A Practical Educational Fatigue Testing Machine

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A PRACTICAL EDUCATIONAL
FATIGUE TESTING MACHINE (EFTM)

Abstract

An experiment and its associated apparatus are proposed to better instill the significance of the Fatigue Failure Phenomenon in undergraduate engineering education. The benchmark for establishing the behavior of engineering materials under dynamic/fatigue loading is the “S-N” diagram. Here, “S” corresponds to the stress level and “N” to the number of cycles. Due to the uncertainties involved in materials’ behavior and characteristics, a large number of specimens are tested at different stress levels for generating the *“S - log N”* diagram. Ideally, the main objective in such tests is two-fold. First, to establish (for a given material), up to what stress levels the material will enjoy an infinite life (Endurance Limit); and second, to correlate the number of cycles at different stress levels that a material will be able to go through before coming to failure. The range of cost for a typical educational fatigue testing apparatus is from $10,500 to $32,500. These units are essentially adaptations of the R. R. Moore Industrial Fatigue Testing Machines which cost in excess of $150,000. The goal is to produce an affordable and a fully functional version of the apparatus that produces dependable results. The time factor for conducting fatigue testing in an educational environment has been incorporated in the design process. The process for the design of the apparatus, its subsystems, and the features of components are discussed. The results of two sets of tests conducted on two different materials are presented. Summary of an assessment reflecting on the positive educational outcomes due to the use of the EFTM is shared with the engineering community.

I- Introduction

Laboratory experimentation is a critical final link for a thorough understanding and appreciation of scientific and engineering theories. Every possible effort should be made not to deprive the future engineers or educators from this vital component of their education [1]. It is therefore necessary to continue development of effective and efficient pedagogical methods and techniques for the engineering laboratory experience [2].

High quality and dependable Laboratory apparatus is generally expensive due to low production levels, specialized features and significantly higher *Design Costs* built into the final cost. For example, the range of cost for a typical educational fatigue testing apparatus is from $10,500 to $35,500. These units are generally adaptations or variations of the R. R. Moore Industrial Fatigue testing devices which may cost in excess of $150,000.

Such high costs may lead to lack of vital laboratory apparatus and in turn deprive the engineering students from being sufficiently exposed to important concepts such as verification of the theory through experimentation, interpretation and analysis of data and gaining sufficient background for designing experiments. However, if blueprints of the designs of a (desired) apparatus are available, and on site machining capabilities exist, a major cut may be expected in the final cost. Such designs and blueprints may be generated in-house in collaboration with undergraduate engineering students [3]. This team hopes that the colleagues and students in other engineering programs would find this effort worthy of replication and potential adaptation in their programs.

II- Background

Roark and Young define Fatigue as "the fracture of a material under many repetitions of a stress at a level considerably less than the ultimate strength of the material" [4]. In a fatigue test, the specimen may be exposed to equal or unequal alternating stresses. When equal positive and negative stresses are applied, it is said that the loading is fully reversed. In this situation, a critical location of the specimen will experience equal levels of both tensile and compressive stresses in one full cycle.

The benchmark for establishing the behavior of engineering materials under dynamic/fatigue loading is the "S-N" diagram. Here, "S" corresponds to the stress level and "N" to the number of cycles. Due to the uncertainties involved in the materials' behavior and characteristics, a large number of specimens are tested at different stress levels for generating the *"S - log N"* diagram. Ideally, the main objective in such tests is two-fold. First, to establish (for a given material), up to what stress levels the material will enjoy an infinite life (Endurance Limit); and second, to correlate the number of cycles at different stress levels that a material will be able to go through before failure.

