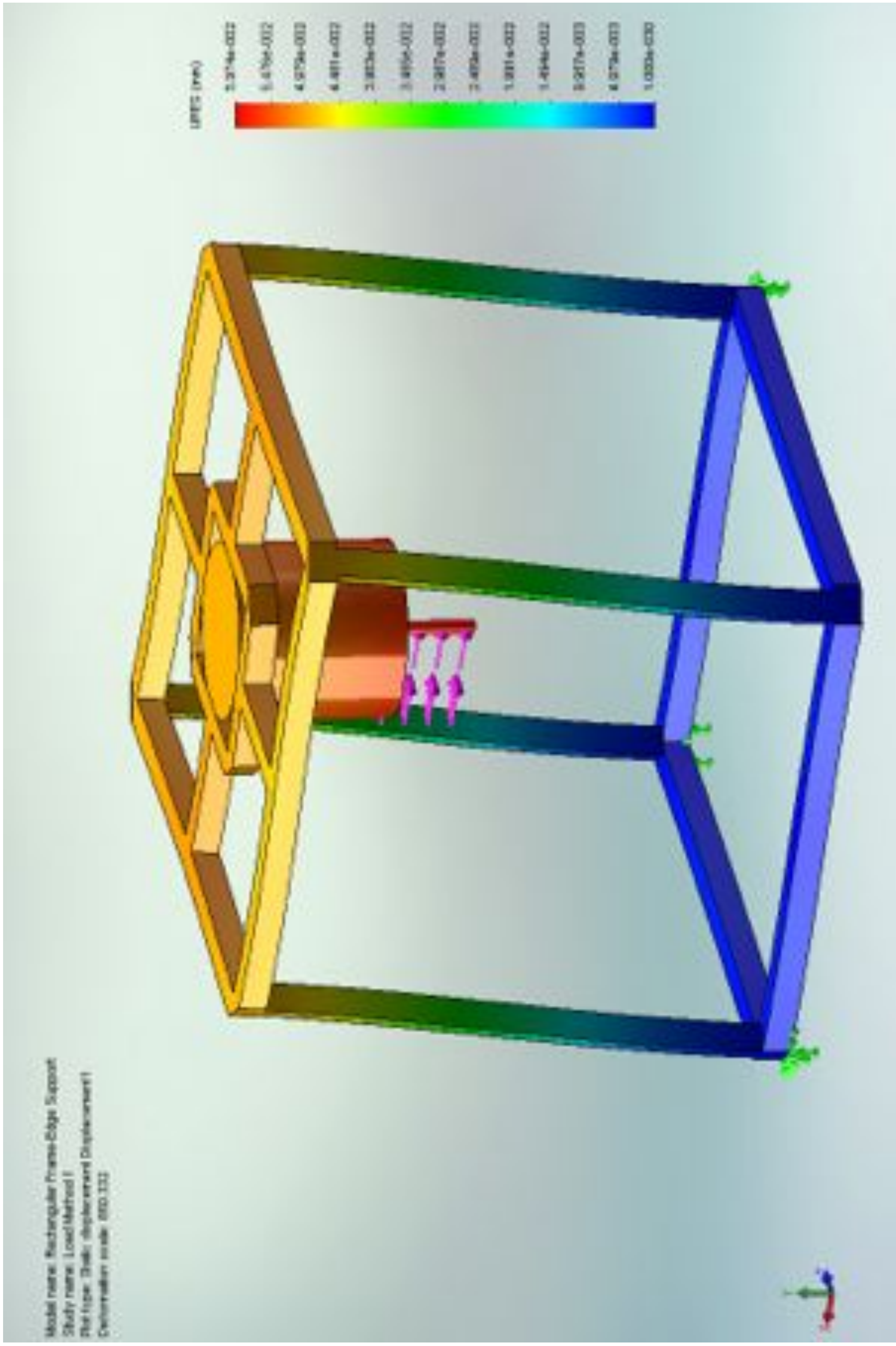
The following images show test results from the 5 frames examined during preliminary frame testing.

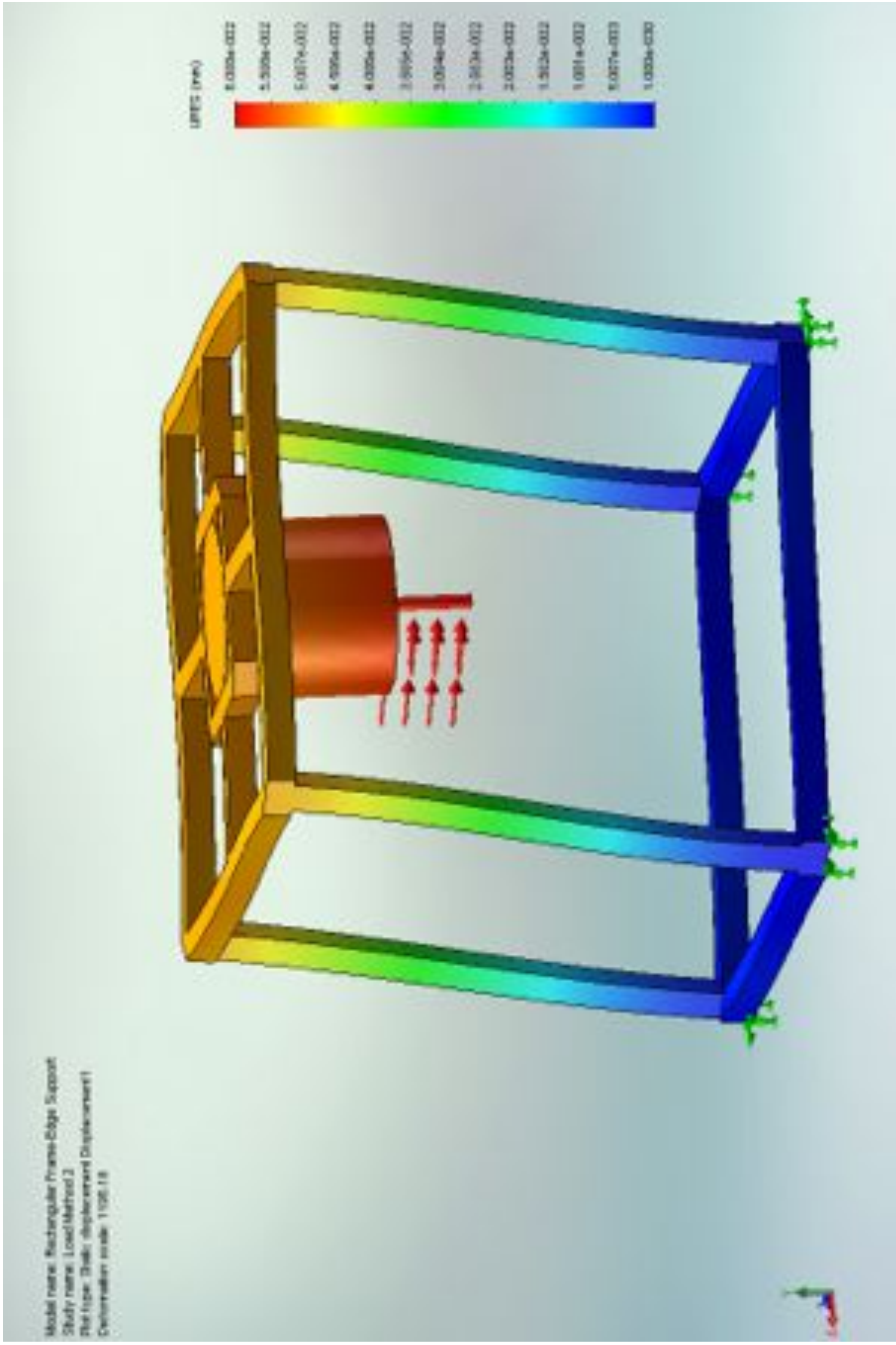
Frame 1: Square Frame (Corner-Supported)



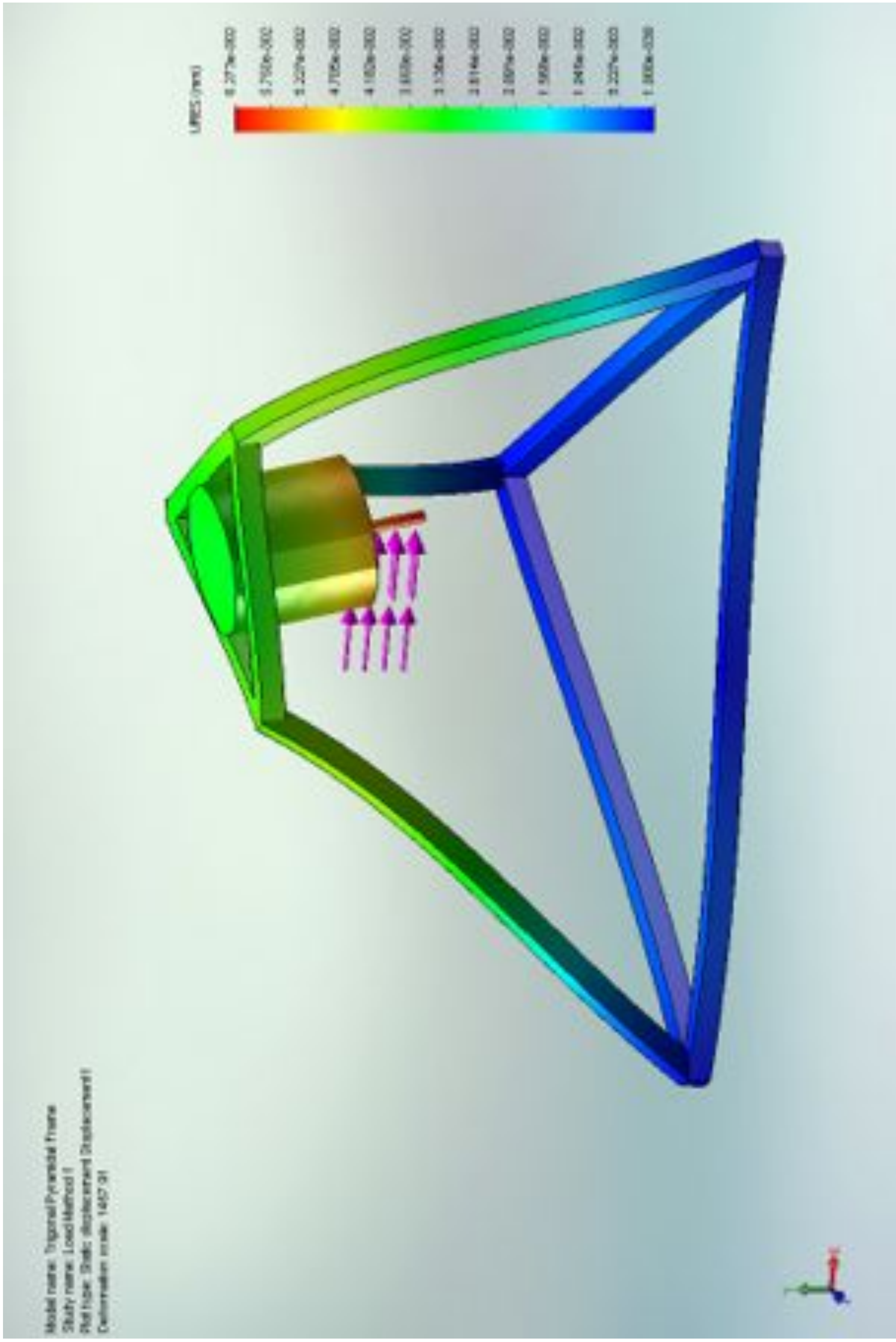


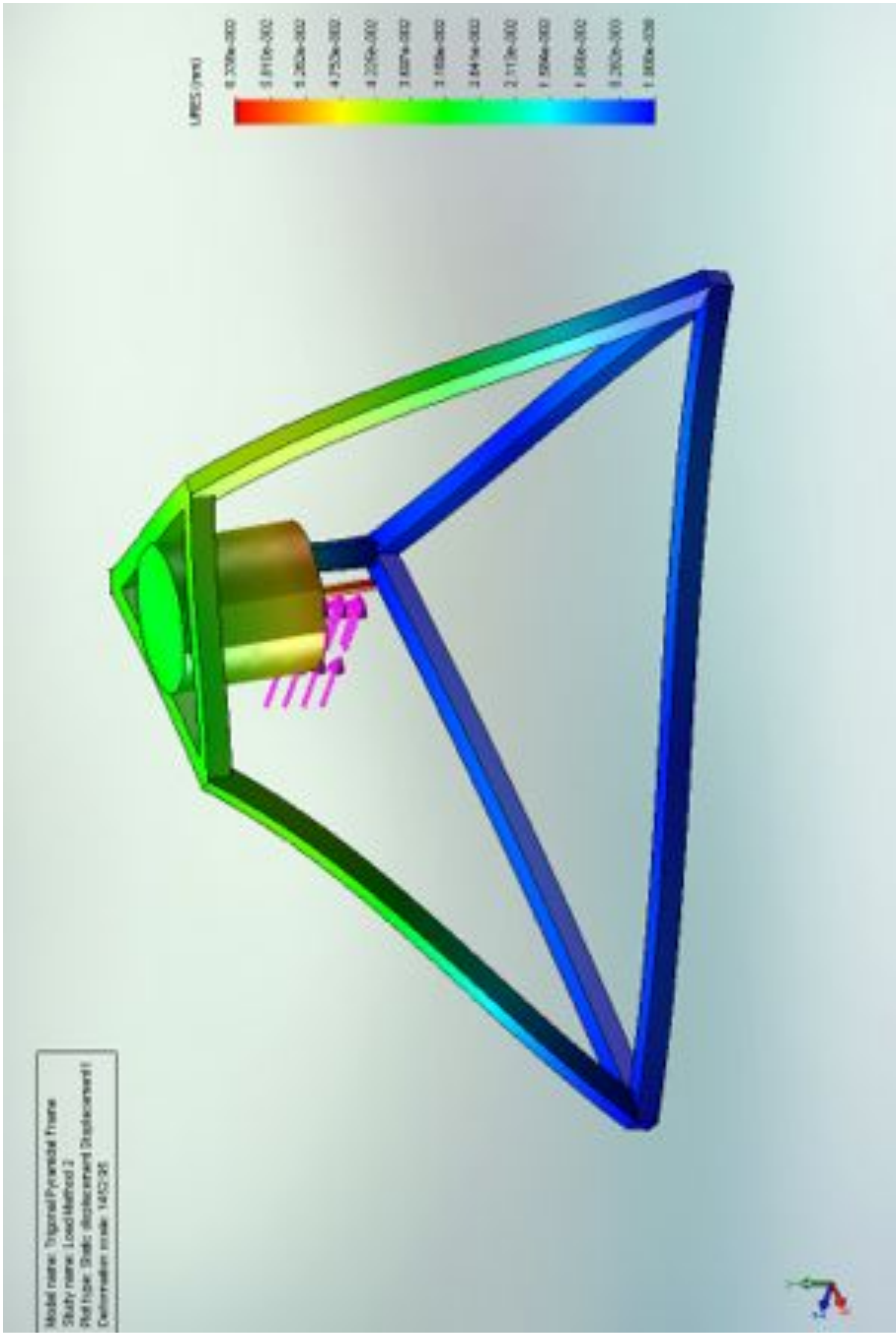
Frame 2: Square Frame (Edge-Supported)



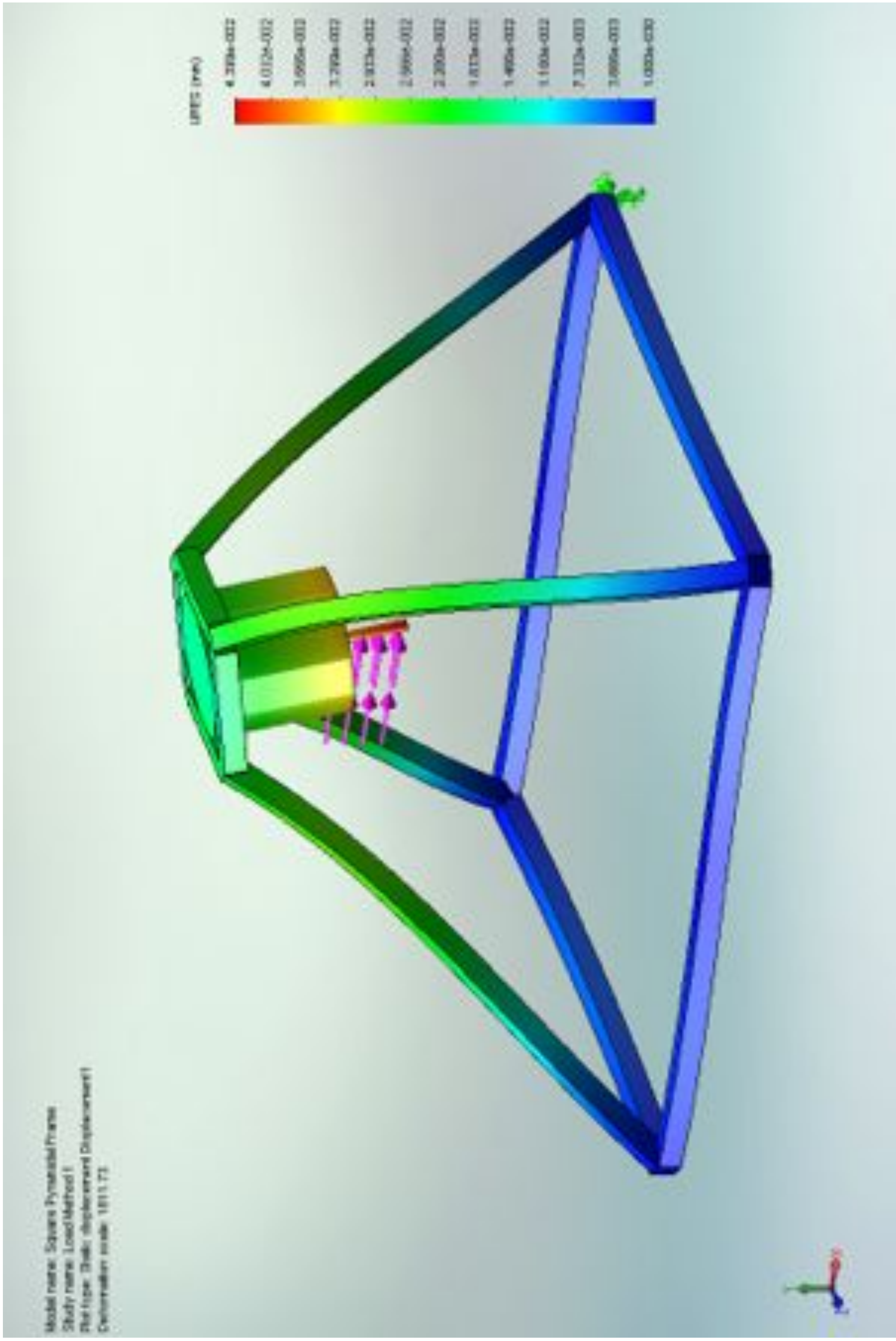


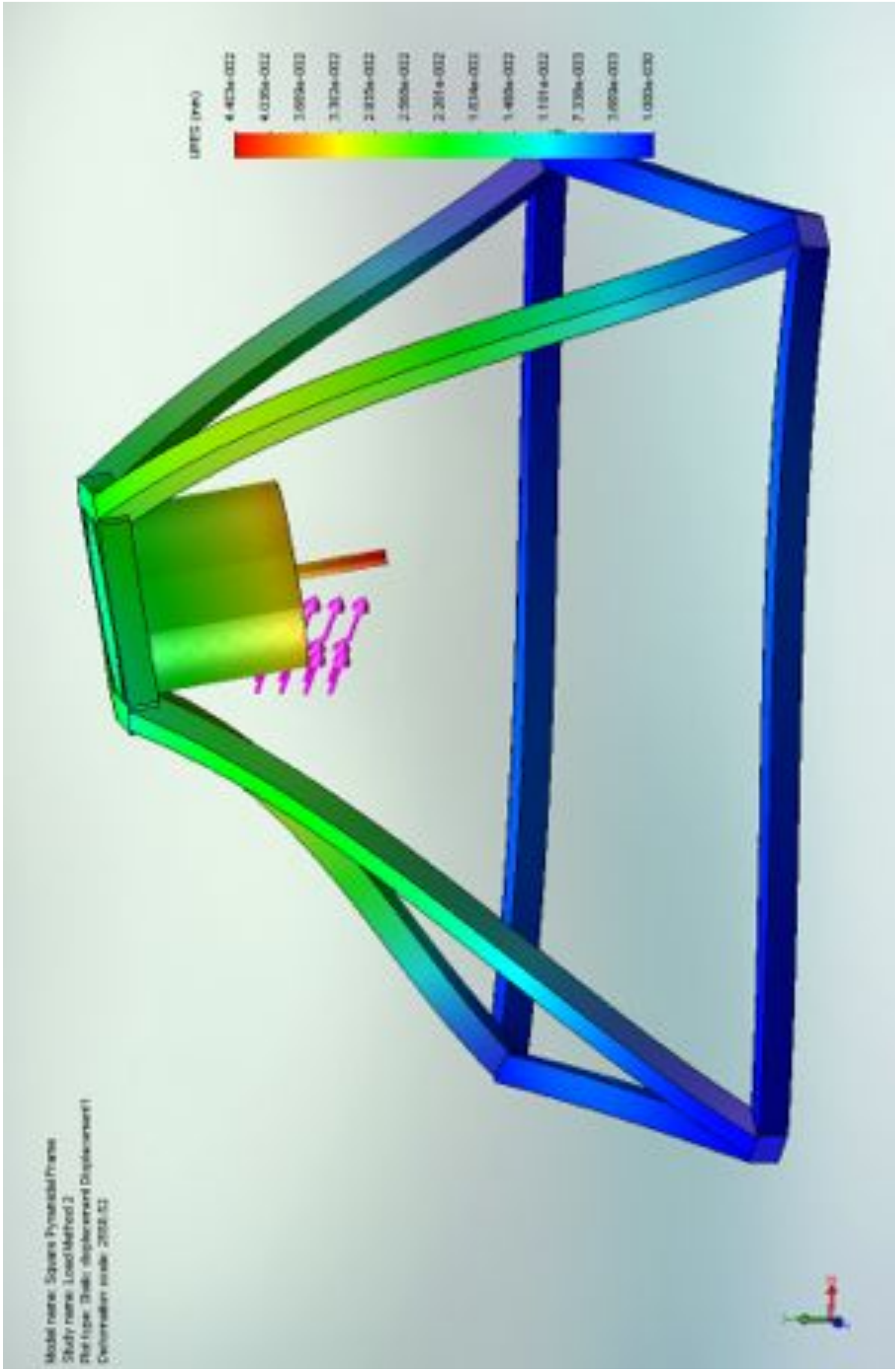
Frame 3: Trigonal Pyramidal Frame



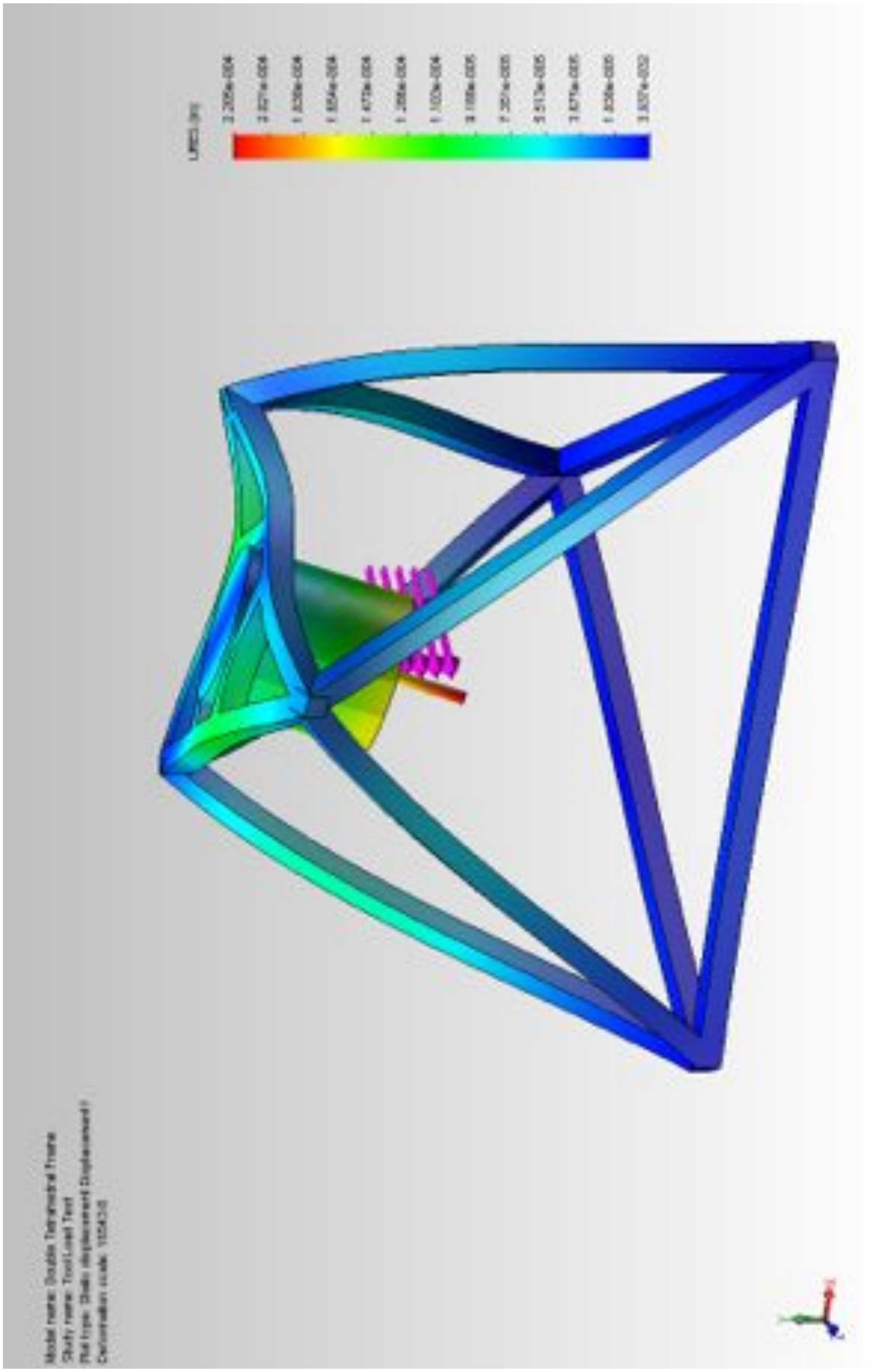


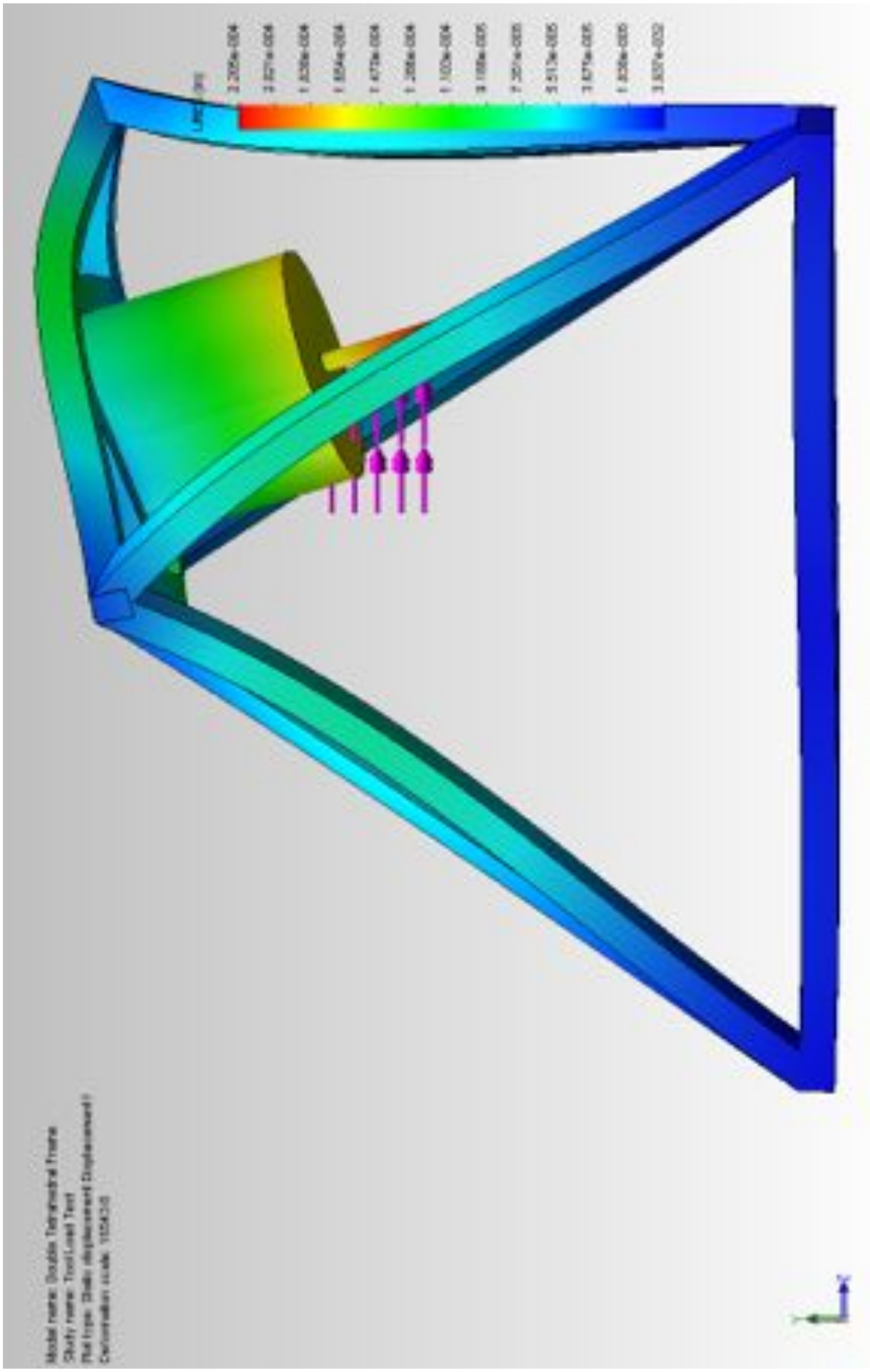
Frame 4: Square Pyramidal Frame




Model name: Square Pyramidal Frame
Study name: Load Method 2
Plot type: Static displacement Displacement1

Frame 5: Double Tetrahedral Frame




Model name: Double Tetrahedral Frame
Study name: Tool Load Test
Plot type: Static displacement Displacement1

Appendix 6: Bearing Load and Moment Calculation Worksheet

The following pages are images of the sheets in the bearing load calculation worksheet developed for this E90. A digital copy of this spreadsheet is available in the attached DVD.

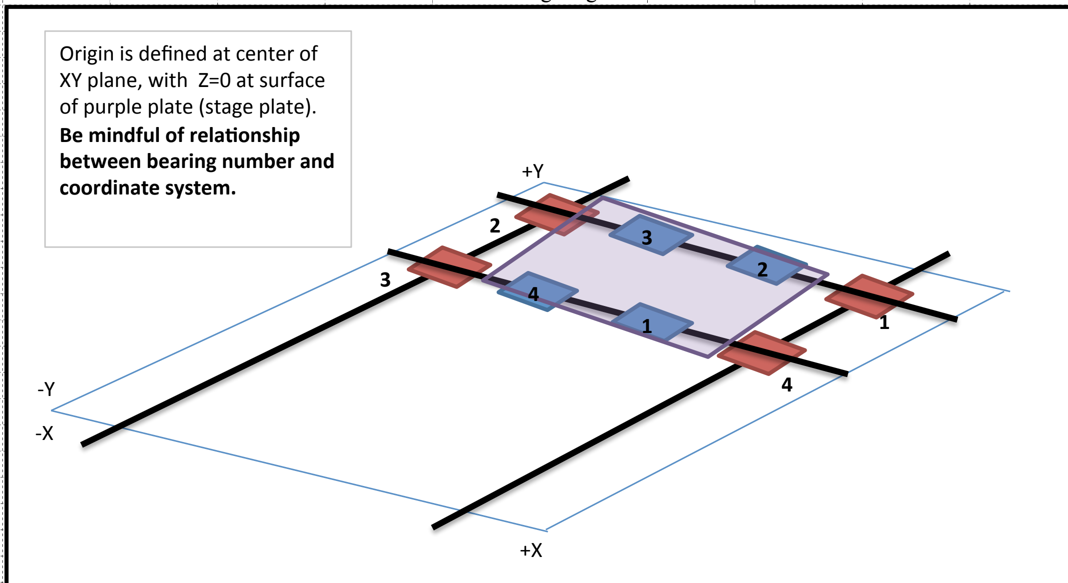
Bearing Load Calculation Spreadsheet - Input

This spreadsheet calculates the loads on linear motion bearings. It takes forces applied at the cutting tool and bearing locations (X,Y,Z) as inputs. The X-Y center of the work volume, at the table height, is defined as the origin. All dimensions in inches. All forces in lbf.

