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ME102B - Mechatronics Design

LINEAR FATIGUE TESTER - FINAL REPORT

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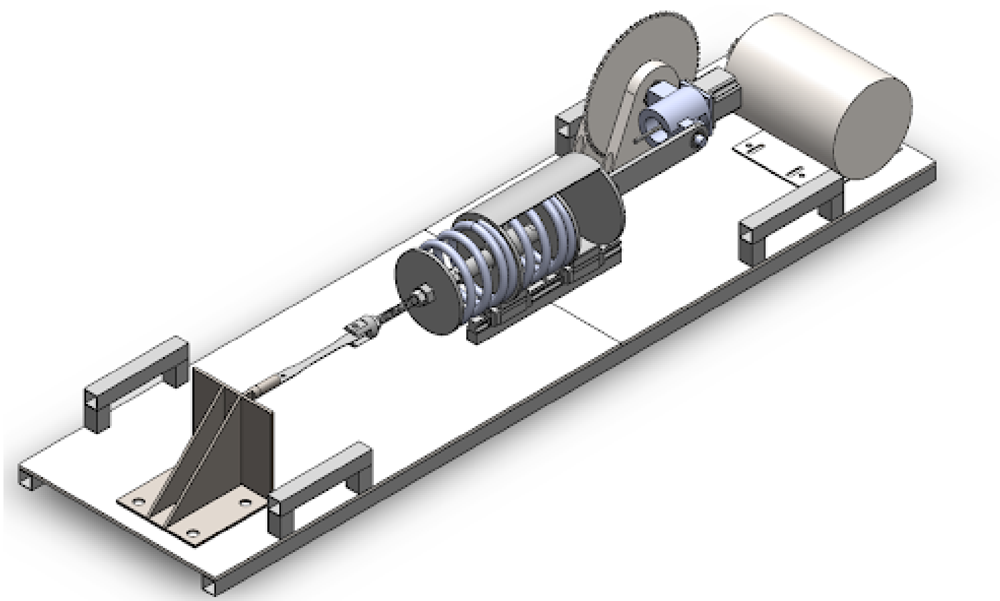
EXECUTIVE SUMMARY

Currently, the materials testing market has limited low-cost testing options for fatigue testing. The central problem to fatigue testing is that the time it takes to fatigue a sample varies directly with the number of cycles it takes the sample to fail. If each cycle takes one second, a test that goes to one million cycles (a common point on material SN curves) will take over 11 days to complete. A higher frequency testing machine is necessary in order to effectively characterize high-cycle fatigue behavior of a material.

The RR Moore test on a spinning loaded cylindrical beam is a good solution for testing raw materials. This tester has a repetition rate of up to 100 Hz or higher, allowing samples to undergo high cycle fatigue in a matter of only minutes. However, because it requires a circular cross section sample, specimens that come out of a sheet or have more complex geometry cannot be tested. One use for fatigue testing that we want to explore is the fatigue life of welded joints. It is possible to program a standard tensile testing machine to perform the repetitions. However, a typical testing machine like the Instron 3300 Series Universal Tester has a max crosshead speed of 20 in/min. [1] At a rate of 10 Hz, the machine could only test samples displacing 0.017 and that assumes infinite crosshead acceleration, as well as the controller being able to operate effectively at these speeds. To test the specimens we would like to test, such machines would operate at a maximum of 1Hz only, causing tests to last multiple days at least.

Our fatigue tester makes use of a powerful AC motor driving an adjustable radius con-rod system through a transmission to push or pull specimens. Tests can be run at different speeds by swapping out sprockets to adjust the gear ratio. By adjusting the radius of the con-rod pin, we are able to adjust the linear displacement at the end of the arm. Finally, we make use of a spring fixture between the con-rod arm and the test specimen to convert our displacement control into force control. Our system is being designed for welded joint testing. However, the flexibility of our system which includes both speed and force control will allow the machine to run a wide variety of tests beyond this first use we conceived. We plan to build to a table top size limitation and under 200 lbs total weight for feasibility of mobility. We believe this is achievable with our parameters.



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1 INTRODUCTION

BFR (Berkeley Formula Racing) is an engineering design competition team that provides ambitious college students the unique opportunity to enhance their engineering design skills through practical application. We create a formula-style, single-seat race car over the course of a school year in order to participate in FSAE Lincoln, a competition between 80 teams in Lincoln, Nebraska, every June. The competition is comprised of several dynamic events to test the vehicles performance and reliability.

The points distribution and track layouts at FSAE Lincoln favor lightweight, agile vehicles over heavy, powerful ones. Our vehicle makes use of a 350cc single cylinder engine. Many of the vehicles we aim to compete with run larger 600cc four cylinder engines. While our smaller, lighter engine gives our car the edge in sharp turns, the more powerful cars can cover a lot of ground quickly during a straight. In order to keep up, we must keep the mass of our car to an absolute minimum to mitigate the effects of having less power.

Our chassis is a welded 4130 space frame weighing 63 pounds. It takes roughly one month and three thousand dollars to build. Given the significant investment of time and resources that it takes to build our chassis, we are unable to plan for a second iteration if something goes wrong. Weld failure is a common chassis failure mode and has happened to us in several past seasons. A failed weld can cause anything from a drop in performance (the chassis exhibiting lower stiffness than was designed) to a serious danger if the engine becomes unconstrained or the structure tears itself apart due to the compensatory loads. For the past two seasons, we have used an extremely conservative estimate of the strength of our welds in order to avoid these types of costly failures.

By using a conservative estimate of weld fatigue strength, we tend to use thicker tubes, additional tubes, or both. All of this adds weight to our vehicle. A more accurate estimate of weld strength would allow us to thin and/or remove tubes from our chassis, decreasing the weight of our chassis by as much as 10 lbs, or 15%. In addition to the direct performance benefit of lower vehicle mass, showing the competition design judges the analytical approach we took to removing that mass can score additional points for our team.

2 SPECIFICATIONS

From our QFD analysis, we determined that the most important specification is our motor power. We need to be able to apply 1000 lbf at at least 10 Hz frequency. Adding a slight safety cushion, we selected a 5hp motor geared down to achieve 12 Hz. Our QFD analysis leads us to believe that a larger heavier machine is acceptable as long as we achieve our performance goals in terms of force and frequency. We would like to set a size requirement such that the machine is able to fit on a standard table, so we will set a limit of 2ftx3ft footprint. In height, we will be limited by high moments on the machine from going too high. We expect this factor to limit our design to less than 12 inches high. In terms of weight, our specification stems from a 2-person lift requirement for mobility of demonstration. To achieve this, we will design handles into the base-plate so it can be lift from a point significantly above its center of mass for stability, and we will also limit the machine to 200lb, for a 100lb/person lift. Standard for these types of materials testing machines is that they require heavy moving equipment and can be more than 500lbs, thus we believe mobility to be an advantage of our project over industrial standard.

Reliability is an important spec for a piece of testing equipment such as this. Because we allow for a larger weight, we expect that there should be a high reliability. All components will be designed to their endurance limit. We expect and plan for some simple electrical failures occasionally, but we will design for a much longer lifetime on the assumption that this prototype can be useful to vehicle teams much past the end of this class. All bearings and motion parts should be designed within their load specifications, and we allow for occasional upkeep such as re-greasing.

For user interface we require that the machine can take input of force amplitude over a simple UI and automatically adjust the radius to this setting. We also want to ensure the test can be stopped automatically after sample failure. All of this should be controlled using a simple digital user interface. We design to allow some user adjustment of simple parts to change between various testing procedures, such as changing from tension to compression fatigue or to a simple monotonic tensile test. This adjustment should be very simple and take less than 5 minutes for an experienced user. Additionally, the stress over time and cycles to failure should be output simply when connected to a computer. We set all these specs with the arduino serial interface in mind, and we expect the user to have some simple understanding of how to download data from the arduino serial port, though the data should be simply formatted and easy to read.

3 CONCEPT GENERATION

All of our concept generation started from the same point: we wanted to produce a machine capable of characterizing the fatigue strength of welded test specimens. We also knew that we would be funding the project ourselves and that the machine had to operate in Jacobs Hall. Establishing these basic constraints was the very first step of our process.

Next, we made a group message. Members would sporadically suggest concepts. Some concepts were shot down immediately. Others were briefly discussed, then shortly ruled out. Ultimately, three concepts were seriously considered as options that could meet our design goals.

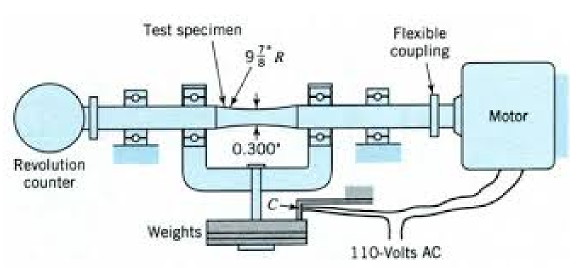

FIGURE 1. RR MOORE DESIGN

The first concept was the RR Moore fatigue testing machine shown in figure 1. There is one in Hesse that is available for students to use shown in figure 2. The tool has been in use for decades and has proven itself to be an effective, reliable fatigue tester. [2] We believed we could build our own tester for less than we could buy one for if we wanted to have a machine at the BFR garage.

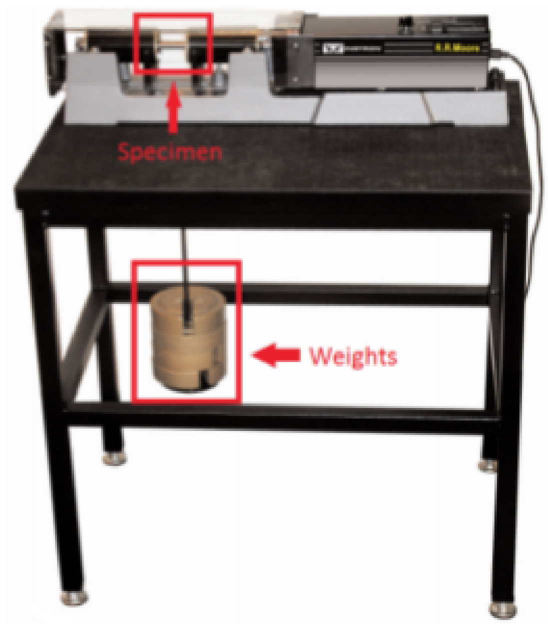

FIGURE 2. RR MOORE MACHINE

The second concept we considered was the use of a pneumatic piston running on shop air or a hydraulic piston driven by a hydraulic pump. The test specimen would be fixed at one end and attached to the cylinder at the other end as shown in figure 3. Force would be set by a pressure-regulator between the fluid supply and the piston. A solenoid would flip the piston between pushing and pulling on the specimen.

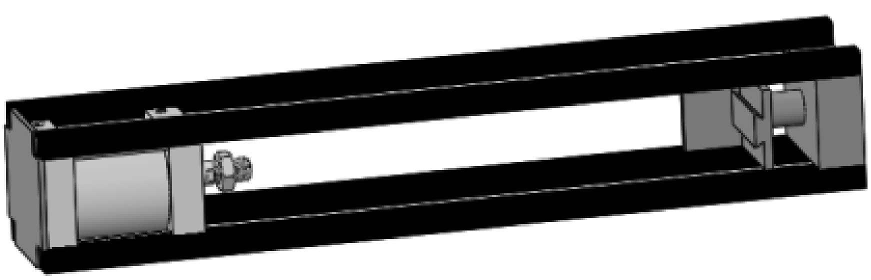

FIGURE 3. PISTON FATIGUE TESTER

The third and final concept was a con-rod spring system. A high power motor drives a con-rod through a transmission. The con-rod pin lies on a lead screw such that the radius at which it rotates can be adjusted. Radius adjustments are used to control the linear displacement of the end of the con-rod arm. Displacement control is converted into force control by means of springs that are fixed to the con-rod arm at one end and the test-specimen at the other end as shown in figure A.7.