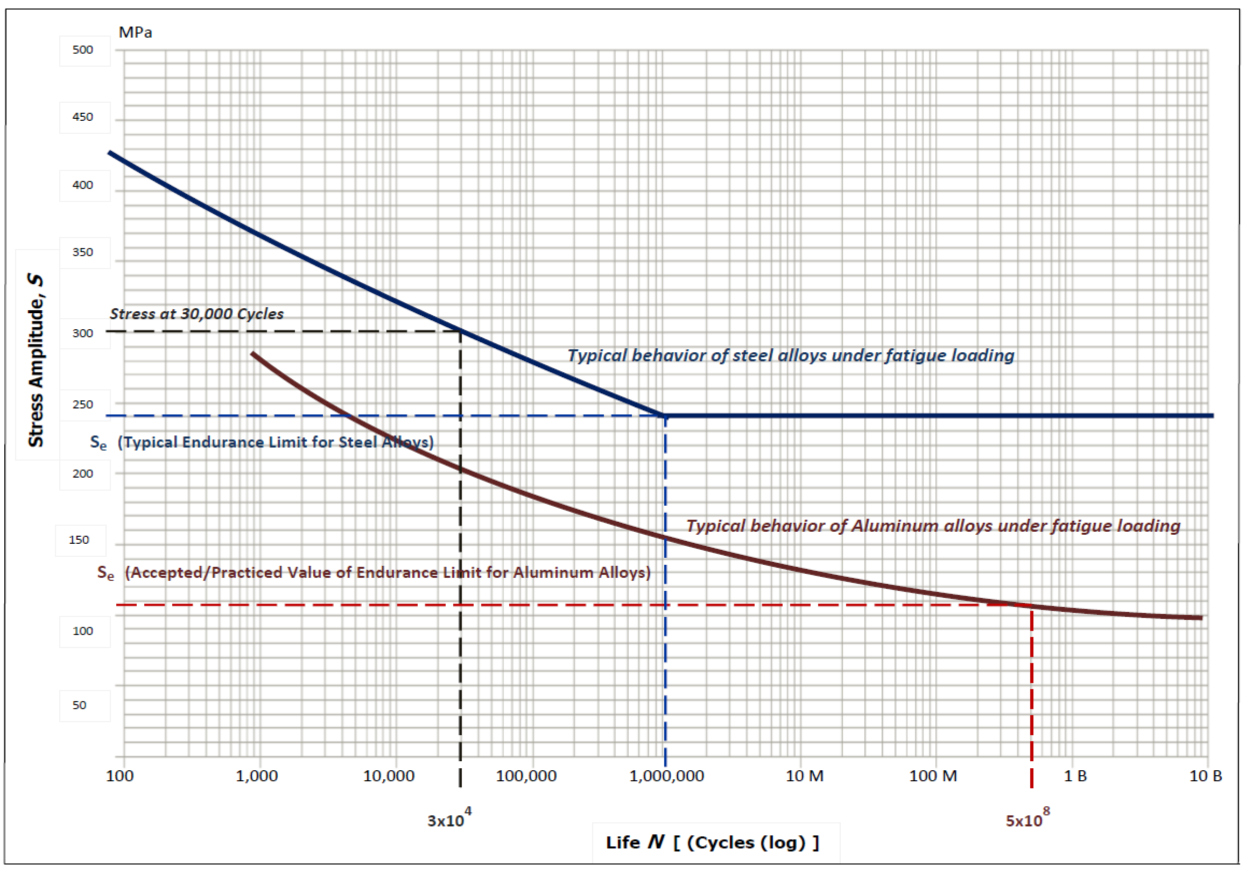

Figure 1. *S-N* Diagram for Typical Behavior of Steel and Aluminum Alloys

The S-N diagrams for several engineering materials have been established as a result of comprehensive and highly time consuming tests. Generally, the results are more reliable for steel alloys as compared with aluminum alloys. Low-cycle fatigue is defined on an S-N diagram as being approximately between zero and 1000 cycles. High-cycle fatigue is generally greater than 10^3 cycles. Finite life is assumed to be below 10^7 cycles [5]. A typical S-N diagram is shown in *Figure (1)*.