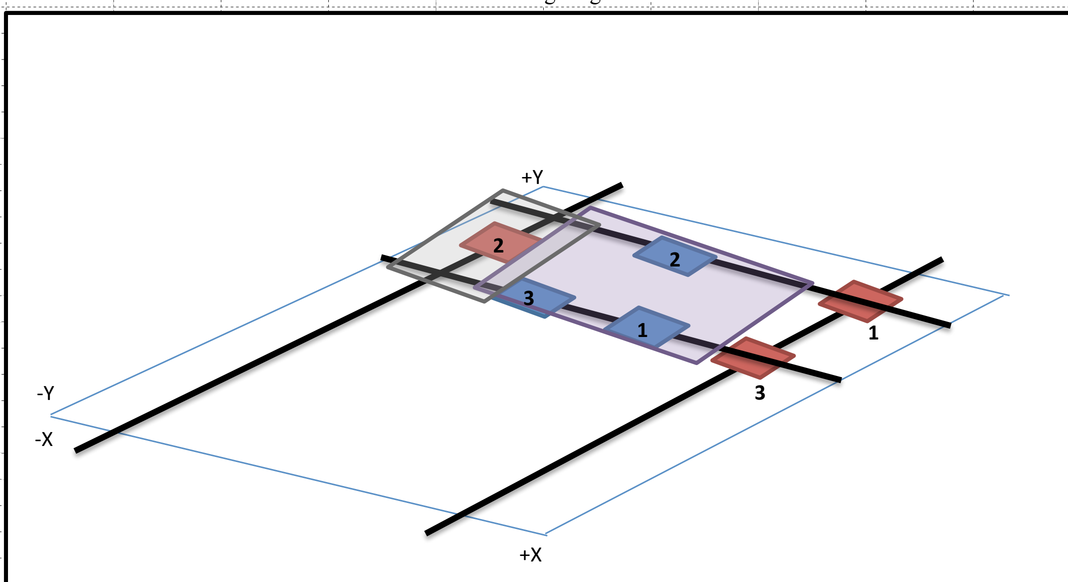
NOTE: If a bearing is not used (e.g. Bearing 4 in 3-bearing stage), clear cells rather than zeroing.

Inputs:		**Value**								
		X	Y	Z						
Spindle:										
	Z Position	0	0	6						
	Cutting Force	-200	0	0						
					Results					
Y Stage (bottom stage)					Fx	Fy	Fz	Mx	My	Mz
	Number of Bearings	3								
	Centroid	0	3	-4						
	Bearing 1 Location	-3.5	4.75	-4	-104.7619	0	-142.8571	NA	NA	NA
	Bearing 2 Location	3.5	3	-4	66.666667	0	285.71429	NA	NA	NA
	Bearing 3 Location	-3.5	1.25	-4	238.09524	0	-142.8571	NA	NA	NA
	Bearing 4 Location				NA	NA	NA	NA	NA	NA
X Stage (top stage)										
	Number of Bearings	3								
	Centroid	-3	3	-2						
	Bearing 1 Location	-1.25	1.25	-2	0	171.42857	-457.1429	NA	NA	NA
	Bearing 2 Location	-3	4.75	-2	0	0	0	NA	NA	NA
	Bearing 3 Location	-4.75	1.25	-2	0	-171.4286	457.14286	NA	NA	NA
	Bearing 4 Location				NA	NA	NA	NA	NA	NA
Z Stage										
	Number of Bearings	2								
	Centroid	#DIV/0!	#DIV/0!	#DIV/0!						
	Bearing 1 Location									
	Bearing 2 Location									
	Bearing 3 Location									
	Bearing 4 Location									

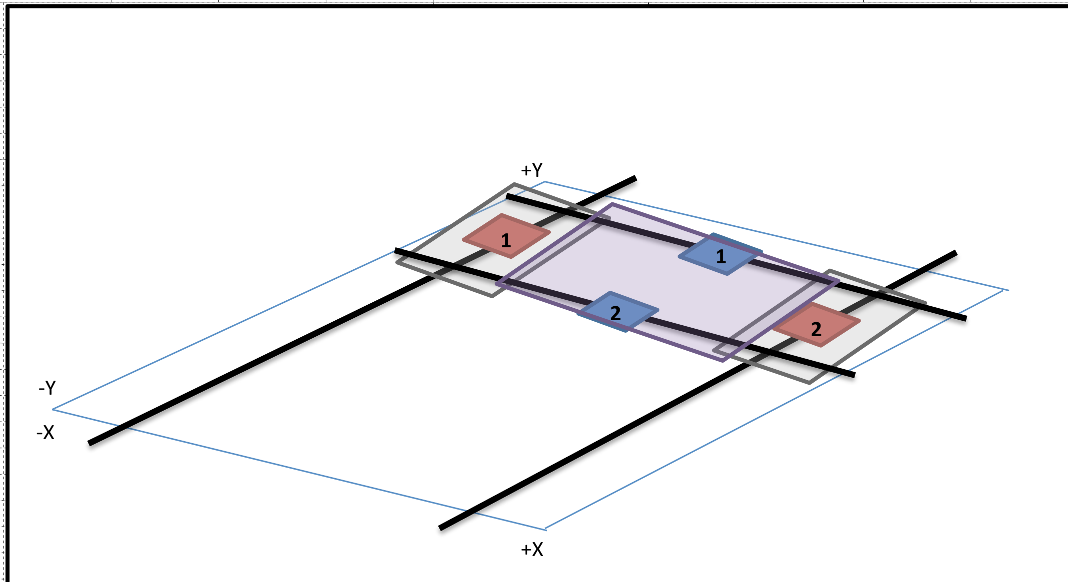
4-Bearing Stages



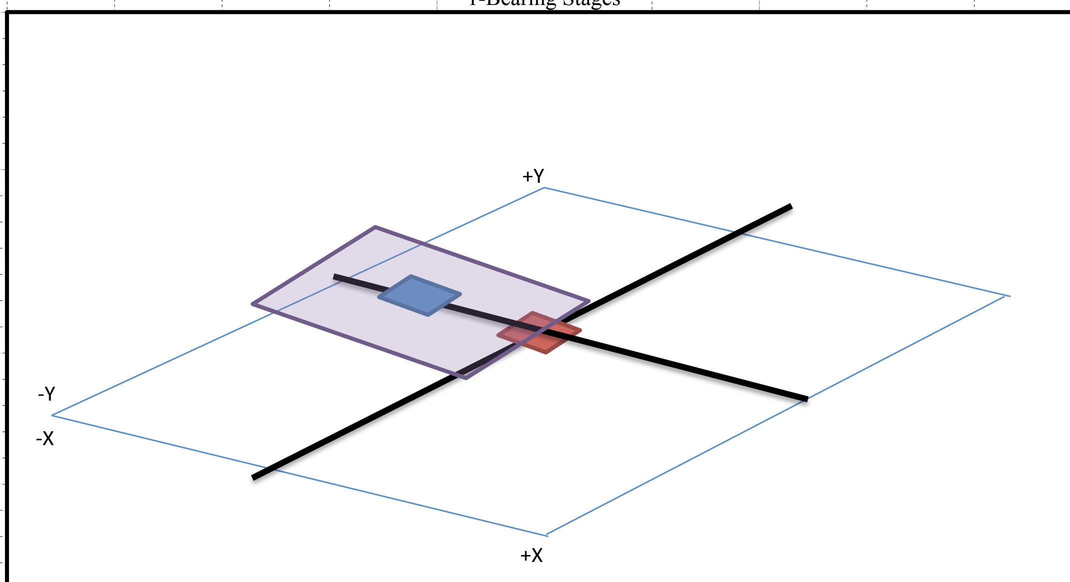
3-Bearing Stages



2-Bearing Stages



1-Bearing Stages



Bearing Load Calculation Spreadsheet - 4 Bearing Values

This spreadsheet calculates the loads on linear motion bearings. **ONLY REFER TO THIS SHEET FOR AXES THAT USE 4 BEARINGS - other calculations are incorrect.**

X Stage Bearings

	Force Due To:	Fx	Fy	Fz	**Total**
Bearing 1	X	0			0
	Y	-240	0		-240
	Z	640	0	0	640
Bearing 2	X	0			0
	Y	-240	0		-240
	Z	640	0	0	640
Bearing 3	X	0			0
	Y	240	0		240
	Z	-640	0	0	-640
Bearing 4	X	0			0
	Y	240	0		240
	Z	-640	0	0	-640

Y Stage Bearings

	Force Due To:	Fx	Fy	Fz	**Total**
Bearing 1	X	0	0		0
	Y		0		0
	Z	114.2857	0	0	114.2857
Bearing 2	X	0	0		0
	Y		0		0
	Z	-114.2857	0	0	-114.2857
Bearing 3	X	100	0		100
	Y		0		0
	Z	-114.2857	0	0	-114.2857
Bearing 4	X	100	0		100
	Y		0		0
	Z	114.2857	0	0	114.2857

Z Stage Bearings

Bearing Load Calculation Spreadsheet - 3 Bearing Values

This spreadsheet calculates the loads on linear motion bearings. **ONLY REFER TO THIS SHEET FOR AXES THAT USE 3 BEARINGS - other calculations are incorrect.**

X Stage Bearings									
	Force Due	Fx	Fy	Fz	**Total**				
Bearing 1	X				0	Z-Force Matrix			
	Y	171.42857	0		171.42857	1	1	1	
	Z	-457.1429	0	0	-457.1429	0.5833333	0	-0.583333	
	Mx				0	0.5833333	-0.583333	0.5833333	
	My				0				
	Mz				0	Inverse Matrix			Input Forces
Bearing 2	X				0	0.25	0.8571429	0.4285714	0
	Y		0		0	0.5	0	-0.857143	0
	Z		0	0	0	0.25	-0.857143	0.4285714	0
	Mx				0				
	My				0	Z-Force Output Matrix:			
	Mz				0	F1	0		
Bearing 3	X				0	F2	0		
	Y	-171.4286	0		-171.4286	F3	0		
	Z	457.14286	0	0	457.14286				
	Mx				0				
	My				0				
	Mz				0				

Y Stage Bearings									
	Force Due	Fx	Fy	Fz	**Total**				
Bearing 1	X	-104.7619	0		-104.7619	Z-Force Matrix			
	Y				0	1	1	1	
	Z	-142.8571	0	0	-142.8571	0.875	0	-0.875	
	Mx				0	1.75	-1.75	1.75	
	My				0				
	Mz				0	Inverse Matrix			
Bearing 2	X	66.666667			66.666667	0.25	0.5714286	0.1428571	0
	Y				0	0.5	0	-0.285714	0
	Z	285.71429		0	285.71429	0.25	-0.571429	0.1428571	0
	Mx				0				
	My				0	Z-Force Output Matrix:			
	Mz				0	F1	0		
Bearing 3	X	238.09524	0		238.09524	F2	0		
	Y				0	F3	0		
	Z	-142.8571	0	0	-142.8571				
	Mx				0				
	My				0				
	Mz				0				

Bearing Load Calculation Spreadsheet - 4 Bearing Values

This spreadsheet calculates the loads on linear motion bearings. **ONLY REFER TO THIS SHEET FOR AXES THAT USE 2 BEARINGS - other calculations are incorrect.**

X Stage Bearings

	Force Due To:	Fx	Fy	Fz	**Total**
Bearing 1	X				0
	Y		0		0
	Z		0	0	0
	Mx				0
	My	800		0	800
	Mz	300	0		300
Bearing 2	X				0
	Y		0		0
	Z		0	0	0
	Mx				0
	My	800		0	800
	Mz	300	0		300

Y Stage Bearings

	Force Due To:	Fx	Fy	Fz	**Total**
Bearing 1	X	100			100
	Y				0
	Z	1000		0	1000
	Mx		0	0	0
	My				0
	Mz	300	0		300
Bearing 2	X	100			100
	Y				0
	Z	-1000		0	-1000
	Mx		0	0	0
	My				0
	Mz	300	0		300

Z Stage Bearings

	Force Due To:	Fx	Fy	Fz	**Total**
Bearing 1	X				
	Y				
	Z				
	Mx				
	My				
	Mz				
Bearing 2	X				
	Y				
	Z				
	Mx				
	My				
	Mz				

Appendix 7: Bearing Load and Moment Calculation Derivations

The following derivation is used in the bearing load and moment calculation worksheet to determine the loads and moments on 4-, 3- and 2- bearing carriages. Bearing orientations are as shown in Appendix 6 above. **All derivations are made for the X-axis carriage only – coordinate transformations must be made to use with other axes.** These derivations are loose approximations, and should be used only as a guideline for bearing system design.

4-Bearing Carriage: This derivation is taken directly from Slocum (Slocum 1992, 507). The orientation of the system coordinate frame has been changed to more accurately match that of the machjine

X direction forces cause the following Z direction forces:

$$F_{1Z,FX} = F_{2Z,FX} = \frac{F_X z_{FX}}{2(x_1 - x_4)}$$

$$F_{3Z,FX} = F_{4Z,FX} = \frac{-F_X z_{FX}}{2(x_1 - x_4)}$$

and the following Y direction forces:

$$F_{1Y,FX} = F_{2Y,FX} = \frac{F_X y_{FX}}{2(x_1 - x_4)}$$

$$F_{3Y,FX} = F_{4Y,FX} = \frac{-F_X y_{FX}}{2(x_1 - x_4)}$$

Z direction forces cause the following Z direction forces:

$$F_{1Z,FZ} = -F_Z \frac{(x_4 - x_{FZ})}{(x_4 - x_1)} \frac{(y_2 - y_{FZ})}{(y_2 - y_1)}$$

$$F_{2Z,FZ} = F_Z \frac{(x_4 - x_{FZ})}{(x_4 - x_1)} \frac{(y_1 - y_{FZ})}{(y_2 - y_1)}$$

$$F_{3Z,FZ} = -F_Z \frac{(x_1 - x_{FZ})}{(x_4 - x_1)} \frac{(y_4 - y_{FZ})}{(y_3 - y_4)}$$

$$F_{4Z,FZ} = F_Z \frac{(x_1 - x_{FZ})}{(x_4 - x_1)} \frac{(y_3 - y_{FZ})}{(y_3 - y_4)}$$

Y direction forces cause the following Y direction forces:

$$F_{1Y,FY} = F_{2Y,FY} = \frac{-F_Y (x_4 - x_{FY})}{2(x_4 - x_1)}$$

$$F_{3Y,FY} = F_{4Y,FY} = \frac{F_Y(x_1 - x_{FY})}{2(x_4 - x_1)}$$

and the following Z direction forces:

$$F_{1Z,FY} = F_{4Z,FY} = \frac{F_Y y_{FY}}{2(y_1 - y_2)}$$

$$F_{2Z,FY} = F_{3Z,FY} = \frac{-F_Y y_{FY}}{2(y_1 - y_2)}$$

Three-Bearing Carriage: This derivation was developed by the author. In this derivation, quantities subscripted "c" indicate the centroid of the stage: for example, x_C is the x-coordinate of the centroid of the stage. Quantities with a bar over them are defined as equal to the difference between the coordinate of the force application point minus the coordinate of the stage centroid: for example, Zbar is equal to the z-coordinate of the tooltip location, minus the z-coordinate of the stage centroid. The subscript n simply indicates to use the coordinate associated with that bearing: for example, when calculating $F_{1Z,\,FX}$, x_n is equal to the x coordinate of bearing 1.

X direction forces cause the following Z forces:

$$F_{1Z,FX} = F_{3Z,FX} = \frac{F_X \bar{z}_F}{2(x_N - x_C)}$$

and the following Y forces:

$$F_{1Y,FX} = F_{3Y,FX} = \frac{F_X \bar{y}_F}{2(x_N - x_C)}$$

Y direction forces cause the following Z direction forces:

$$F_{1Z,FY} = F_{3Z,FY} = \frac{F_y \bar{z}_F}{4(y_N - y_C)}$$

$$F_{2Z,FY} = \frac{F_y \bar{z}_F}{2(y_2 - y_C)}$$

and the following Y forces:

$$F_{1Y,FY} = F_{3Y,FY} = \frac{F_y \bar{x}_F}{2(x_N - x_C)}$$

Z direction forces produce an indeterminate system in the carriage. Using the following equations, where F_Z' is the effective Z load at the centroid of the carriage, $x_s$bar and $y_s$bar are the distances between the bearing carriages and the centroid of the stage in the x and y directions, and M_Y' is the resultant moment produced at the centroid of the carriage:

$$F_Z' + F_{1Z} + F_{2Z} + F_{3Z} = 0$$

$$F_{1Z} \cdot \bar{x}_s - F_{3Z} \cdot \bar{x}_s + F_Z' \cdot \bar{x} = 0$$

$$-(F_{1Z} + F_{3Z}) \cdot \bar{y}_s + F_{2Z} \cdot \bar{y}_s + F_Z' \cdot \bar{y} = 0$$

the following matrix can be produced, which allows calculation of Fz for all three bearings.

$$\begin{bmatrix} 1 & 1 & 1 \\ \dfrac{\bar{x}_s}{\bar{x}} & 0 & -\dfrac{\bar{x}_s}{\bar{x}} \\ -\dfrac{\bar{y}_s}{\bar{y}} & \dfrac{\bar{y}_s}{\bar{y}} & -\dfrac{\bar{y}_s}{\bar{y}} \end{bmatrix} \cdot \begin{bmatrix} F_{1Z} \\ F_{2Z} \\ F_{3Z} \end{bmatrix} = \begin{bmatrix} -F_Z' \\ -F_Z' \\ -F_Z' \end{bmatrix}$$

Two-Bearing Carriage: This derivation was developed by the author.

X direction forces produce the following moments about the Y axis:

$$M_{1Y,FX} = M_{2Y,FX} = \frac{-F_X \bar{z}_F}{2}$$

and the following moments about the Z axis:

$$M_{1Z,FX} = M_{2Z,FX} = \frac{F_X \bar{y}_F}{2}$$

Z direction forces produce the following moments about the Y axis:

$$M_{1Y,FZ} = M_{2Y,FZ} = \frac{-F_X \bar{x}_F}{2}$$

and the following Z direction forces:

$$F_{1Z,FZ} = \frac{-F_Z(y_2 - y_{FZ})}{(y_2 - y_1)}$$

$$F_{2Z,FZ} = \frac{F_Z(y_1 - y_{FZ})}{(y_2 - y_1)}$$

Y direction forces produce the following moments about the Z axis:

$$M_{1Z,FY} = M_{2Z,FY} = \frac{-F_Y \bar{x}_F}{2}$$

as well as the following Z and Y direction forces:

$$F_{1Z,FY} = \frac{F_Y \bar{z}_F}{2}$$

$$F_{2Z,FY} = \frac{-F_Y \bar{z}_F}{2}$$

$$F_{1Y,FY} = F_{2Y,FY} = \frac{-F_Y}{2}$$

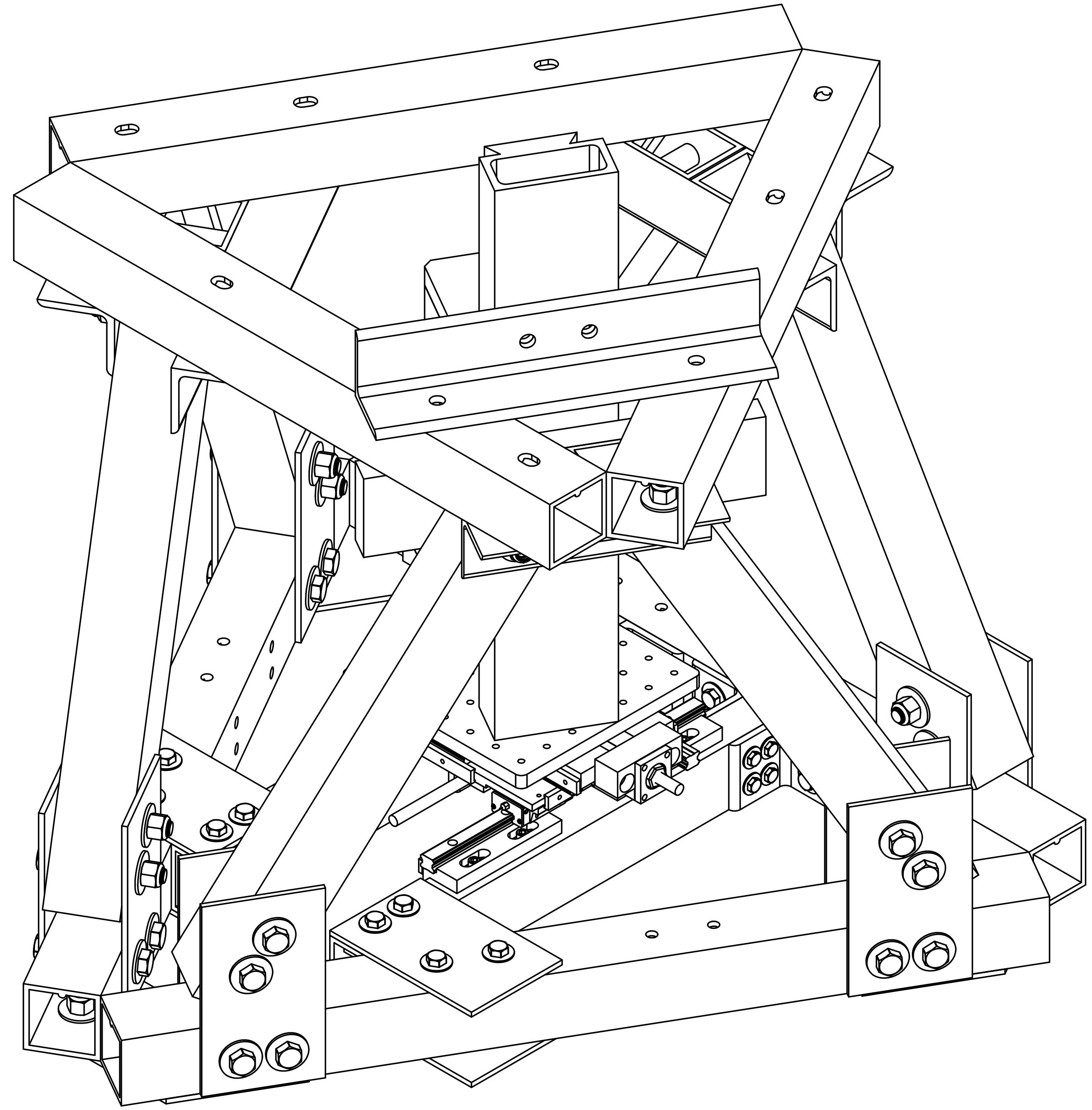
Appendix 8: Machine Part & Assembly Drawings

The following drawings detail all manufactured parts involved in the construction of this machine. Additionally, three drawings of the entire assembly are also attached, with part numbers called out and referenced in a partial bill of materials.

Note: Only some stock fasteners/hardware are included in the bill of materials: others have not been featured in the model. These drawings are primarily intended for use when creating the parts required for the machine, not as an assembly guide.

Self-Replicating Milling Machine

Julian Leland, 2012
Swarthmore College
E90 Senior Design Project



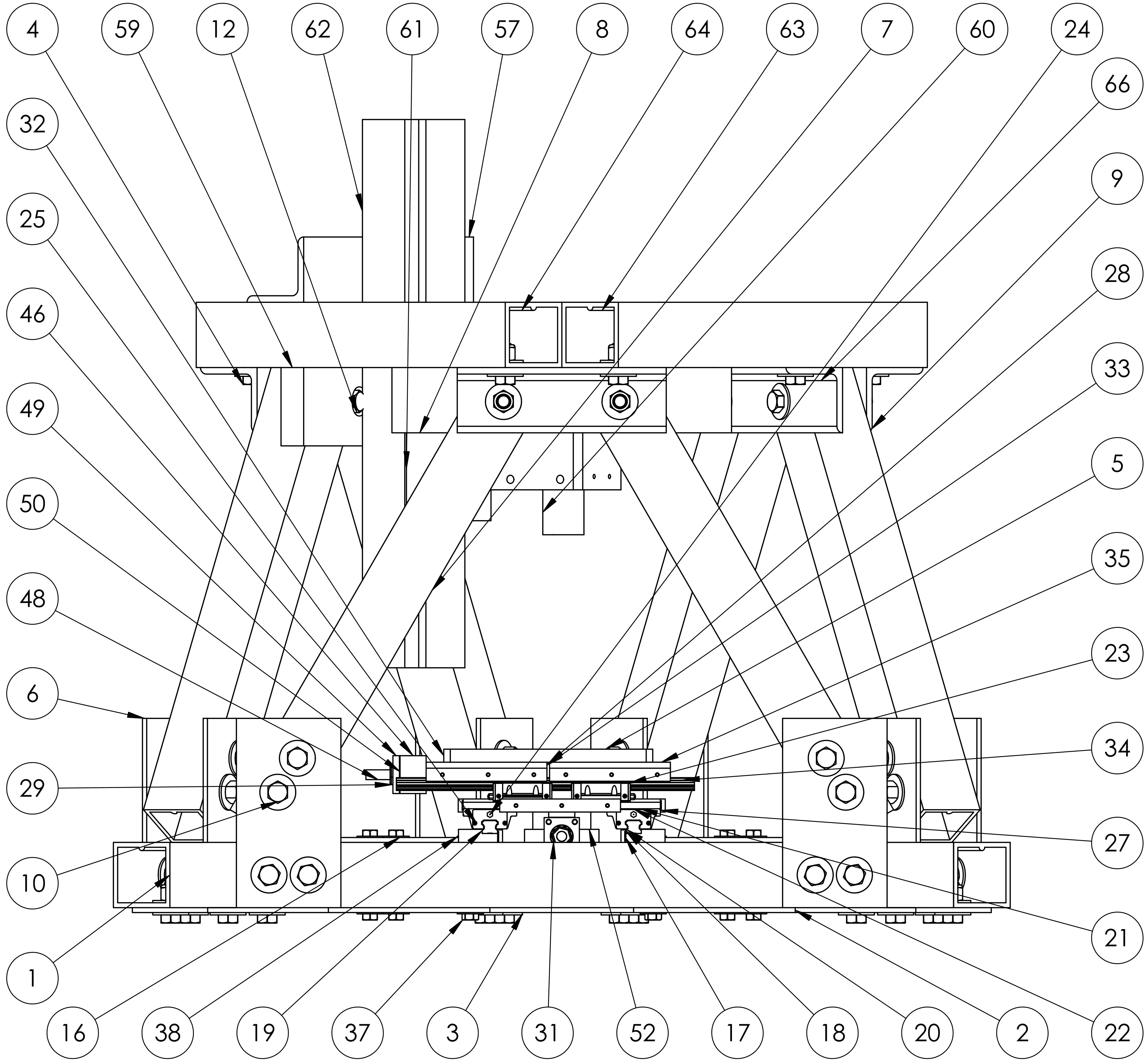
		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	5/3/12
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE:

FullAssembly - Final Design

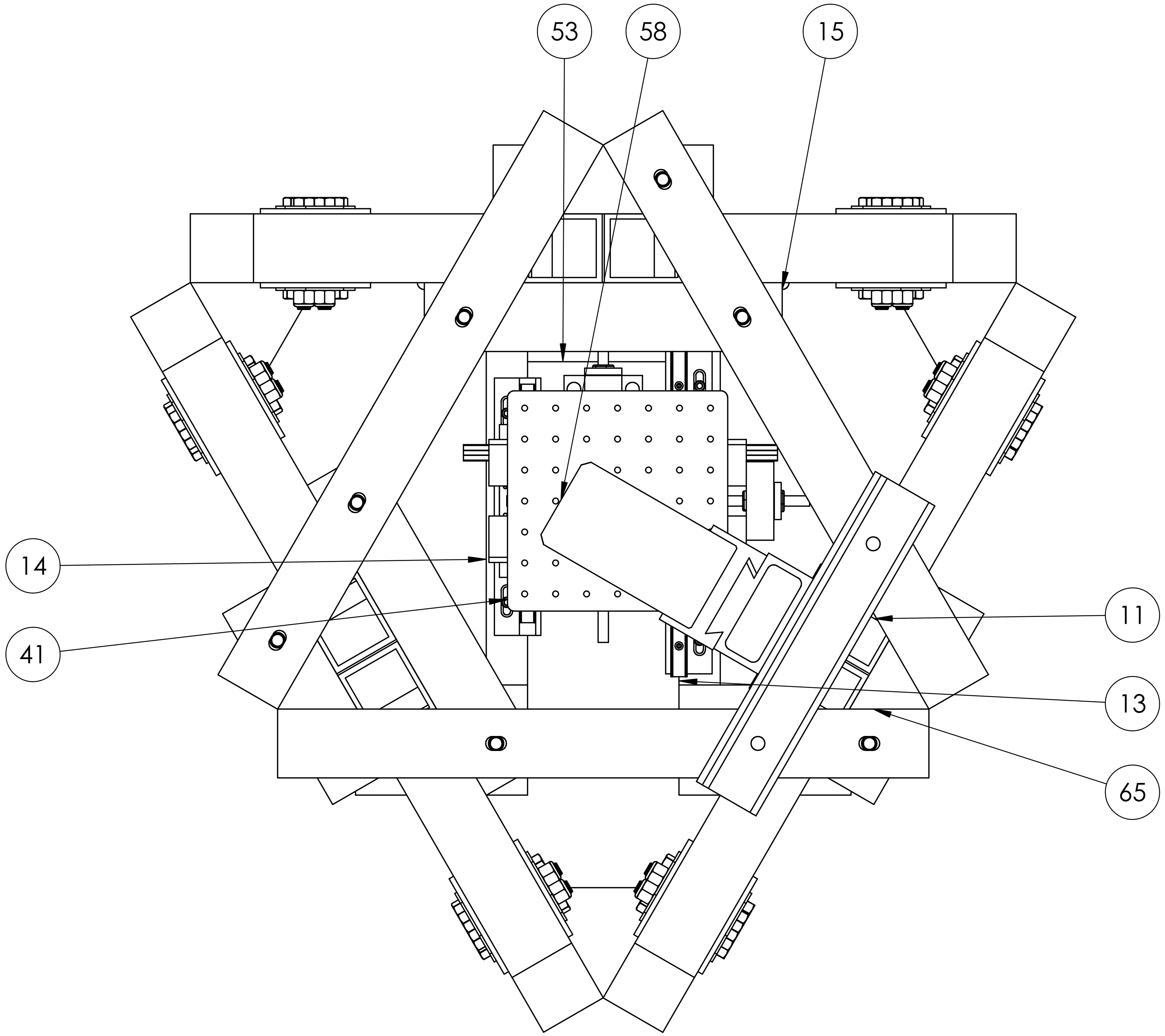
SIZE	DWG. NO.	REV
B		

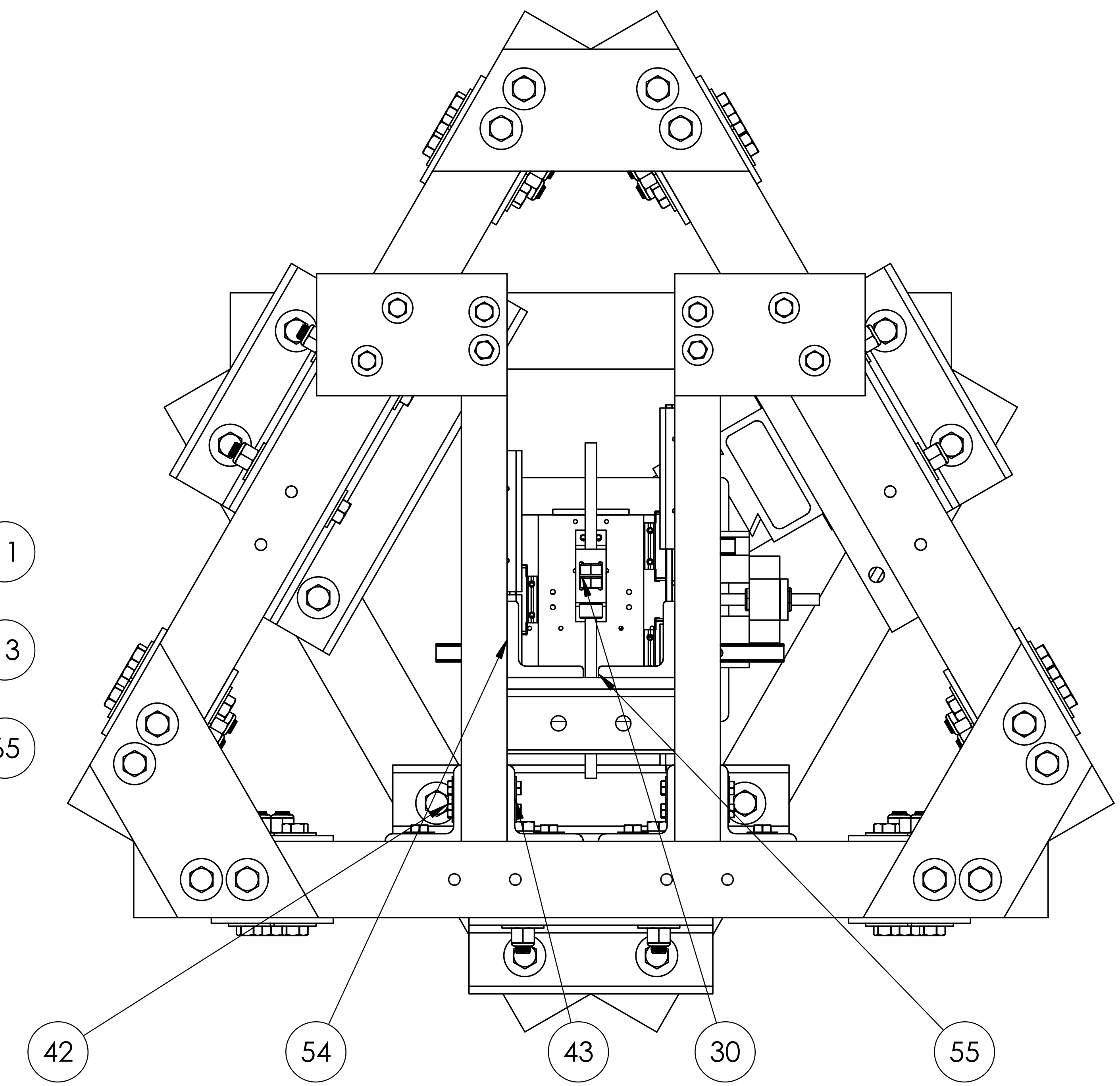
SCALE: 1:5 WEIGHT: SHEET 1 OF 4



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	5/3/12	TITLE:
	CHECKED			FullAssembly - Final Design
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS:			SIZE B / DWG. NO. / REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:5 / WEIGHT: / SHEET 2 OF 4