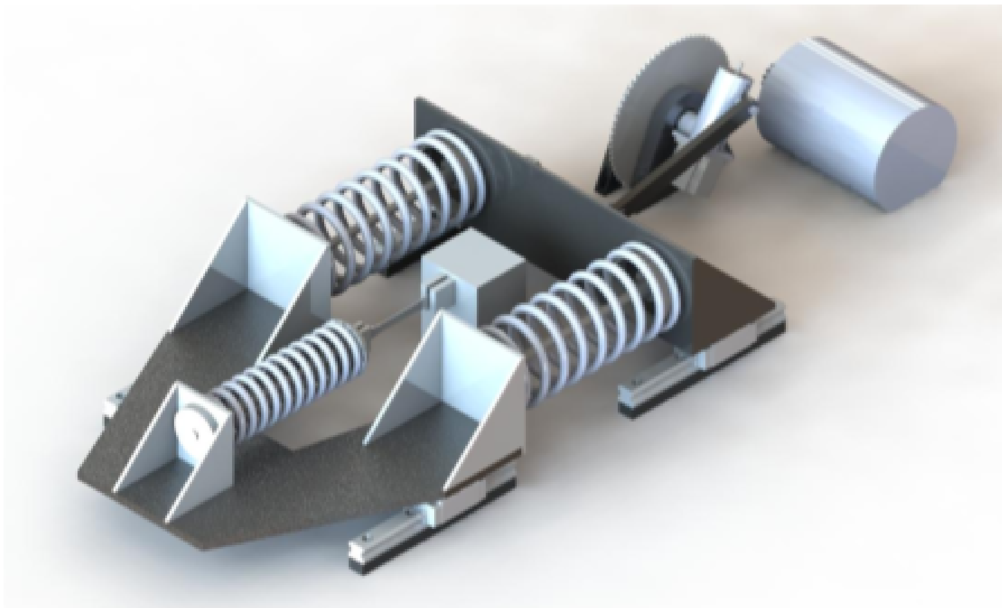

FIGURE 4. CON ROD FATIGUE TESTER

4 CONCEPT SELECTION

Now that we had some concepts brainstormed, we needed to choose our best one. The most important elements we decided to use in our scoring matrix were machinability, design simplicity, support for testing of non-cylindrical parts, novelty, cost and robustness. First, we wanted a design that we could easily manufacture, and the largest component of the process is machining. Thus, choosing a design that we could machine was an important factor. Another aspect of manufacturing is assembly, so choosing a simple design with fewer assembled components was also desired. One of the main motivations for this fatigue tester was the ability to test line welds and parts with a non-circular cross section, so this was a critical criteria for our design. Similarly, broadening on this idea, we wanted to choose a design that was novel with unique capabilities compared to existing fatigue testers. We needed a robust design that would be able to withstand high frequency and high force loads. Finally, since our project is self-funded, we needed a design we could afford to build. See the scoring matrix below for the breakdown of each design, 1 being the worst, 5 being the best for each category.

Concept	Machinability	Simplicity	Non-Cylindrical	Novelty	Robustness	Cost	Total
R.R. Moore	3	2	1	2	5	4	17
Pneumatic Piston	3	4	5	4	2	3	21
Hydraulic Piston	3	4	5	4	1	2	19
AC Motor Con-Rod and Spring	3	3	5	4	4	4	23
DC Motor Con-Rod and Spring	3	2	5	5	4	2	21

TABLE 1. Scoring Matrix

As shown in table 1, we determined the AC motor driven con-rod and spring design the best all-around design.We had more confidence in the durability of the con-rod and spring displacement to apply load compared to the pneumatic or hydraulic piston. The main concern with the pneumatic and hydraulic piston designs was their durability in running numerous tests over time which applying high loads. The R.R. Moore machine is a well established and overall strong design, but it was not novel and we could not think of a way for it to support non-cylindrical parts. Thus, the final decision is the con-rod and spring design with an AC motor. The design had no major weaknesses, and its main advantage over the other designs was its robustness. The small increase in novelty of a DC motor does not outweigh the cost of an AC motor. We estimate our cost to be \$230 for the motor, \$1000 for steel and raw materials, and \$100 for electrical hardware. We hope to refine the design to make the assembly even more simple as well, which would improve perhaps its weakest attribute.

5 CONCEPT DESCRIPTION

5.1 Motor and Transmission

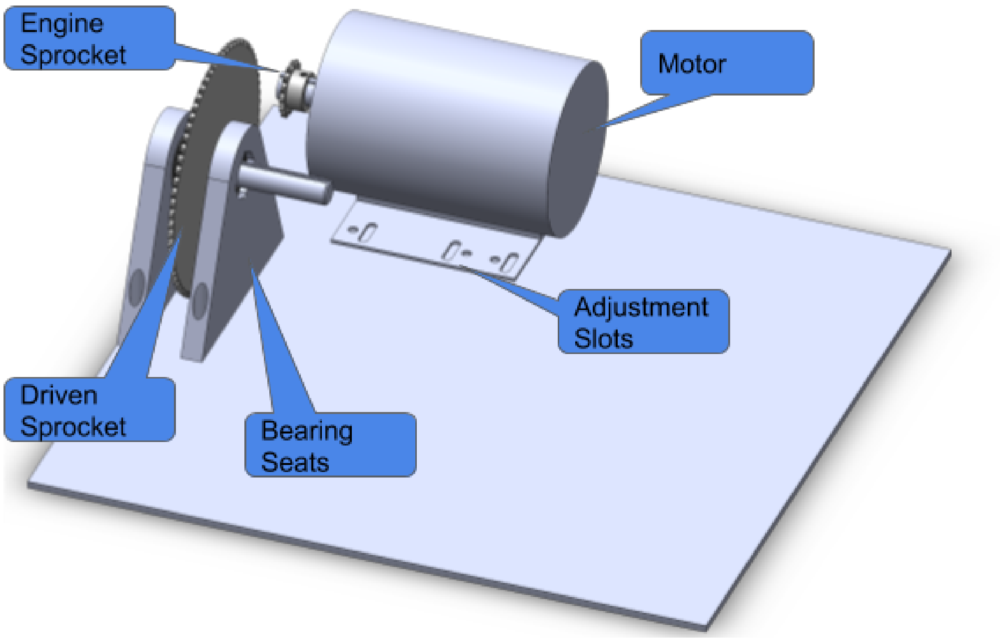

FIGURE 5. MOTOR AND TRANSMISSION

Our fatigue tester is driven by a 5 HP AC compressor motor bolted to a base plate and spins at 3450 RPM. Although DC motors are much simpler to control than AC motors, our project does not require motor velocity control. For a given power DC motor, we were able to find an equivalently powerful AC motor that was less expensive. The only major drawback to the AC motors we found beyond requiring complex controllers if we want to go beyond on/off behavior is the fact that almost all are driven at 230 V, common in shop environments but not as common as 120 V. We felt the cost benefit of an AC motor outweighed the controllability of a DC motor.

Our motor has a 7/8 output shaft with a 3/16 key. McMaster sells a wide range of sprockets that are compatible with this shaft. We knew that a testing frequency of 11.6 Hz would allow us to hit 1 million cycles in a day. While a smaller final drive ratio would increase the rate of testing, it decreases the torque that drives our con-rod, which in turn decreases the maximum force we can subject our sample to. We hope to make use of our machine without constantly changing sprockets, so we aimed for a reduction that would get us as close to 11.6 Hz as possible.

The smallest chain type McMaster sells sprockets for is ANSI 25. The smallest engine sprocket we could find has 20 teeth, and the biggest sprocket has 72 teeth. This reduces our speed to $3450RPM * \frac{60s}{1min} * \frac{20T}{72T} * \frac{1cycle}{revolution} = 16Hz$. Moving up to ANSI 35, McMaster sells a 15T engine sprocket and a 72T driven sprocket. [3] This can test down to 12 Hz, which we agreed is close enough to our desired rate. The 15T sprocket has an effective radius of 0.9. Driven by our 5 HP motor at 3450 RPM, this makes for a chain force of 103 lbs. The chain is rated to 190 lbs so our chain load factor is 1.85.

The driven sprocket is attached to a shaft supported by two ball bearings pressed into steel bearing seats that are bolted to a base plate. The bearings have a dynamic load rating of 2250 lbs and a static load rating of 1300 lbs. [3] Our dynamic load factor is [2250 lbs/(103 lbs/2 bearings)] approximately 43, and the static load factor is not of much concern as each bearing only needs to statically support half of the weight of our driven sprocket, driven shaft, and adjustable radius pin combined.

Chains are known to stretch with use. [4] This could introduce slack into our chain and dramatically decrease our driveline efficiency, wear out our sprockets, or even throw the chain off in the most extreme case. The motor mounting bracket has slotted holes that we will use to move our engine away from the driven sprocket, removing slack in the chain. The slots could also be used to take up slack that could be introduced if we move to a smaller driven sprocket.

5.2 Variable Radius Con-Rod Pin

The variable radius con-rod pin is pivotal to the utility of this device because it allows the user to automatically adjust the force amplitude applied by the machine to the test sample. Obviously for fatigue testing this is a critical parameter, and the adjustment should be simple and repeatable. Additionally, this pin will see very large force in repeated loading. Therefore the pin needs to be constructed with a high factor of safety to the endurance limit of the pin while being adjustable. In the final design, we set a factor of safety of 2.49. Finally, the adjustment of the pin must be motorized in a way that allows for the pin to be adjusted and then not move for the duration of the test.

The adjustable radius pin will be designed to be supported by a wide thick walled tube to take all the bending moments seen by the rod. The only load that should be taken by the power screw adjuster is a direct axial load. To ensure that the pin does not move during the test, we employ a self locking screw. The motor needs to be easily controllable to a specific length, and for this reason we use a stepper motor. An additional advantage of using a stepper motor is that it can be held at a dead stop by force, an additional constraint that will maintain a constant force amplitude.

5.3 Con-Rod Driven Carriage

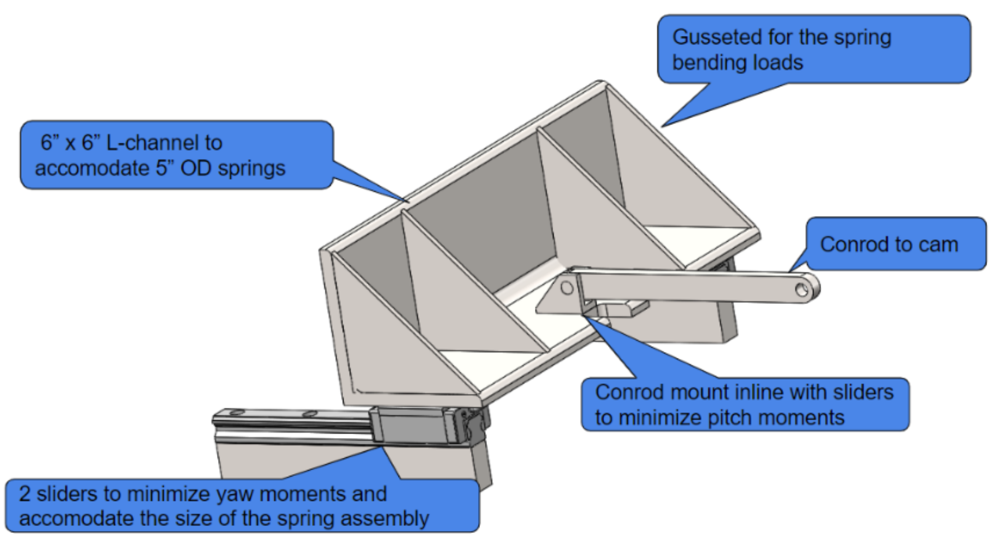

FIGURE 6. Carriage

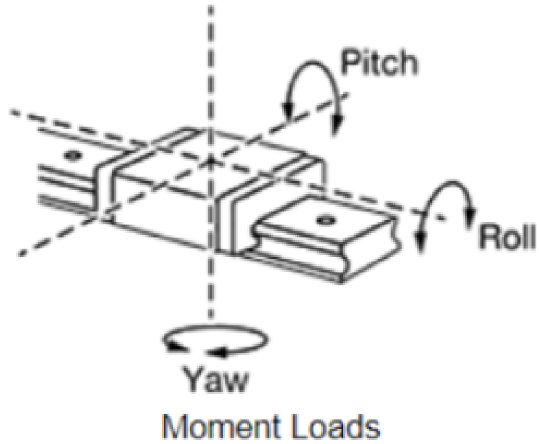

FIGURE 7. PITCH, ROLL, AND YAW

The con-rod driven carriage is responsible for converting rotational motion of the motor into linear motion in the most efficient way possible. This requires a design that minimizes forces and moments reacted by the bearings. Because there is a large variation in the forces the con-rod applies to the carriage, the carriage design also depends on reliable, well-built linear bearings that will minimize efficiency losses due to friction. We have chosen a ball bearing carriage as opposed to a cylindrical linear bearing because of the carriages greater ability to react moments. Two side by side rails are required so that the force from the con-rod can be shared by two bearings and prevent the creation of yaw moments on either bearing. In the case of a single linear bearing, one would end up with a solution that works less efficiently because of the large yaw moment applied. It is not possible without very large diameter springs to have the carriage in the same axis as the primary forces. These off axis forces create the driving load case for the linear bearings, the pitch moment. Given the size of our springs the 1000 lb load would have to be applied at 6 inches above the axis of the linear bearings. Therefore, the linear bearings have to be rated for 500 ft-lbs of pitch moment. The ball bearing carriages selected are designed for 305 ft-lbs each and work with 28mm rail widths.[10]

The linear bearings sit on mounts so that the their axis is at the same relative height as the center of the adjustable radius. This makes it such that the distance traveled by the carriage forward and backward are the same. On top of the carriages sits a piece of 6 x 6 steel L-channel which also acts as the flat surface to apply load to the springs. The L-channel is reinforced with welded steel gussets to brace against the high bending loads the L-channel will see from the springs. The steel con-rod mount was placed in line with the two linear bearings to minimize their pitch moments from the con-rod. The mount is designed as a reinforced clevis that can take the bending loads from the con-rod.