Ferrous materials usually show a definite breaking point on the S-N diagram around 10^6 cycles, whereas nonferrous metals show no such point. As shown in *Figure (1)*, for nonferrous metals, a value of $5x10^8$ cycles is usually assigned as the fatigue limit. There are several theories available for prediction of failure due to cyclic loading [6]. Depending on the situation at hand, the designer must apply the suitable theory as no one theory may optimally address all of the design requirements. However, all of them converge on the fact that this type of failure is not yet completely understood and extra care must be taken when dealing with fatigue phenomenon. Shigley and Mischke present a rather comprehensive view of the issues involved with the variations of behavior of different materials in the fatigue analysis process [7]. *The goal in the current experiment is to create and simulate the conditions that allow students to test the reliability of such (S-N) diagrams and gain a better understanding of the statistical and probabilistic nature of the Fatigue Failure Theories.*

III– Design of the Experiment and its Associated Apparatus

The following criteria have been incorporated in the design of the experiment and the associated apparatus:

- *Safety*
- *Simplicity and Practicality in Fabrication (at other institutions)*
- *Affordability/Control of Cost*
- *Use of Reliable Sources for Components*
- *Durability*
- *Use of Non-Corrosive & Aesthetically Pleasing Materials*
- *Simplicity of Operation*
- *NO use of Discontinued Parts/Components*
- *Time Factor in Conducting the Experiment*

The requirement of having a modular design and stopping the motor (when the specimen fails) presented some interesting challenges. Additionally, the size, weight, and other physical characteristics of the experiment were not defined at the inception of the project. Initially, this lack of constraints may have been a blessing (for the students) since it did free the design process to vary these factors. However, later, it became clear that the price for such a freedom is dealing with the lack of starting points/values in the process. *Table (1)* provides a synopsis of the steps and the parameters involved in the implementation of the project.

Table 1. Steps and the Parameters involved in the successful implementation of the project.

#	*TYPE OF ACTIVITY*
1	Brainstorming for Design of the Experiment and the Apparatus
2	Meeting Minutes and Progress Reports
3	Prototyping
4	Generation of Technical Drawings for all (Home Made) Components
5	Selection of (commercial) Components and Identification of Suitable Sources
6	Fabrication and Compilation of Notes on Best Approach for Machining
7	Electro-mechanical control system
8	Testing, Calibration, Generation of Data and Measure of Precision and Accuracy
9	Generation of the Laboratory Manual for the Experiment
10	Loading of All Necessary Information and Helpful Links on a CD

IV- Theories of Fatigue Failure

To better appreciate the complexity of the fatigue phenomenon, *Theories of Fatigue Failure* were comprehensively reviewed (by the collaborating students). These were further examined and used as a visual platform to decide on the degree of sensitivity of the apparatus. Further, they serve as indicators by which a laboratory coordinator/instructor may make more informed decisions about the time required for conducting the experiment/ demonstration. A summary of these models are presented in *Appendix (A)* [8].

V- Design of the Components and the Subsystems of the Apparatus

A simple schematic of the proposed Educational Fatigue Testing Machine (EFTM) is shown in *Figure (2)*. The first completed (full scale) prototype of the EFTM is shown in *Figure (3).* We note that [as compared with the schematic shown in Figure (2),] the final design of the apparatus eliminates the need for use of gears for reduction/control of the RPM. This was achieved by employing a motor with the desired RPM (of 1725) allowing for a Direct-Drive system.

EFTM is comprised of the following major components and subsystems. The role and design characteristics of each of these components are briefly discussed.

<u>1. The Base</u>

In order to construct the apparatus, a suitable base needed to be acquired. A 0.5 inch thick by 6.75”x23” plate of 6061-T6 anodized aluminum proved to be strong enough to provide room for installation of all of the components.