NEXT ASSY	USED ON
APPLICATION	





		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	5/3/12	TITLE:
			CHECKED			FullAssembly - Final Design
			ENG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	MFG APPR.			
		MATERIAL	Q.A.			
			COMMENTS:			SIZE B · DWG. NO. · REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:5 · WEIGHT: · SHEET 3 OF 4

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	BaseMember - Final Design		3
2	BasePlate - Final Design		3
3	98026A033		128
4	92620A714		46
5	97135A250		64
6	SidePlate - Final Design		12
7	SideMember - Final Design		6
8	TopInsidePlate - Final Design		1
9	TopOutsidePlate - Final Design		3
10	91257A727		12
11	p75x2p125Sleeve		18
12	91257A728		6
13	LeftRailSupport - Final Design		1
14	RightRailSupport - Final Design		1
15	RailFrameAttachmentAngle - Final Design		4
16	RailFrameAttachmentPlate - Final Design		4
17	ShortBearingRailClampV2 - FinalDesign		4
18	ShortPushPlate - FinalDesign		10
19	12 in - EGR15U		1
20	220mm - EGR15U		2
21	BearingBlockMountPlate - FinalDesign		1
22	Block_EGW15CCZ0HHIHI		6
23	End_Seal_EG15HIHI		12
24	HGT5_34310002HIHI		6
25	Bolt_EG15HIHI		24
26	HGT5_PlugScrewHIHI		6
27	SingleHolePushPlate - FinalDesign		2
28	TopStageNutMount		2
29	Acme.375x10Nut		8
30	9713K72		2
31	90128A840		2
32	TopTable - FinalDesign		1
33	TopShortBearingRailClampV2 - FinalDesign		2
34	12 in TEST- EGR15U		1
35	TopShortBearingRailClampV2 - Left - FinalDesign		2
36	RailSupportMountBolt - BaseSupport		4
37	92620A655		24
38	98026A031		24
39	RailSupportMountBolt - BaseSupportV2		4
40	RailClampMountPlate		4
41	91251A442		8
42	92620A563		16
43	98026A029		32
44	94895A805		16
45	RailSupportMountBolt - RailMountSupport		4
46	LeadscrewThrustBearingMount		1
47	94669A084		2
48	18inAcmeLeadscrew		2
49	ThrustBearingPushPlate		2
50	6680K11		4
51	16x2mmSpacer		2
52	LeadscrewThrustBearingMount - BottomStage		1
53	BottomStageMountAngle1		1
54	BottomStageMountAngle2		1
55	BottomStageMountAngle3		1
56	RailSupportMountBolt - LeadScrewSupport		2
57	TopInsideAngleZMount - Final Design		1
58	Spindle Motor Mount		1
59	TopInsideZMountv2		1
60	Spindle Base		1
61	Spindle Gib Mount		1
62	SpindleColumn		1
63	Copy of Copy of TopMember - Final Design^FullAssembly - Final Design		1
64	Copy of TopMemberSideB - Final Design^FullAssembly - Final Design		1
65	TopMemberSideC - Final Design		1
66	Copy of Copy of TopInsidePlate - Final Design^FullAssembly - Final Design		1

NEXT ASSY | USED ON
APPLICATION

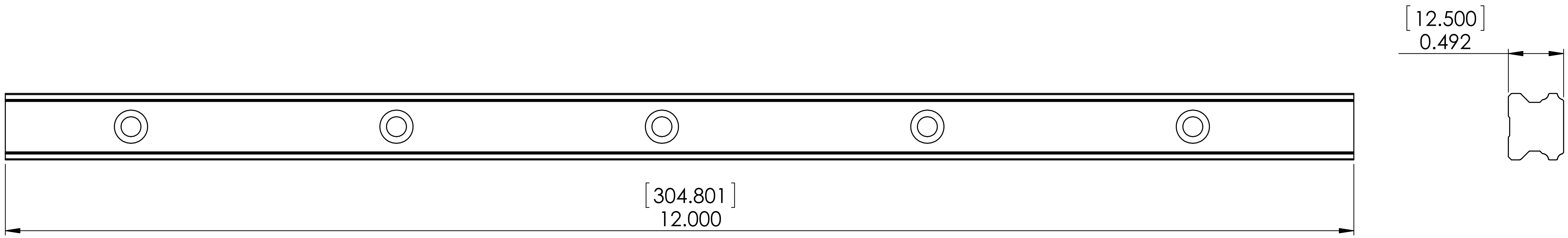
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	5/3/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:
Part descriptions are either descriptions listed in part files, or are McMaster-Carr part numbers. Only some fasteners are included.

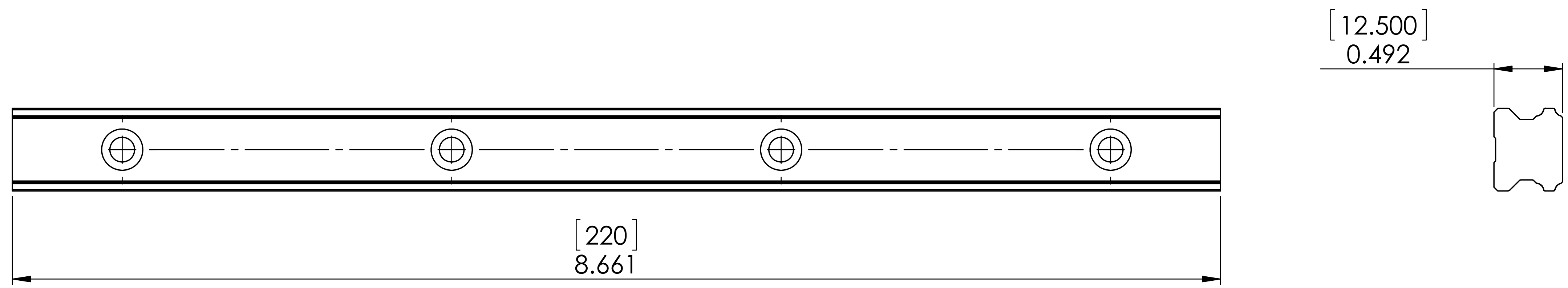
TITLE:
FullAssembly - Final Design

SIZE B | DWG. NO. | REV
SCALE: 1:5 | WEIGHT: | SHEET 4 OF 4



UNLESS OTHERWISE SPECIFIED:		NAME	DATE		
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	5/3/12	TITLE:	
	CHECKED			12 in - EGR15U	
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL	COMMENTS: Cut rail to roughly 12". Note that sides of rail are hardened - difficult to cut. Make 2x			SIZE B DWG. NO.	REV
FINISH					
DO NOT SCALE DRAWING				SCALE: 1:1 WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	



UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	5/3/12
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS: Pre-cut size - 220mm long. No machining required.		
FINISH			
DO NOT SCALE DRAWING			

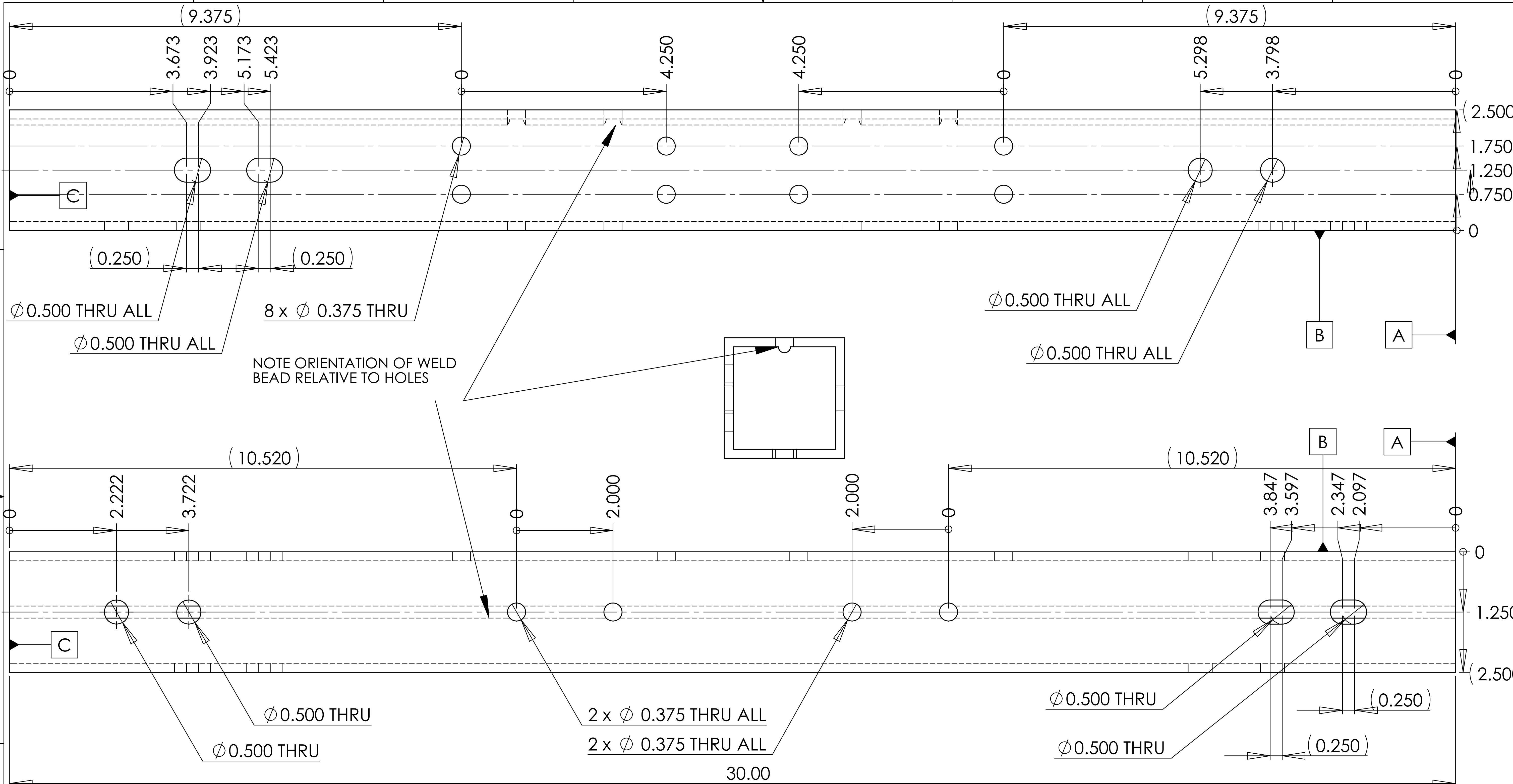
NEXT ASSY	USED ON
APPLICATION	

TITLE:

220mm - EGR15U

SIZE B | DWG. NO. | REV

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1



MAKE 1 COPY OF PART WITH SET OF 8 .375"
HOLES, BUT NO .375" HOLES ON TOP OF PART.
MAKE 2 COPIES WITH .375 HOLES ON TOP OF
PART, BUT WITHOUT 8 X .375 HOLE PATTERN.

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING

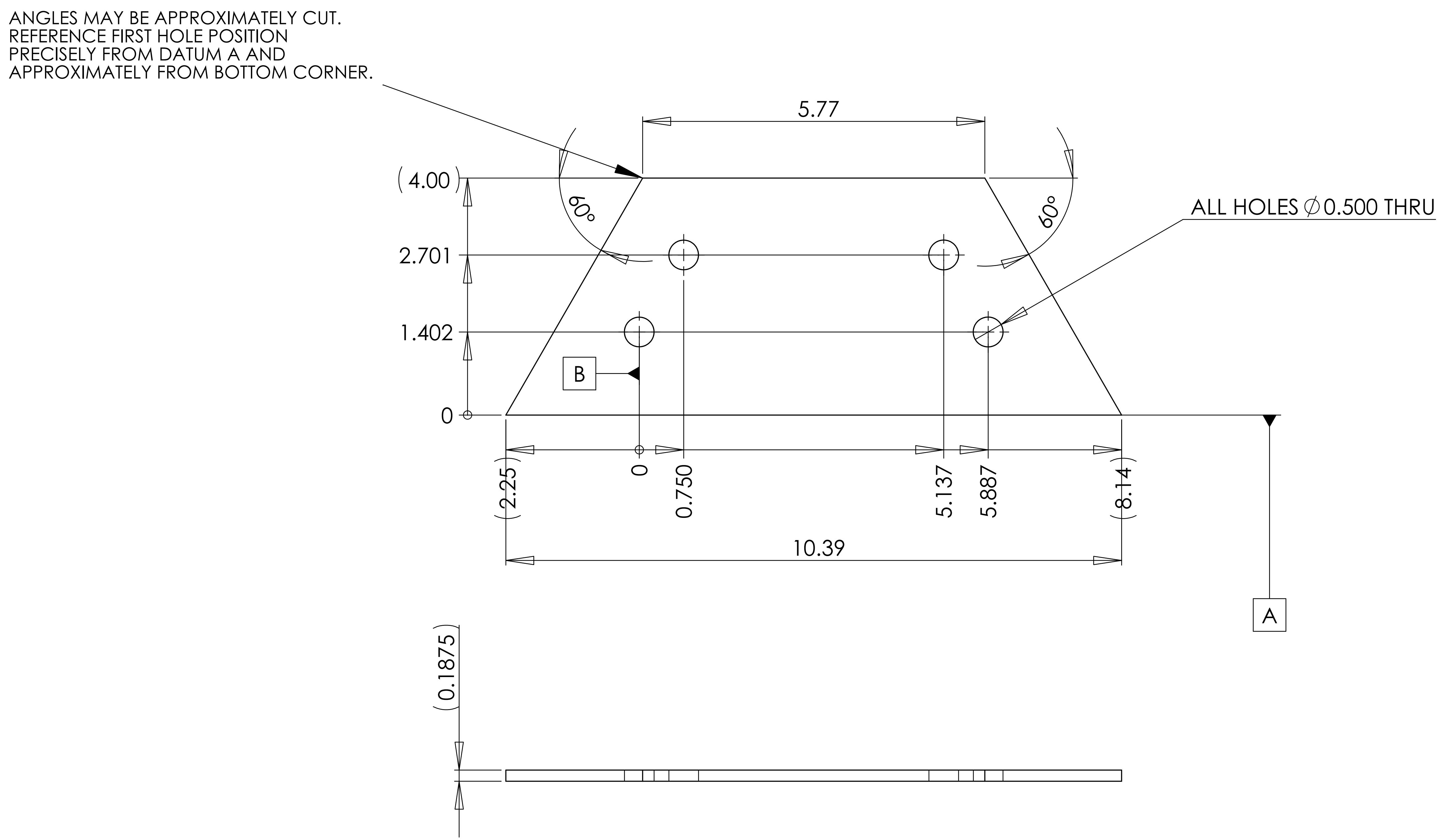
	NAME	DATE
DRAWN	Leland	3/12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:
Make 3x according to instructions (see note)

TITLE:
BaseMember - Final Design

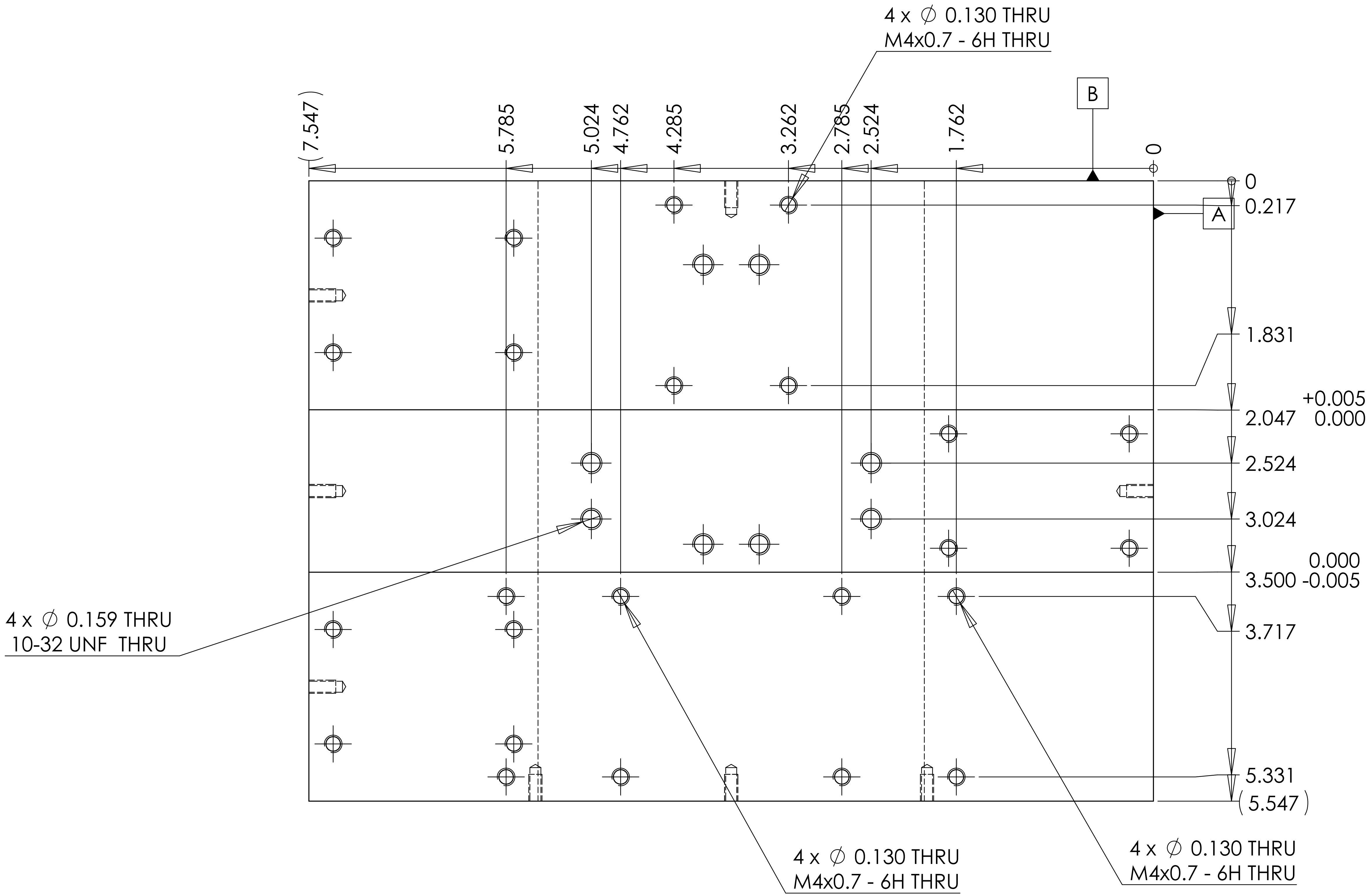
SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE		
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	3/12/12	TITLE:	
	CHECKED			BasePlate - Final Design	
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL	COMMENTS: Make 3x			SIZE **B** DWG. NO.	REV
FINISH					
DO NOT SCALE DRAWING				SCALE: 1:2 WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	



UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/17/12
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS: Make 1x		
FINISH			
DO NOT SCALE DRAWING			

NEXT ASSY | USED ON

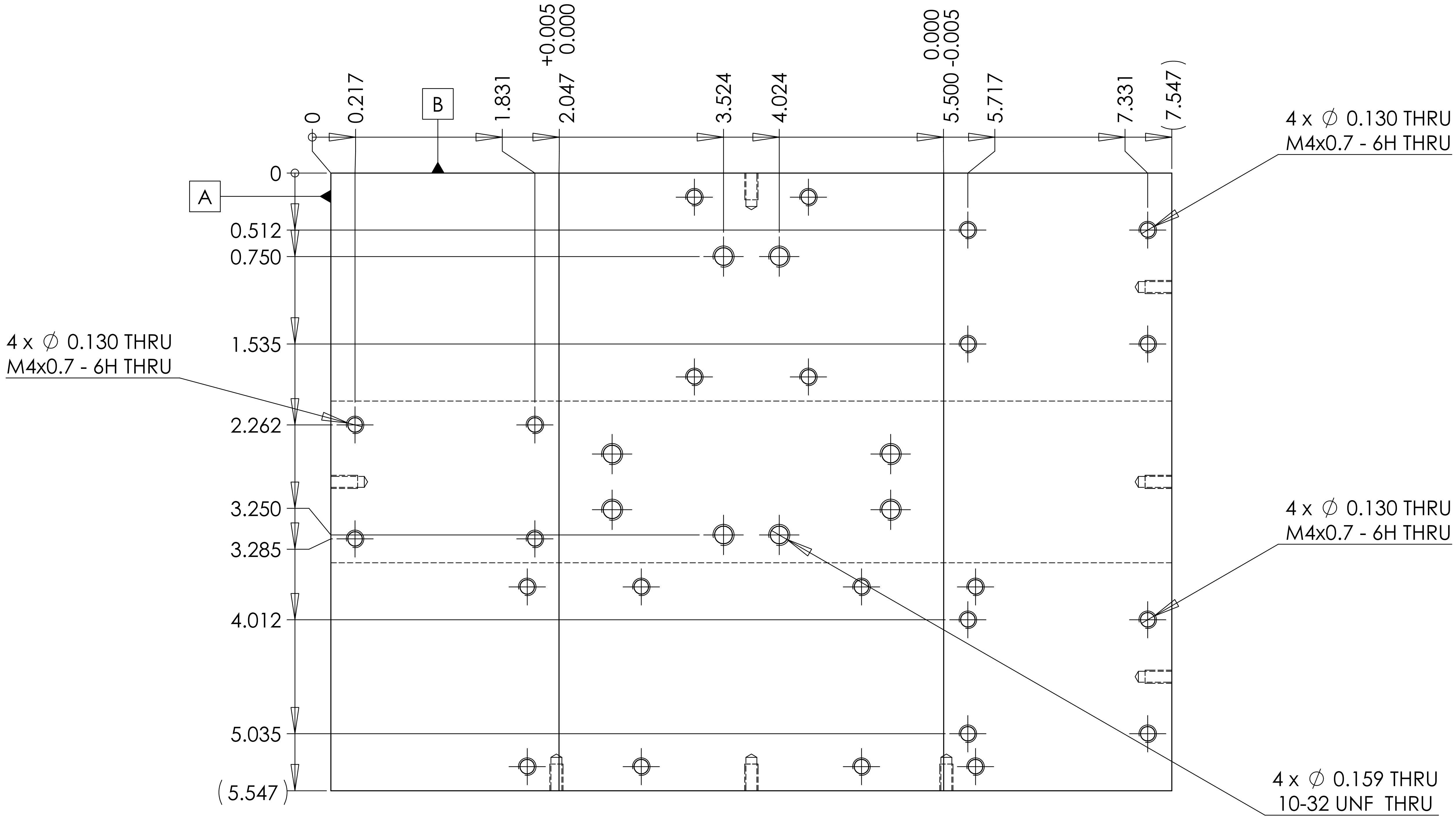
APPLICATION

TITLE:

BearingBlockMountPlate - FinalDesign

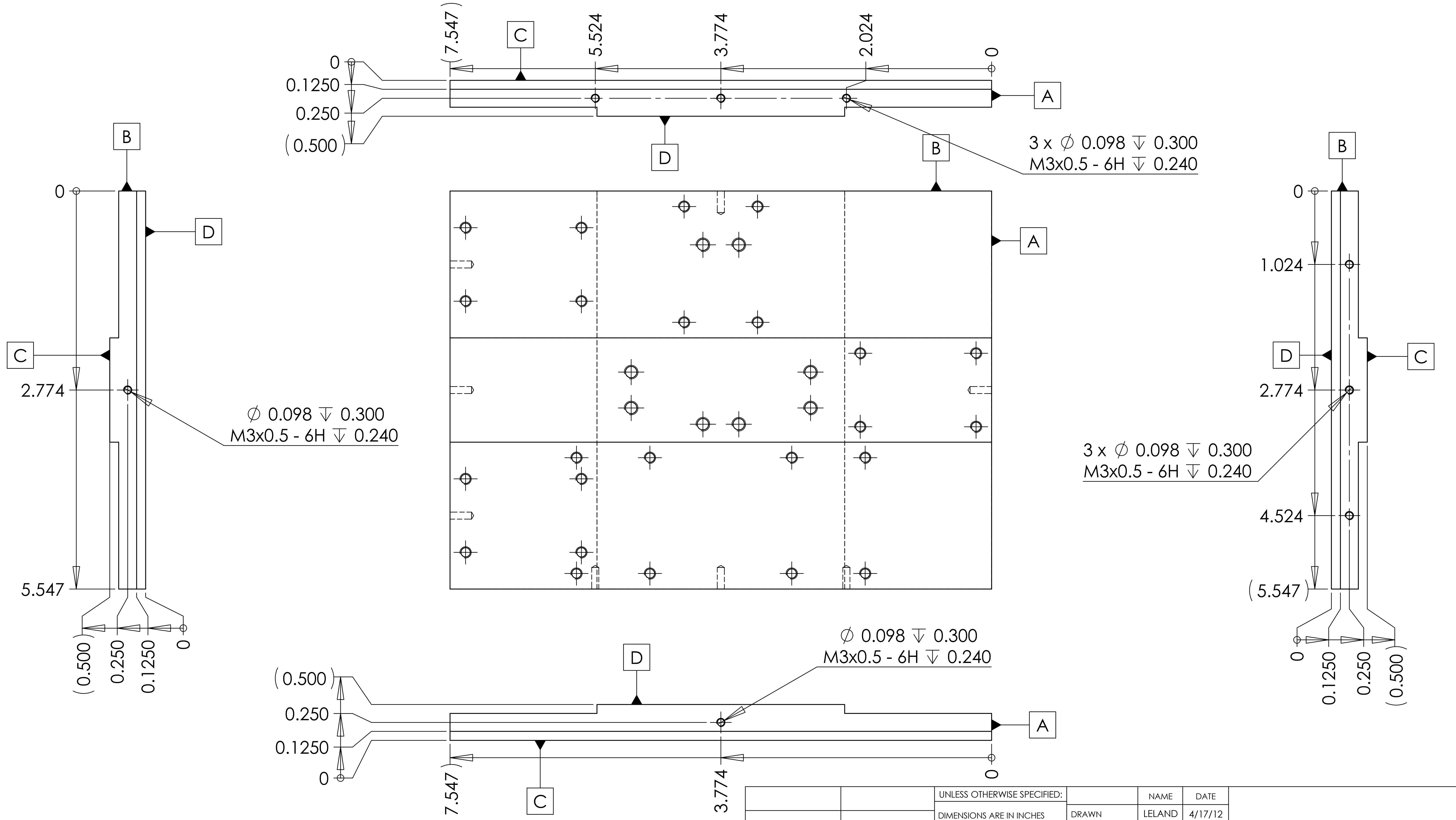
SIZE B | DWG. NO. | REV

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 3



UNLESS OTHERWISE SPECIFIED:		NAME	DATE		
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/17/12	TITLE:	
	CHECKED			BearingBlockMountPlate - FinalDesign	
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL	COMMENTS:			SIZE B	DWG. NO. REV
FINISH					
DO NOT SCALE DRAWING				SCALE: 1:1 WEIGHT:	SHEET 2 OF 3

NEXT ASSY	USED ON
APPLICATION	



UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

NEXT ASSY | USED ON

APPLICATION

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	4/17/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

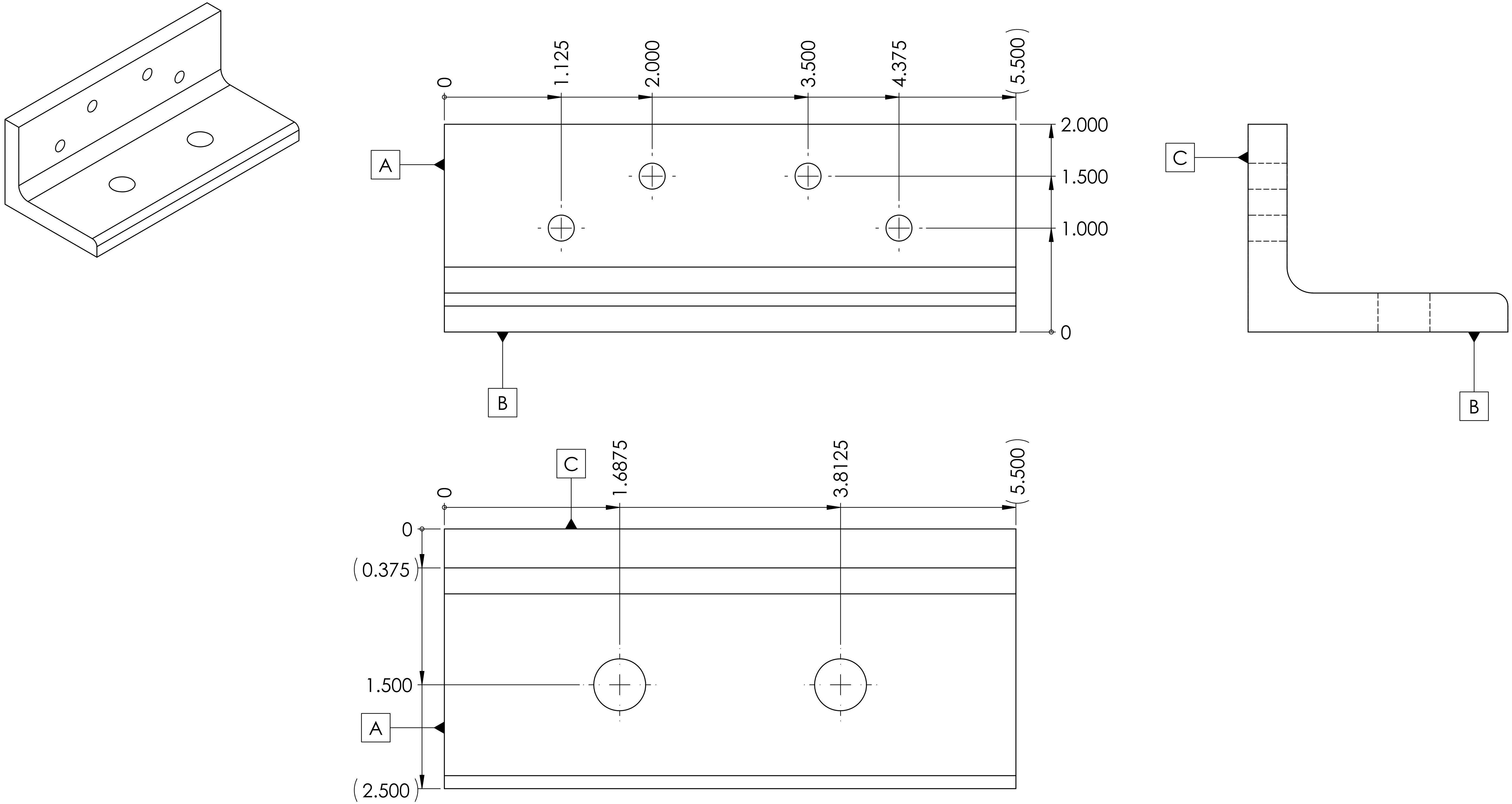
COMMENTS:

TITLE:

BearingBlockMountPlate - FinalDesign

SIZE B | DWG. NO. | REV

SCALE: 3:4 | WEIGHT: | SHEET 3 OF 3



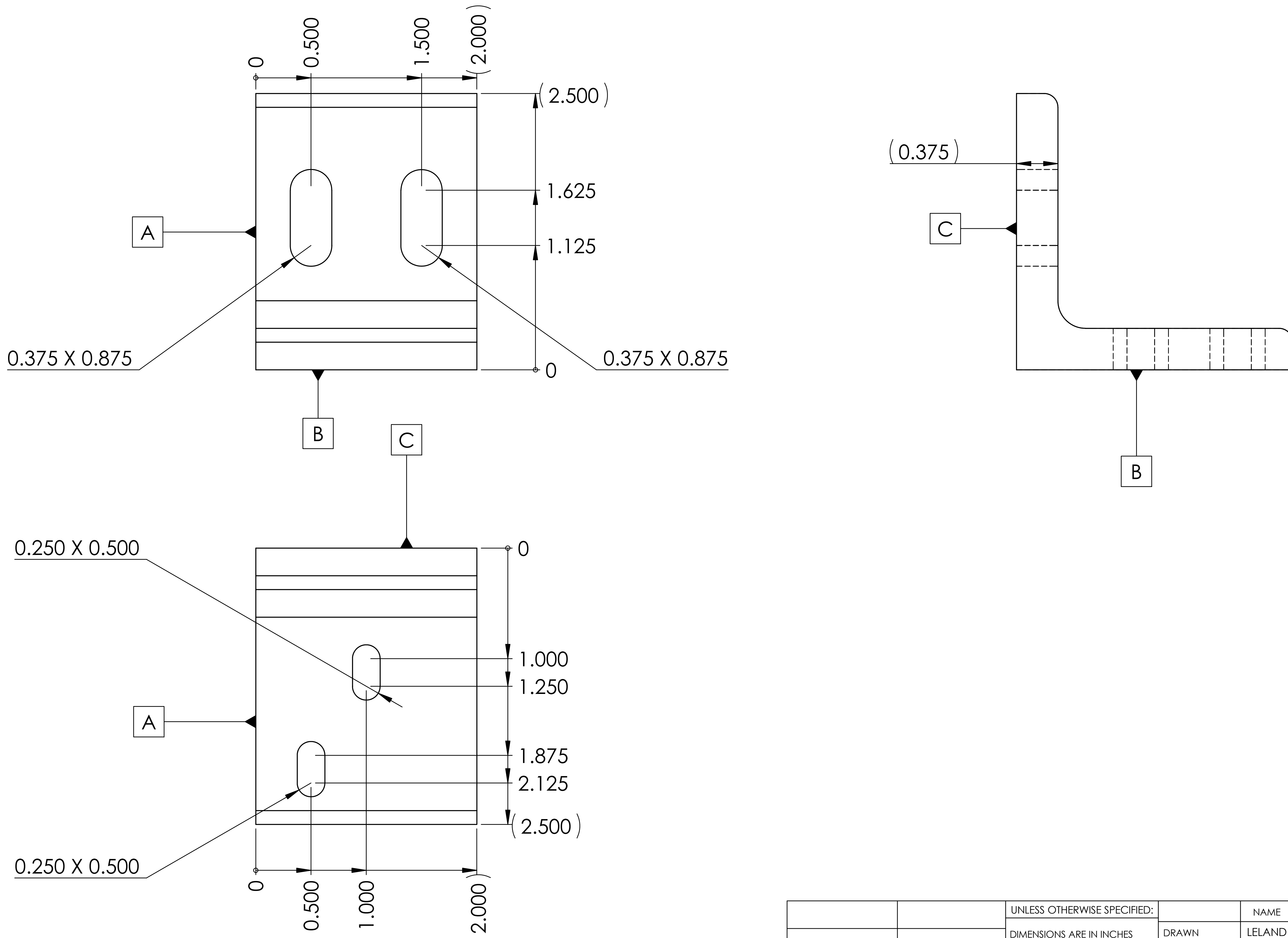
		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/12/12
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS: Make 1x		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE:

BottomStageMountAngle1

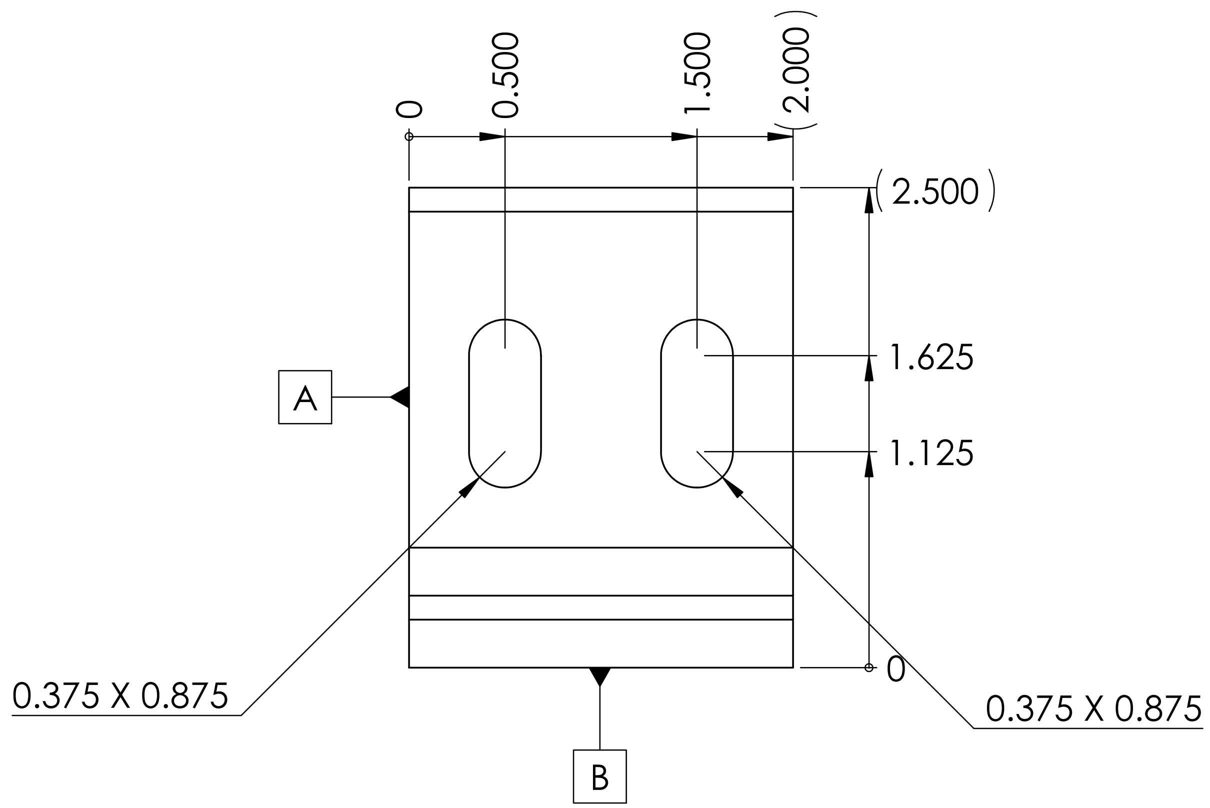
SIZE	DWG. NO.	REV
B		

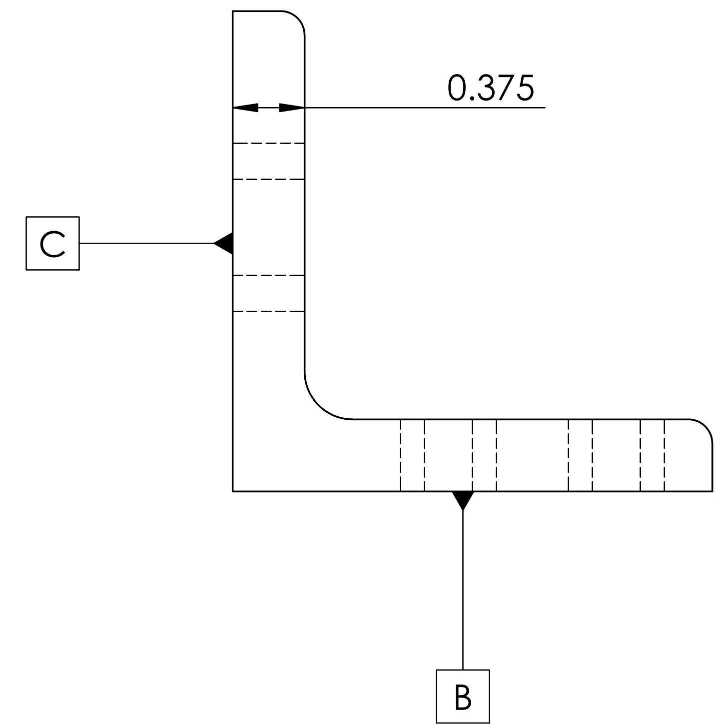
SCALE: 1:1 WEIGHT: SHEET 1 OF 1

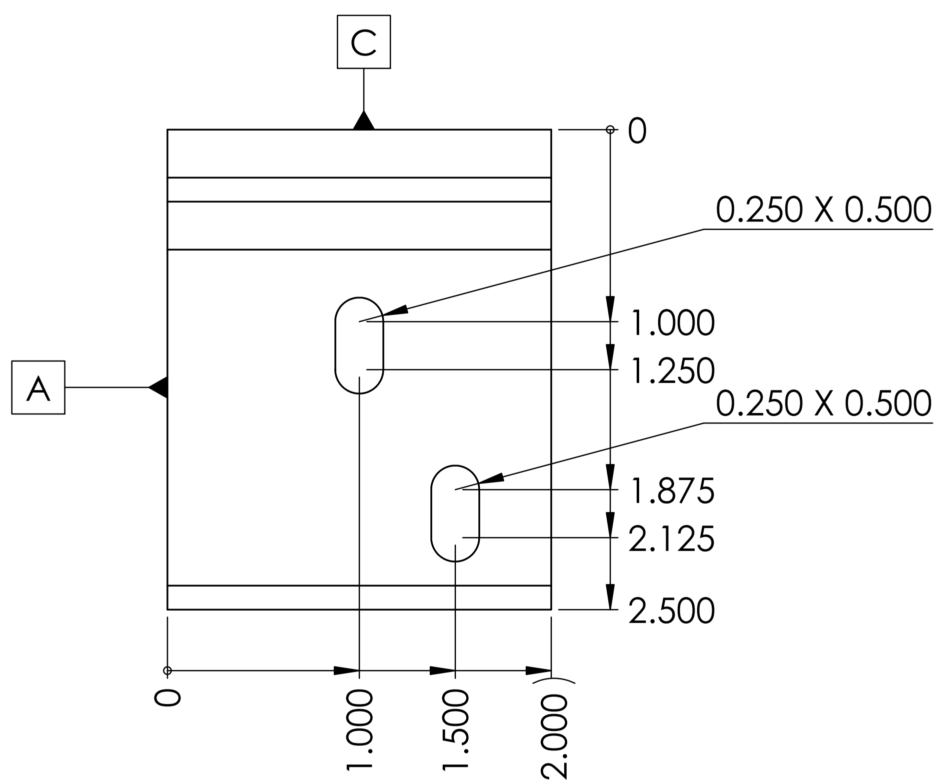


UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/12/12	TITLE:
	CHECKED			BottomStageMountAngle2
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS: Make 1x			SIZE B · DWG. NO. · REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	







UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH

	NAME	DATE
DRAWN	LELAND	4/12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

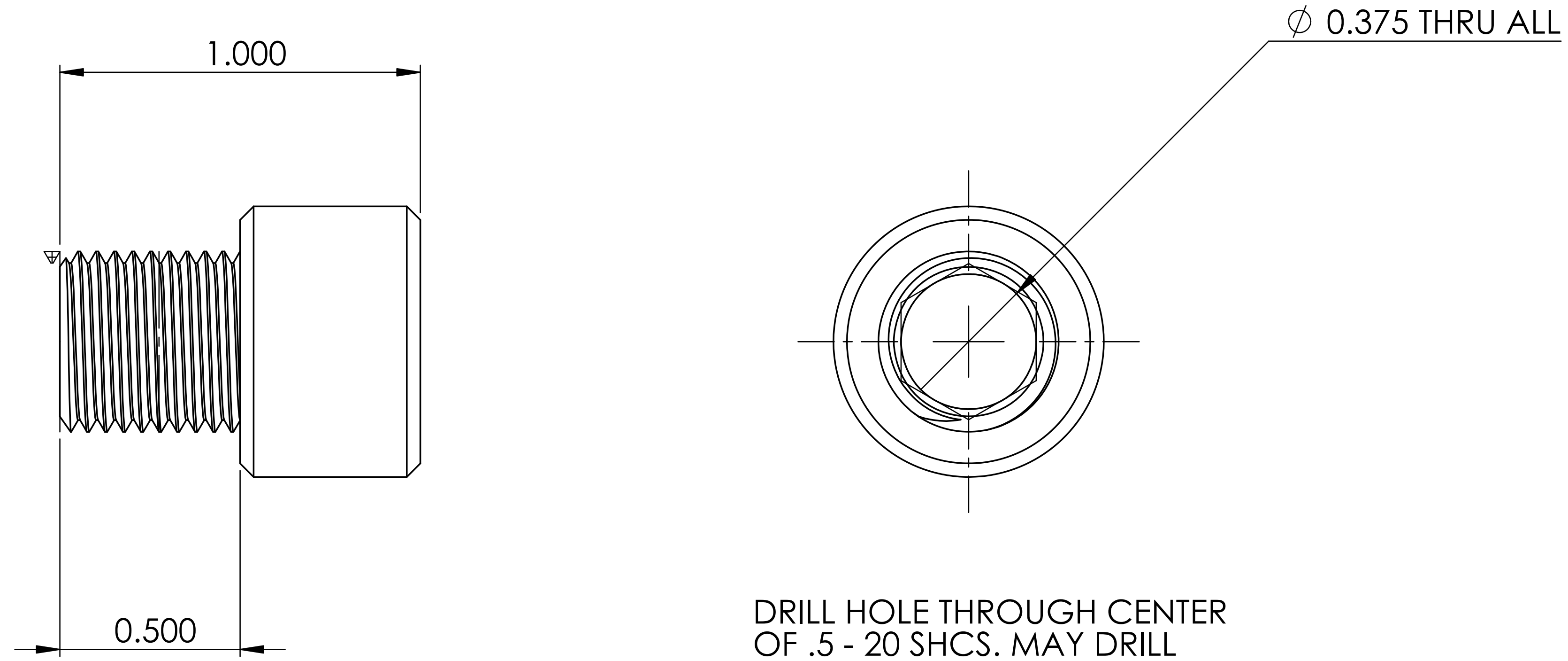
COMMENTS:
Make 1x

TITLE:
BottomStageMountAngle3

SIZE **B** DWG. NO. REV

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

NEXT ASSY | USED ON
APPLICATION
DO NOT SCALE DRAWING



DRILL HOLE THROUGH CENTER OF .5 - 20 SHCS. MAY DRILL SLIGHTLY OVERSIZE TO ALLOW CLEARANCE FOR LEAD SCREW.

UNLESS OTHERWISE SPECIFIED:	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	
INTERPRET GEOMETRIC TOLERANCING PER:	
MATERIAL	
FINISH	
DO NOT SCALE DRAWING	

	NAME	DATE
DRAWN	LELAND	5/3/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:
Make 2x

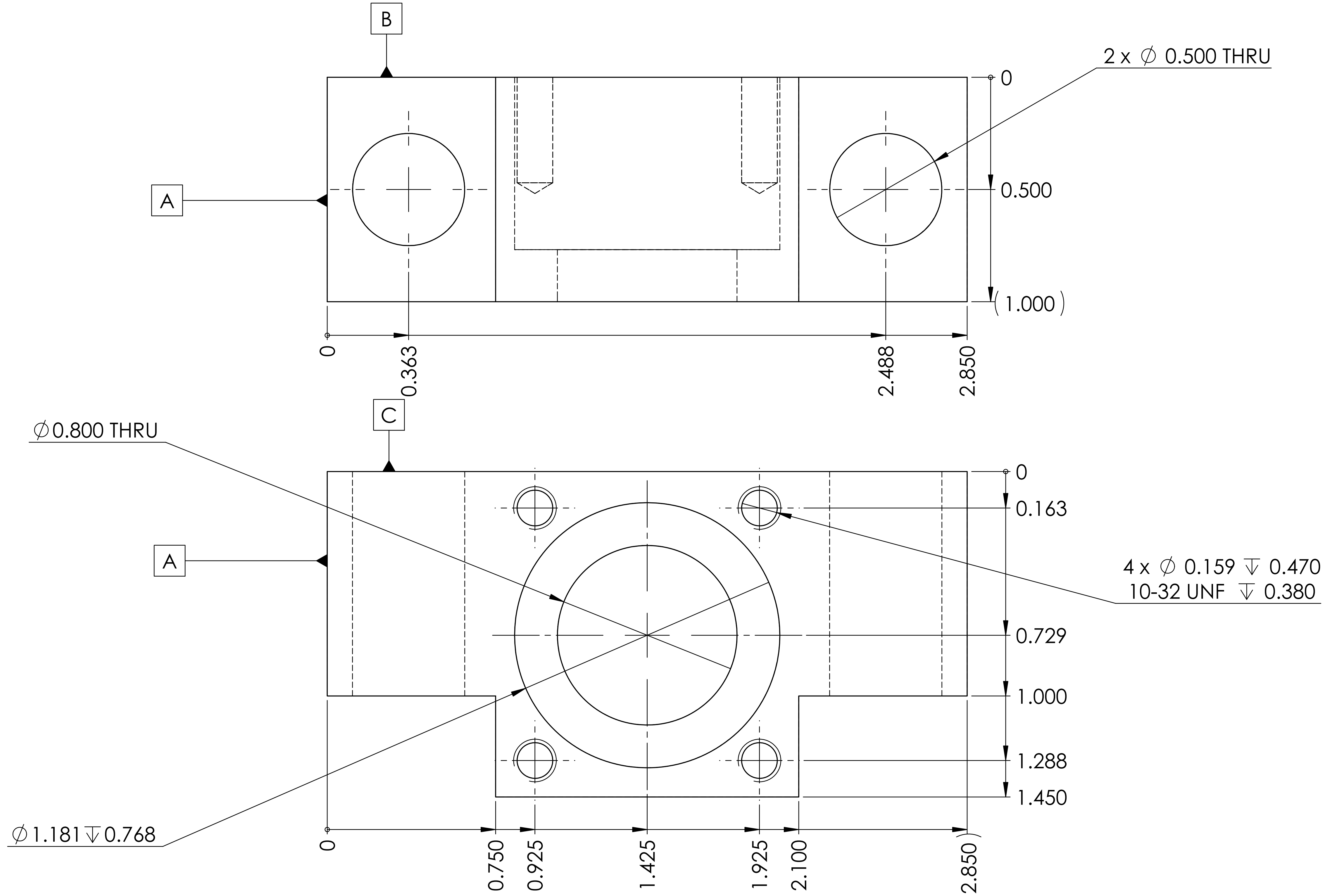
NEXT ASSY	USED ON
APPLICATION	

TITLE:

.5x20PushSCHS

SIZE	DWG. NO.	REV
B		

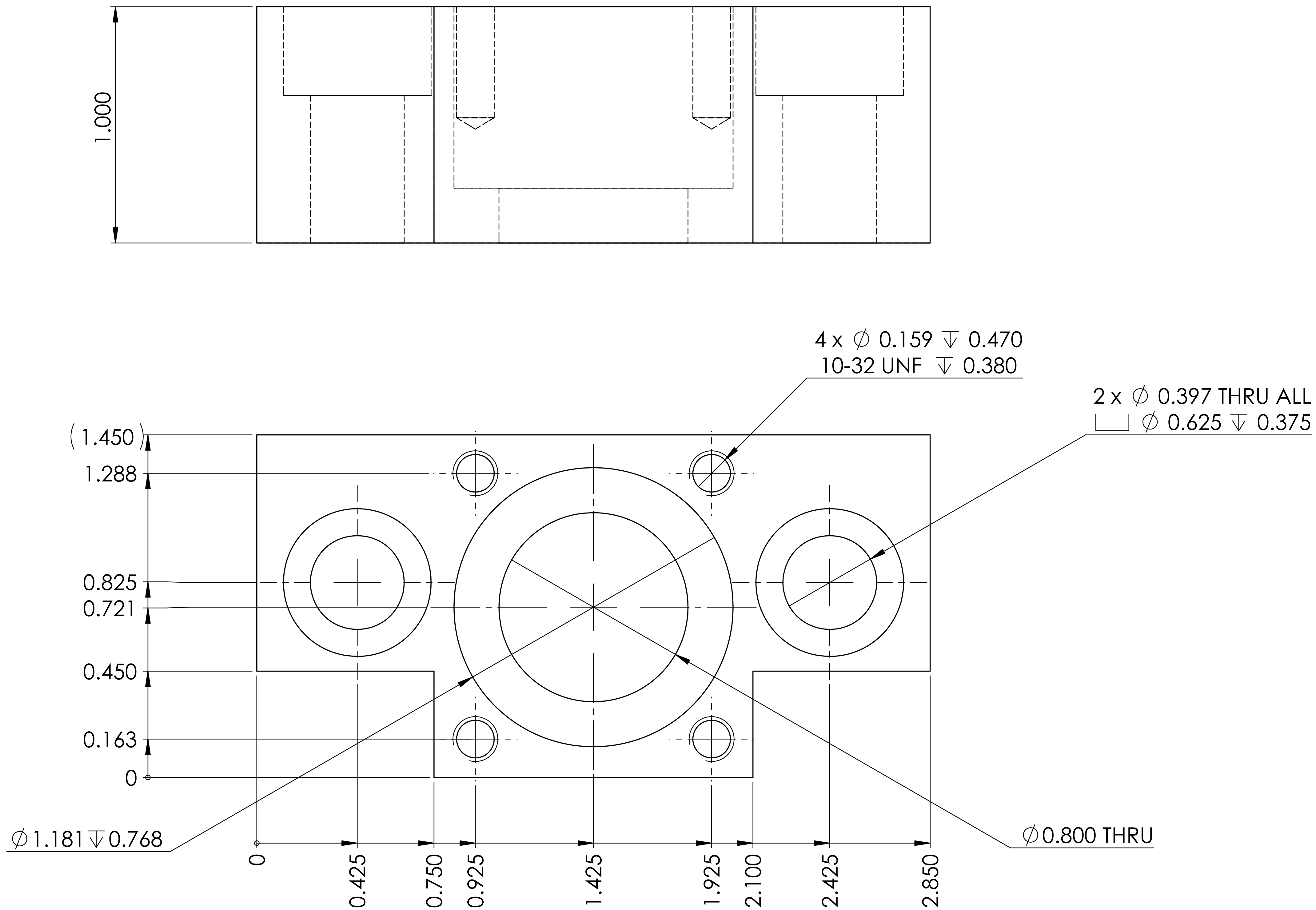
SCALE: 2:1 | WEIGHT: | SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/12/12	
	CHECKED			TITLE:
	ENG APPR.			LeadscrewThrustBearingMount - BottomStage
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS: Make 1x. Part is similar to LeadscrewThrustBearingMount			SIZE B · DWG. NO. · REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 2:1 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY | USED ON

APPLICATION



NEXT ASSY	USED ON	
APPLICATION		

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	4/11/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

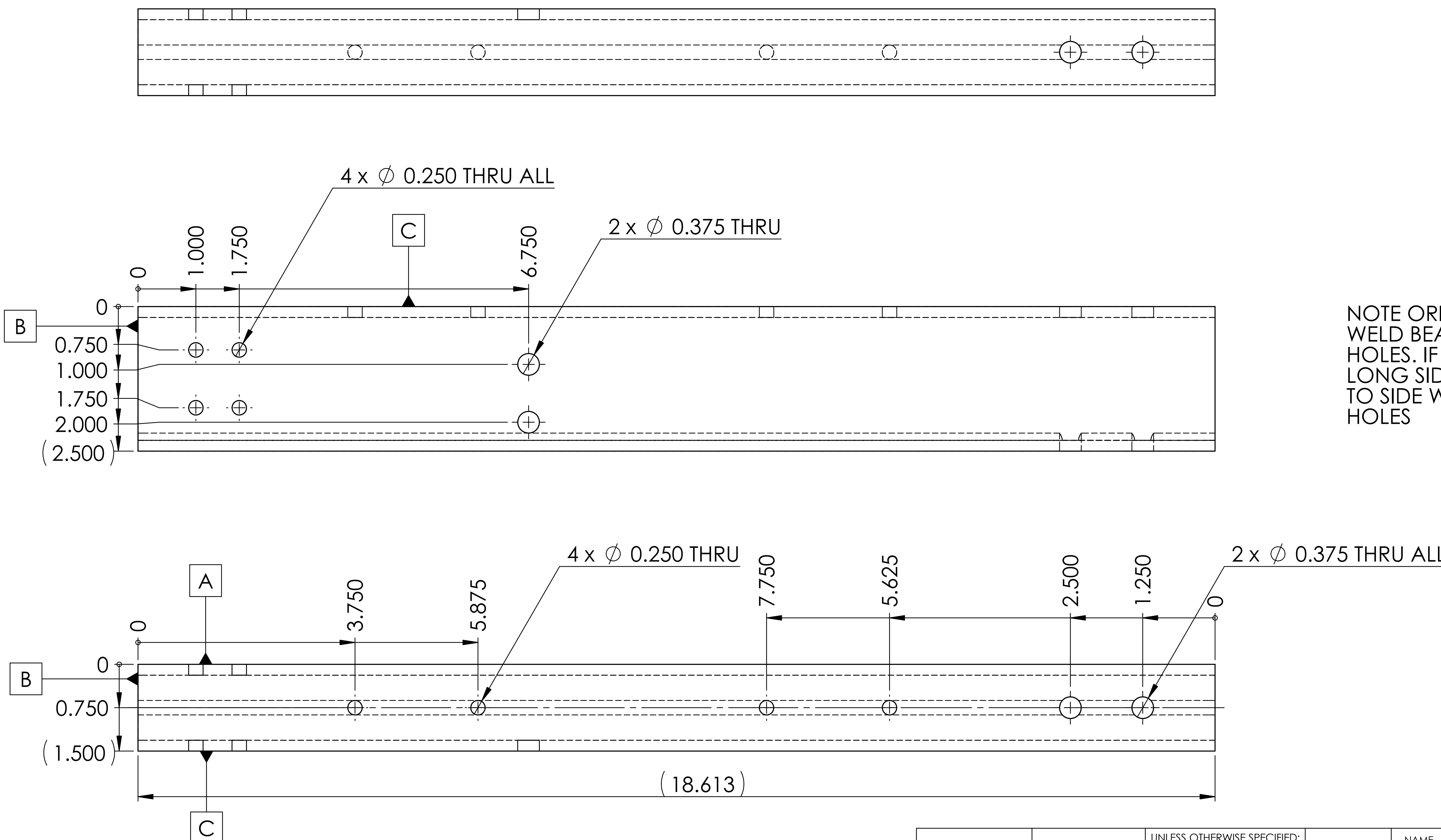
COMMENTS:
Make 1x. Part is similar to LeadscrewThrustBearingMount - BottomStage

TITLE:

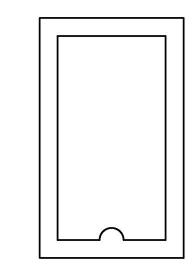
LeadscrewThrustBearingMount

SIZE	DWG. NO.	REV
B		

SCALE: 2:1 | WEIGHT: | SHEET 1 OF 1



NOTE ORIENTATION OF WELD BEAD RELATIVE TO HOLES. IF BEAD IS ON LONG SIDE OF PIPE, ORIENT TO SIDE WITH ONLY 4 HOLES



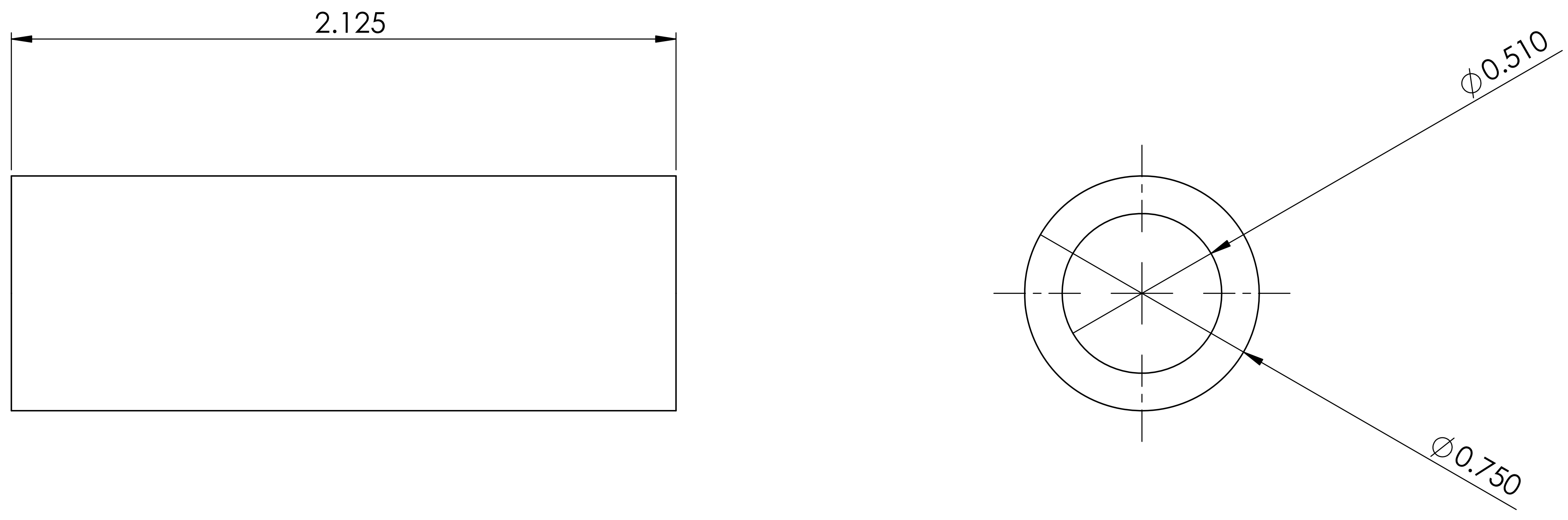
UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/6/12
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS: Make 1x		
FINISH			
DO NOT SCALE DRAWING			

NEXT ASSY	USED ON
APPLICATION	

TITLE:

LeftRailSupport - Final Design

SIZE	DWG. NO.		REV
B			
SCALE: 1:5	WEIGHT:	SHEET 1 OF 1	

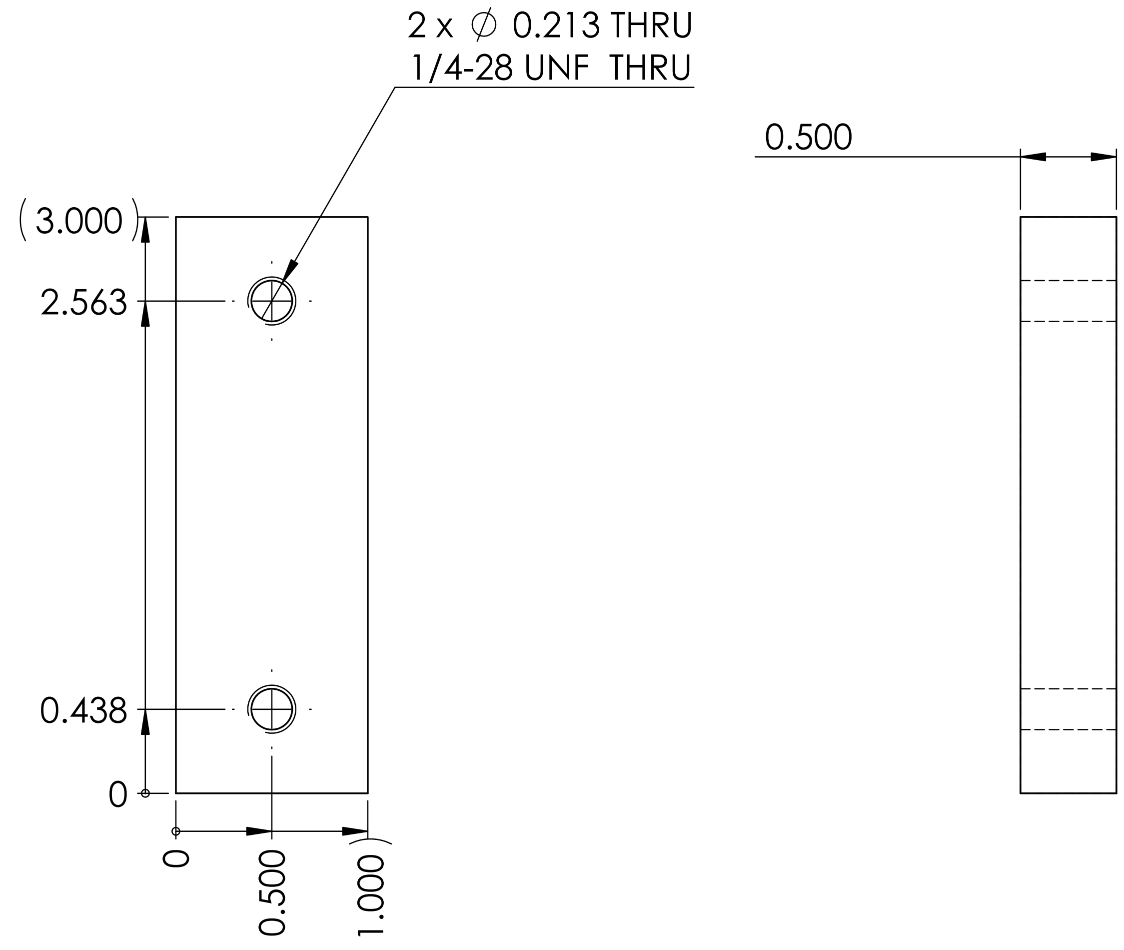


		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	5/3/12
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS: Make 18x		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE:

p75x2p125Sleeve

SIZE	DWG. NO.	REV
B		
SCALE: 2:1	WEIGHT:	SHEET 1 OF 1



NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	4/6/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