The con-rod itself is designed to be as short as possible without having the carriage collide with the transmission assembly so that buckling is less of an issue under high compression loads. The two ends of the con-rod contain simple press fit roller bearings that will react radial loads and allow the ends of the con-rod to rotate with ease.

The components of this system are exclusively made of steel because they need to not fatigue. All components were designed to see stresses below their respective endurance limit with a minimum factor of safety of 1.5.

5.4 Spring Specimen Fixture

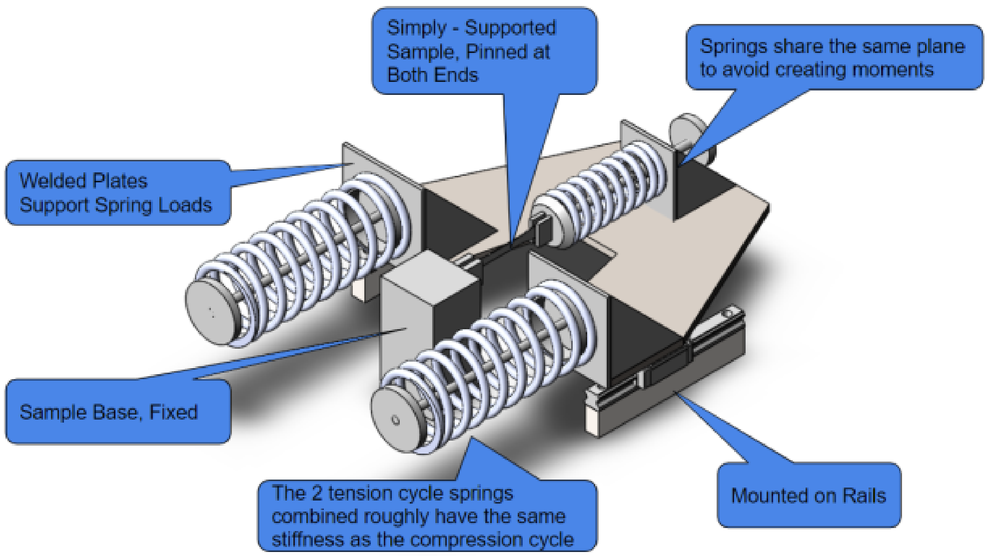

FIGURE 8. SPRING FIXTURE

The spring specimen fixture shown in figure 8 is designed to use compression springs that can apply compression and tension loads to a simply supported specimen. Only compression springs are used because of the lack of tension springs on the market that have high enough springs rates for our application. Of those springs that do, none of them are rated to the high displacements that are required by our machine. Therefore, this design relies on compression springs that are loaded in different directions to create tension forces on the sample. In order to load the springs in different directions the spring carriage is connected to the con-rod carriage by shafts that ride through cylindrical linear bearings. These shafts have shaft collars on the end which allows the con-rod carriage to rigidly pull on the spring carriage. This compresses the single compression spring by the chosen radius and applies a compression load to the sample as seen in figure 9.

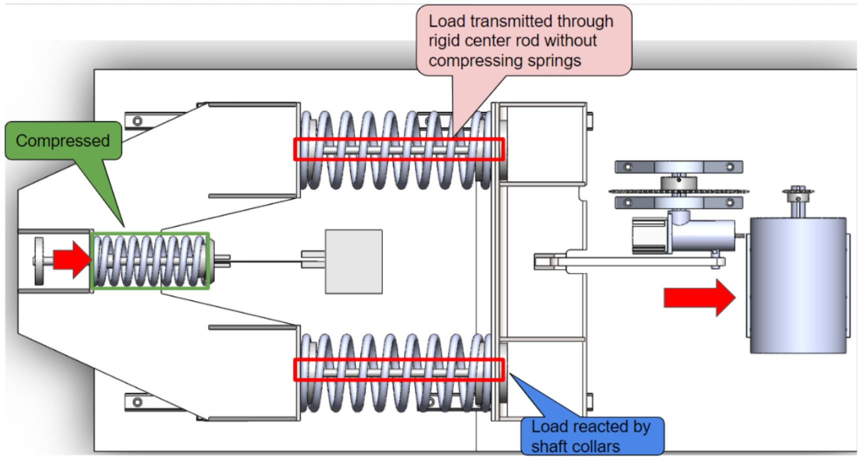

FIGURE 9. COMPRESSION

The tension cycle is performed very similarly to the compression cycle as seen in figure 10. In the tension cycle the upper and lower springs are both compressed and rigid shaft and specimen are put in tension. Similarly, spring load is reacted through the shaft collar to create the desired load.

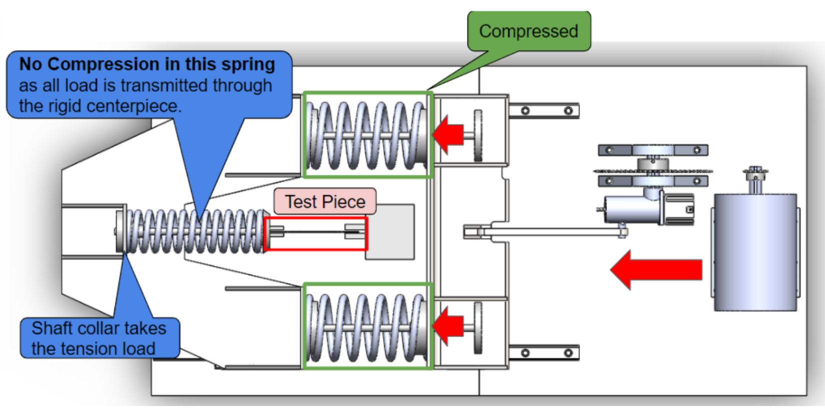

FIGURE 10. TENSION

The two springs were chosen such that their combined spring rate was within 7% of the single compression spring. Ideally they would have the same spring rates, but this slight loss of accuracy was considered acceptable given that custom springs would cost an additional $250.[9] Spring choice was primarily driven by a spring rate that gives 1000 lb at our max radial displacement. Because of the relatively high displacement required by the springs, springs that are rated up to 1000 lbf are no shorter than 10 and largely determine the overall size of this machine. Spring perches sit on either side of each spring so the springs are retained to a single axis.

The test specimen is fixed in place by rigid steel mount. This mount is made adjustable by slots in the baseplate to accommodate different sample sizes. Currently, the maximum specimen length is 6.5 which is equivalent to the ASTM standard material testing dogbone size.[6] The specimen is pinned to clevises on either side because it is much more simple to have test pieces that interface with bolts as opposed to a clamping design. Fixtures that clamp constrain the specimens that can be tested to primarily rectangular samples. Machining clamping fixtures would also be significantly more complicated than a clevis.

The design methodology of the spring carriage is largely the same as the con-rod carriage. In this case the steel baseplate has plates welded to it to brace against the springs. Those spring plates have welded gussets to support against the bending loads from the springs. In this case the rails arent as critical and in a perfect world would not see any load. Regardless, rails are necessary to maintain alignment and ensure that the test specimen doesnt see any off axis loads that would result in bending.

5.5 Load and Displacement Measurement and Control

Based on the input of the user for a desired load, we will need to adjust the radius of the flywheel to ensure we have the proper load applied. Thus, we will need a control system for the radius of the flywheel, which is trying to meet a reference load, and we can feedback our actual load being applied.[7] The required components for this control system are an Arduino, strain gauge, optical encoders, and a stepper motor. The strain gauge is to give us the actual applied load to the specimen that we can use for feedback. The optical encoders will give the position of the lead screw, and thus our radius. The stepper motor will move our radius. Some advantages of a stepper motor are that it is quantized making for potentially easier control, and additionally will resist changes from its current slot which will keep the lead screw in place when desired. While we recognize that stepper motors are prone to skipping steps, we feel like it is a strong solution and we can account for that issue if it arises. The Arduino brings it all together, as it will contain the code for the control system, read in the strain gauge output, apply the proper voltage to the stepper motor, and read from the encoders to determine the position of the lead screw.

6 PARAMETER ANALYSIS

6.1 Motor and Transmission Design

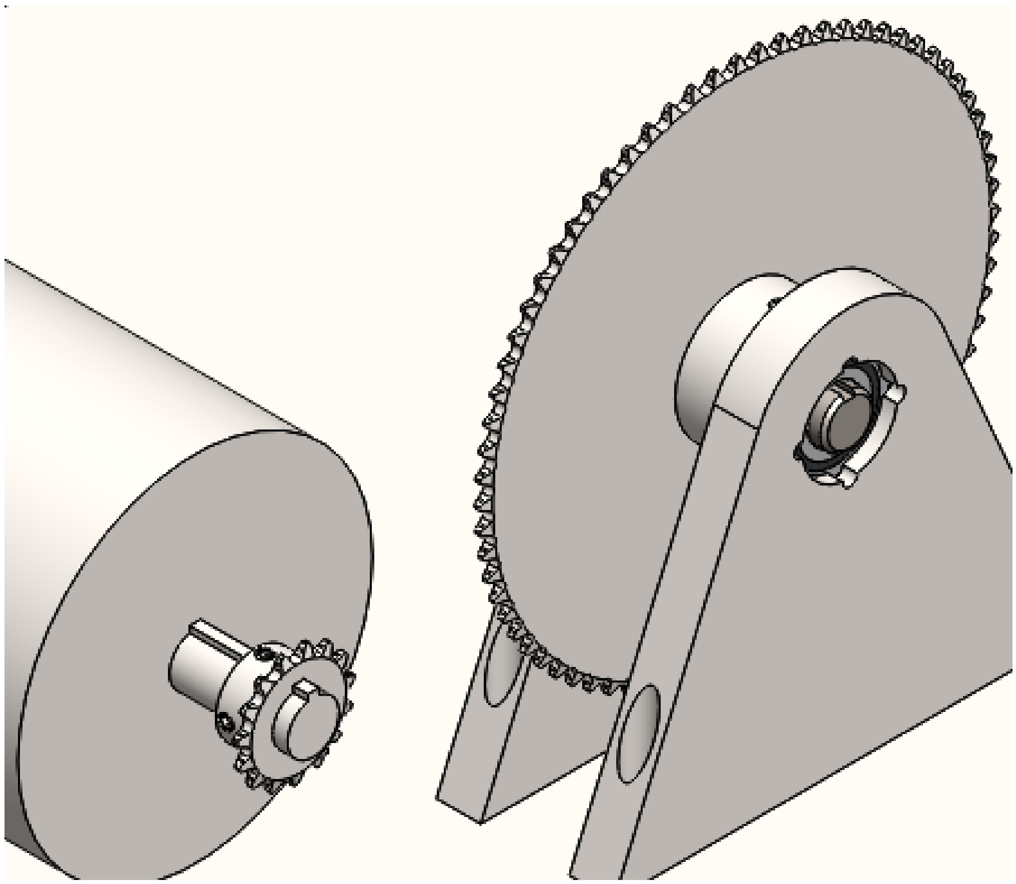

FIGURE 11. MOTOR AND TRANSMISSION

We chose a 5 HP AC compressor motor to power our fatigue tester because it was a low cost (¡ $250), high power motor that was well suited to our tester. The motor spins at 3450 RPM, which would correspond with fatigue testing specimens at 57.5 Hz. Our con-rod setup can apply the most force when the con-rod is tangential to the path of the adjustable radius pin. Unfortunately, this point very nearly coincides with the point where the spring is at its relaxed length during full tension/compression cyclic testing. When the springs are fully compressed or extended, they are normal to the con-rod pin path and the motor can exert no force on the spring. The system must rely on inertia to get through these extreme points ($\theta = 0, \theta = 180$), and a high motor force to reach them (the restoring force of the spring works with the rotation of the assembly as the spring goes from fully extended to relaxed and fully compressed to relaxed) as the force exerted on the spring by the motor is $sin(\theta) * \frac{T}{r}$

With this in mind, we decided to choose a gearing reduction such that our machine would run at the lowest acceptable rate, which would correspond with the maximum achievable torque at the driven shaft that our motor could supply at our chosen frequency. We wanted to be able to accomplish 1 million cycles in a 24 hour period, which corresponds with testing at a frequency of 11.57 Hz. We found 15T and 72T ANSI 35 sprockets on

McMaster. This reduces the frequency to $57.5Hz * \frac{15T}{72T} = 11.98Hz$, which we deemed close enough for our needs. 5 HP being delivered by a shaft rotating at 3450 RPM corresponds with an output torque of $5HP * \frac{396000}{2*\pi*3450RPM} = 91.3in-lbs$. Torque at the driven shaft is increased by our gearing reduction to $91.3in-lbs * \frac{72T}{15T} = 438in-lbs$.