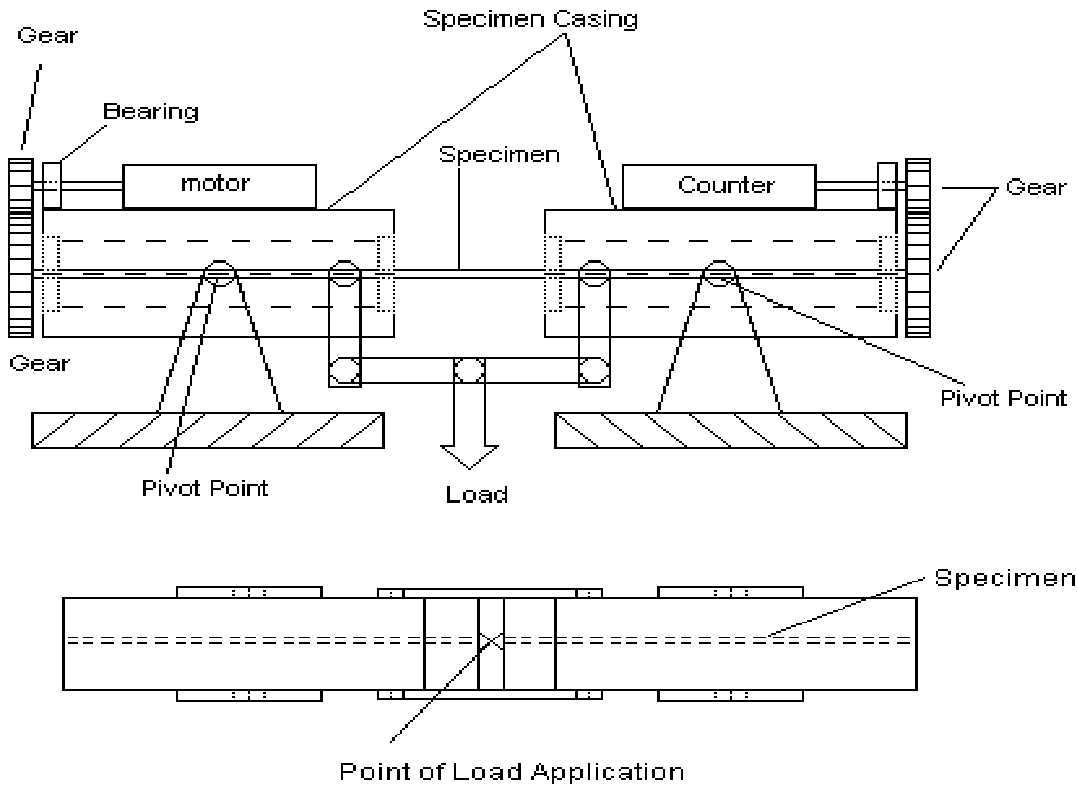

Figure 2. Basic configuration of the proposed Educational Fatigue Testing Machine *(EFTM)*