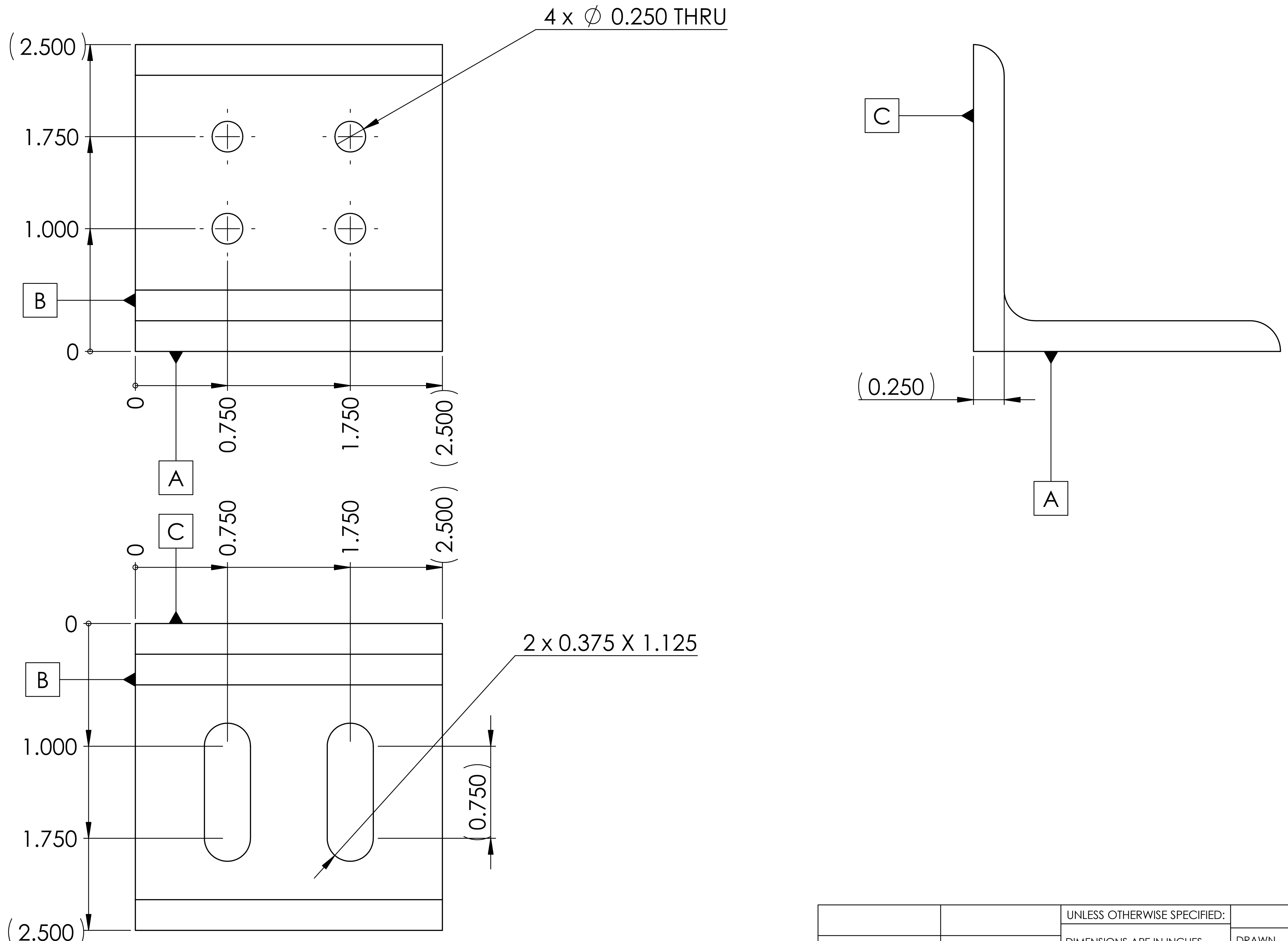
COMMENTS:
Make 4x

TITLE:

RailClampMountPlate

SIZE	DWG. NO.	REV
B		

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1

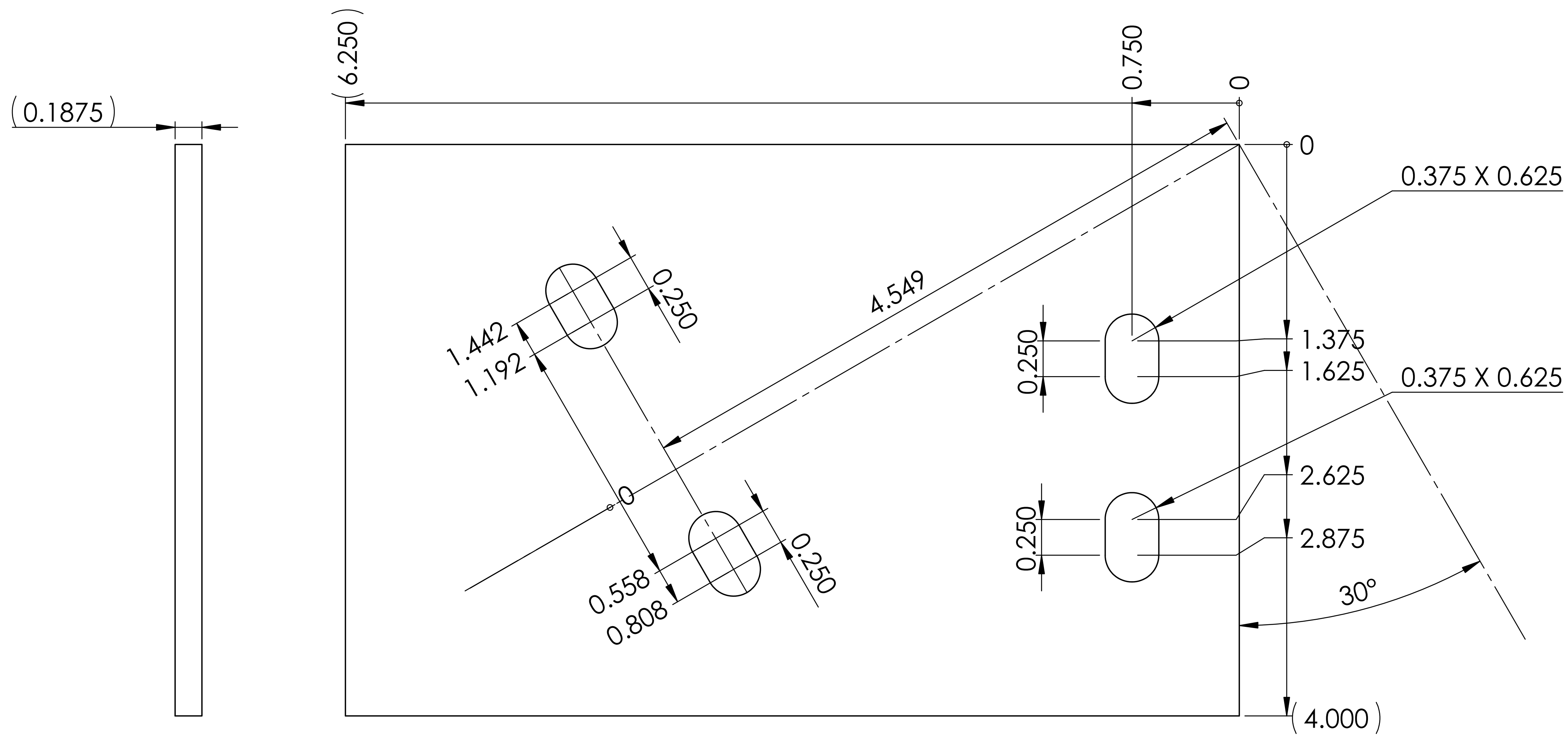


		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/6/12
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS: Make 4x		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE:

RailFrameAttachmentAngle - Final Design

SIZE	DWG. NO.		REV
B			
SCALE: 1:1	WEIGHT:	SHEET 1 OF 1	



ANGLE MEASUREMENT AND ACCOMPANYING CONSTRUCTION LINES ARE FOR USE WITH MANUAL ROTARY TABLE. ORIGIN OF PLATE SHOULD BE SET AT CENTER OF ROTARY TABLE, AND TABLE SHOULD BE ROTATED 30 DEG. TO CUT ANGLED SLOTS.

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/6/12
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS: Make 4x		
FINISH			
DO NOT SCALE DRAWING			

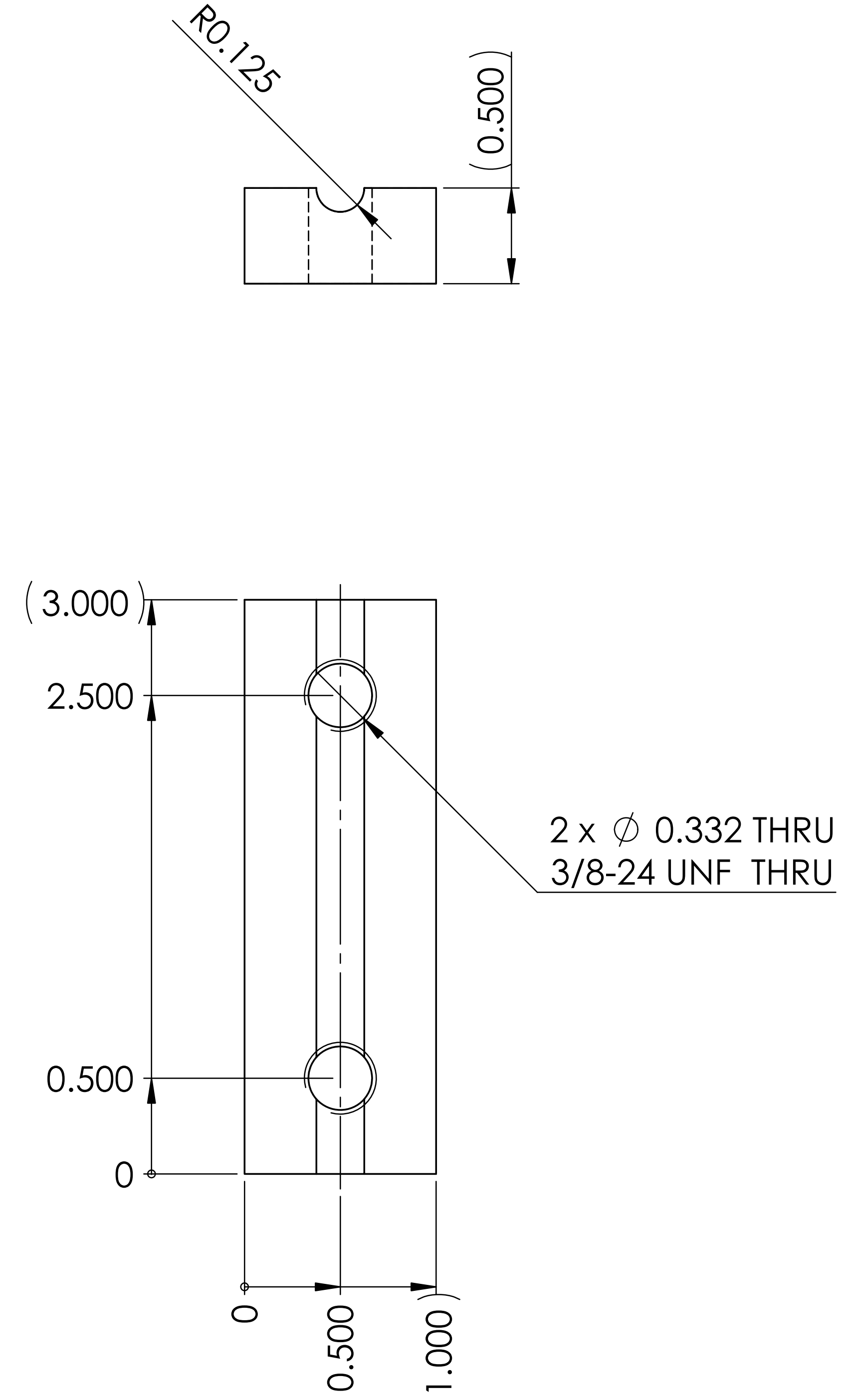
NEXT ASSY	USED ON
APPLICATION	

TITLE:

RailFrameAttachmentPlate - Final Design

SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		
INTERPRET GEOMETRIC TOLERANCING PER:		
MATERIAL		
FINISH		
NEXT ASSY	USED ON	
APPLICATION		DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	4/6/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

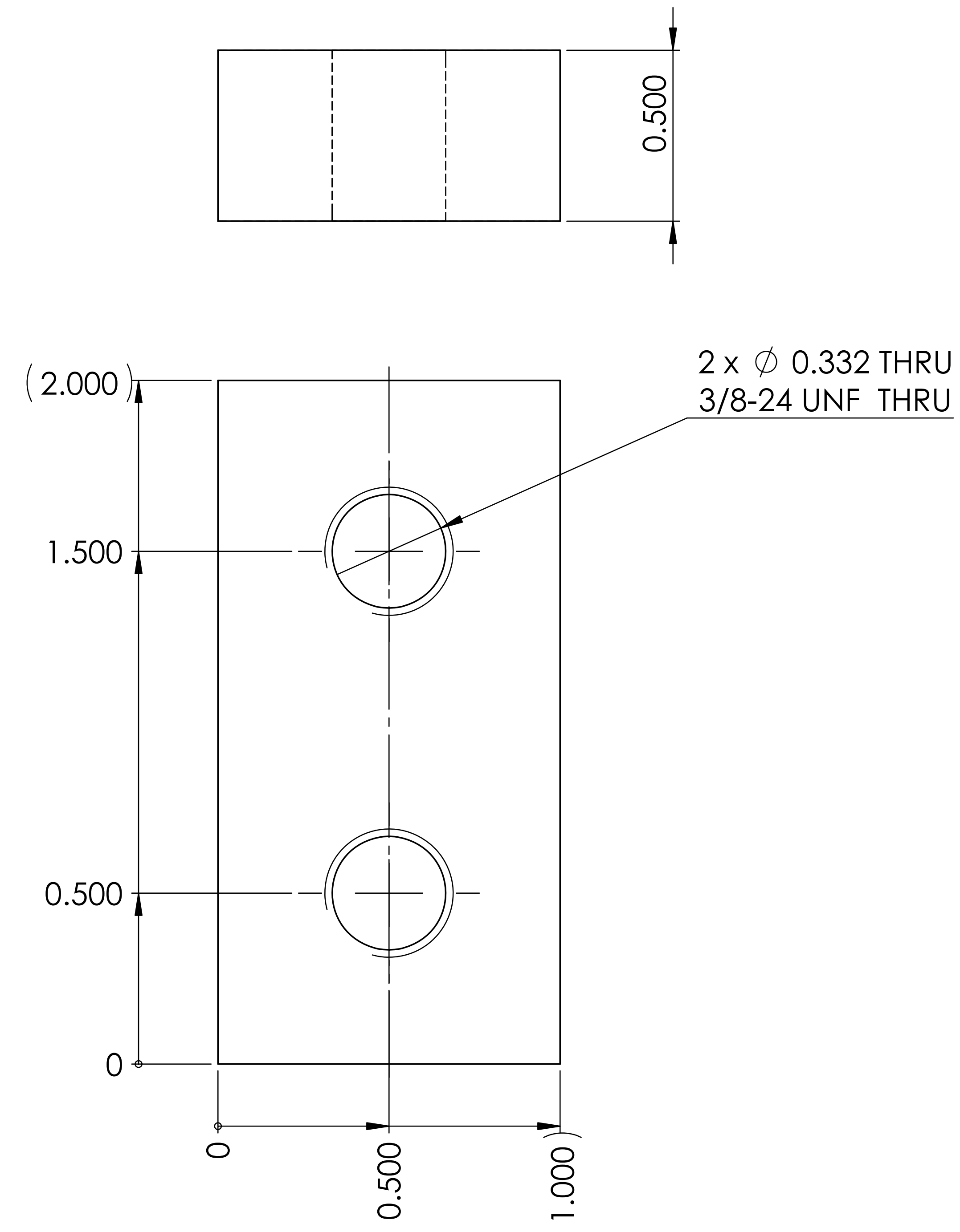
COMMENTS:
Make 4x

TITLE:

RailSupportMountBolt - BaseSupport

SIZE	DWG. NO.	REV
B		

SCALE: 1:1 WEIGHT: SHEET 1 OF 1



NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	4/6/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

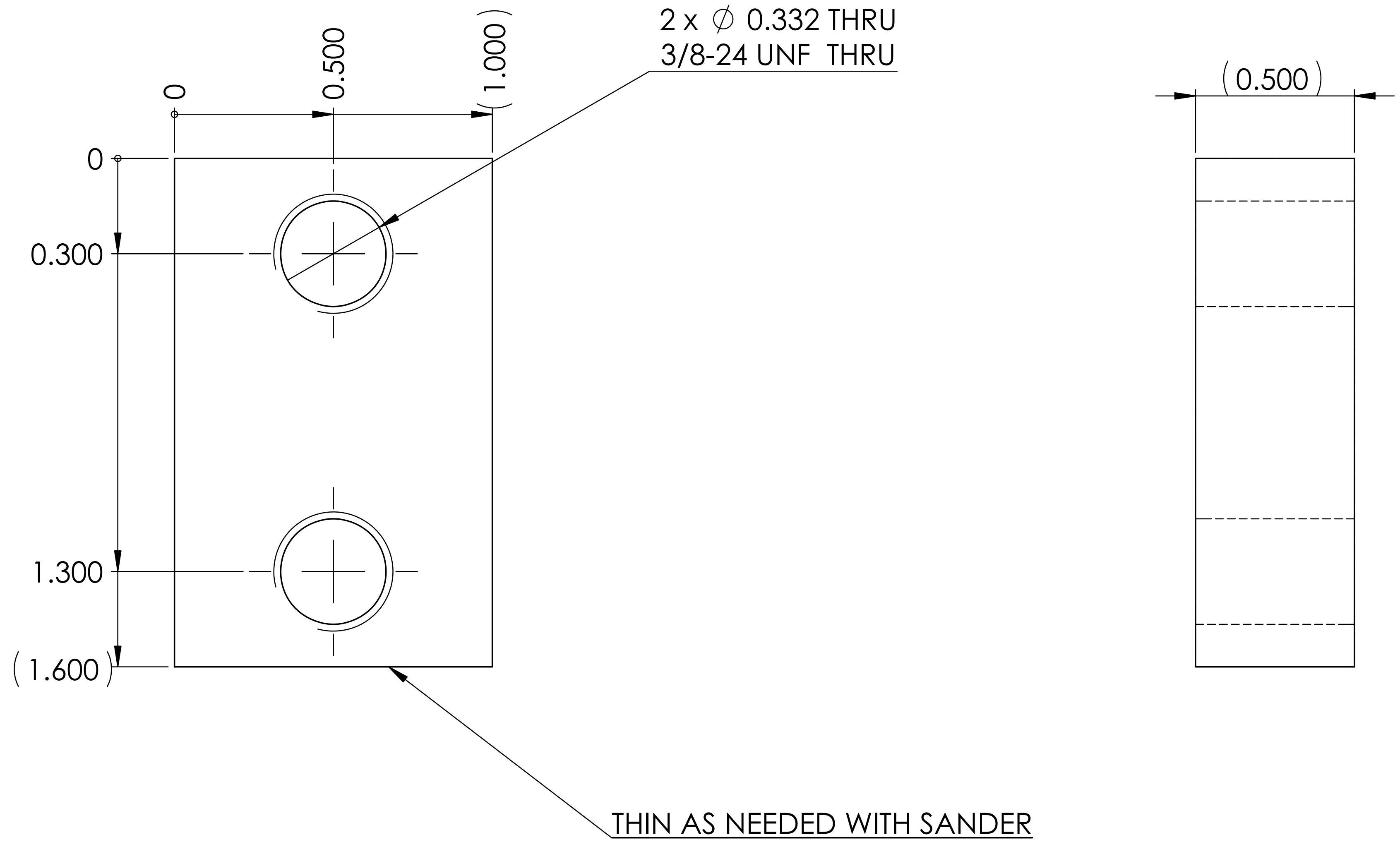
COMMENTS:
Make 4x

TITLE:

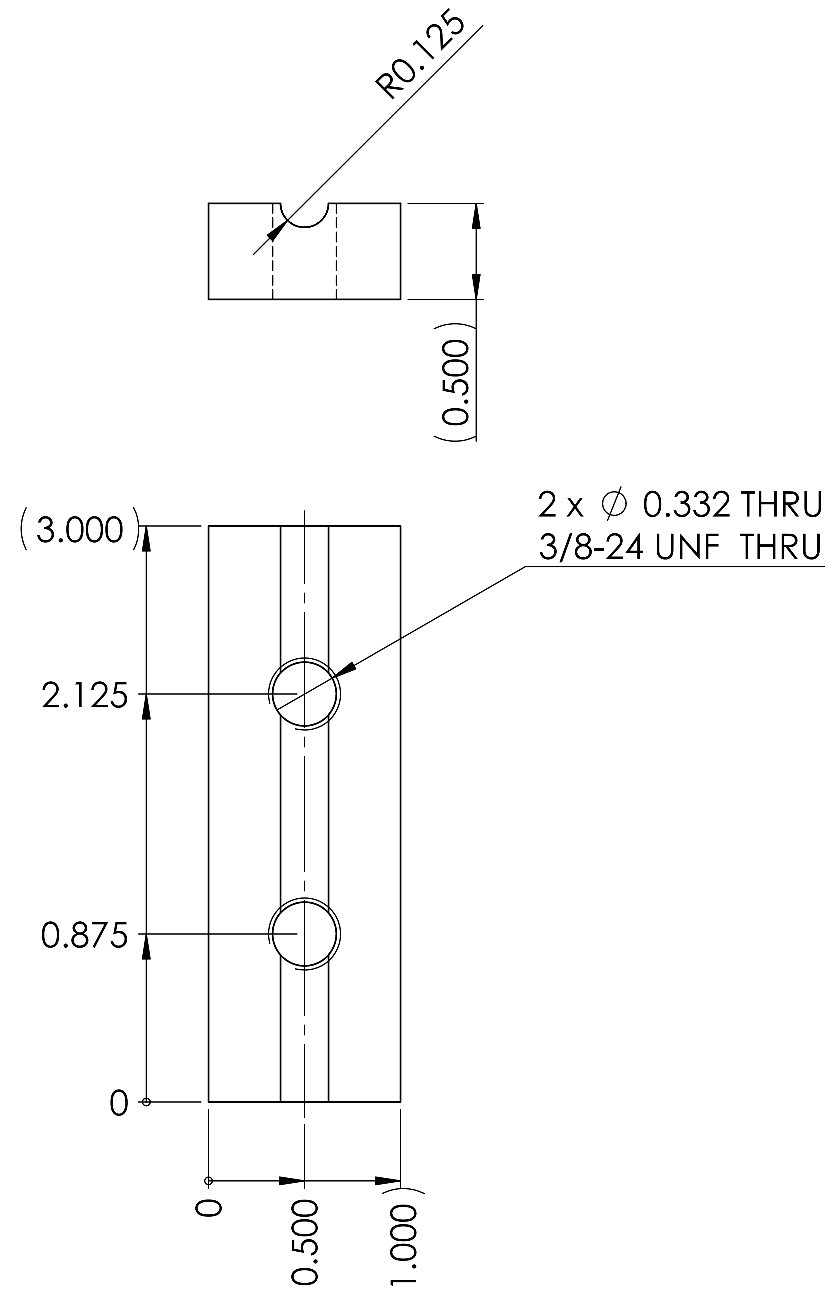
RailSupportMountBolt - BaseSupportV2

SIZE	DWG. NO.	REV
B		

SCALE: 2:1 WEIGHT: SHEET 1 OF 1

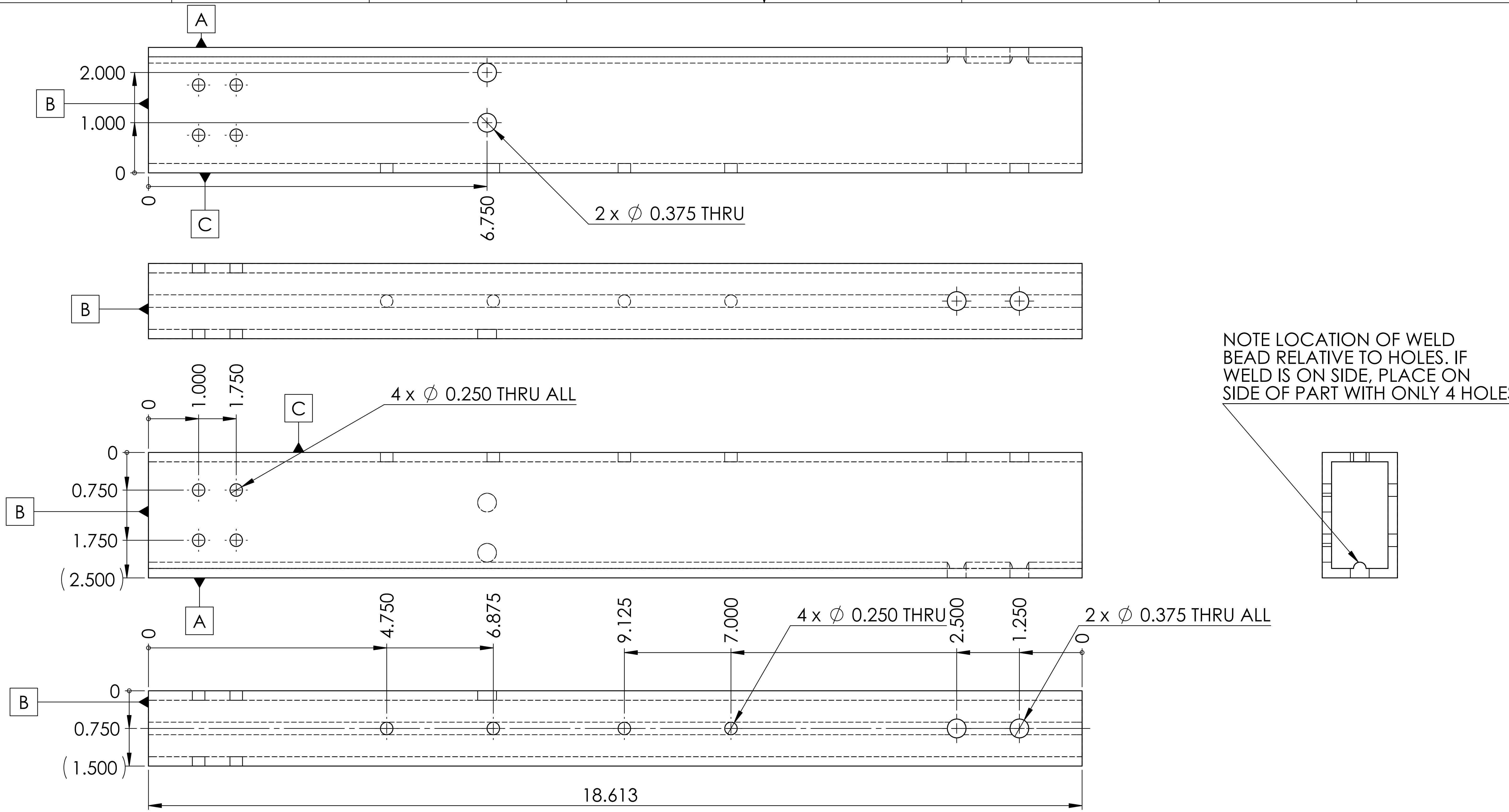


		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/10/12	TITLE:
			CHECKED			RailSupportMountBolt - LeadScrewSupport
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS: Make 2x			SIZE B · DWG. NO. · REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 2:1 · WEIGHT: · SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/6/12	TITLE:
	CHECKED			RailSupportMountBolt - RailMountSupport
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS: Make 4x			SIZE B · DWG. NO. · REV
FINISH				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1
DO NOT SCALE DRAWING				

NEXT ASSY	USED ON
APPLICATION	



	NAME	DATE
DRAWN	LELAND	4/6/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

NEXT ASSY | USED ON

APPLICATION

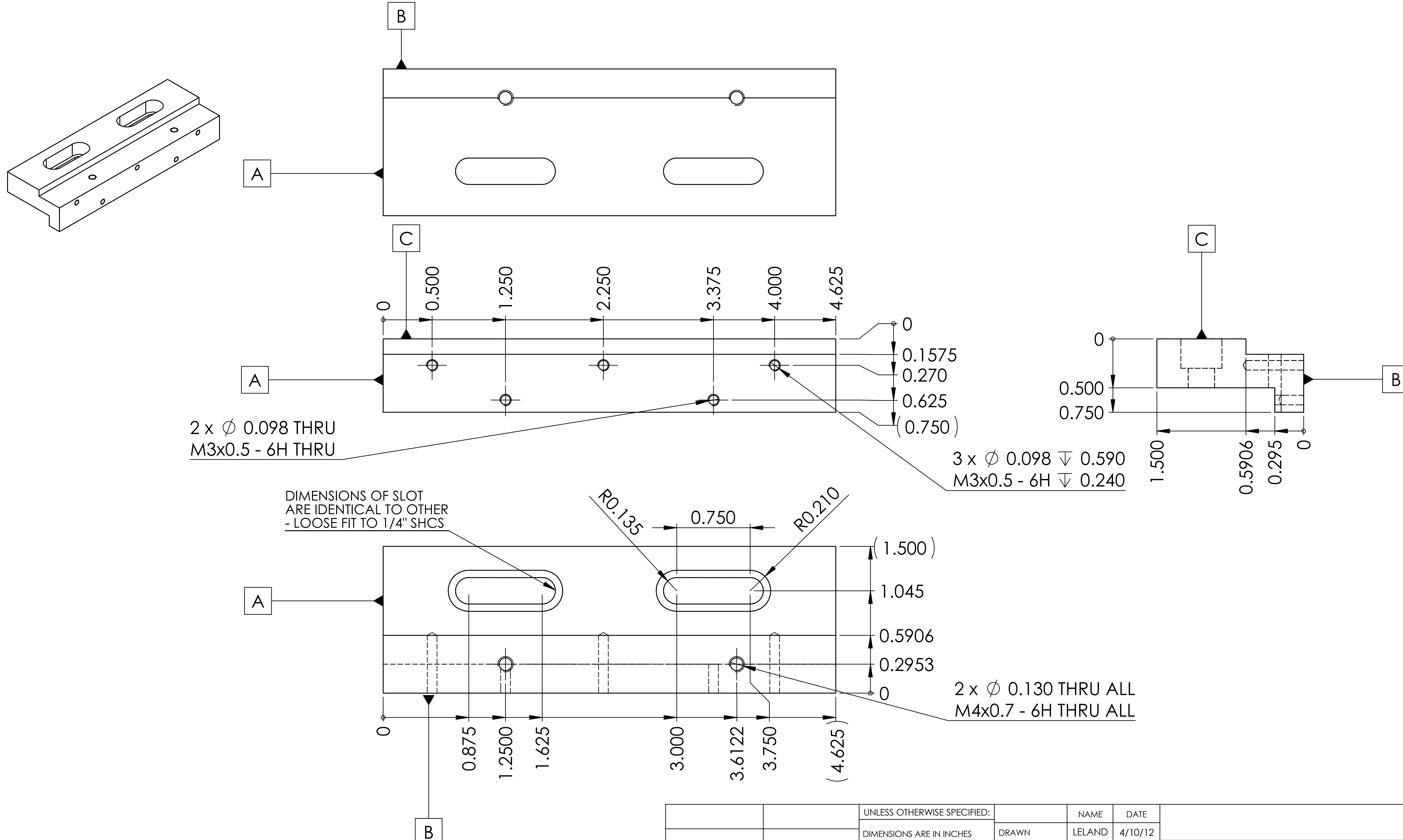
DO NOT SCALE DRAWING

COMMENTS:
Make 1x

TITLE:
RightRailSupport - Final Design

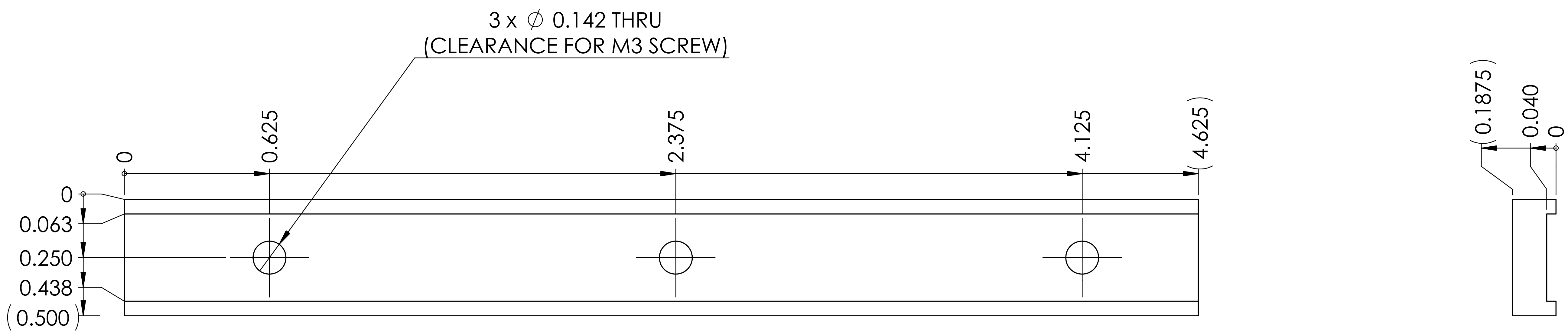
SIZE **B** | DWG. NO. | REV

SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE		
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/10/12	TITLE:	
	CHECKED			ShortBearingRailClampV2 - FinalDesign	
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL	COMMENTS: Make 4x			SIZE **B**	DWG. NO. / REV
FINISH					
DO NOT SCALE DRAWING				SCALE: 1:1	WEIGHT: / SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	