ANSI 35 was initially selected because there was a wide range of set-screw style sprockets available on McMaster. The motor shaft outputs 91.3 in-lbs of torque. A 15T ANSI 35 sprocket has a pitch diameter of 1.8037. The load in the chain is therefore $\frac{91.3in-lbs}{1.8037"/2} = 101.5lbs$. ANSI 35 chain sold on McMaster is rated to 190 lbs, so we have a relatively comfortable load factor of 1.8 for our chain. While McMaster does not list a spec for the strength of their ANSI 35 sprockets, we assume that the sprockets are stronger than the chain theyre designed to work with.

Both the cantilevered motor shaft and cantilevered end of the driven shaft are themselves subjected to cyclic loading during the operation of our device. Both are steel parts with endurance limits, so we wanted to verify that neither shaft would be subjected to stresses above their limits. They see a cyclic bending moment in addition to a cyclic torque (when there is little to no load on the motor, there is low torque in the shafts). Because there are a wide variety of ways we could use the machine, we settled on a worst-case model where the shafts constantly output peak torque and must deal with cyclic peak bending moments. The stress calculations are outlined in the appendix. Between Goodman and Gerber fatigue failure criteria, Goodman was the more conservative model. The motor shaft sees an effective Goodman stress of 2752 psi. While the grade of steel for the output shaft of our motor is not listed, we went with the lowest UTS steel spec we could find on matweb which was 42000 psi. Using the rule of thumb that for most steels, endurance limit is half of UTS, we found a fatigue load factor of $\frac{42000/2}{2752} = 7.6$. For the driven shaft, we calculated an effective Goodman stress of 33432 psi. We estimated the UTS of the hardened 1045 shaft to be 98000 psi (an EMJ metals hardenability chart for lower carbon content 1042 steel showed a UTS of 98000 psi for the same hardness). Knowing this, we found a fatigue load factor of $\frac{98000/2}{33432} = 1.4$. While this is a much tighter margin than for the motor shaft, it still puts it below the endurance limit.

Finally, a quick check was performed to verify that the chosen bearings would hold up to the dynamic loads of the tester. The calculations are shown in the appendix. The maximum load reacted by the transmission bearings is 1811 lbs. They are rated for 2250 lbs of dynamic load, which gives us a bearing load factor of $\frac{2250}{1811} = 1.2$. If the bearings end up having a low life because they are operating near their rating, we can simply replace them. They are less than $20/bearing and replacement only requires a simple press out/press in operation.

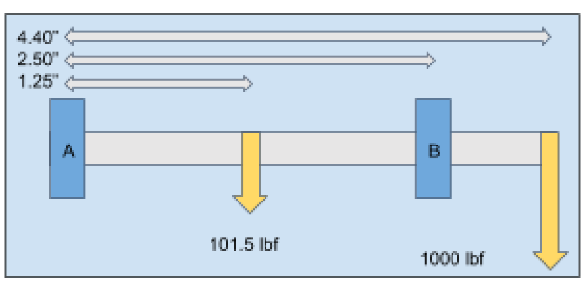

FIGURE 12. BEARING REACTIONS

6.2 Adjustable Radius Assembly

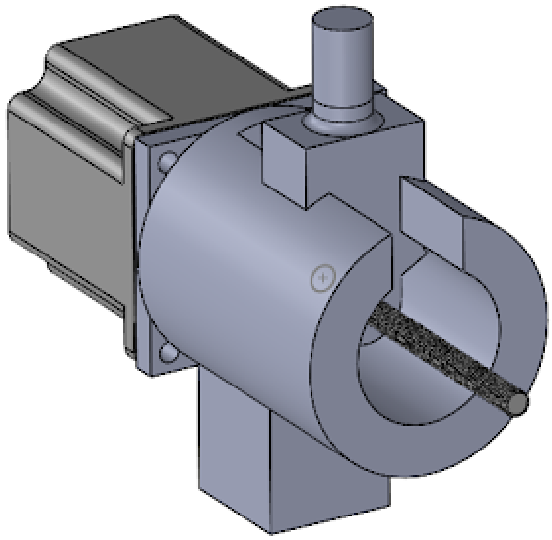

FIGURE 13. ADJUSTABLE RADIUS

The adjustable radius design sees very high stresses in the base of the pin on the slider. The two variables we can adjust are the thickness or shape of the pin and the location of the applied force. Because the shape we are analyzing is very short thick, we cannot use beam theory. Thus we must iterate using FEA analysis. We designed this part to a conservative safety factor of 2 for the endurance strength of 4130 steel. We chose 4130 steel for this part because it has high fracture toughness and fatigue properties, while still being relatively cheap. We also want to choose material that we can weld easily, because there are two welded parts on this assembly. We want to maintain clearance between the conrod and the pin allowing for space for possible deformation, so it is not optimal to place the conrod right at the base of the pin. We were able to achieve the desired safety factor with a 5/16 offset, which we deemed acceptable. The deformation and stresses in the remaining cylinder and shaft are very small relative to the pin, so these parts were excluded from the final FEA analysis to allow more detail in the more critical pin.

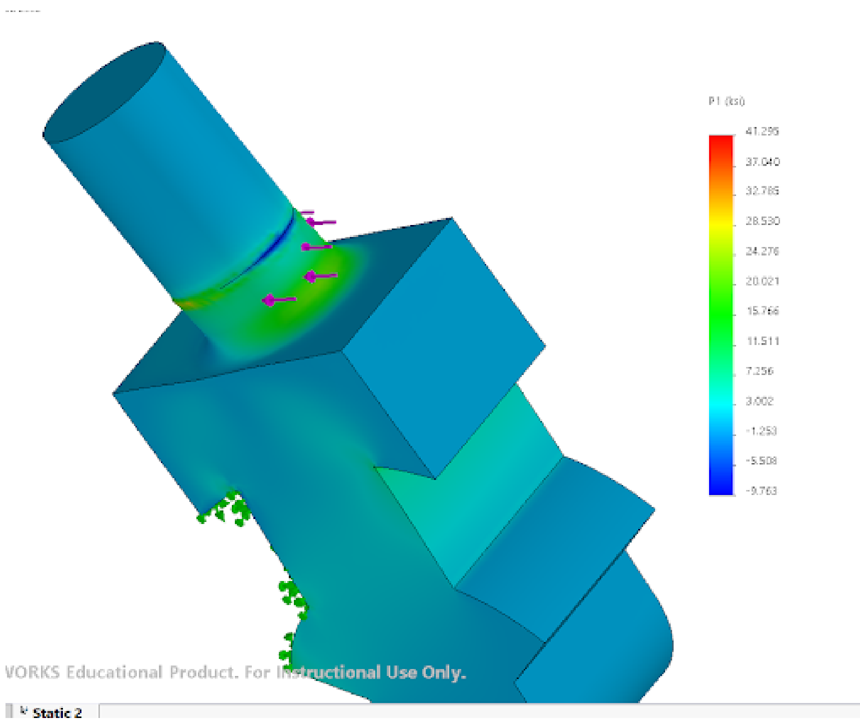

FIGURE 14. MOTOR AND TRANSMISSION

The next important design parameter is designing the slider interface. It must be highly toleranced on the contact surface between the adjustment tube and the slider because this will be a sliding wear surface, and the conrod force could impact loads if they are allowed to develop separation. Thus, these parts will designed to be waterjet cut and then CNC milled to finish. We believe that with machine shop capabilities available and the experience we have machining on our team we can achieve a .005 tolerance on this part. This should be sufficiently tight to prevent impacts.

6.3 Con-Rod/Spring Assembly

The most important design choice for this assembly is spring selection. Springs were chosen such that they can be preloaded to our desired max load on the sample and still compress to enough to generate the max compressive and tensile forces on our sample. This is to ensure that the springs never break contact with the spring perches and create impact forces as well as a ton of noise. Because the assembly is moving at 10 Hz, impacting every cycle would prove to be detrimental for fatiguing our machine itself. The resulting requirements are that the [maximum rated load ¿ 1.5 * spring rate]. The other requirement is that the spring can achieve our desired max load. The max load the spring can generate is dependent on the displacement of the adjustable radius and its spring rate, but is ultimately limited by the available torque from the motor. Our motor has an output of 442 in-lbs. Therefore if we want to the load the specimen to 884 lbs the radius would need to be set to an inch and the resulting spring rate would need to be 1768 lb-in. This is a very large spring rate but is slightly alleviated by the fact that the preloaded springs have effectively double their original spring rate. As you load one side of the spring at its original spring rate you are also reducing load in the opposing direction at that same spring rate. Spring selection was resultantly very limited to high spring rate, high max load springs, that were long enough to compress significantly. After compiling a spreadsheet of springs that fit theses specs we decided on the smallest possible springs so that our machine doesnt become needlessly large.

After springs were chosen, everything else was designed around that size. The cylinder body that applies load to the springs was designed for minimum clearance with the springs. It needed to become a half cylinder to make assembly possible and was also required for any maintenance that would potentially need to occur. The main load constraints on the cylinder body are the large compressive loads that could potentially buckle the body that potentially at risk due to being open on one side. A SolidWorks buckling FEA was run to ensure that the structure would not buckling in this situation. This was performed by fixing the fixed specimen side of the cylinder and applying a 1000 lbf compressive load on the con-rod side of the cylinder . Because the full load was used in the simulation, the resulting load factor is essentially the buckling factor of safety.

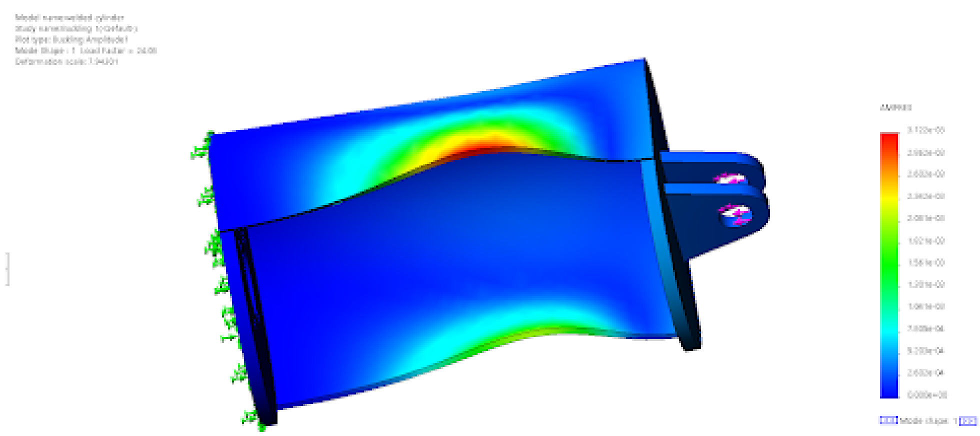

FIGURE 15. CYLINDER BODY

The linear rail carriages are attached to the cylinder body by small blocks that are welded to the cylinder body. 2 Carriages were necessary because 1 carriage would not be able able to react the full pitch moment applied to it. The 1000 lbf load applied 5 above the carriage axis results in 416 ft-lbs of pitch moment, which was compared to the rating of 305 ft-lbs. By using 2 carriages this moment is reacted by load sharing between the 2 carriages and therefore the smallest available carriages and rails were chosen.