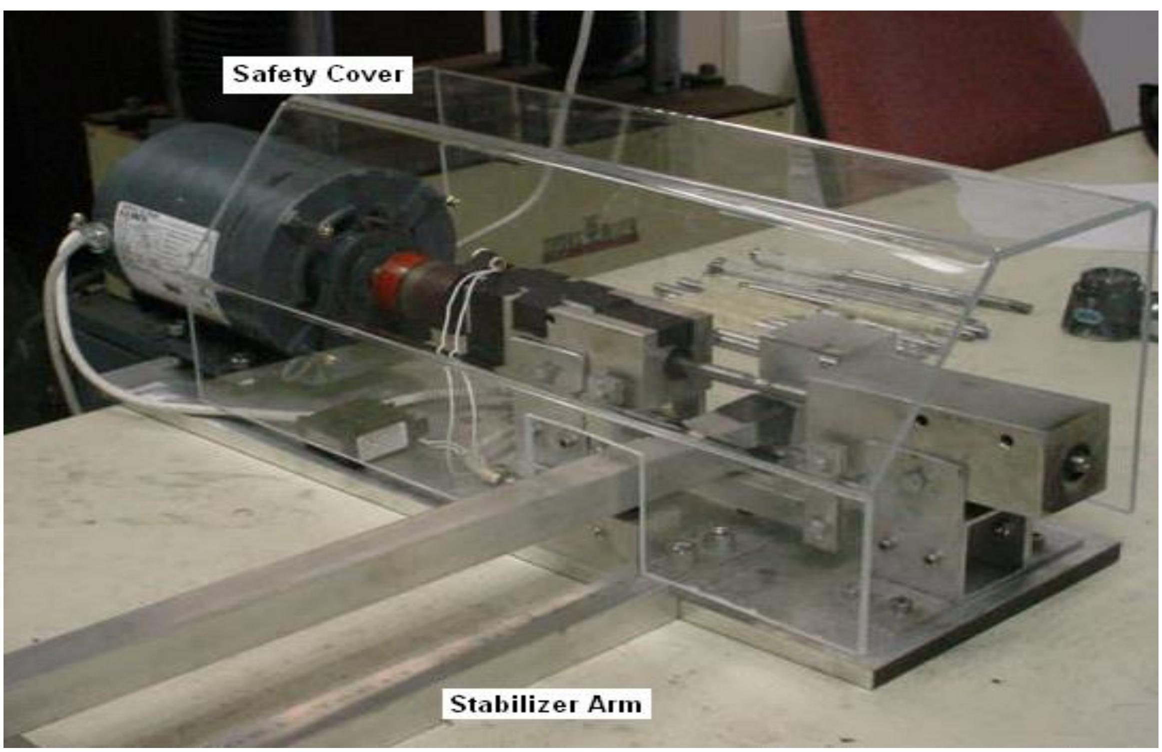

Figure 3. The First Full Scale Prototype of the Educational Fatigue Testing Machine *(EFTM)*

2. The Motor

The choice of the most reliable and safe motor was critical. The research led to the selection of an AC powered ¼ hp motor that included thermal protection and bearings.

3. The Gears and the Drive System

The importance of the gear system is much more than simply to rotate the specimen. The gears also function as clamps to secure the specimen from translating inside the specimen holders during testing. Since the specimen needs to be easily removed and replaced, the selection of the clamping style of the gears, and the way that they mesh is important. However, in the current design, the selection of the motor enabled the team to take advantage of a "direct drive" system resulting in the elimination of a good number of components and significant reduction in cost.

4. The Lever Mechanism

From the Calculations [enclosed in *Appendix (D)]*, one can see that the loading of the specimen was rather high, and could not be readily (and safely) accomplished with conventional means of loading. In order compensate for this, a leverage mechanism will be used for creating the desired deflection of the specimen. In *Figure (4)*, one can see a rudimentary diagram of the proposed loading mechanism.

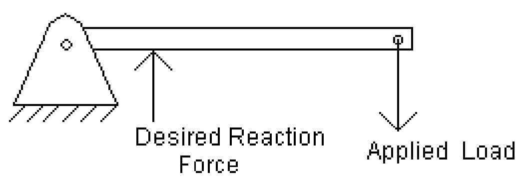

Figure 4. High Mechanical Advantage Leverage Mechanism

The advantage of this system is that using a small input weight, depending on the moment arm, a much greater output force can be generated. For example, if the desired reaction force is located 1 inch from the fulcrum, and the lever arm is 10 inches long, a 10 pound load can produce a 100 pound reaction force. This system ensures the safety of people in the laboratory environment, and ease of applying such a large load by an individual. With reference to *Figure (5), and a*s shown in *Appendix (D),* the (final) expression for "R" is:

R= The Applied Load to the Specimen = [5.79+10*W] lb

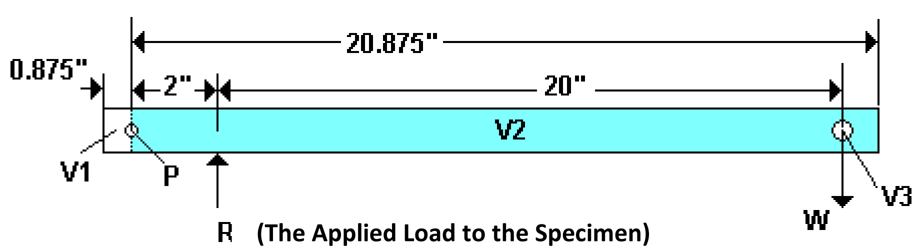

Figure 5. Modeling of Lever Arm and Determination of Calibration Equation

5. The Kill Switch

A stopping mechanism must be in place in order to cut off power to the motor once the specimen has failed. This may be achieved by placing a kill switch underneath one of the specimen holders or loading carriage as shown in *Figure (6)*.