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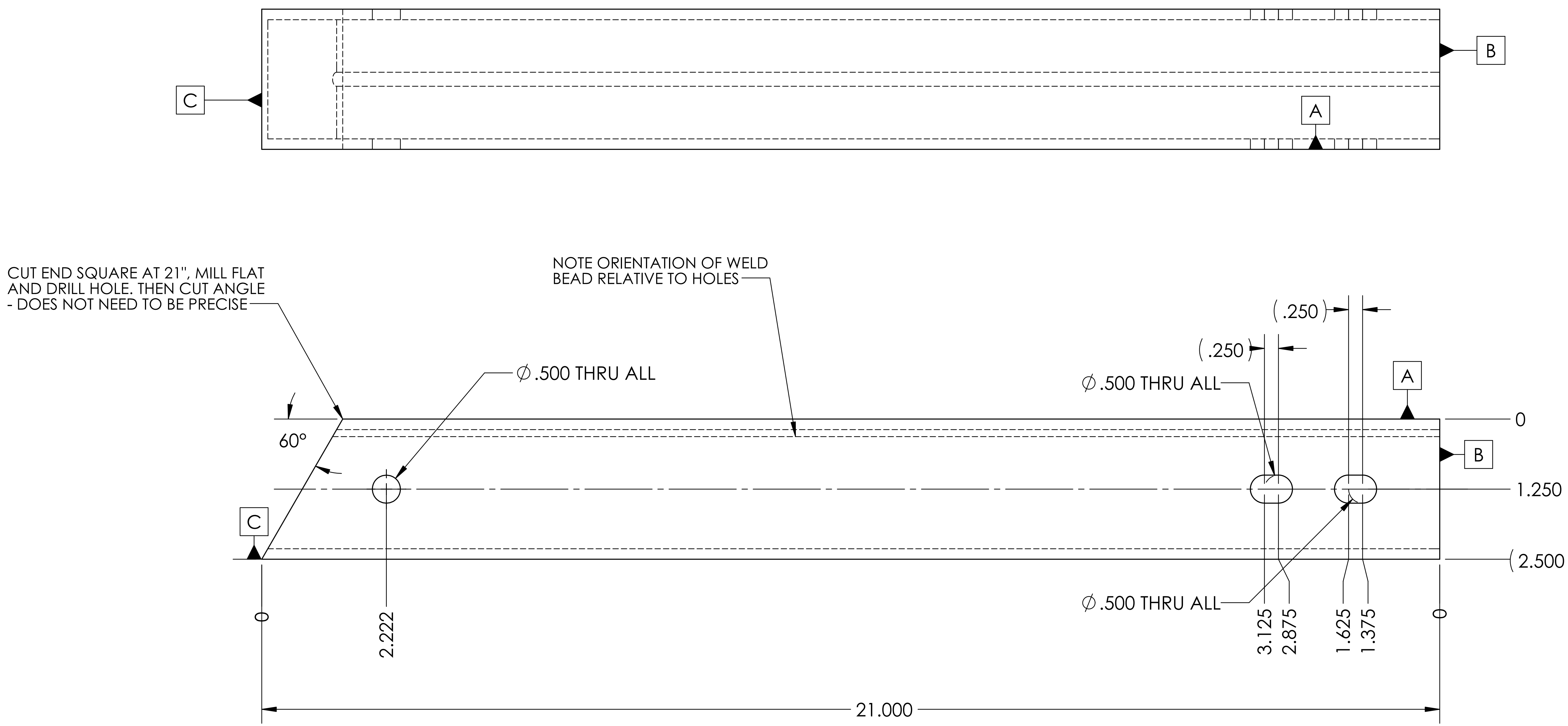
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/10/12	TITLE:
			CHECKED			ShortPushPlate - FinalDesign
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS: Make 10x			SIZE B / DWG. NO. / REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 2:1 / WEIGHT: / SHEET 1 OF 1



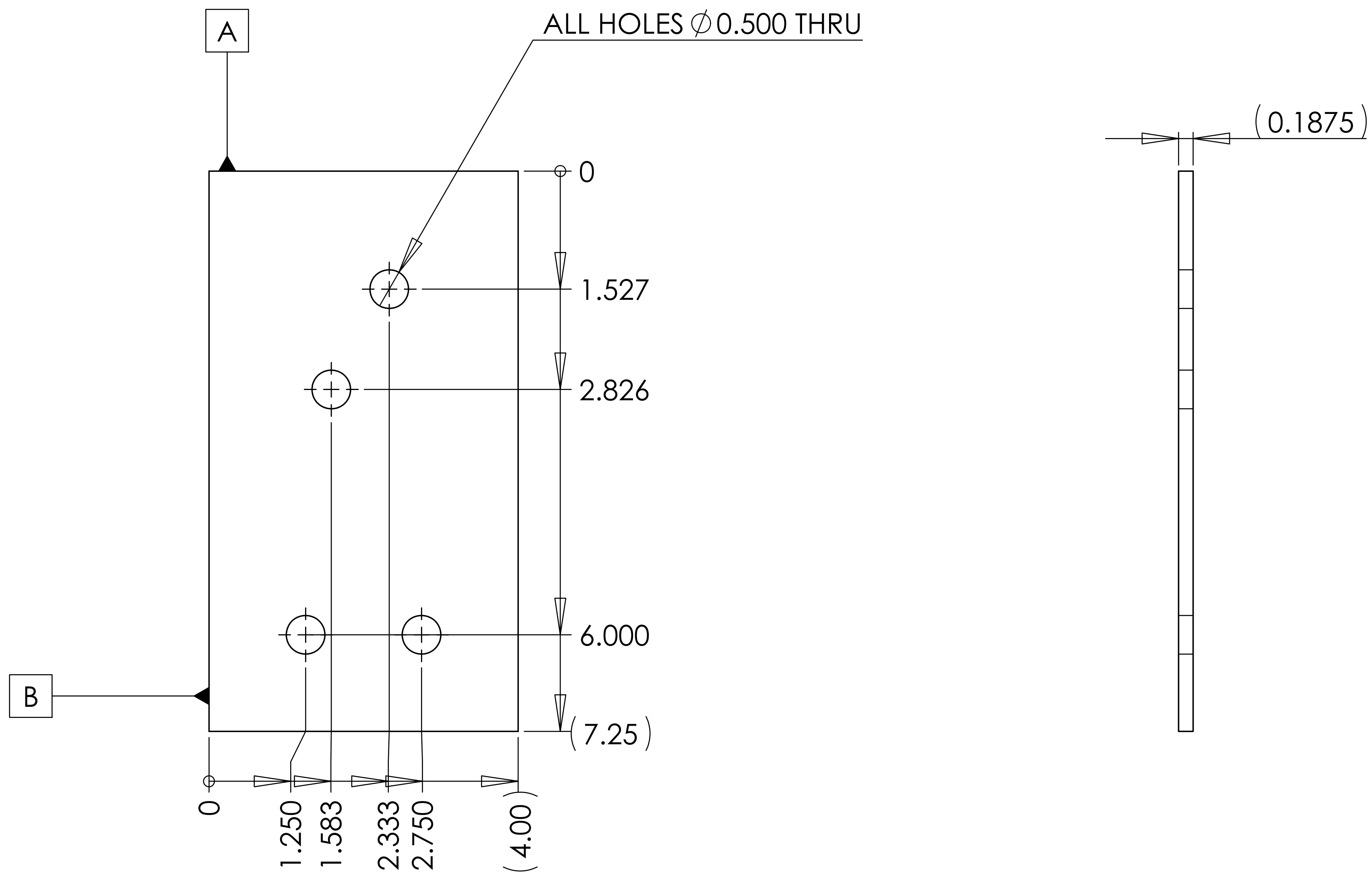
		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	Leland	3/12/12
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS: Make 6x		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE:

SideMember - Final Design

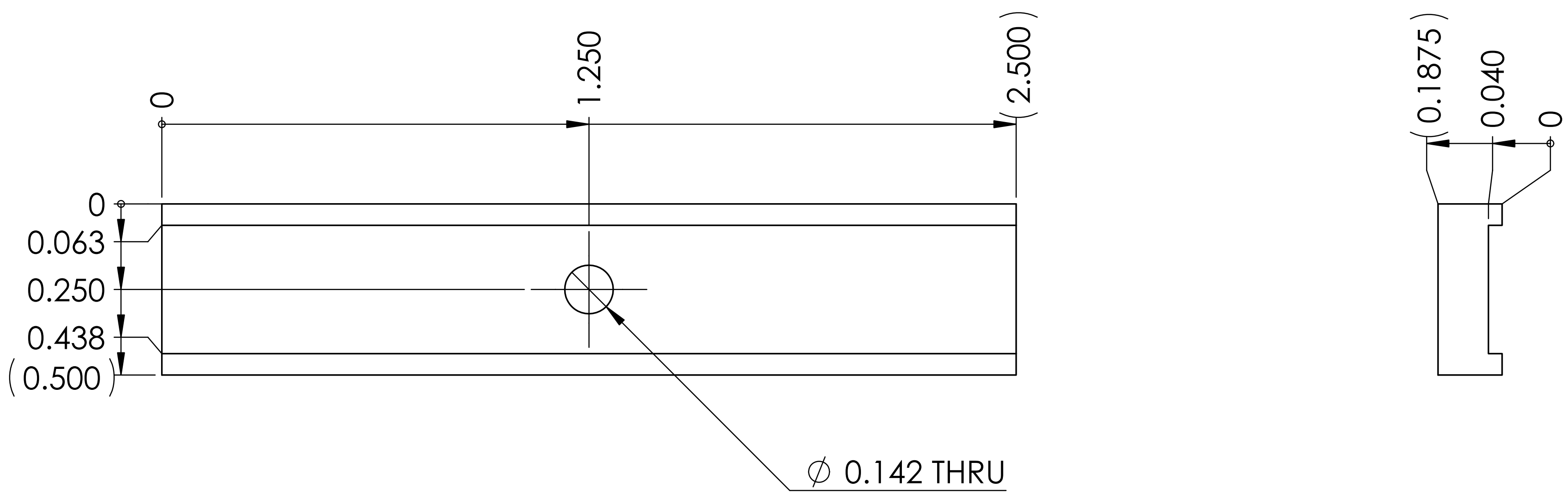
SIZE	DWG. NO.	REV
B		

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

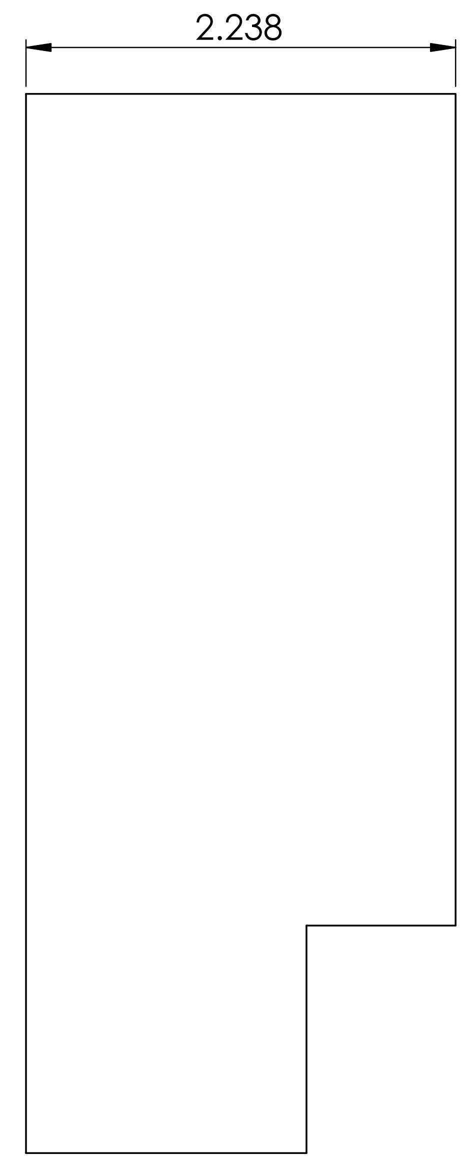


UNLESS OTHERWISE SPECIFIED:		NAME	DATE			
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	3/12/12	TITLE:		
	CHECKED			SidePlate - Final Design		
	ENG APPR.					
	MFG APPR.					
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.					
MATERIAL	COMMENTS: Make 12x			SIZE **B**	DWG. NO.	REV
FINISH						
DO NOT SCALE DRAWING				SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

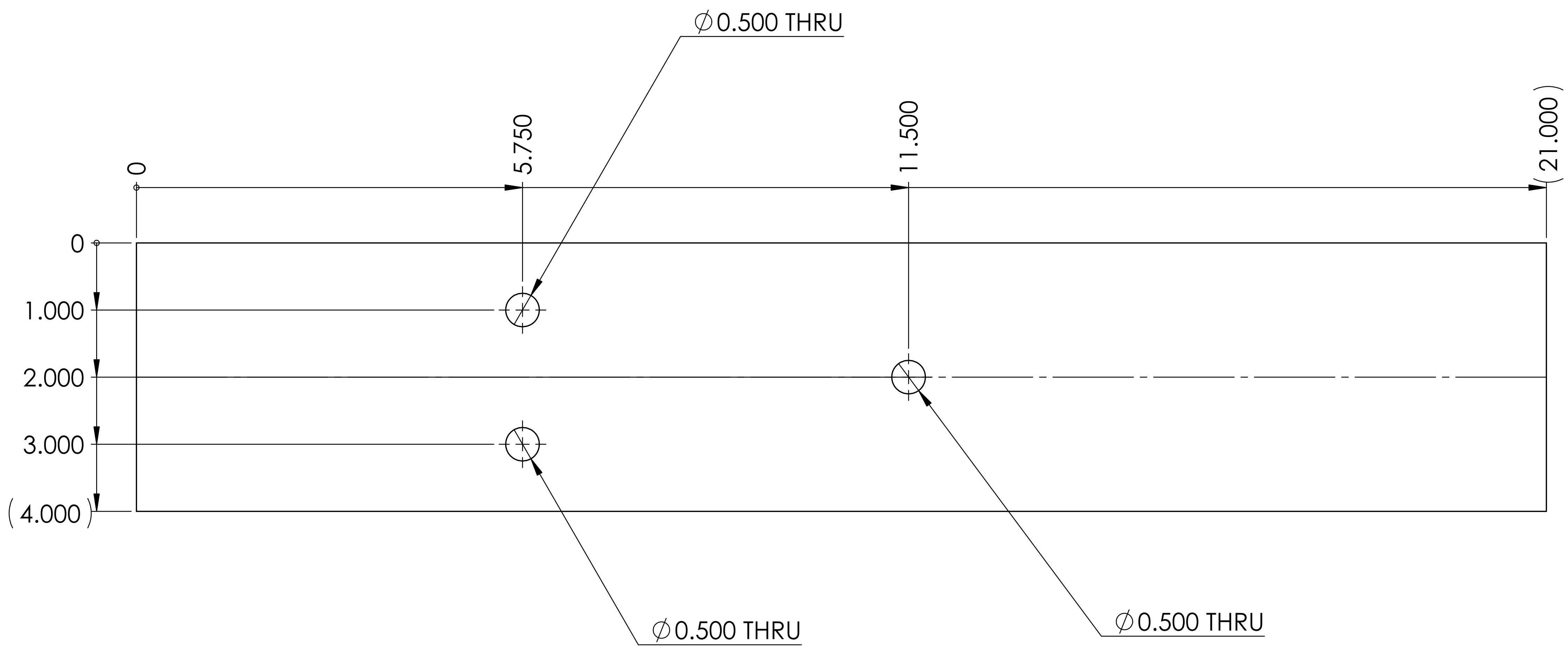


		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/12/12	
			CHECKED			TITLE:
			ENG APPR.			SingleHolePushPlate - FinalDesign
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS: Make 2x			SIZE B | DWG. NO. | REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 2:1 | WEIGHT: | SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE			
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	5/3/12	TITLE:		
	CHECKED			Spindle Gib Mount		
	ENG APPR.					
	MFG APPR.					
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.					
MATERIAL	COMMENTS: Make 1x. Cut existing gib mount to specified thickness. Purchase appropriate fasteners (M8 x 50)			SIZE **B**	DWG. NO.	REV
FINISH						
DO NOT SCALE DRAWING				SCALE: 1:1	WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	



CUT THIS END - SHOULD BE END OPPOSITE TO END WHERE RACK ATTACHES TO COLUMN

	NAME	DATE	
DRAWN	LELAND	4/6/12	TITLE:
CHECKED			SpindleColumn
ENG APPR.			
MFG APPR.			
Q.A.			

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

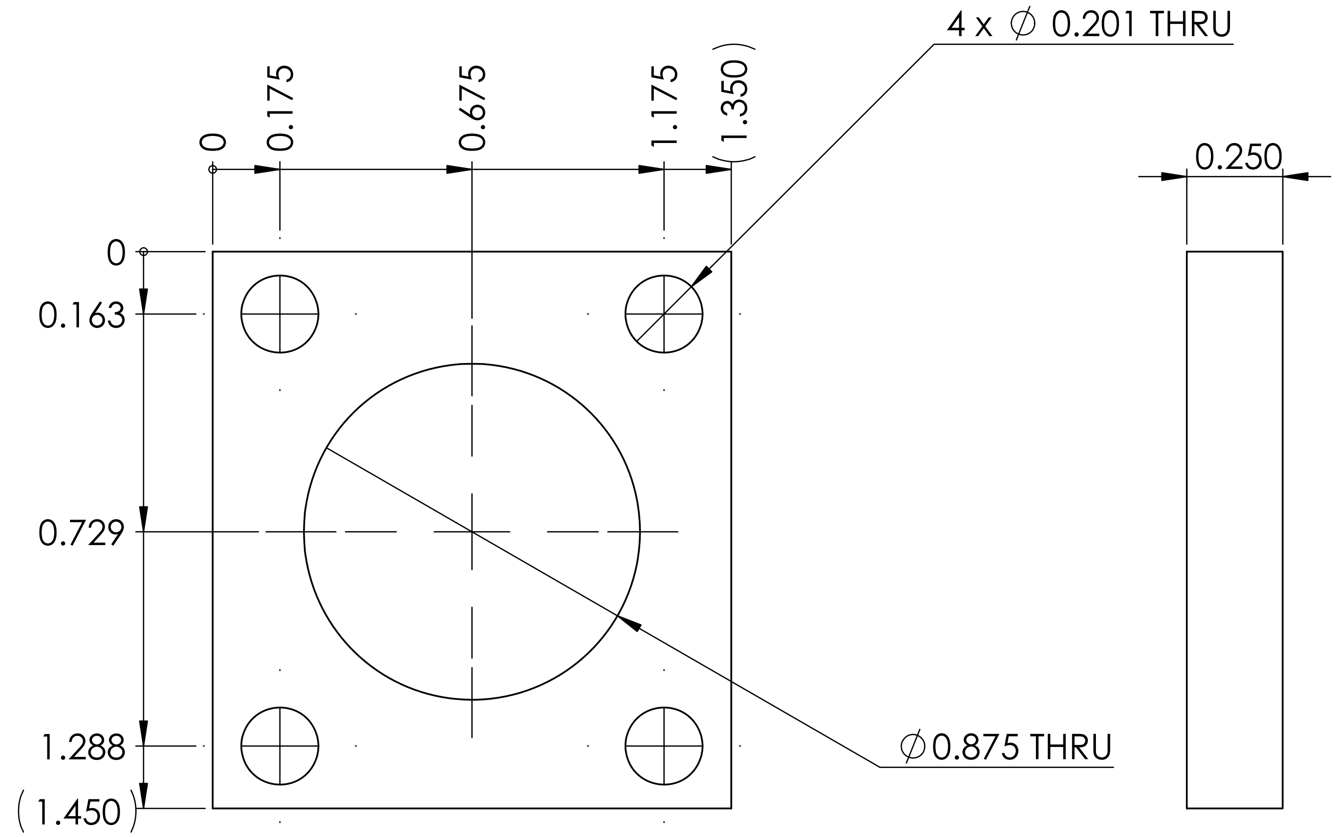
COMMENTS:
Make 1x

NEXT ASSY | USED ON

APPLICATION

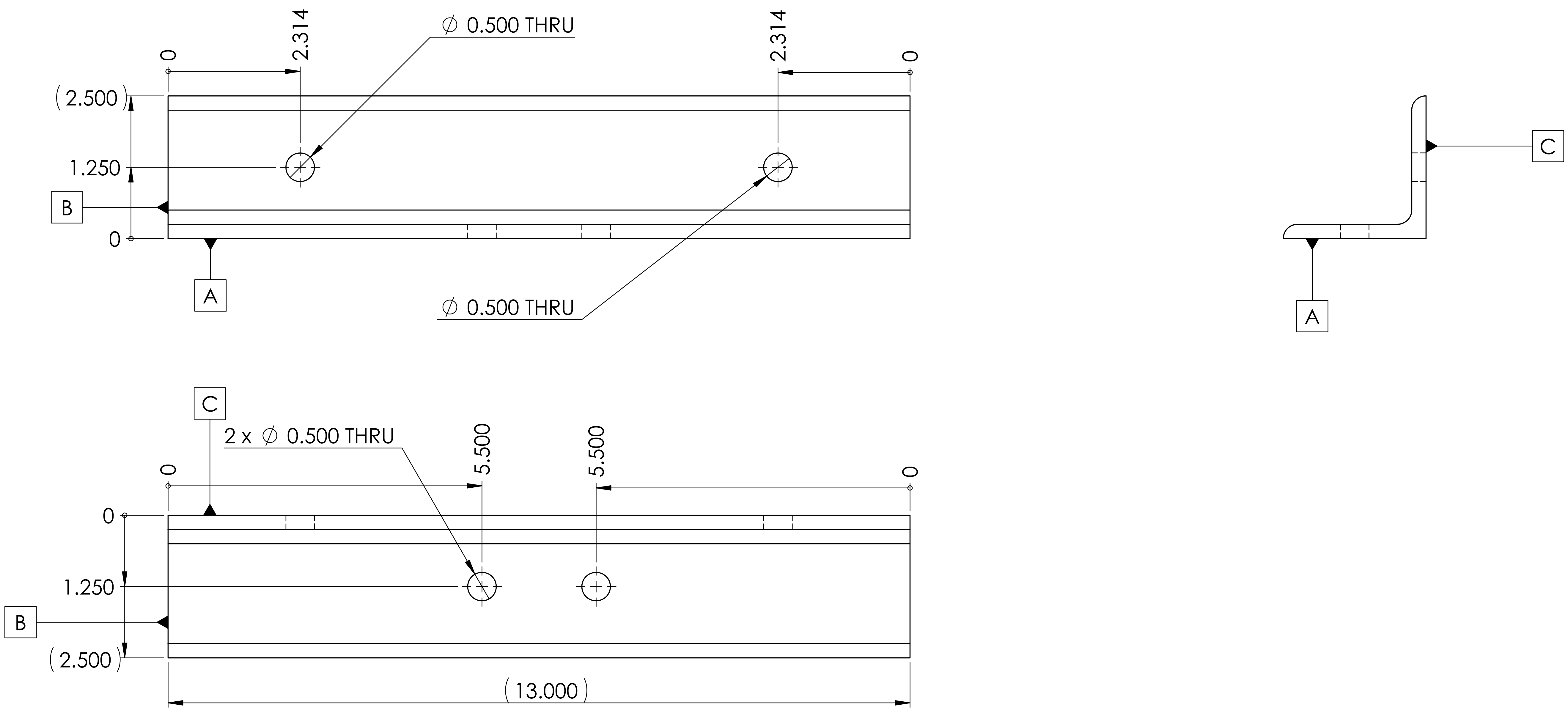
SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE		
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	5/3/12	TITLE:	
	CHECKED			ThrustBearingPushPlate	
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL	COMMENTS: Make 2x			SIZE B DWG. NO.	REV
FINISH					
DO NOT SCALE DRAWING				SCALE: 2:1 WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
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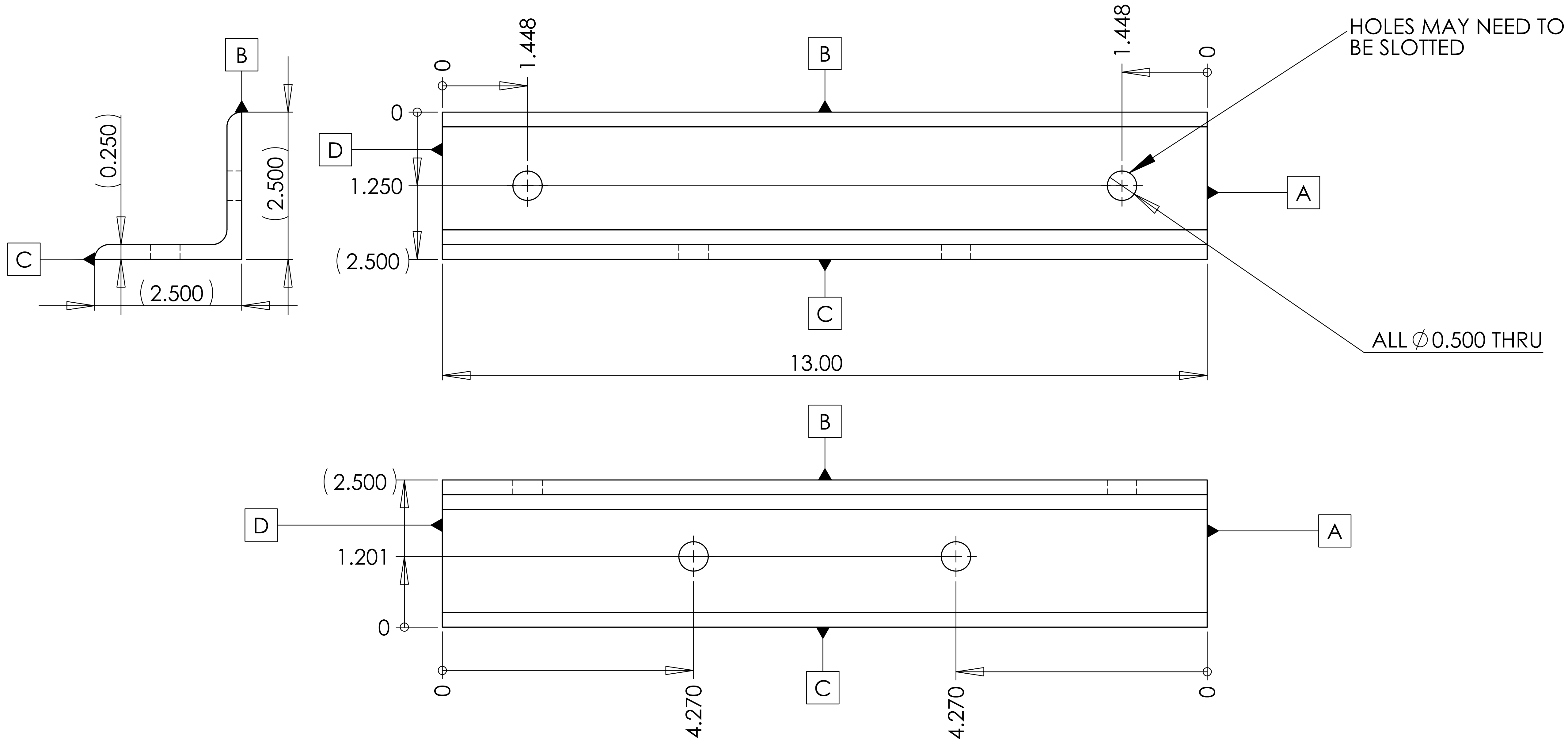
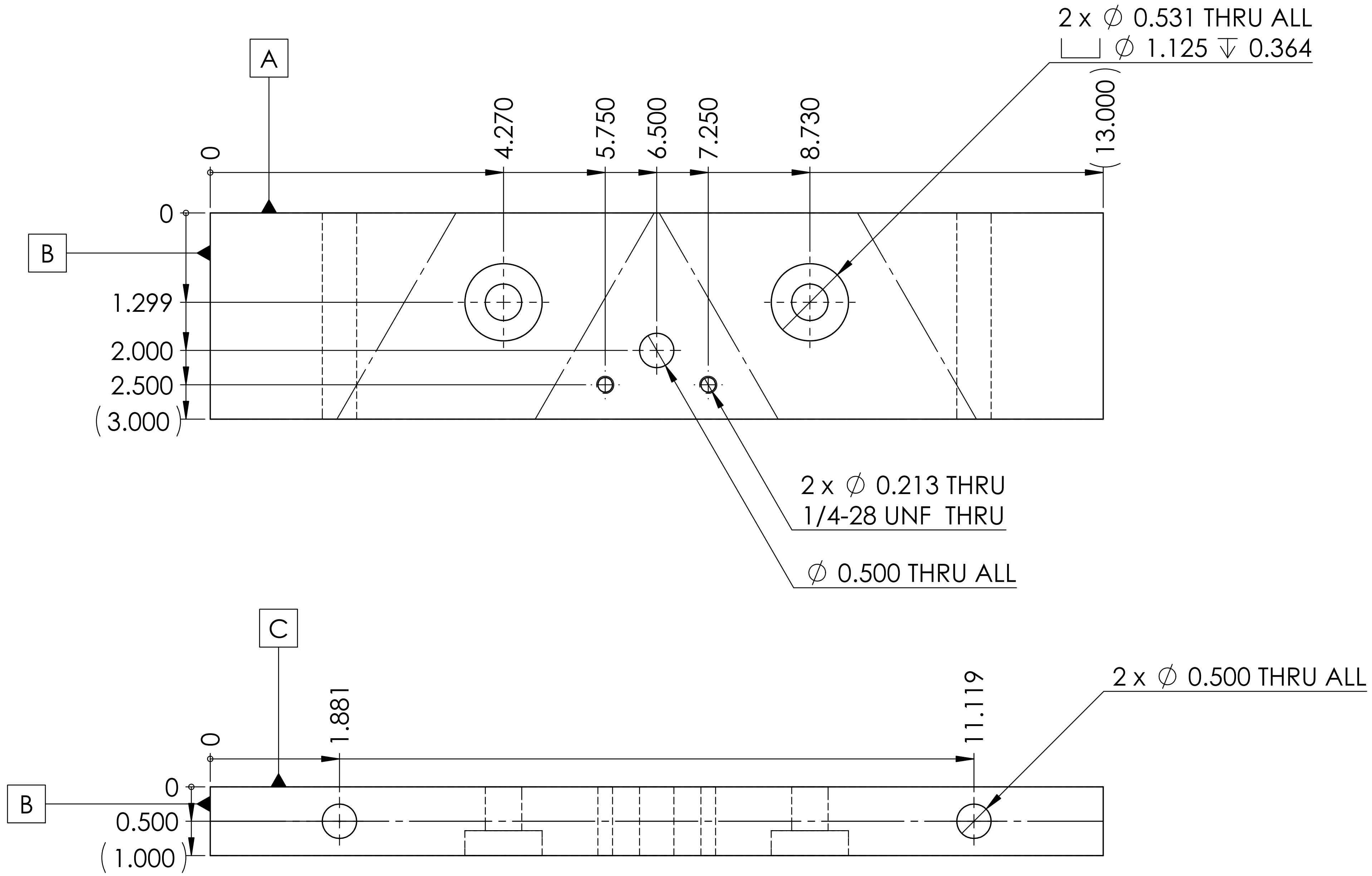
			CHECKED			TopInsideAngleZMount - Final Design
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS: Make 1x			SIZE B DWG. NO. REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 WEIGHT: SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE		
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	3/12/12	TITLE:	
	CHECKED			TopInsidePlate - Final Design	
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL	COMMENTS: Make 2x			SIZE B	DWG. NO. / REV
FINISH					
DO NOT SCALE DRAWING				SCALE: 1:2 WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	



UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

NEXT ASSY | USED ON

APPLICATION

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	4/4/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

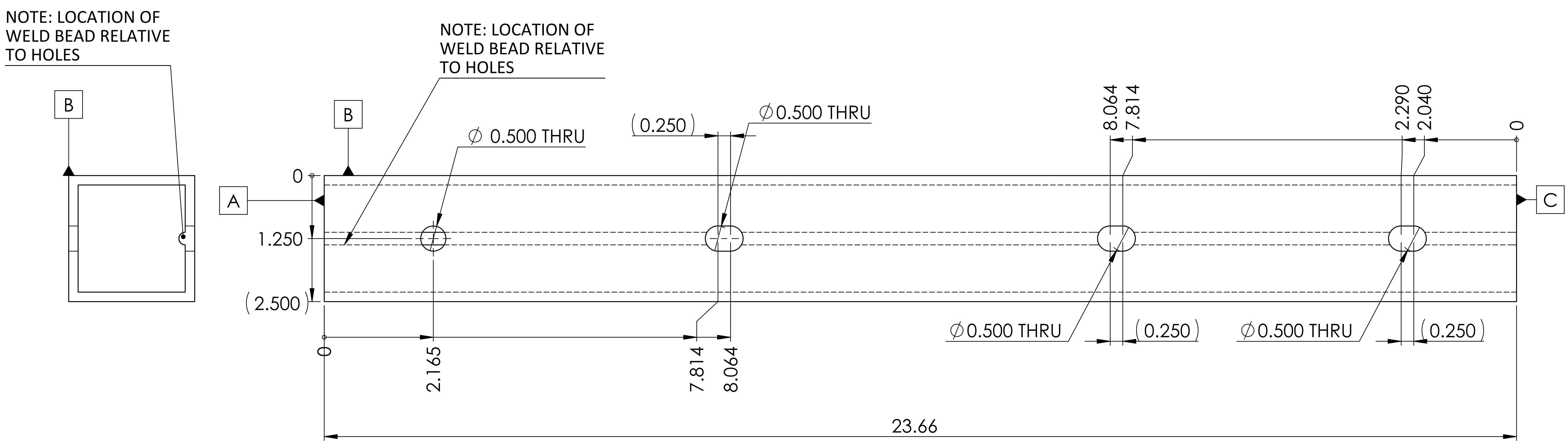
COMMENTS:
Make 1x

TITLE:

TopInsideZMountv2

SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1



NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	3/12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

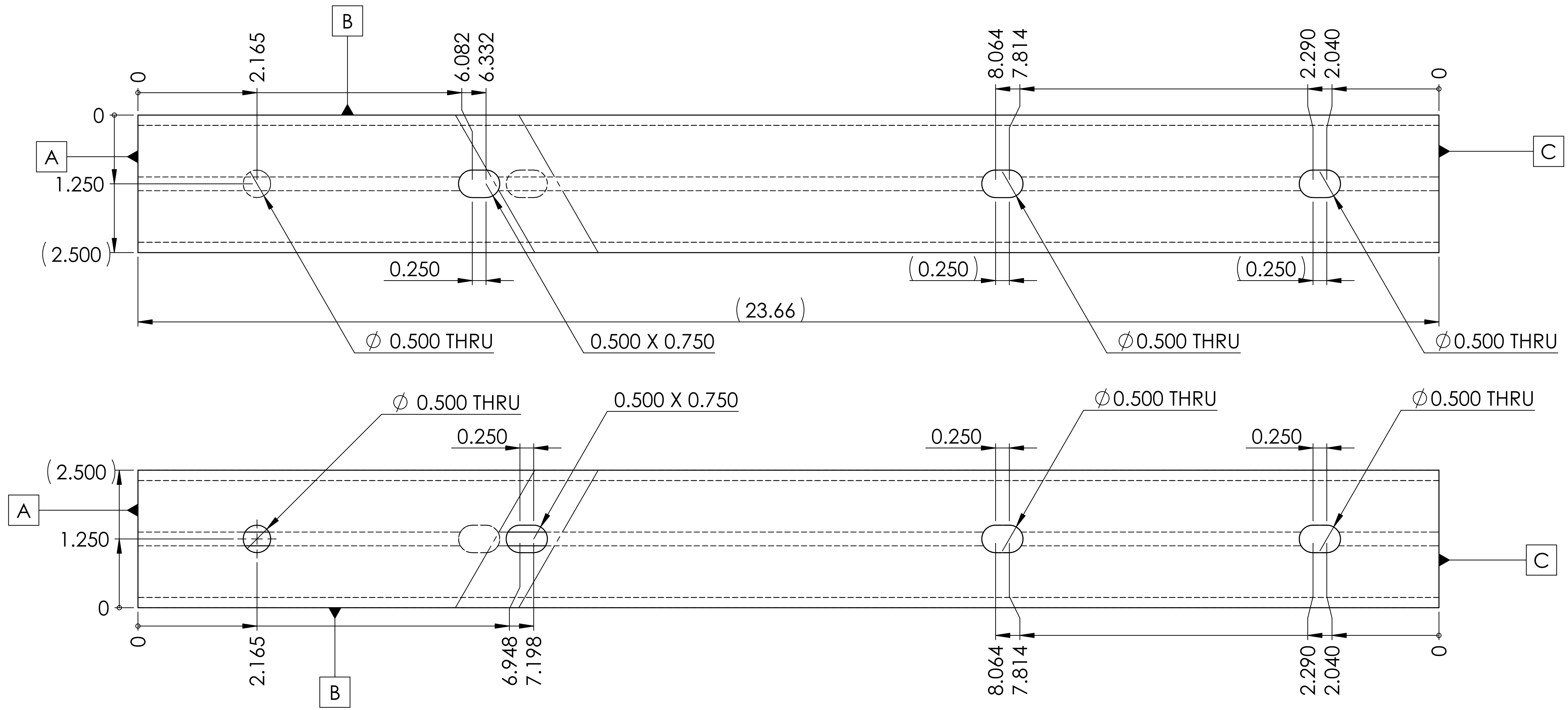
COMMENTS:
Make 1x

TITLE:

TopMember - Final Design

SIZE	DWG. NO.	REV
B		1

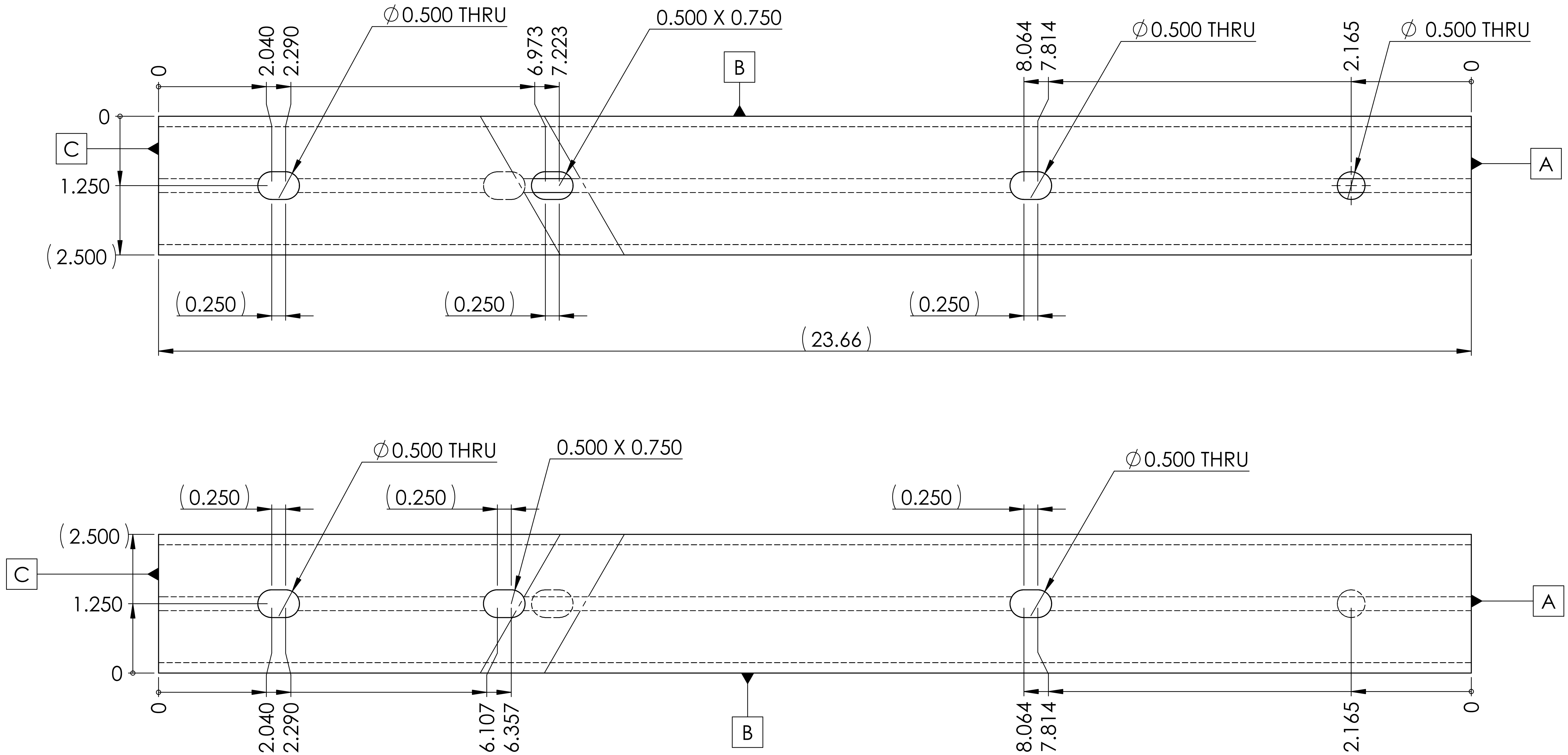
SCALE: 1:2 WEIGHT: SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/4/12	TITLE:
	CHECKED			TopMemberSideB - Final Design
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS: Make 1x			SIZE B / DWG. NO. / REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:5 / WEIGHT: / SHEET 1 OF 1

NEXT ASSY | USED ON

APPLICATION



NEXT ASSY	USED ON	
APPLICATION		

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	4/4/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

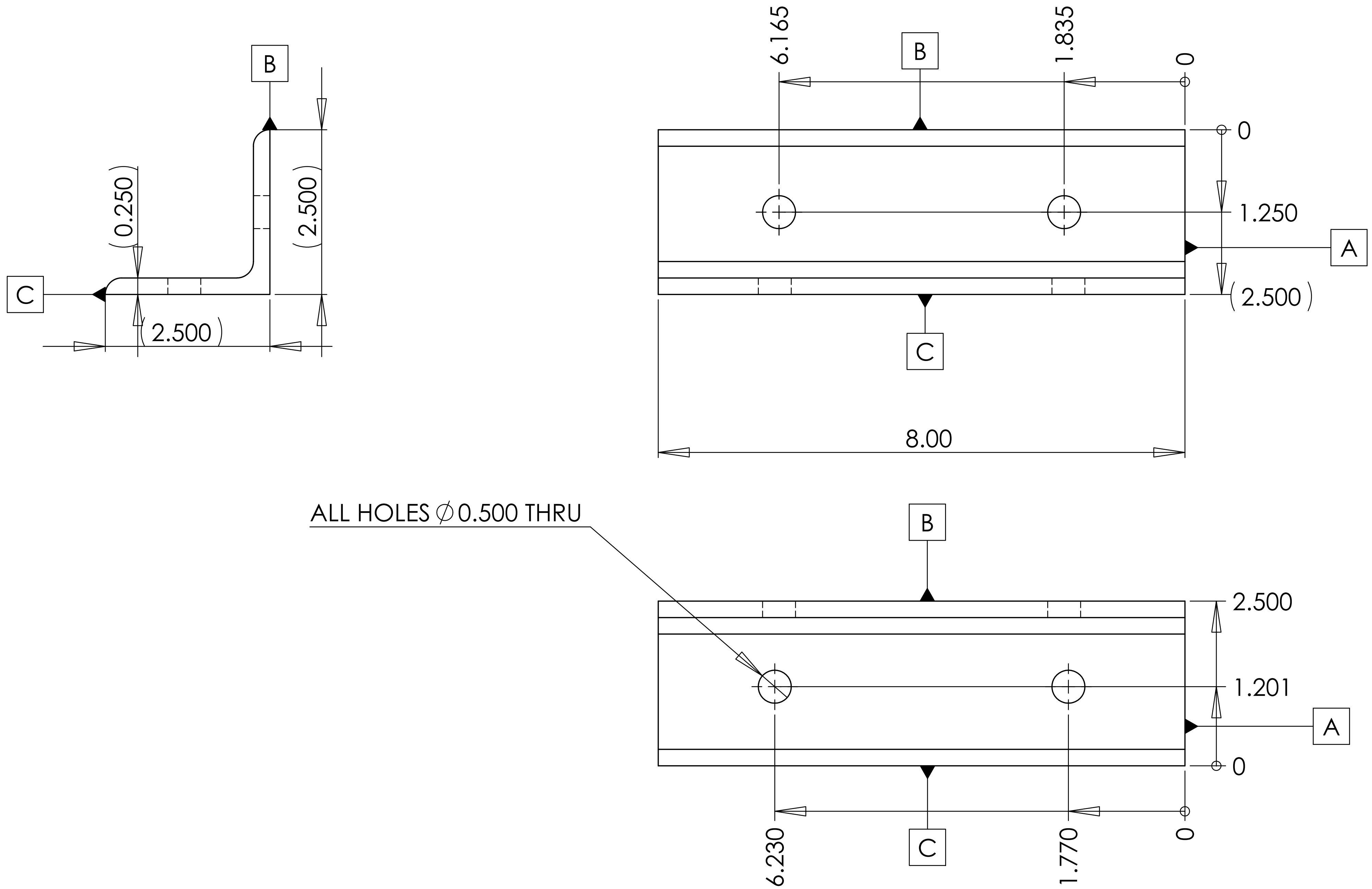
COMMENTS:
Make 1x

TITLE:

TopMemberSideC - Final Design

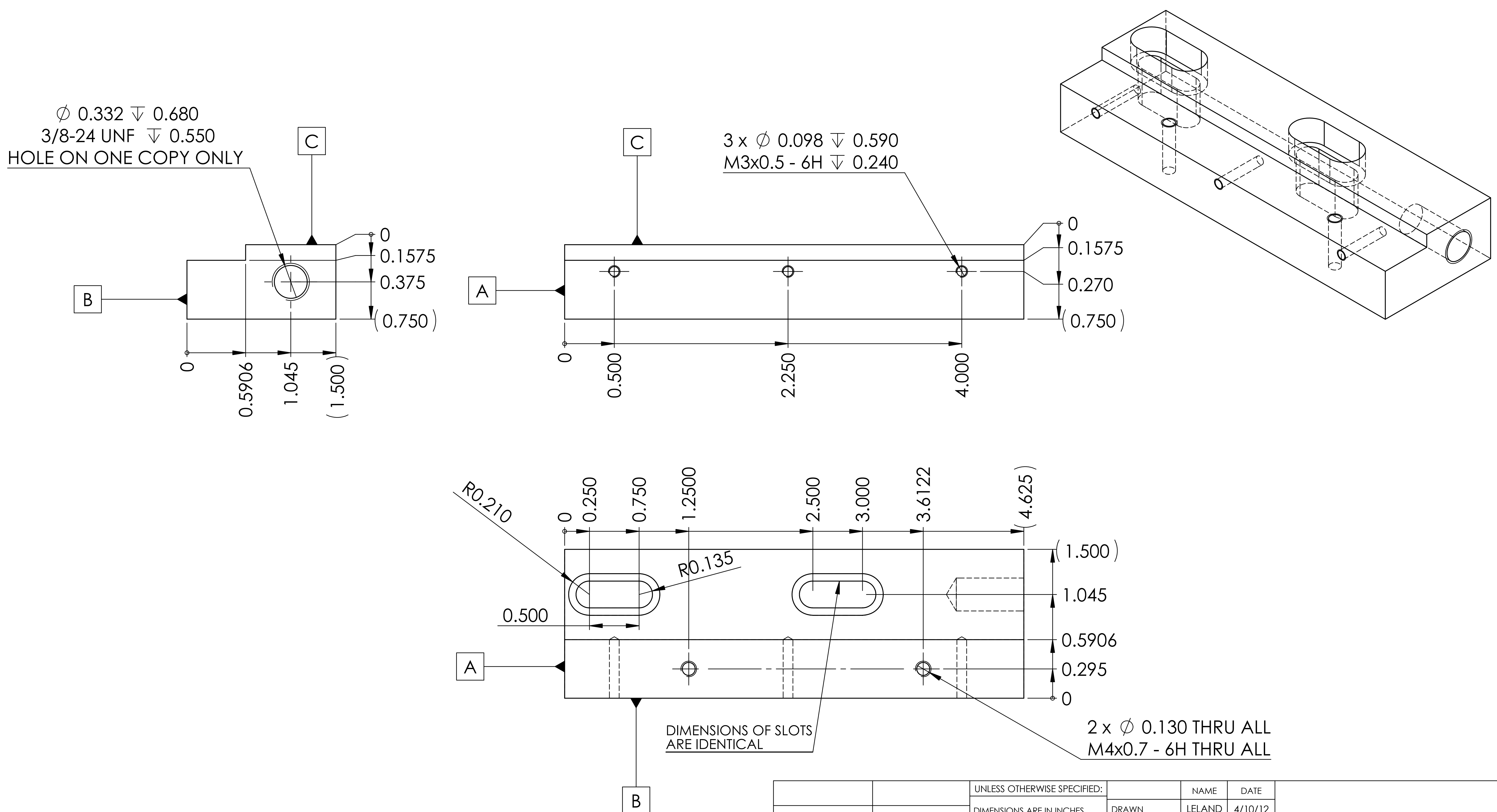
SIZE	DWG. NO.	REV
B		

SCALE: 1:5 WEIGHT: SHEET 1 OF 1

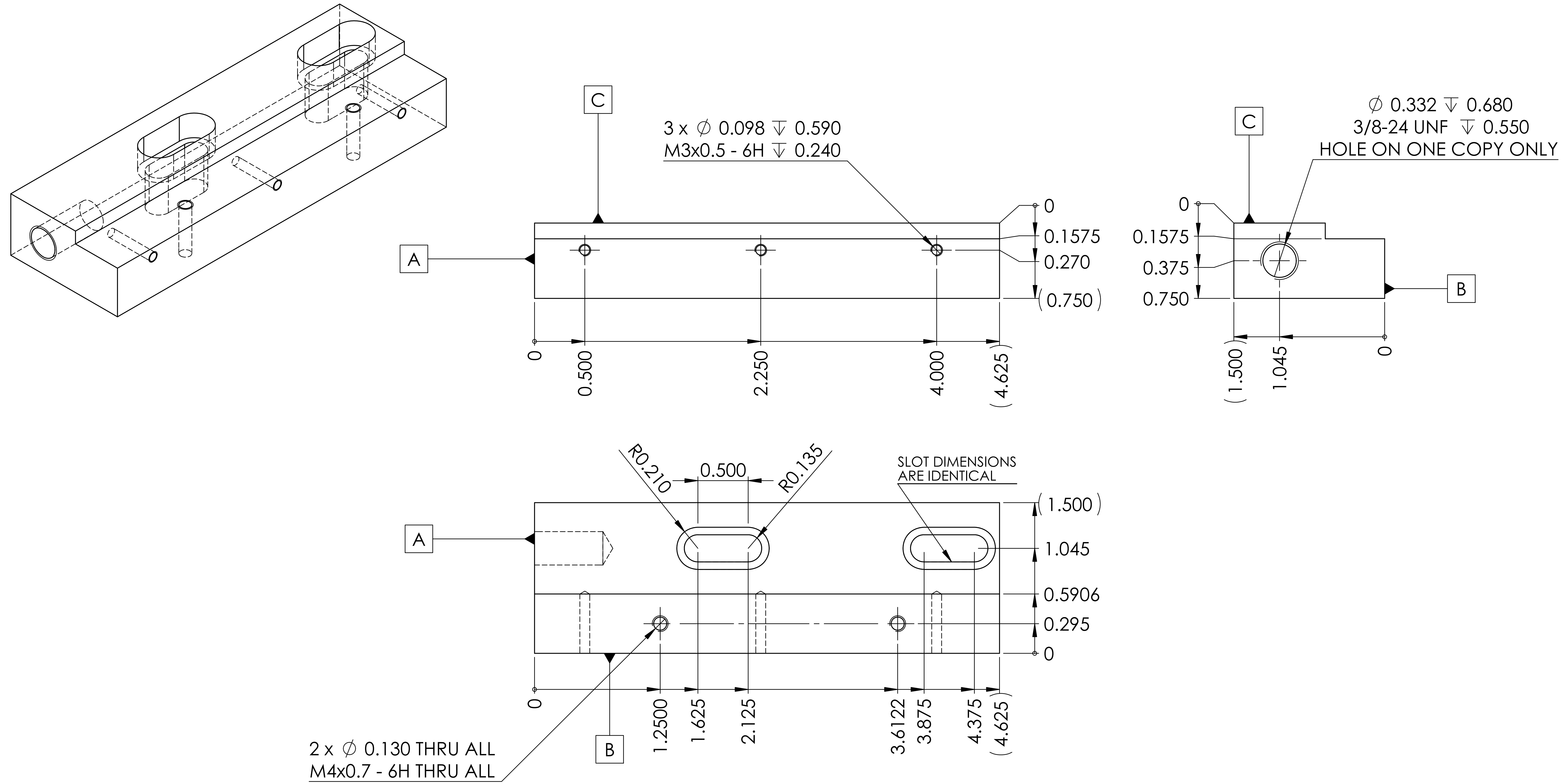


UNLESS OTHERWISE SPECIFIED:		NAME	DATE		
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	3/12/12	TITLE:	
	CHECKED			TopOutsidePlate - Final Design	
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL	COMMENTS: Make 3x			SIZE B	DWG. NO. / REV
FINISH					
DO NOT SCALE DRAWING				SCALE: 1:2 / WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	



				NAME	DATE	
		UNLESS OTHERWISE SPECIFIED:				
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	LELAND	4/10/12	TITLE:
			CHECKED			TopShortBearingRailClampV2 - FinalDesign
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS: Make 2x. Observe differences between parts (detailed above)			SIZE B	DWG. NO.	REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:1	WEIGHT:	SHEET 1 OF 1



NEXT ASSY	USED ON	
APPLICATION		DO NOT SCALE DRAWING

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC
TOLERANCING PER:

MATERIAL

FINISH

	NAME	DATE
DRAWN	LELAND	4/10/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

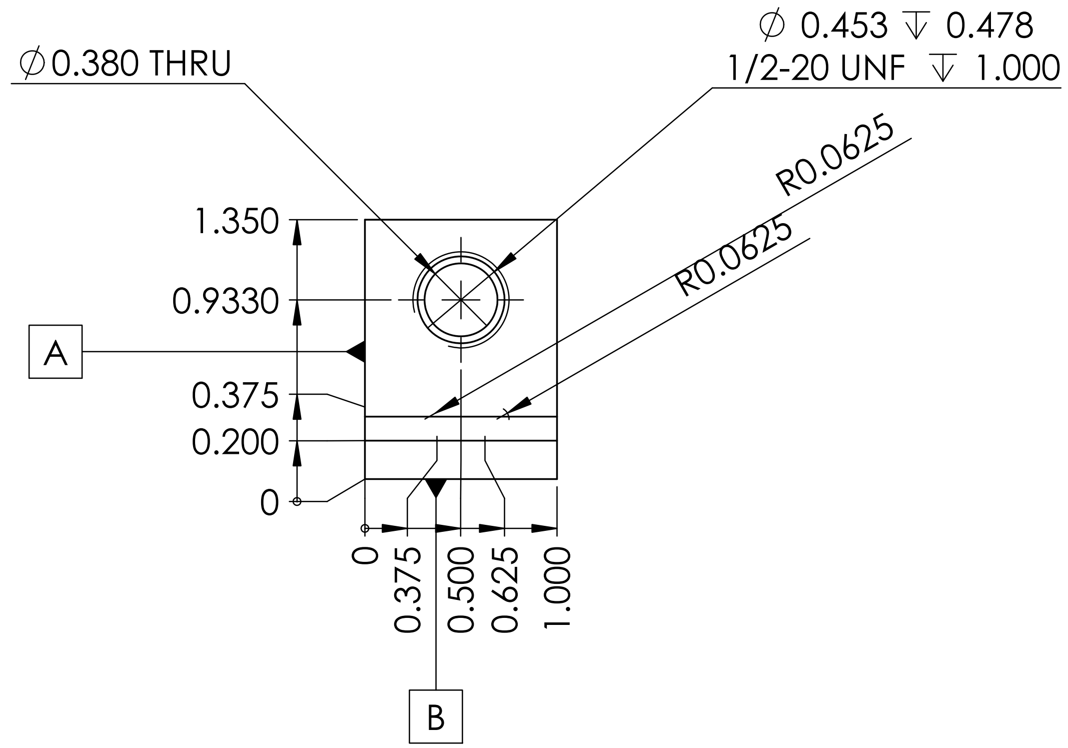
COMMENTS:
Make 2x. Observe differences between parts (detailed above)

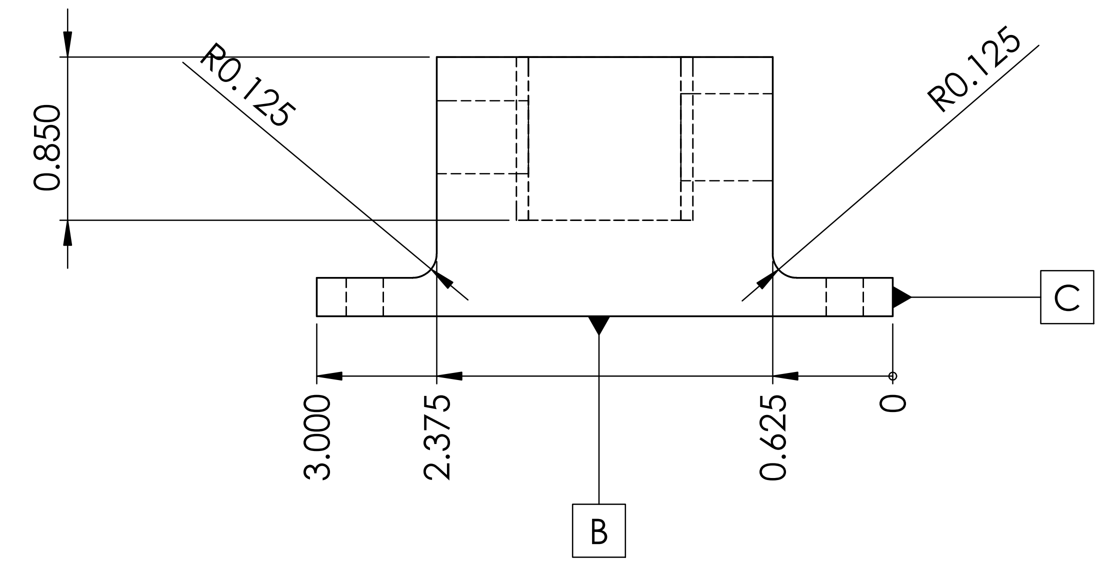
TITLE:

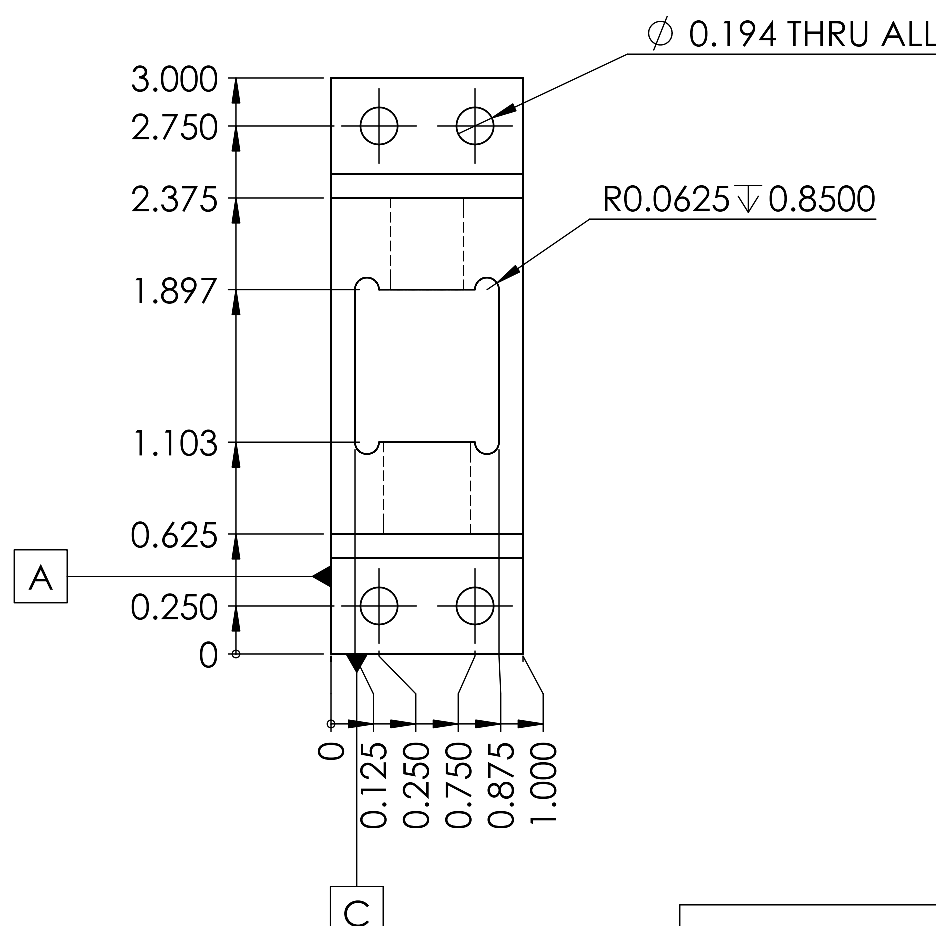
TopShortBearingRailClampV2 - Left - FinalDesign

SIZE	DWG. NO.	REV
B		

SCALE: 1:1 WEIGHT: SHEET 1 OF 1







UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	4/12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:
Make 2x

NEXT ASSY | USED ON

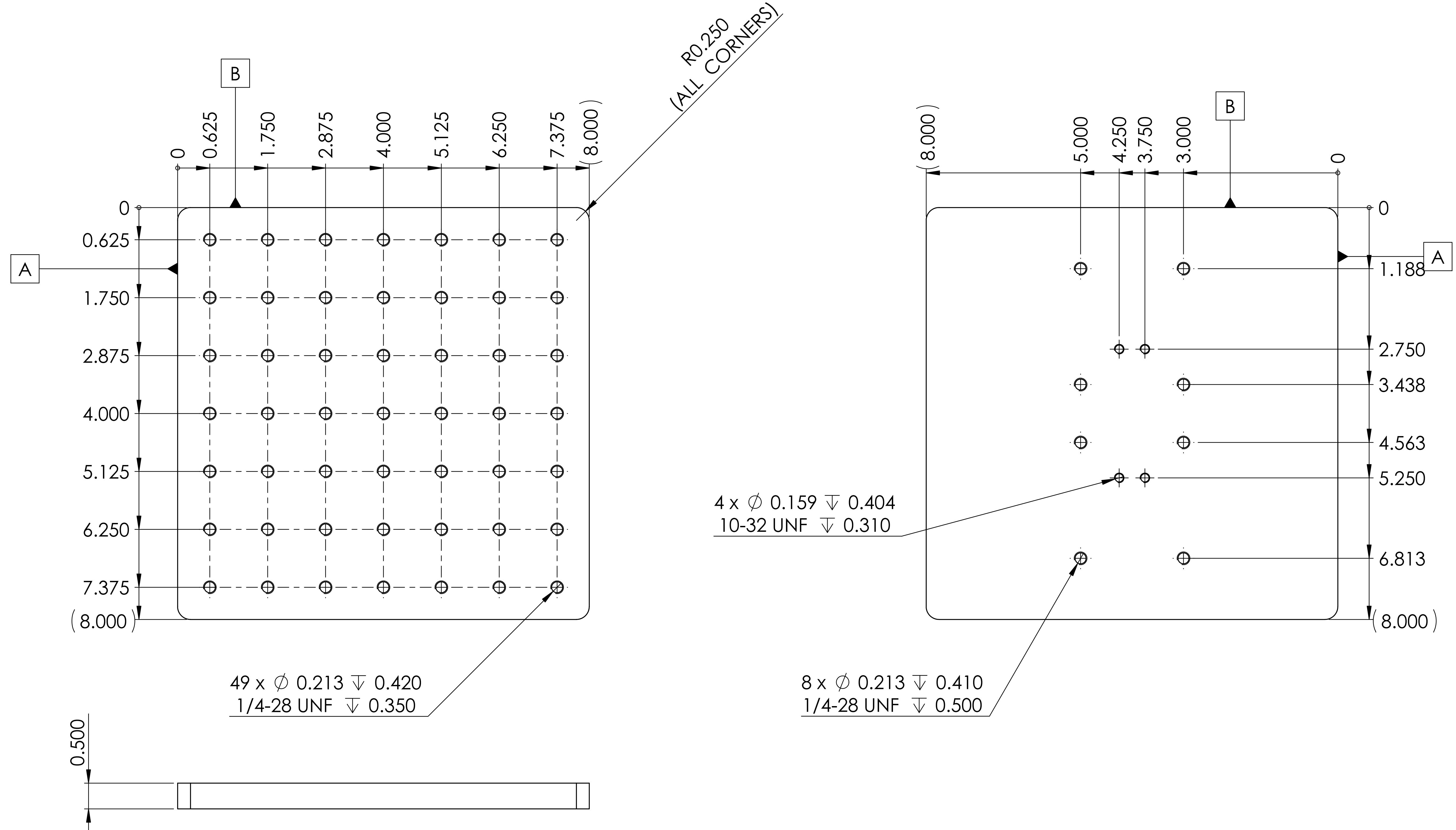
APPLICATION

TITLE:

TopStageNutMount

SIZE B | DWG. NO. | REV

SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	LELAND	4/21/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:
Make 1x

NEXT ASSY | USED ON

APPLICATION

TITLE:

TopTable - FinalDesign

SIZE B | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

Appendix 9: Assembly Notes

The following notes were taken during the first assembly of the machine. These instructions should be regarded as guidelines for assembly only: the assembly process as described here is highly imperfect.

Bottom Frame

1. Attached rail supports to back base member (through angle bracket)
    a. Requires long 3/8 – 28 screw to hold nuts inside back base member while assembling.

2. Attached corner plates to back base member

3. Attached connection plates (between side base member and rail supports) to side base members

4. Attach side base members to back base member and to rail supports
    a. Attach first to back base member with corner plates, and then to rail supports
    b. I attached these parts with the base on its side (sticking up in the air). Probably better to attach side base members to back base member while assy is flat, and then flip on to side to attach to rail supports.