The con-rod attachment to the cylinder body is a double shear pinned connection. The tabs were designed such that the shear stresses generated in the pin were below the fatigue limit [1000 lbf/ (0.79in^2 *2) = .63 ksi]. Hertzian contact stresses determined the pin hole size such that the hole would not ovalize or fatigue. A .005 clearance hole resulted in 13 ksi which was deemed adequate. The con-rod bearings were sized to be the smallest bearings that were rated to 1100 lbf or greater. Although a factor of safety of 1.1 isnt great for this application, manufacturer ratings are almost always under specced.

The con- rod was designed to minimize its length to mitigate buckling under max compressive loads. To provide 0.25 minimum clearance between moving part the minimum con- rod length was 7 bearing center to bearing center. Using a pin- pin boundary condition for Euler buckling we get a critical load of 73300 lbs. The resulting factor of safety becomes 73.3.

6.4 Specimen Fixture

The purpose of the specimen fixture is to mount the specimen and collect real-time load data that can be fed into the control loop. It is rigidly attached to the machine base and does not move. Because of this, weight is much less of a concern than other parts in the machine. The main objectives for this assembly are resistance to fatigue, ability to accurately and reliably measure loads, and low cost and ease-of-assembly.

Steel is the obvious choice to meet the first requirement as it has an endurance limit. Stresses under the limit do not damage the part. For 4130 steel, the endurance limit is 49 ksi. All stresses will be kept below this limit. Secondly, strain gauges are an easy and capable method for getting load data, and one we have considerable experience in. Strain gauges work best when there is a thin section to deform relatively highly compared to the rest of the part. Thin sections under compression carry a risk of buckling, so we will need to ensure that this does not happen. We will be using a full bridge setup. This allows us to negate bending and heating effects, although both are unlikely to occur. It also allows us to amplify the output, meaning we can get the same resolution with a stiffer load cell.

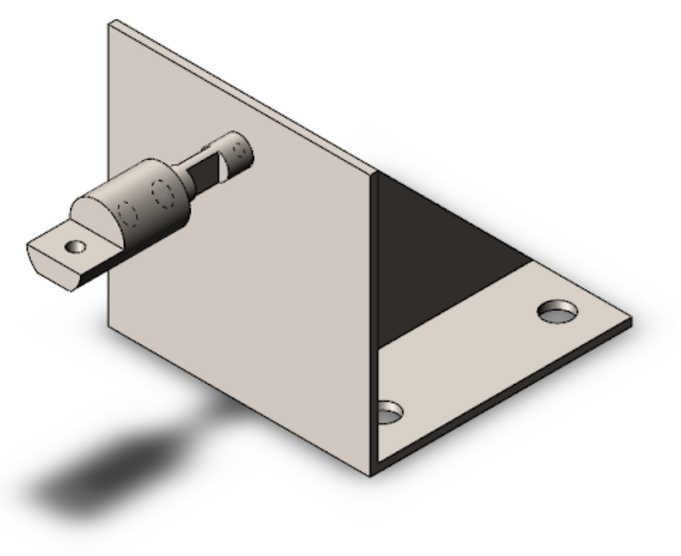

FIGURE 16. SPECIMEN FIXTURE

7 FINAL DESIGN

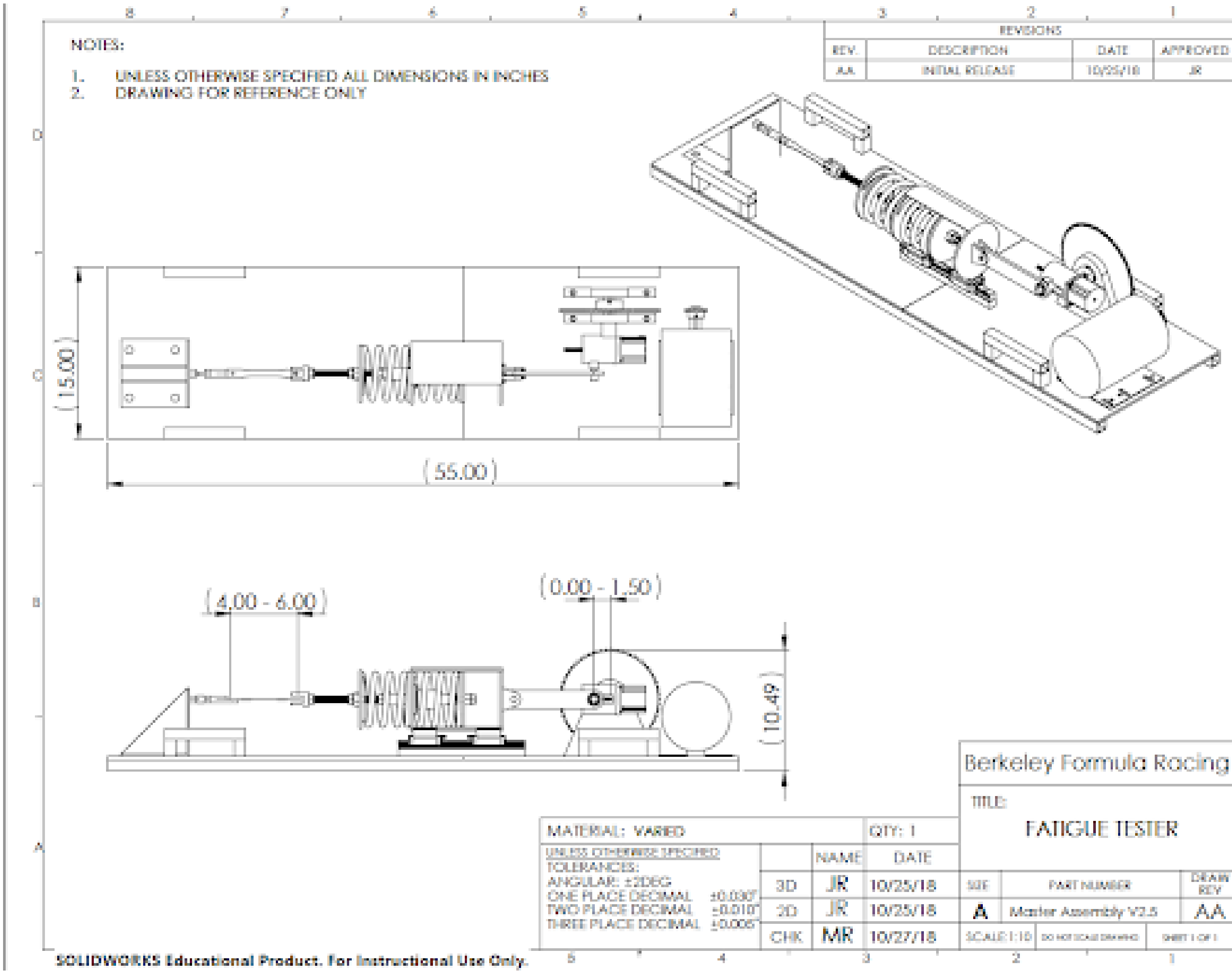

FIGURE 17. FINAL ASSEMBLY DRAWING

7.1 Motor and Transmission Assembly

ANSI 35 sprockets are mounted to the engine output shaft as well as the driven transmission shaft. The two sprockets are connected by ANSI 35 roller chain. While there are calculators for chain wrap and estimating the correct center to center distance for two sprockets of known pitch diameter and chain of known pitch, we are simply connecting the motor to the base plate with 4 slotted holes. We can break our 5 foot chain down to any number of even links, connect the two, then take the slack out of the chain by moving the motor as far as possible from the transmission shaft with the chain on, then tightening the motor down. We will use 1 slots which can take up several chain-pitches worth of slack. The engine sprocket will be constrained radially by set screws, and axially by snap rings. The axle sprocket will be constrained in a similar fashion. Snap rings will also be used to constrain the transmission shaft relative to the transmission bearings, which will be pressed into their seats which are in turn bolted to the base plate.

7.2 Adjustable Radius Assembly

The adjustable radius assembly will be powered by a self locking power screw to adjust the location of a pin that attaches to the con-rod. This will be actuated to change the applied force on the specimen. There will be a large cylinder to take the high bending loads off the power screw. The motor will adjust the radius when the machine is not running, so it will need a low torque to overcome only the friction on the slider along the cylinder.

7.3 Con-Rod/Spring Assembly

The con-rod spring assembly is driven by the adjustable radius output from the motor. Through the con-rod it converts rotary motion to linear motion in a way that reduces as much friction as possible. In operation the cylinder body moves with the displacement of the adjustable radius and the spring axis stays in place loading the specimen depending on how much the cylinder body displaces the springs. The springs will be assembled in the cylinder body and preloaded to the absolute max load of adjustable radius output. Assembly is made possible by the cutout in the cylinder wall and preloading is performed by screwing the spring perch further down the threaded rod that is used as the spring axis. After the springs are preloaded in place the clevis that attaches to the specimen is attached and this assembly with run without any user input.

7.4 Radius and Motor Control System

The role of the control system is to integrate the user interface with the adjustable radius and motor control. The system centers around an arduino, which is connected to three components. It is connected by a relay to the motor to turn the motor off and additionally count the number of cycles of the motor shaft by use of an optical encoder. The arduino is also connected to a stepper motor, which will control the radius position also using an optimal encoder for a position readout. Lastly, the arduino will read the output of the strain gauge and strain gauge amplifier coming from the specimen. The arduino will be powered by a user laptop, where the user interface will appear. See figure 18.

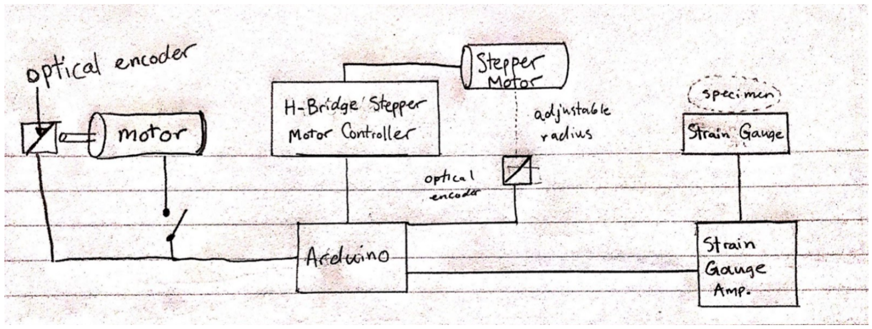

FIGURE 18. CONTROL SCHEME

Functionally, the control system is integrated with the interface of running a test. First the user will input a desired force amplitude to test the specimen, and additionally choose whether the test is pure tension, pure compression, or both tension and compression. After these inputs, the user will start the test, and the first calibration cycle will begin. The stepper motor will move the radius to be a very small but non-zero distance away. Once the stepper motor is done, the main motor will activate for one second. The goal of this calibration has two main goals. The first goal is to check if the user has loaded the proper springs to achieve the desired stress ratio specified. If not, the calibration test will return an error, and instruct the user to make changes to the springs. Secondly, this test will give a data point for the output force at a given radius for the test specimen and spring combination. Using this, we can move the radius further away to achieve the inputted force amplitude. Once the radius has moved, the motor will be turned back on for a second calibration test, confirming that the new radius position results in the desired force amplitude. The fatigue test will then begin, running until the specimen fails. The control system will detect that the specimen fails when the strain gauge outputs a sudden drop. At this point, the arduino will send a signal to the relay to kill the motor. Once the motor is killed, we can read the number of cycles from the optical encoder on the motor. This will then be fed to the user interface, where we will output the number of cycles and a cycle count vs force amplitude plot, the latter of which to show the user the validity of the test. Finally, the radius will be set back to zero to prepare for the next test.

7.5 Specimen Fixture

The purpose of the specimen fixture is to mount the specimen and collect real-time load data that can be fed into the control loop. It is made up of the specimen attachment, the loadcell, and the base. The thin section in the loadcell will have a full bridge strain gauge setup bonded onto it. It will see 260 microstrain at 1000 lbf applied load, enough for a strain gauge to detect. A thin section will be at risk of buckling under compressive loads, but this piece has a factor of safety of 13 against buckling.

The base is made from welded 0.25 thick 4130 sheet steel. It bolts to the machine base with four bolts and to the load cell with one 1/4-28 grade 8 bolt. As this does not need to be removable, it will be secured with a threadlocker to prevent it being undone by vibrations. The specimen attachment has a -20 internal thread to mate to the load cell. The specimen will be bolted to the attachment with a 5/16-24 grade 8 bolt. Properly torqued, it will provide conservatively a 2000 lbf force against the applied force. It will be secured with a locknut that will be replaced after each test.