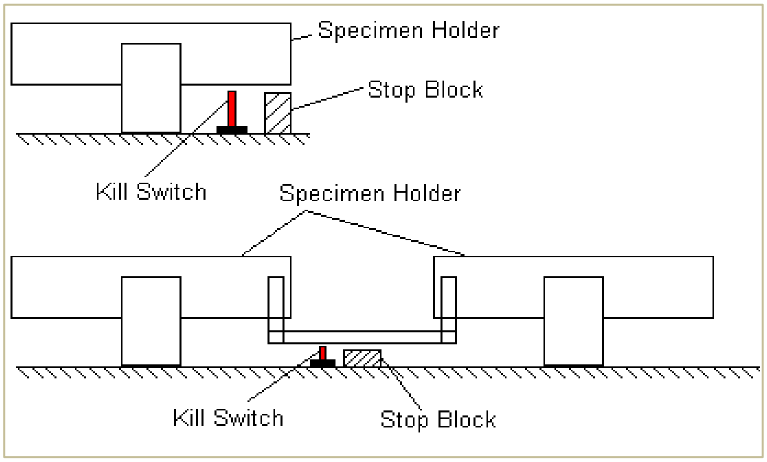

Figure 6. Different possibilities for the kill switch configurations

The stop block is necessary so that the kill switch is not damaged by the descending specimen holder or load carriage that will be supporting the entire force generated by the lever arm. Due to the tight clearance between the specimen holder support and loading bridge, the stop block and kill switch were positioned under the lever arm as shown in *Figure (7)*. This modification improved the symmetry of the apparatus while providing adequate clearances.

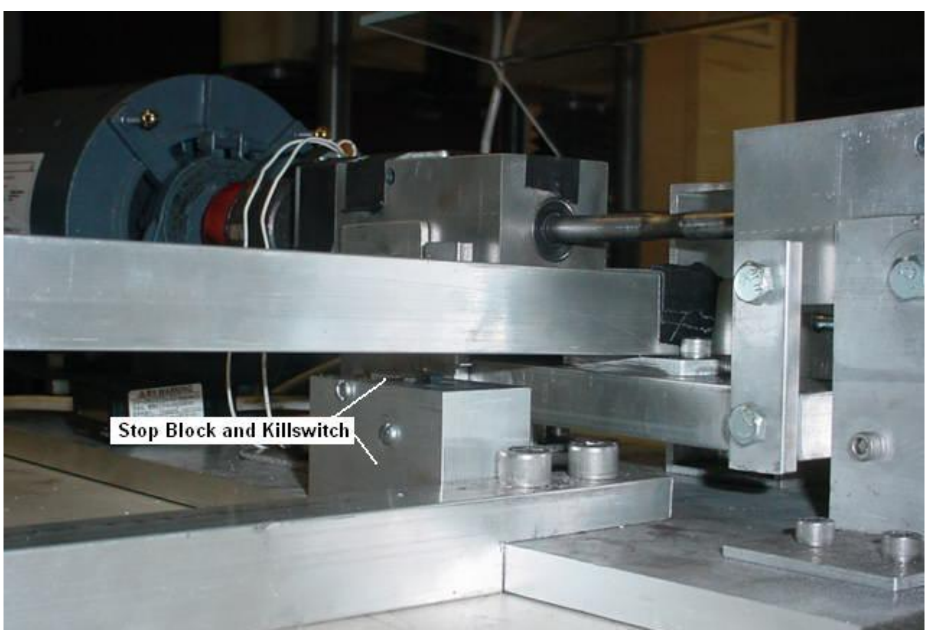

Figure 7. Kill Switch and the Stop Block Locations

6. The Counter

A digital counter with sufficient level of sensitivity was obtained from McMaster-Carr. As shown in *Figure (8)*, it is an electronic counter with an eight (8) digit display allowing registration of cycles near 100 million. The counter operates by having two sensors pass by one another, and complete an electrical signal, which adds 1 to the display for each time the circuit is completed. With such a system, the two gear design [as shown in *Figure (2)*], need not be used. Rather, a hub, such as the one in *Figure (9)* could be employed. This hub would clamp on to the end of the specimen, and turn at the same rate as the specimen.