5. Attach final corner plate

6. Place face-down and begin aligning base members
    a. Tighten all bolts going through corner plates and HOLES (not SLOTS) on base members
    b. Then set base as close to specified angle as possible (60 deg)
    c. Tighten down all bolts going through corner plates and SLOTS

7. Flip frame over onto correct side

Top Frame

1. Attach side triangles
    a. ONLY install innermost bolts on both sides. Do not snug tight – just get close to tight.

2. Attach top members
    a. Start with sides B and C (sides that attach to the spindle mount plate). Attach innermost bolts
    b. Attach topmost spindle mount plate
    c. Insert bolts into all HOLES at ends of top members. Do NOT install nuts.
    d. Then place bolts into SLOTS at opposite ends of top members (Inner slots should still be empty except for slots attaching to spindle mount plates.
    e. Attach spindle Z axis LOOSELY
    f. Insert bolts into remaining slots. Attach nuts loosely to ALL BOLTS.

g. (May have to remove and reinsert various bolts to get top frame to align properly)

Leveling, Carriage Installation, and Motion System Installation

1. Leveling procedure
    a. Support base at three corners – for instance, with back base member on table and third corner on stool or chair.

    b. Level base frame by shimming at corners.
        i. Start by leveling back base member. Place level along back base member and shim under corner bolts.
        ii. Then level third corner. Place level alternately between back base member and opposing side base members to determine levelness.

    c. Level top frame
        i. Slack off bolted connections between side frame triangles and top frame triangle. Slack off slotted connections between base members and side frame triangles. Keep thru hole connections relatively tight
        ii. Iteratively, place level across corners of top frame (parallel to base frame members). Lever/shim sides until top corners are level. Slowly tighten connections between base members and side frame triangles as top frame is leveled
        iii. Tighten connections between side frame triangles and top frame triangles.
        iv. Install bolts in currently empty holes and snug tight (never mind – don't do this yet)
        v. Snug tight all connections in top frame (between top frame and angle iron connection members)
        vi. Check alignment again
        vii. Strain-tighten all connections (never mind – don't do this yet)

2. Rail installation
    a. Snug tight ALL rail mounting block bolts and all rail mounting screws.

    b. Check distance between inner faces of rails. It should be roughly 4.909 in. If not, with ALL rail support beam bolts slacked completely, adjust positions of rail supports so that distance is correct (ideally, so they are roughly in the center of their range of motion).

    c. With level and square, begin aligning longer rail. Slowly tighten 3/8" bolts at both ends of rail support beam, adjusting position with hammer. Check that:
        i. Top surface of rail is level along long axis
        ii. Top surface of rail is level along short axis (check as best as possible)
        iii. Rail is roughly perpendicular to back base member

    d. Once rail is close to level, strain-tighten all bolts

e. Check distance between inner faces of rails again. With all bolts on SHORT RAIL support slacked off completely, move rail until it is roughly 4.909 from other rail.

f. With level and square, begin aligning shorter rail. Slowly tighten 3/8" bolts at both ends of support beam, adjusting position with hammer. Check that:
    i. Top surface of rail is level along long axis
    ii. Top surface of rail is level along short axis (check as best as possible – this will be harder because the other rail is in the way)
    iii. Rail is roughly perpendicular to back base member. Check this by checking that distance between rails is roughly 4.909" – this is the critical measurement
    iv. Top surfaces of rails are coplanar (lay level across them and check levelness. This does not actually guarantee that the two top surfaces are coplanar, but it means that they're pretty close → close enough to be shimmed into place.

g. Once rail is close to level/in position, strain-tighten all bolts.

h. Carriage installation
    i. Attach bearing blocks to bearing block plate. Tighten down all push plates, and securely attach block plate mount screws. Tighten to 40 kgf-cm (34 lbf-in)
    ii. Slack rail mounting block bolts, push plates, and rail mounting screws.
    iii. Slide bearing block plate onto rails.
    iv. Begin leveling longer rail. Move carriage back and forth along rail. Shim under rail mounting blocks, and then snug-tight rail mounting blocks. If any binding is felt (if carriage becomes harder to move), loosen bolts and shim further.
    v. Tighten rail mounting block bolts on long rail side fully.
    vi. Tighten push plates on long rail side.
    vii. Begin further leveling of longer rail. Begin to tighten rail, shimming under rail as needed.
    viii. Tighten rail mounting block bolts on short rail side fully.
    ix. Begin shimming under shorter rail, slowly tightening bolts. Also shim against rail side mount, slowly tightening push plate. If a precision level is available, check that the rail is level in both directions.
    x. Insert rails into top side bearing blocks.
    xi. Attach top side rail support blocks. Fully tighten rail bolts and push plates.
    xii. Attach top plate. Shim between top side rail support blocks and top plate as needed, while sliding top plate to ensure that no binding occurs.
    xiii. Tighten top side rail support blocks fully. Attach thrust bearing mount loosely.

3. Lead screw units (These instructions apply to both lead screw assemblies)
    a. Create circular shim using .003" shim stock to fit gap between bearings and lead screw.

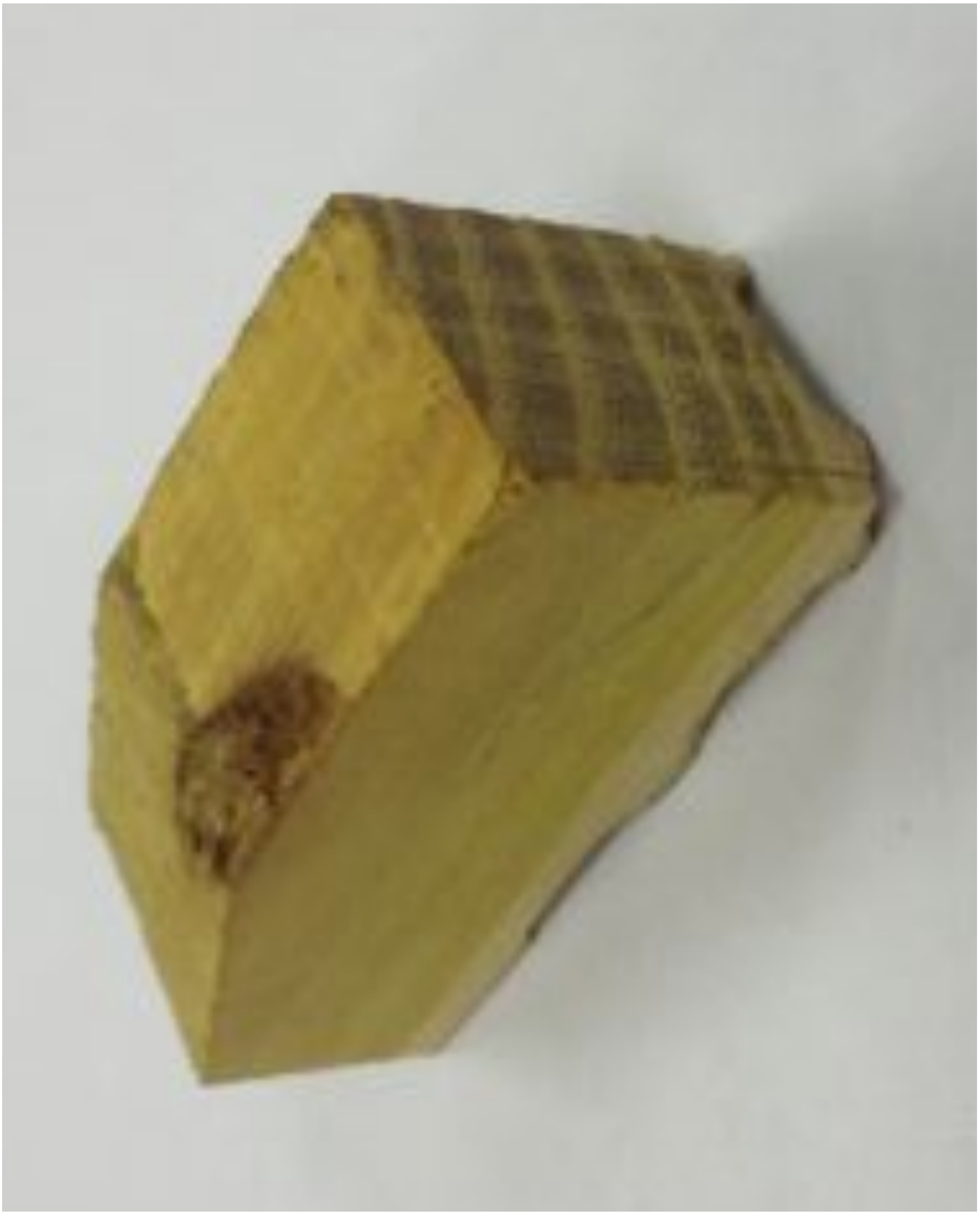
b. Thread single nut roughly 6” onto screw.
c. Place spacer washer between bearings, ensuring that bearings are placed in DF configuration. Fit circular shim inside of inner bearing diameters.
d. Push bearing stack onto lead screw until it makes contact with nut.
e. Thread second nut onto lead screw. Tighten nuts against each other to compress inner races together.
f. Place screw assembly into thrust bearing mounts.
g. Grease lead screw.
h. Thread screw into lead nut assembly. Place two nuts, separated by spring washers, inside nut housing. Thread screw through nuts and washers.
i. Push screw assembly fully into thrust bearing mount. Attach thrust bearing push plate, and tighten screws to snug tight.
    i. Be careful when tightening thrust bearing push plate, as overtightening has caused binding.
j. Turn lead screw to bring stage as close to thrust bearing mount. Tighten mounting bolts fully to ensure that lead screw is aligned to lead nut axis of travel.

Appendix 10: Machining Test Results

The following images are of parts machined during cutting performance tests.

Stainless Steel – Drilling Test

Osage Orange – Milling Test

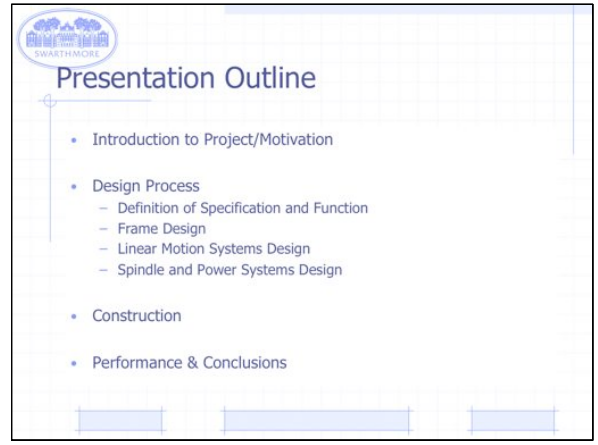


6061-T6 Aluminum – Milling Test

Appendix 11: Final Presentation Slides

The following are slides used at the final presentation of this ENGR 090 project, conducted on May 2nd, 2012.

Hello everybody; thank you all for coming. My name is Julian Leland, and today I'm going to be presenting my E90 – the design and construction of a self-replicating milling machine.



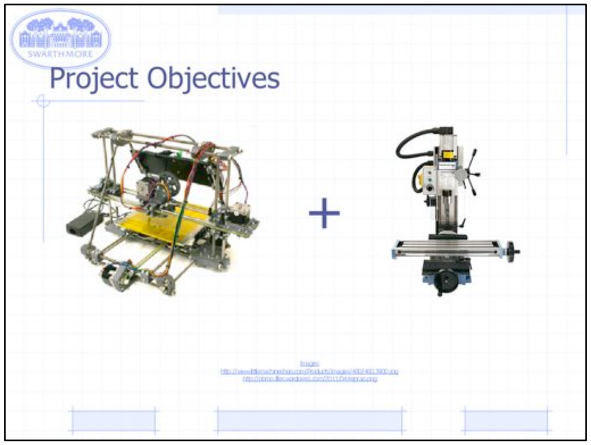
Briefly, I'd like to outline my presentation today.

First, I'm going to introduce my project, and explain why I think it's worth pursuing.

Second, I'm going to describe the process I went through to design this machine, focusing on these four major sub-tasks I had to complete.

Third, I'll talk a little about the construction and assembly processes.

Finally, I'll discuss where the machine is now, how close it comes to achieving my design objectives, and talk about avenues for future work.



• In recent years, small, affordable, digital rapid fabrication tools like 3D printers (see above) have made desktop manufacturing a buzzword, even outside of hobbyist and maker communities

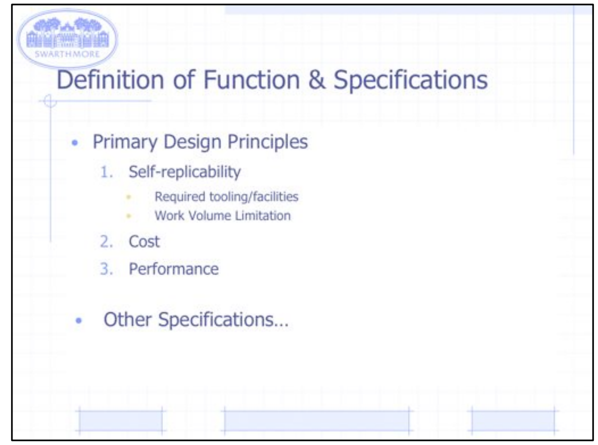
- One major aspect with SOME OF THESE MACHINES is "self-replication" – idea that if you have one of these machines, you can build all the parts that you can't just buy off the shelf to build another.

• Most of these tools are either 1) additive manufacturing tools (3D printers), 2) non-contact subtractive (plasma cutters), or 3) light-duty contact subtractive (wood routers)

- **Nothing addresses the need traditionally filled by milling machines – working of metals at reasonably rapid rates while still maintaining cut precision measurable in .001"**
- Currently available small milling machines are expensive (upwards of $600 w/o CNC functionality), limited in their functionality, and often of poor quality – finally, cannot self-reproduce.

• **Goal of this E90**: Design and construct a 3-axis machine tool, which can mill basic materials including mild steel and can create all parts that can't be easily purchased which are needed for its construction.

- Work volume of 6" x 6" x 6"
- No advanced machining processes
- Comparable in cost to its competitors ($500-$700)
- Intended for CNC use



The first step in the design process was to fully define the machine's mechanical specifications and intended function.

Three primary design principles were defined at the beginning of the project, to serve as a compass for selecting between competing design goals

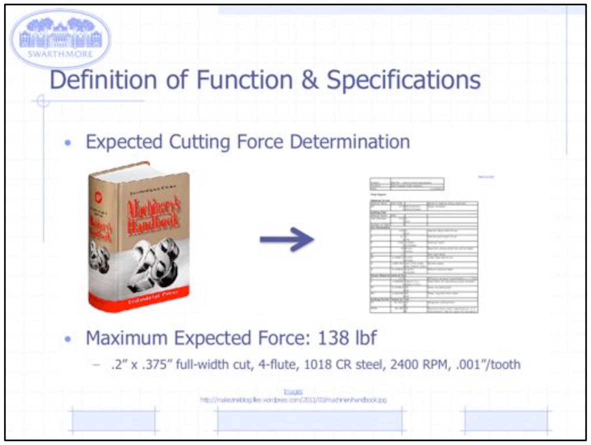
• Self-replicability: a user should be able to create an exact copy of the machine, using only a preexisting machine as well as basic hand tools/measuring equipment. This statement has a lot of ramifications.

- No advanced manufacturing processes – no casting, scraping, grinding or welding
- No fancy metrology equipment – intended to only require 3' scale, 6" dial calipers, combination square and shim stock.
- All precision-manufactured parts must fit within the machine's work volume (6" x 6" x 6") – if they cannot, they must be designed for compliance (e.g slots instead of holes to allow for more generous manufacturing tolerances).

• Cost: More time- or manufacturing- intensive design options should be selected over more costly design options.

• Finally, the performance goals stated earlier – capable of light milling in mild steel – should be achievable

In addition to these basic principles, a number of other aspects of the machine were outlined, including the basic geometry of the frame, the machine's kinematic type, the positioning ability of the machine, the required translation rates of the machine, the materials to be used in the machine, and the maintenance and repair requirements of the machine.

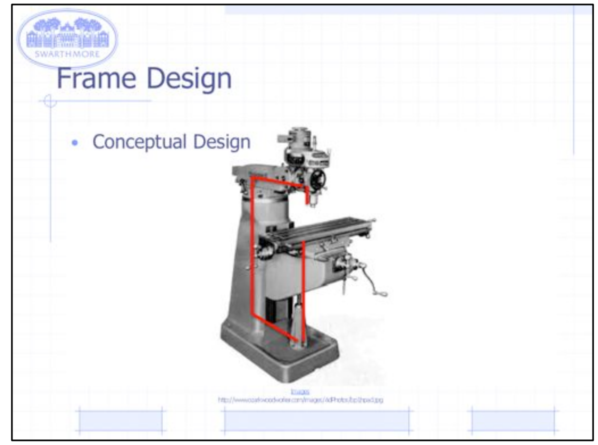


In order to fully define the performance goals of the machine, as well as to give myself loads to use for design purposes, I had to determine the maximum expected cutting load.

I developed a method based off of information in Machinery's Handbook and a variety of other sources to estimate maximum cutting force experienced by 2-, 3- and 4-fluted end mills for specified cut geometries, feed/speed combinations and materials.

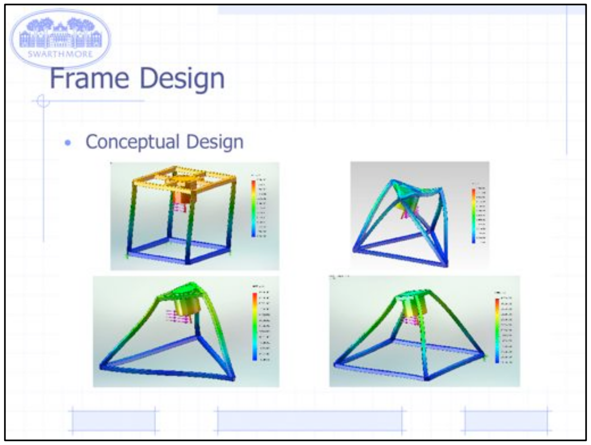
I programmed this into a spreadsheet, and simulated a variety of cuts. I found that the maximum expected cutting force was 138 lbf, for a .2" by .375" full-width cut in 1018 steel, at 2400 RPM and with a feed of .001" per tooth. I then applied a factor of safety of 1.5 to this expected load, leading to my expected design load of 200 lbf.

With this design load, I stipulated that my machine should be able to perform a .125" x .1" full-width cut in 1018 steel without experiencing total error motion greater than .001"



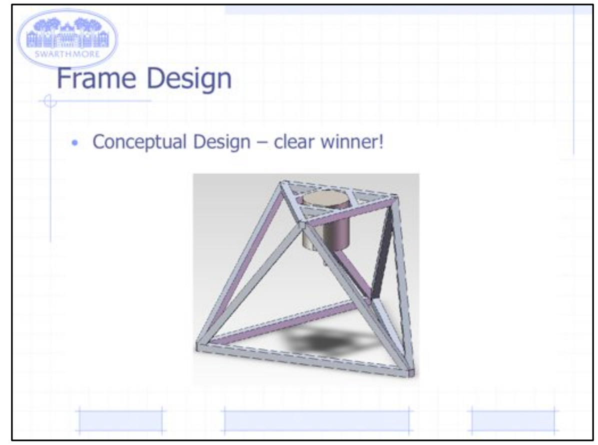
With the machine basically outlined, I turned my attention to my first major design task - designing the frame of the machine.

Most milling machines rely on a C-frame design. Although this design allows for easy operator access, all of the frame members are cantilevered off of each other – prone to flexure.



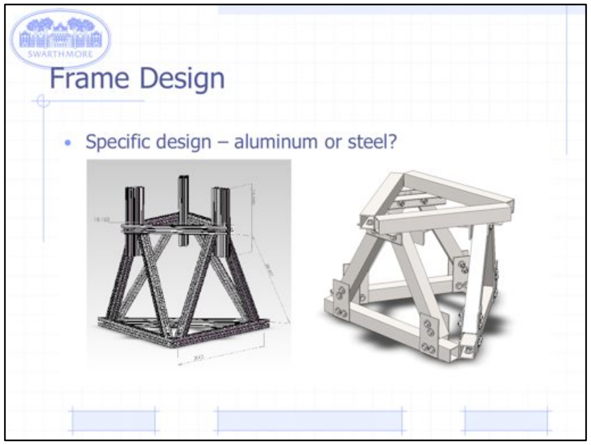
I instead decided to opt for a closed frame. This allows greater flexibility in design, and typically yields better dynamic and static performance. Additionally, since I'm intending for my machine to eventually be CNC controlled, I don't care as much about accessibility.

I developed a number of concept "test frames" in Solidworks and simulated them under static loading conditions. Each frame was designed to fit a certain "work volume cube" within it, and used identical frame members to isolate the "innate stiffness" of the design.



A winner quickly emerged – what I'm calling a double tetrahedral frame.

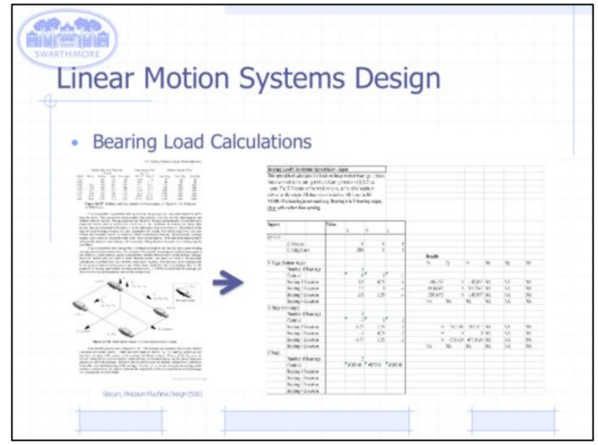
This frame design proved to be an order of magnitude stiffer than the other designs I'd come up with. It also allows one "long" axis – basically, parts can protrude through one of the triangular side frames to allow long parts to be machined.



I then moved on to the specific design of the frame.

I developed models of the frame using two different construction materials: 80/20 aluminum extrusion framing, and standard hollow steel section with structural bolted connections.

As you can see, I ultimately elected to go with the steel. My reasons for this included:
• the lower cost of the steel – initial estimates showed it would be significantly less expensive than the aluminum (although this would later prove to not be completely accurate).
• greater dynamic and static performance of the steel. Simulations showed that the steel frame was significantly stiffer than the aluminum, and was also more vibration-resistant due to its greater weight and natural damping characteristics.



The second major design phase of my project was the design of the linear motion systems – the bits that slide back and forth within the machine.

Although I briefly entertained exploring non-Cartesian motion systems – for example, combined rotary and linear motion systems - I ultimately elected to stick with the tried-and-true stacked linear motion systems for X, Y and Z motion.

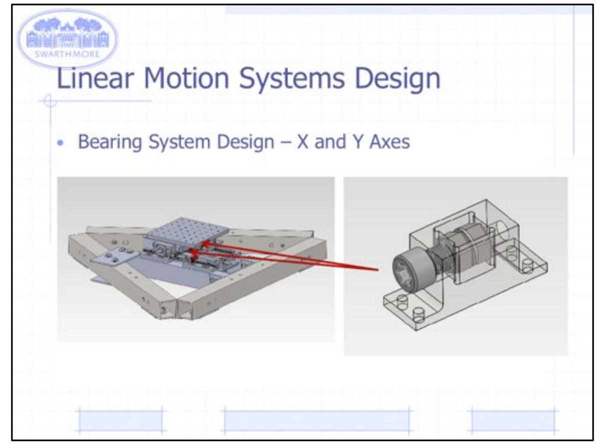
Before I could select a bearing system, I needed to know what sort of forces I could expect at my bearings. Working from a derivation by Dr. Alex Slocum, I developed a spreadsheet which would approximate the reaction forces in 4-, 3- and 2-bearing carriage systems given a point load at any point in 3-space.

This spreadsheet not only allowed me to determine what kinds of loads I could expect at my bearings, but also allowed me to evaluate the relative performance and cost tradeoffs between using the different bearing systems – more bearings produces lower loads on individual bearings, but increases cost.

Because of time constraints and manufacturing difficulty, I elected to purchase a premade bearing system rather than design one from scratch.

The variety of bearing systems commercially available is incredible, ranging from simple sliding element systems to "drawer-slide" caster systems to sophisticated recirculating ball-bearing carriage systems.

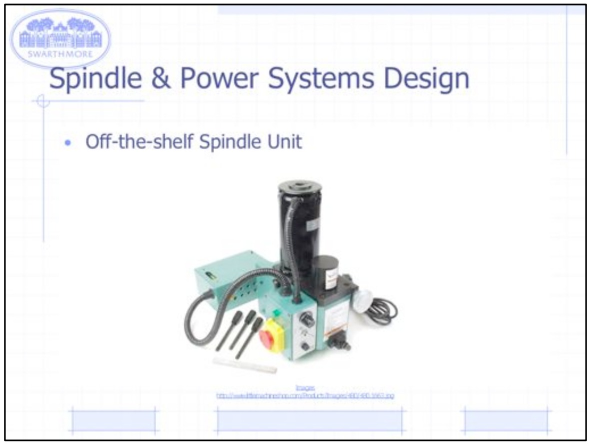
Ultimately, I selected a recirculating ball-bearing system. These are the standard among both hobbyist and industrial machines: they are extremely stiff, very low friction, and resistant to impact loading/oscillating loads (which frictional bearing systems are not). However, they are highly sensitive to mounting misalignment, and are EXTREMELY expensive.



I designed the X and Y bearing systems in Solidworks. To accommodate the high mounting precision requirements of the linear rail system I had selected, all precision components were required to fit within the work volume of the machine – i.e. be no greater than 6" in any direction. Additionally, leveling screws and slots were used to allow for compliance in assembly, to prevent internal strains from being developed by the machine.

I also designed a lead screw drive system based around a 3/8"-10 Acme threaded rod with a backlash-reducing nut system. I elected to use standard-, rather than precision-grade Acme hardware for cost reasons. The lead accuracy – the distance traveled per amount turned – of the precision and standard grades are the same, and the lower quality thread fit of the standard grade thread is accounted for by the backlash-reducing nut.

The lead screws are singly supported, using two angular contact bearings in a DF configuration. This allows for slight angular misalignment, but produces high axial stiffness.

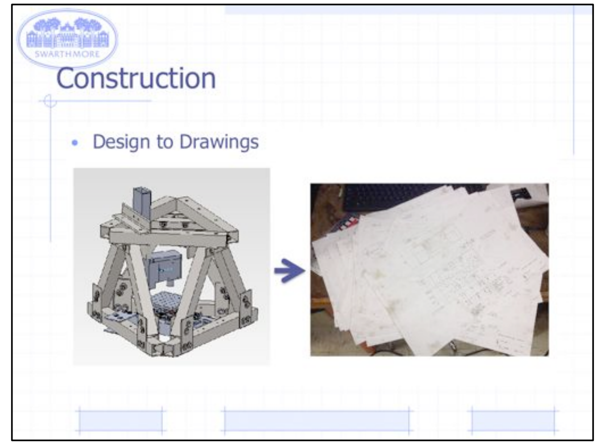


The final design task in this project was the design of the spindle and power systems.

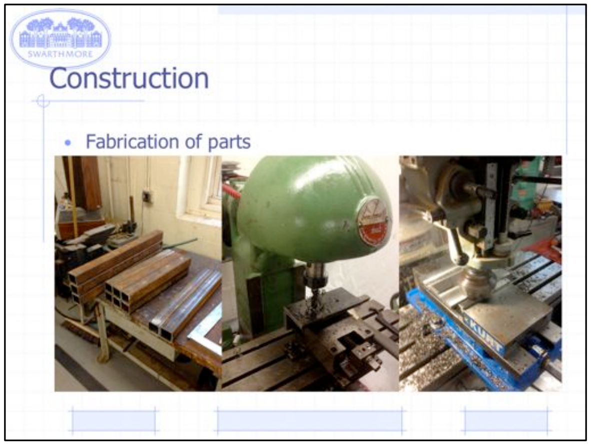
Early on in the project, I investigated a variety of options for the spindle and power system, including off-the-shelf units, engraving spindles, and self-designed spindles. Unfortunately, spindles are among the highest-precision components in a machine tool, and typically are hardened and ground.

Consequently, I elected to use an off-the-shelf unit. The most cost-effective option I was able to find was a combined spindle and motor unit used in a variety of benchtop milling machines. It incorporates variable speed control and uses R8 collets.

I also selected to use the dovetail-mount column that this spindle unit is designed for, which simplified attachment of the spindle. It incorporates a rack-and-pinion gear giving Z-axis motion.



All told, this machine requires 94 machined pieces, 37 of which are unique. I transferred my Solidworks parts to paper drawings, and began machining.



Fabrication took a long, long time. I didn't keep close track of the hours I spent in the shop, but I would estimate it at roughly 200-250 hours of machining time.

All machining was done either in the Swarthmore machine shops, or at my house in Washington, DC. To simplify the fabrication process, I used the full capabilities of the machines I was using – I didn't limit myself to a 6" x 6" x 6" work volume.

Assembly took surprisingly little time, considering the amount of time required to machine the parts. I completed assembly in a period of roughly 12 hours.

The majority of this time was spent attempting to level the X and Y axes. This proved to be an extremely difficult process, and one which is still not completed to my satisfaction - I'm intending to go back and realign the axes, to make sure that the bearings don't fail.

Assembly was easily completed using basic hand tools – box wrenches, socket wrenches, hex wrenches and hammers. I did use a precision level and square for some alignment tasks – however, these could be replaced with a high-quality combination square or carpenter's level.

So, does it work?

In short…kinda.

Although no quantitative measurements of cutting performance have been made, the samples I'm passing around now, some of which you can see here demonstrate the machine's cutting performance.

• The wooden block is osage orange, a particularly hard and fibrous wood. The machine has no problem making cuts in this material.
• The aluminum bar shows the machine's performance under both end- and side-milling conditions. As you can see, chatter is clearly a serious problem – indeed, the machine was beginning to chatter so seriously it started to loosen some of the screws in the stage. There is believed to be some loose component that is the root source of this chatter – it is being investigated now.
• The stainless steel bar shows that the machine is capable of executing basic drilling operations.

The work volume wound up actually exceeding the specified limit in the Z direction, yielding a total volume of nearly 6” x 6” x 12”

There are a number of other metrics beyond cutting performance which can be used to evaluate the performance of this machine

• Cost: The total cost for raw materials, fasteners and parts, not including shipping costs or tooling costs, came to approximately $1,100. Additionally, this amount is not reflective of the true cost to the average consumer – many of these items were purchased using educational discounts, or with Swarthmore's tax-exempt status.

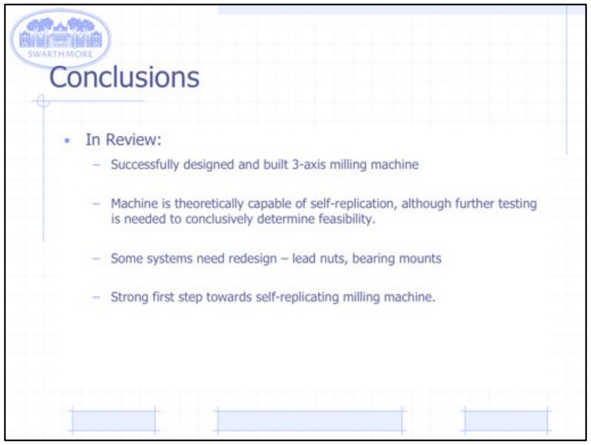
• Self-replicability: With the exception of possibly 2 parts, this machine is theoretically self-replicable – all parts can be machined within its work volume. However, since it has yet to be determined that the machine can mill mild steel with reasonable accuracy, it cannot be said conclusively if the machine is capable of self replication.

Conclusions

- Immediate Improvements/Future Work
  - Tighten all bolts
  - Re-aligning XY Carriage (especially Y axis)
  - Redesign lead nuts
    - Use bronze nuts
  - Rebuild spindle (has been dropped on its head, literally)

Conclusions

- Design Improvements
  - Design spindle, linear motion systems to reduce cost
  - Redesign X axis mounting system to reduce misalignment
  - Redesign Z axis to use custom linear motion system – take better advantage of frame stiffness
  - Develop a better method for aligning/leveling X and Y axes

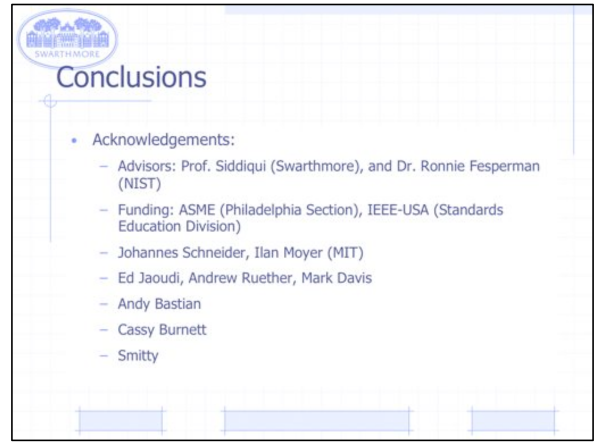


In review, I have:

• successfully designed and built a 3-axis milling machine, which has been demonstrated to be capable of milling and drilling wood and aluminum, although not yet mild steel.

• This machine is theoretically capable of self-replication, but more testing will be needed to conclusively determine this.

• A number of systems within the machine need redesign, but overall, this is a strong first step towards developing a truly self-replicating milling machine.



Finally, before I close, I'd like to thank some of the many people who made this project possible.
• First, I'd like to thank Professor Siddiqui, who was my faculty advisor for this project, as well as Dr. Ronnie Fesperman at the National Institute of Standards and Technology, who also advised me.
• I would also like to thank the Philadelphia Section of ASME as well as IEEE-USA, who generously provided funding for this project.
• I would like to thank Ilan Moyer and Johannes Schneider at MIT for their consultation and for materials they provided.
• I would also like to thank Ed Jaoudi, Andrew Ruether and Mark Davis for technical assistance.
• I would like to thank Andy Bastian for consultation and 3D printing services
• I would like to thank Cassy Burnett for her patience and help
• Finally, but certainly not least of all, I would like to thank Smitty. This machine would not have been built without his unending patience and assistance, and I would have never thought to pursue this project without his introducing me to the machine shop, and teaching me over the past four years.

Questions?

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