7.6 Bill of Materials

Name	Dimensions	Source	Cost
	Motor and Transmission Assembly		
5 HP Compressor Duty Electric Motor	14.85” long, 8.65” tall, 6.6” wide 7/8” shaft , 3.5” up	Walmart	$228.95
15T ANSI 35 Roller Chain Sprocket	0.75” wide, 1.99” OD, 7/8” shaft	McMaster Carr	$14.10
72T ANSI 35 Roller Chain Sprocket	1” wide, 8.81” OD, 7/8” shaft	McMaster Carr	$68.29
ANSI 35 Roller Chain	5 feet of chain 3/8” pitch, 0.2” roller diameter, 3/16” roller width	McMaster Carr	$19.50
ANSI 35 Roller Chain Connecting Link		McMaster Carr	$0.82
R14-2Z Ball Bearing	7/8” ID, 1 7/8” OD, 1/2” wide 2250 lb dynamic load rating	McMaster Carr	$24.96
1045 Carbon Keyed Rotary Shaft	7/8” OD, 6” long	McMaster Carr	$15.45
Black-Oxide Alloy Steel Socket Head Screw	5/16-32, 1.5” long, 0.469” cap diameter	McMaster Carr	$13.20
Low-Carbon Steel Sheet	12” x 24” x 3/4”	McMaster Carr	$201.69
External Retaining Ring	0.821” groove diameter, 0.042” thick	McMaster Carr	$8.93
ASSEMBLY COST			$595.89
	Adjustable Radius Assembly		
Adjustable radius Cylinder	2.5” OD, .5” wall thickness	Mcmaster Carr	$63.25
Radius adjustment motor	NEMA 11	Mcmaster Carr	$85.00
power screw	1/4-20 self locking precision threads	Mcmaster Carr	$23.40
slider	.75” Steel plate	Mcmaster Carr	$30.00
Motor support plate	.125” steel plate	Jacobs Hall	$24.42
ASSEMBLY COST			$226.07
	Con-rod Spring Assembly		
External Retaining Ring	1/2” Shaft	Mcmaster Carr	$10.54
R8-2Z roller bearings (2x)	1/2” Shaft	Mcmaster Carr	$14.16
SAE 841 Bronze Bearings (2x)	1/2” Shaft	Mcmaster Carr	$2.94
Black-Oxide Alloy Steel Socket Head Screw	M5 x 0.8mm, 8mm long	Mcmaster Carr	$9.48
Guide Rail for Ball Bearing Carriage	20 mm Wide, 280 mm long	Mcmaster Carr	$109.20
Ball Bearing Carriage (2x)	20 mm rail width	Mcmaster Carr	$255.18
Flange-Mounted Linear Ball Bearing	1” Shaft	Mcmaster Carr	$63.91
Low-Carbon Steel Bar	2” x 6” x 0.5” THK	Mcmaster Carr	$8.27
Low-Carbon Steel Bar	5” x 12”x 0.5” THK	Mcmaster Carr	$34.77
1075 Spring Steel Sheet	8” x 24” x 0.109 THK	Mcmaster Carr	$56.32
B7 Medium-Strength Steel Threaded Rod	1/2” - 13, 8” long	Mcmaster Carr	$4.80
Grade 5 Steel Hex Nuts	1/2” - 13	Mcmaster Carr	$14.19
Spring-Tempered Steel Compression Spring (2x)	6” long, 4.906” OD, 3.906” ID	Mcmaster Carr	$56.52
Low-Carbon Steel Bar	1.75” x 12” x 5/16” THK	Mcmaster Carr	$11.93
4140 Steel Rod	1.25”OD x 6” long	Mcmaster Carr	$12.87
Low-Carbon Steel Bar	0.75” x 0.75” x 12” long	Mcmaster Carr	$11.03
Black-Oxide Alloy Steel Socket Head Screw	M5 x 0.8mm, 40mm long	Mcmaster Carr	$11.79
4140 Steel Rod	0.5” OD x 6” long	Mcmaster Carr	$2.77
Low-Carbon Steel Sheet	24” x 24” x 0.25” THK	Jacobs Hall Store	$24.42
ASSEMBLY COST			$715.09
	Specimen Fixture Assembly		
Baseplate (2x)	1/4” THK	Jacobs Hall Store	$48.84
LoadCell	0.625” OD	McMaster-Carr	$5.00
Specimen Attachment	1.25” OD	McMaster-Carr	$15.00
ASSEMBLY COST			$68.84
	Full Project		
PROJECT COST			$1605.89

8 PLAN

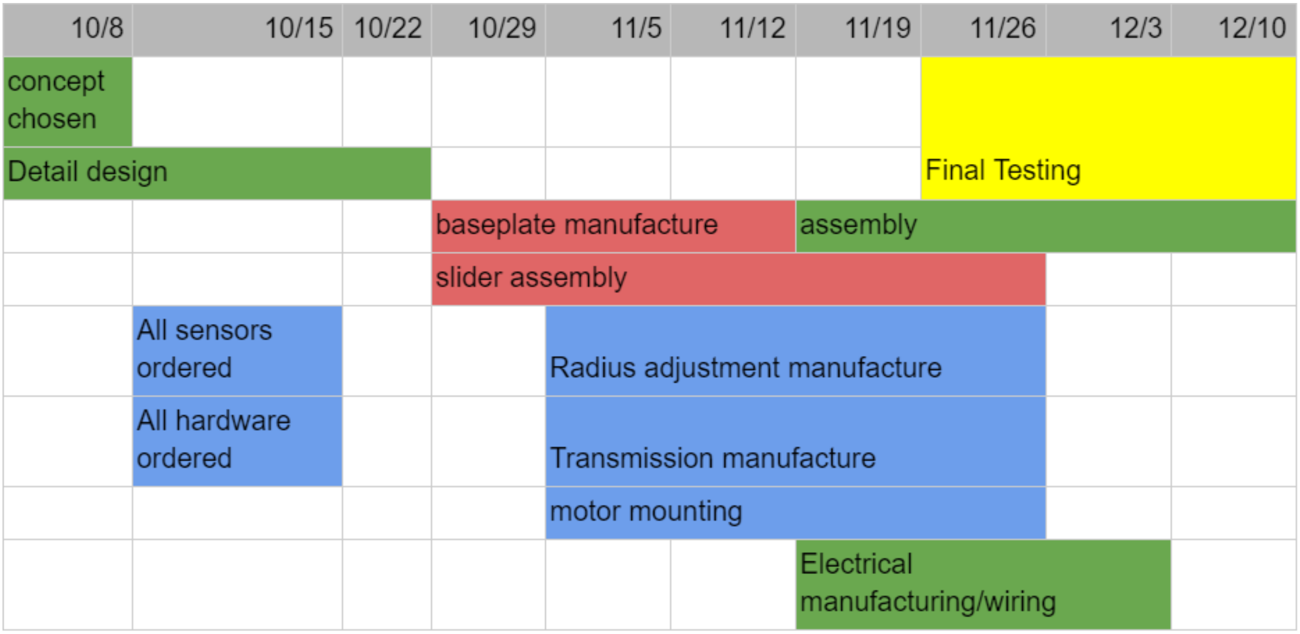

FIGURE 19. GANTT CHART

Above is our gantt chart for completion of various portions of the project. Due to the high loads this project will generate, we will allow three weeks too validate our designs with FEA analysis. Additionally, safety may be a concern for this project with a high power motor and a powerful energy storage device in the spring, so design time must be spent to ensure mechanical robustness. The sensors and major dimensions of the material we need should be complete partway through the design cycle, so we plan to order all sensors and material by October 15.

The Baseplate is the first thing we need to manufacture because all other subassemblies will be assembled to it, so this happens first. The slider assembly is a complex independent component that should also start being manufactured immediately. The radius adjustment, transmission, and motor mount, are all dependent on the other features, so they can be started slightly later. Assembly of all mechanical components can begin following the completion of the baseplate. Wiring and electrical system assembly will be concurrent with mechanical assembly. We want to have at least three weeks of testing of the completed components so that we will have time to fatigue test steel components.

9 INFORMATION SOURCES

The single most significant source of information for our project is the McMaster website. While nearly every part available from McMaster can be found from another source, the amount of time and effort sourcing outside of McMaster would take is non-trivial. While we could probably save money by spending more time optimizing our BOM, we expose ourselves to the risk that we might get a low quality, defective part or run into shipping issues. These are of little concern working with McMaster. The online store also does an excellent job of documenting the dimensions and capabilities of everything in their catalogue. Taking the transmission for example, we started with a motor outputting a known velocity that we wanted to reduce. We also knew the torque at the output shaft. Without the use of any sources outside of McMaster beyond a basic understanding of statics and a calculator, we were able to choose an engine sprocket, axle sprocket, chain, drive shaft, and bearings.

Our machine is a fatigue tester. It is critical that it can make the specimens we load into it fail in fatigue without itself failing due to cyclical loading. Much of the information for the design comes our coursework, including ME C85, ME 108, ME 134, and ME 103. In particular, we use our knowledge of material failure from ME 108. We have to design all the tensile stresses to be below the endurance limit, and the knowledge of how to do this was covered in ME 108 and the textbook[5]. Our groups knowledge of structural mechanics is mostly from ME C85. Knowledge of controls is very important to the radius control and we extensively plan to use knowledge from ME 134[7]. Finally, our sensing and data acquisition skills, especially for the strain gauge, were heavily directed by ME 103 and the textbook[8].

10 PROBLEM ANALYSIS

From our specification, we require a 5HP motor and components that can handle repetitive stresses at 1000 lbs. Additionally we hope to achieve very high reliability and a long life span of the machine. This is likely to be the root cause of the design challenges we face. Most circular and linear bearings are very expensive when the dynamic load specification exceeds a few hundred pounds, so we must minimize all unnecessary moments and stresses wherever possible. Additionally, because many parts will fail after a high number of cycles, rooting out failure points could be time intensive. To head off this problem, it is imperative to do accurate analysis of expected failure points and reinforce these areas with a significant factor of safety in addition to allowing extra time to run and repair failures.

A second related issue with this project is the issue of sensor and measurement accuracy over the course of a single test. Fatigue tests are standardized and performed with constant loading conditions, so it is extremely important to the usefulness of this project to the customer that the sensors be accurate. Because we will be performing long term high power tests, heat and repetitive stress may cause errors in our sensors, especially our load cell strain gauges. Luckily, many members of our team have significant experience with these type of sensing and data acquisition techniques. This should allow us to complete the basics of these sensors quickly and early so that we can test the long term issues we may face.

Finally dealing with the high power high strength components will yield and inevitably heavy and unwieldy device that will need to be displayed and be mobile, and of course safe. This points to an extra close examination of the packaging and weight specifications that we set. The mounting and covers for this machine must therefore be very strong in order to contain any possible sample pieces that may be ejected. It also must have some sort of easy handholds for transportation, or possibly be mounted on locking wheels. It will also need to be arranged in such a way as to display the test sample clearly to the user. This requirement is the less important and more manageable than the others, however it should not be ignored and must be designed rigorously.