Figure 8. Digital Counter

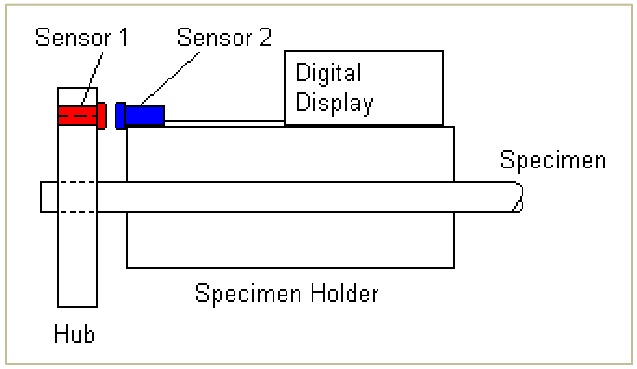

Figure 9. The Sensor and the Hub setup of a typical digital counter

7. Safety Cover

A safety cover was constructed out of ¼ inch shatter-proof transparent material that was 19 inches wide and 21.5 inches long prior to bending. It was bent utilizing a plastic bending machine and fastened to the base with two hinges. The main purpose of the cover was to primarily protect the users of the machine from bodily injury resulting from either flying debris that could result from the specimen breaking, or coming into contact with the moving parts of the machine.

8. Stabilizer Arm

A stabilizer arm was developed in order to prevent the base of the apparatus from tipping over once a load had been applied to the end of the lever arm. The addition of the stabilizer arm negated the need for a C-clamp and allowed for placement of the machine on any level surface that was high enough for weights to be hung from the lever arm.

Characterization and Redesign

The initial tests of the EFTM revealed the need for increase in the size of certain components as well as coming up with means to eliminate unacceptable levels of vibrations. For example, it was noted that after machining, the central axis of some of the critical components were not perfectly co-linear. These included the specimen sleeves, bearing blocks, and specimen holders. This resulted in an eccentric rotation of the specimen and possibly attributed to the wearing away of the specimen sleeve. Adding to this issue was the imperfections of the extruded square stock tubing. With the necessary modifications, and proportional increase in size (and in turn, the moment of inertia) of the components in the chain of motion, the eccentric rotational issues were marginalized and vibrations fully controlled.

Size, Geometry, and the Surface Finish of the Specimen
The specimen diameter is 3/8 of an inch and they are cut from 4 foot sections of solid round bars. This length was chosen so that three 16-inch specimen could be cut from a single four (4) foot length of material without any waste. The specimen is then machined so that a 5/16 of an inch diameter may be achieved at the mid-span between the two specimen holders with "practically no fillet effect". Next is to sand them with 150 grit sand paper so that they would easily slide through the bearings. Finally, they are polished with steel wool in order to achieve the finest (practical) surface finish possible. A set of polished and unpolished specimen that have already been cut to length with the required radii are shown in *Figure (10)* below.