11 MANUFACTURING AND TESTING

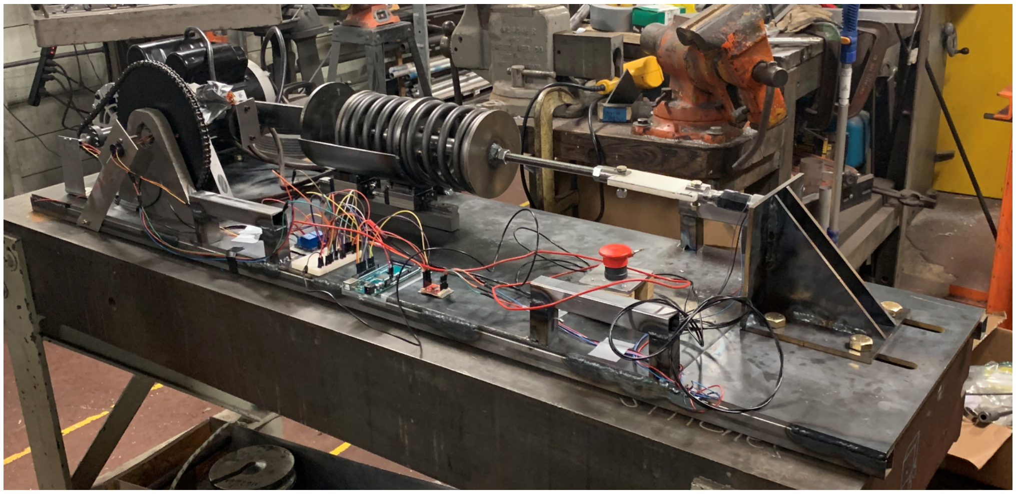

FIGURE 20. Linear Fatigue Tester: Full Assembly

Manufacturing our project required very high skill manufacturing techniques, such as TIG welding and cnc machining, as well as standard milling and turning operations. Due to the manufacturing skill of the members of our team, these processes were accomplished quickly and accurately. The majority of our manufacturing time was spent machining the individual components, such as the baseplate/motor mount, adjustable radius cam, spring assembly, and the specimen fixture load cell assembly. The final assembly of the components turned out to be relatively quick due to the accurate design and tolerancing. The electrical design and sensory acquisition systems needed some basic debugging. One unanticipated issue we had was the strain gauge amplifier circuits inability to sample at a sufficient rate to keep up with the driving motor frequency. We used a HX711 strain gauge amplifier/ADC board. The system was labeled at having two modes, a 10 Hz mode and a 80 Hz mode, that was switchable by digital input from the microcontroller. What we did not foresee was that the PCB that the IC was sold attached to limits this sampling capability, causing a failure of the circuitry at high sampling rates. To solve this, we ended up using a custom strain gauge amplifier circuit, with the ADC that is standard on the arduino. This interface allowed us to sample at a very high rate, almost 1kHz, however the 10-bit adc caused the reading to be quite coarse. This is can be solved by adjusting gain, and an optimally set gain for a given stress amplitude applied to a load cell has more than enough resolution with 10-bits, however by using a much higher resolution adc, we could have removed the necessity of adjusting gain from sample to sample. From the beginning, we set out that the success of our project would be determined by the ability of our project to fatigue steel specimens without itself fatiguing, and as a secondary objective, have an integrated sensory acquisition system to measure the loads and cycles dynamically. Our final criteria is that the system be very user friendly, with the user only needing to perform minimal work to change force amplitude and replace the specimen. We have succeeded on the first two counts, and currently we have a minimal amount of manual adjustment necessary to change the radius. In our testing, we were able to fatigue multiple notched steel specimens, capturing the failure process in slow motion video as displayed on demo day and detailed in figure 25. We were able to document the crack propagation in the sample, growing slowly and finally failing in unstable crack growth. Because of this, we consider the project over all to be a success. Further, we believe that we will be able to accurately and rapidly test welded specimens in fatigue tests that will be useful for the FSAE team. Below we discuss the success and shortcomings of manufacturing methods employed for this project

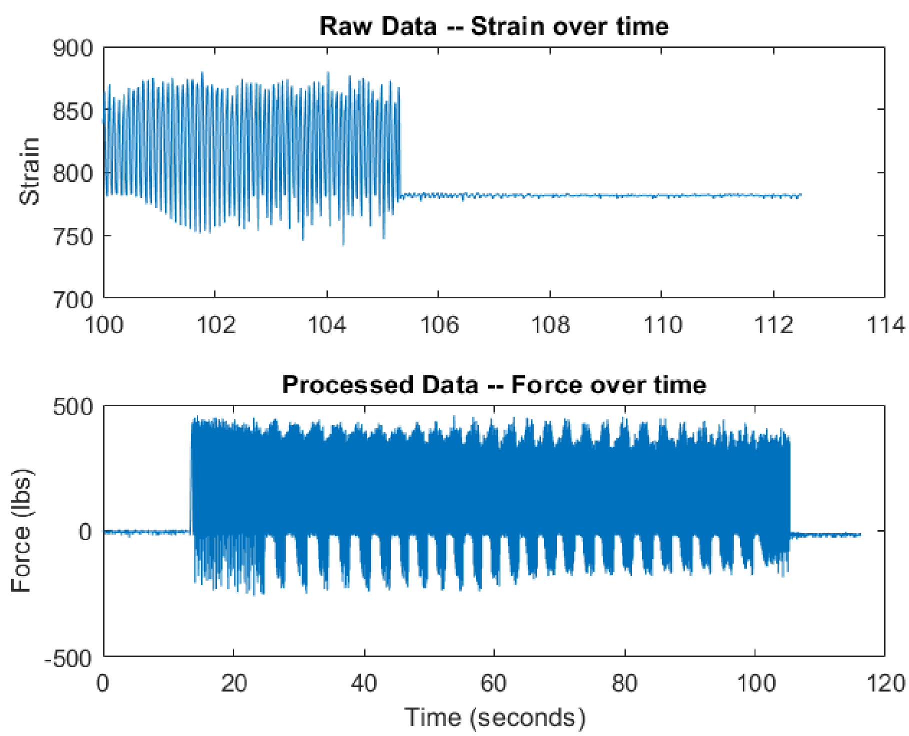

FIGURE 21. Fatigue Test of a Notched Steel Specimen: Strain Gauge Data

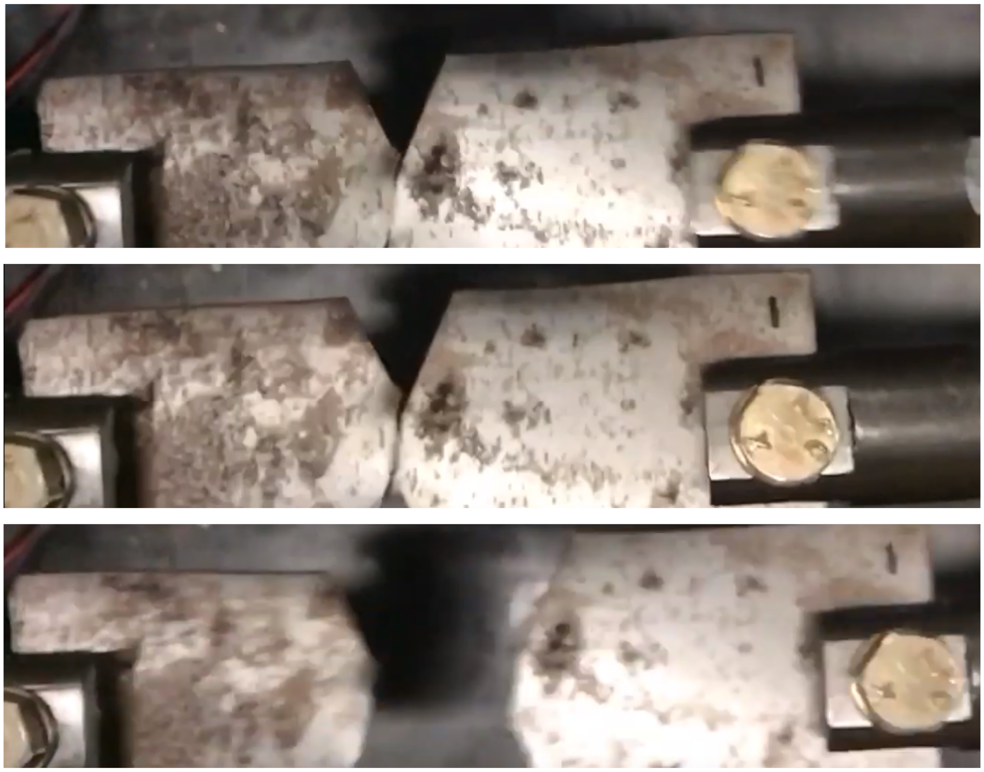

FIGURE 22. Fatigue of a Notched Steel Specimen

11.1 Baseplate and Specimen Fixture

The baseplate is a simple part to make. Hole locations are the only precision features, and these can be made on a waterjet cutter. The locations of the structural support tubes have wide tolerances and can be welded by hand with no particular difficulty. The welding is easy as the steel is considerably thick, but not so thick as to require special attention (/procedures?). The same is true of the handles, as there is plenty of space for them to go. (more space for handles for better carrying balance, include wires in cad so handles didn't have to be moved to accommodate them)

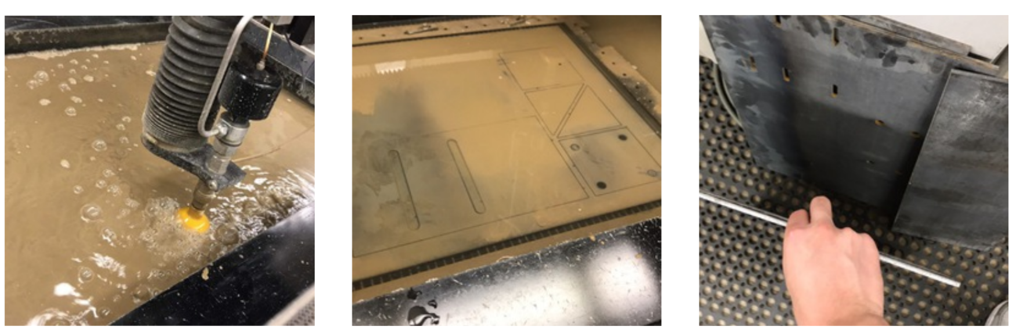

FIGURE 23. Water Jet Component Manufacturing

The specimen fixture had three parts: two clevises and and a mounting base. The clevises were made with lathe and mill operations. The load cell was integrated with the base side clevis. Neither clevis was particularly difficult to make, as once again there was no need for high tolerances. Any variation in manufacturing could be accounted for with the calibration of the load cell. The mounting base was made of waterjet and welded steel. Here the waterjet allowed us to easily achieve the correct locations for bolt holes. Welding the plates together was likewise an easy operation because of the steel thickness. (more structural support)

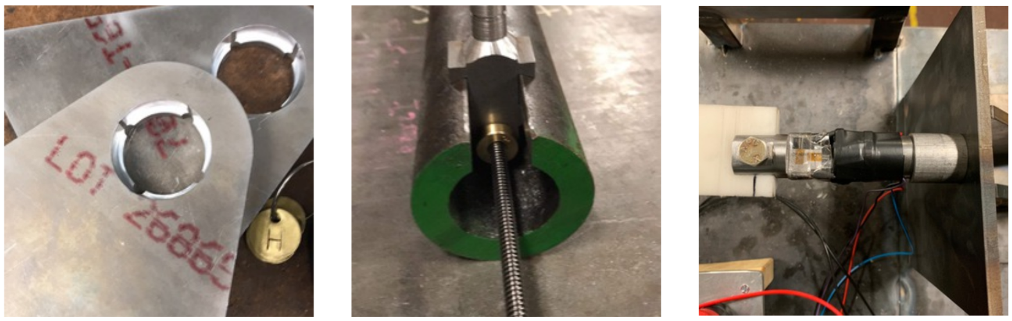

FIGURE 24. Mill and Lathe Component Manufacturing

All of these parts were simple to make and considerably overbuilt were possible. None have failed nor shown any sign of failing. These facts point to the design of this assembly being a success.

11.2 Adjustable Radius Cam Assembly

The cam assembly saw the highest loads and moments out of any part on the project, as well as needing to be wired with a slip ring to the plate with which it had a high relative angular velocity. All of these factors resulted in this part taking the longest to manufacture and debug. First, we had to machine very large circular and plate steel stock to shape which proved time consuming. Then, we welded together these parts. The weld joining the slider cylinder to the shaft socket was a high stress weld joining very thick members. Although we had skilled welders on our team, we had not had experience with this type of weld. We needed to cut deep chamfers in our parts and take multiple passes with the TIG welder. Eventually, we learned this technique and joined these parts, with no evidence of joint weakness or failure.

FIGURE 25. Welded Component Manufacturing

We purchased slip rings to tie the stepper motor to the controller electronics. The only available route to pass these wires through was through the keyed slot in the shaft. Although this route seemed circuitous,we experienced no wire chafing, and no electrical issues with the stepper motor. Currently, the main issue with the stepper motor is that it is under-powered to adjust the radius while a sample is attached to the specimen mount. This issue can be solved by simply performing adjustments before sample prep, however if this is forgotten, more work will be necessary to manually adjust the position of the adjustable radius cam.

11.3 Spring Cylinder Assembly

The cylinder was one of the most involved manufacturing operations for this project. The sides of the cylinder were water jet and then welded to a rolled piece of steel. Then the con-rod tabs and linear bearing mounts were welded to the cylinder body. These components had to be jigged while welding to ensure that the tabs were perfectly perpendicular with the linear rail.

Spring perches were water jet cut and then turned down on a conventional lathe to a friction fit on the springs. Weld nuts were then welded to the spring perches to allow preloading of springs on the threaded central axis.

11.4 Motor Assembly and Transmission

The motor was attached to the base plate with 4 slots. Slots were chosen instead of holes so that the motor's position relative to the transmission could be adjusted to remove slack from the chain. The engine sprocket was a standard keyed sprocket for 7/8" shafts, and was secured to the keyed 7/8" motor shaft by means of set screws.

The transmission axle required a few simple machining operations on a manual lathe to add snap rings for locationing and facing one end of the shaft to length. Although the shaft was mildly hardened to Rockwell B95, it could still be machined with standard HSS tools. The snap rings were placed outboard of the 2 bearing locations and outboard of either side of the transmission sprocket.

The bearing carriers were the most complex part to machine in this assembly. The general shape and bearing relief holes were done on the Etcheverry waterjet. The countersunk bolt holes were drilled from the bottom of the plate, then the counterbores were machined with a counterbore cutter. The bearing bores were done on a 3-axis Haas CNC VMC. The bearings were pressed into the bearing seats with a hydraulic press, while the shaft was pressed through the bearings with a mallet. The transmission sprocket was secured to the shaft axially with snap rings and radially with set screws. The chain running from the engine to the axle sprocket was completed with a master link. The entire transmission was attached to the base plate with 4 bolts.

12 DISCUSSION

As discussed above, we have achieved a very successful project, able to successfully fatigue notched steel specimens, capture data using a load cell and hall effects sensor, and report the data simply with the Arduino interface. The biggest strength of our project is its physical robustness, experiencing essentially no mechanical failures during testing. The motor we selected was powerful enough to apply the necessary forces for the tests,and versatile to e able to use the 110V supplies that are standard, as well as the less common 220V supplies for an even higher power output. One drawback of the motor was its high weight, which ended up a significant portion of the project weight. While weight was a design factor for us, it was much less significant than the test performance, and as we stayed below or 200lb design max weight, motor weight was not an issue. The design of the roller chain step down and the simple slotted tensioning system functioned very well. The Radius Adjustment feature, although it had a complicated wiring system, ended up functioning well. The stepper motor was, however, underpowered. It had enough torque to reliably control the radius, but only without a specimen mounted. The solution to this would require a larger stepper motor, and therefore a larger spacing between the cam and the electric motor. All of this is feasible, and necessary for consumer product design. For experimental testing, however, this step is unnecessary as long a standard operating procedures are clear that the radius must be adjusted before specimen mounting.

One major issue with the project was flexure in the baseplate. The loads generated by the motor/spring transducer were high relative to the bending stiffness of the baseplate, causing significant bending in the plate. We had ignored these design issues during design with the idea that the project would be bolted to studs in the ground, clamped to a table, or otherwise fixed in some way. What we did not foresee was that it was often not feasible to securely fix the project due to the size and strength of the tables we had or the lack of studs in the ground at demonstration locations. This issue can be fixed in a number of ways. Because we would like to actually use this project for testing of welds for the FSAE vehicle, we propose a solution incorporating materials available to the FSAE team. This includes a very stiff solid steel welding bench with threaded bolt holes. Our plan will be to drill holes in locations around the base plate and manufacture standoffs from steel to increase the available clamping force without bending the plate. Then, we will bolt into the table to fix it in place. Barring this, we would need to add stiffening struts down the length of the baseplate. This would likely result in a higher height of the plate and a higher overall center of mass and total weight, which may present its own issues. Therefore, it is far superior to assume access to a structural welding bench and fasten to that.

The adjustable radius mechanism is our limiting factor when it comes to achieving higher loads with our machine. As we plan to create a S-N curve for our materials, it is likely that we will need access to higher loads to put through the sample. We can work around this by replacing the adjustable radius with a series of waterjet cams, and pressing bearings into them at the point of attachment with the con-rod. In addition, the cams will have a lower profile than the adjustable radius does, and this will put less of a moment on the bearings supporting the driven shaft; these bearings are the secondary limiting factor to achieving higher loads. The lower profile will require new bolt holes to be drilled for the linear rail and specimen mounting base. These can be located off of the original holes with lasercut wood pieces.

Another way to increase available load is to increase power. This can be done relatively easily by purchasing another compressor motor and attaching it to the original motor shaft. It would need to be wired in reverse of the original motor so it can spin in the proper direction.

Currently we have wired our motor to 110V so that it can be run pretty much anywhere. The motor is rated to 240V, and we have a 240V plug at the garage. The motor will be rewired to take advantage of this extra power before tests are run.

One thing that we saw during testing was ovalization of bolt holes in the sample. This has the potential to be problematic when testing for long periods of time. In order to avoid this, we will design the sample geometry so that stresses in the tapered section are higher than Hertzian stresses at the bolt holes. If needed, we can invest in proper clamps to hold the sample.

From an electronics and software standpoint, one major improvement that can be made is automating as the experience as much as possible. For example, the relay we used was not rated for high enough current to kill the motor, so the user had to physically turn off the motor at the end of the test. This can be easily fixed by replacing the current relay with a higher rated one, and would allow for the arduino to kill the motor automatically when it detects the test is complete. Another improvement can be made with the stepper motor position calibration. In trying to reset the position each time, we noticed that at times when the motor is stopped, it draws more than 1 amp of current. The H-Bridge used to control the stepper was only rated for 1 amp, so finding a power supply that can power the motor voltage and limit current would be an easy fix and allow for more automated and precise control of the radius. Lastly, an integrated GUI from MATLAB that would successfully pass variable assignments and data to and from the arduino would make the user experience much easier and unlock MATLAB's vast tools for data analysis. Given the time constraints and other more performance critical efforts, this was not made a priority for this project but would certainly be helpful to implement moving forward.

13 CONCLUSION

This product fulfills the low-cost, variable-geometry niche in the material testing market. A competitor to the R. R. Moore machine, it is designed to test a much greater range of material geometries that no other machine in its class can match. Before its introduction, engineers and students looking to understand the fatigue qualities of their materials were forced to contract with testing companies or hope that their university had access to equipment with the ability to handle varied test sample shapes. As of now that capability is available at an exceptionally accessible cost.

This machine is capable of applying up to 1000 pounds at 12Hz in tension and compression. At this rate 1 million cycles can be reached in less than a day. It uses a force control system, with a user defined set point. The ability to handle complex geometries is what sets it apart from competing machines, such as the R. R. Moore tester. It accepts material samples with a wide range of widths, thicknesses, and lengths. Additionally, being able to put tensile and compressive forces on the test piece (i.e. fully reversed loading) ensure that the sample is subjected to the worst-case condition. We have been able to successfully fatigue notched steel samples in low cycle fatigue, measuring the stress amplitude applied and the rotations with a strain gauge and hall effect sensor respectively. We had some minor issues with base plate flexure and stepper motor torque. Both of these problems in fact have simple solutions that we plan to implement to continue our tests of weld strength for FSAE applications

This fatigue tester will be of interest to anyone looking to complete serious engineering analysis while operating on a budget. It will allow for student teams, early-stage start-ups, and similar funding-limited organizations to make better products and pursue loftier ambitions than they could before. They ability to design to the manufacturing capabilities that a company or group actually possesses is extremely valuable, and will continue to be.

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A APPENDIX

A.1 Motor Shaft Stresses

- 15T ANSI 35 Sprocket has pitch diameter of 1.8037in
- 91.3 in-lbf/(1.8037in/2) = 101.5 lbf
- Peak bending moment: 101.5 lb * 1.67in = 169 in-lbf
- I = π*(7/8in)^4/64 = 0.028775 in^4
- σ = 169*(7/16)/0.028775 = 2570 psi
- J = π*(7/8in)^4/32 = 0.014387 in^4
- τ = 91.3*(7/16)/0.014387 = 2777 psi

A.2 Motor Shaft Fatigue Life

- σ_{mean} = 2777/2 psi (from shear and Mohrs circle)
- $\sigma_{amplitude}$ = 2570 psi (from bending)
- Minimum UTS for steel of 295 MPa = 42000 psi
- Endurance limit = 42000/2 = 21000 psi
- Goodman effective stress = 2752 psi
- Gerber effective stress = 2582 psi
- Most materials lie between Goodman and Gerber
- Fatigue Life factor = 21000/2752 = 7.6 FoS

A.3 Driven Shaft Reactions

- -101.5*1.25 - 1000*4.4 + RB*2.5 = 0
- RB = 1810.75 lbf
- RA - 101.5 + 1810.75 - 1000 = 0
- RA = -709.25 lbf

A.4 Driven Shaft Stresses

- Peak bending moment: 1000 lb * 1.in = 1900 in-lbs
- I = π(7/8in)$^4/$64 = 0.028775 in^4
- σ = 1900*(7/16)/0.028775 = 28888 psi
- J = π*(7/8in)^4/32 = 0.014387 in^4
- τ = 438*(7/16)/0.014387 = 13320 psi

A.5 Driven Shaft Fatigue Life

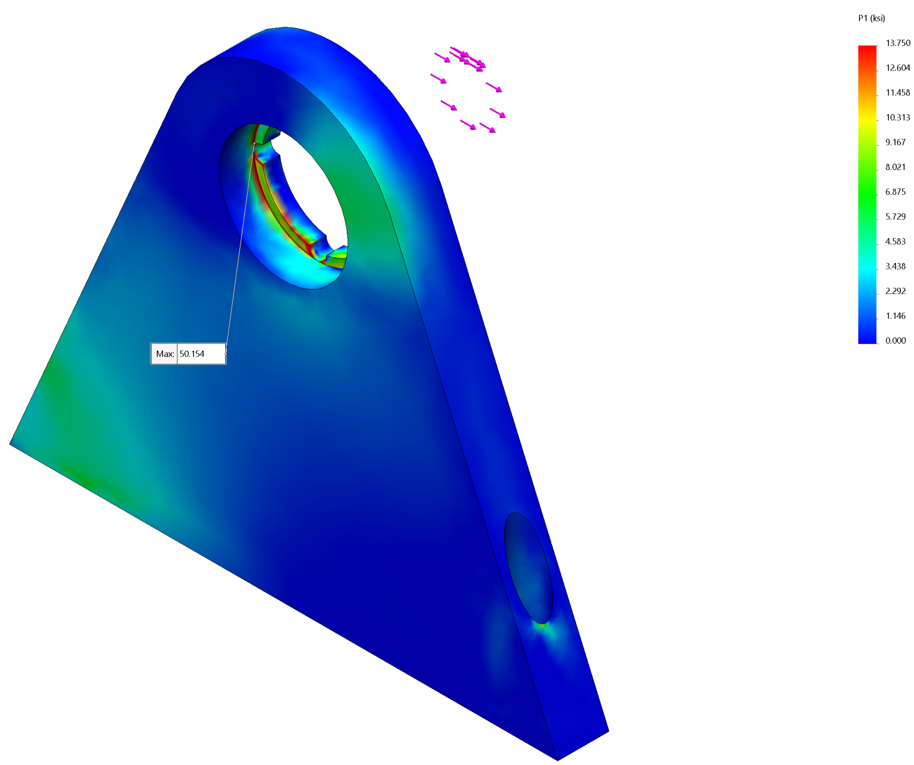
- σ_{mean} = 13320/2 psi (from shear and Mohrs circle)
- $\sigma_{amplitude}$ = 28888 psi (from bending)
- UTS = 98000 psi
- Endurance limit = 98000/2 = 49000 psi
- Goodman effective stress = 33432 psi
- Gerber effective stress = 29432 psi
- Most materials lie between Goodman and Gerber
- Fatigue Life factor = 49000/33432 = 1.4 FoS

A.6 Driven Shaft Bearings

- Bearings rated to 2250 lb dynamic load
- Static rating is 1300 lb - bearings support shaft, sprocket, adjustable radius, and a little chain
- Load Factor = 2250/1810.75 = 1.2 FoS

A.7 Driven Shaft Bearing Seats

- Minimum UTS of 1018 is 55 ksi, isoclip done at 13.75 ksi (bearing contact incorrectly modeled)



A.8 Bearing Seat Bolts

- F1 = RB = 1810.75 lbf
- F2 = 1900 in-lbs/3.2in = 593.75 lbf
- Fnet = 1906 lb
- Mu (steel) = 0.6
- Clamping load = 5623 lb (iEngineer app)
- Friction Load Factor = 5623*0.6/1906 = 1.7 FoS