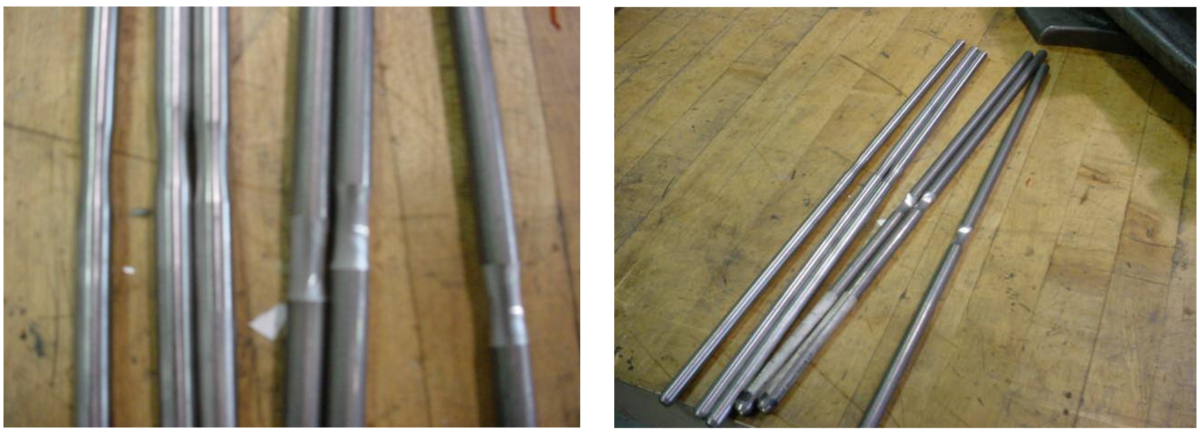

Figure 10. Sample Test Specimens. To demonstrate the effect of the "Surface Finish", (in each figure,) the three specimens on the left are polished, while the three on the right are only ground.

VI- Experimental Program and Collection of Data

Although the testing program has resulted in some promising data, in general, the tested specimens consistently fail at higher number of cycles than those predicted by text book models. The authors speculate that this may be due to the combinational effects of the statistical/ conservative nature of the text book models and the conservative listings of the strength of materials in tables. This is a critical issue in that when running the experiment, the (hidden) actual difference(s) may offset the result by tens of thousands of additional (unexpected) cycles. The tested specimens (chosen) from the certified bars have shown significantly better results compared to the non-certified samples [9].

Testing

Testing was conducted on two sets of specimens with different materials. One set was A-36 structural steel and the other 1018 cold rolled steel. The A-36 stainless steel had a yield strength of 36,300 psi and an ultimate strength between 59,000 psi and 79,000 psi. The large variation in the ultimate strength was a concern and the effects from this variance were demonstrated in testing. The second material tested was 1018 cold rolled steel. According to Matweb.com, the yield strength for 1018 cold rolled steel was 53,700 psi and the ultimate strength was 63,800 psi.

A-36 Structural Steel

For the A-36 Structural Steel, 11 tests were conducted along with a tensile test in order to corroborate the data given by the manufacturer. For the tensile test, the ultimate strength was determined to be 72ksi, which fell within the manufacturer's specifications. The trial number, alternating stress level, and cycles till failure at a particular stress can be seen in *Table (2).*

Table 2. Fatigue testing results for *A-36 steel*

A-36 Structural Steel			
Trial #	**Cycles**	**Weight** (lbs)	**Stress** (psi)
1	1036172	15	44865
2	59518	16	47785
3	89587	18	53630
4	74690	20	59470
5	254199	20	59470
6	9630	21	62390
7	74665	21	62390
8	43678	22	65310
9	15104	23	68230
10	20738	24	71150

This data in graphical form can be seen in *Figure (11).* As scattered as it may seem at the first glance, in general, it is very comparable to the results found in textbook displays of tests conducted for this class of materials.

1018 Cold Rolled Steel

For the 1018 specimen, 7 tests were completed. The data for the testing on 1018 specimen is recorded in *Table (3).* The fatigue strength diagram for 1018 steel can be seen in *Figure (12).* Here, the trend follows a much more linear pattern when compared to the data for wrought and structural steel.

It is important to note that using the Tinius-Olsen Tester at TCNJ, the tested ultimate strength was determined to be 84,400 psi (which is well over the listed value of 63,800 psi). This discrepancy does not invalidate the fatigue tests, but makes use of the acquired yield strength difficult. For the 1018 specimen, it especially difficult to demonstrate breakage of specimen at stress levels significantly under the yield strength with such a low and most probably questionable listed yield strength.

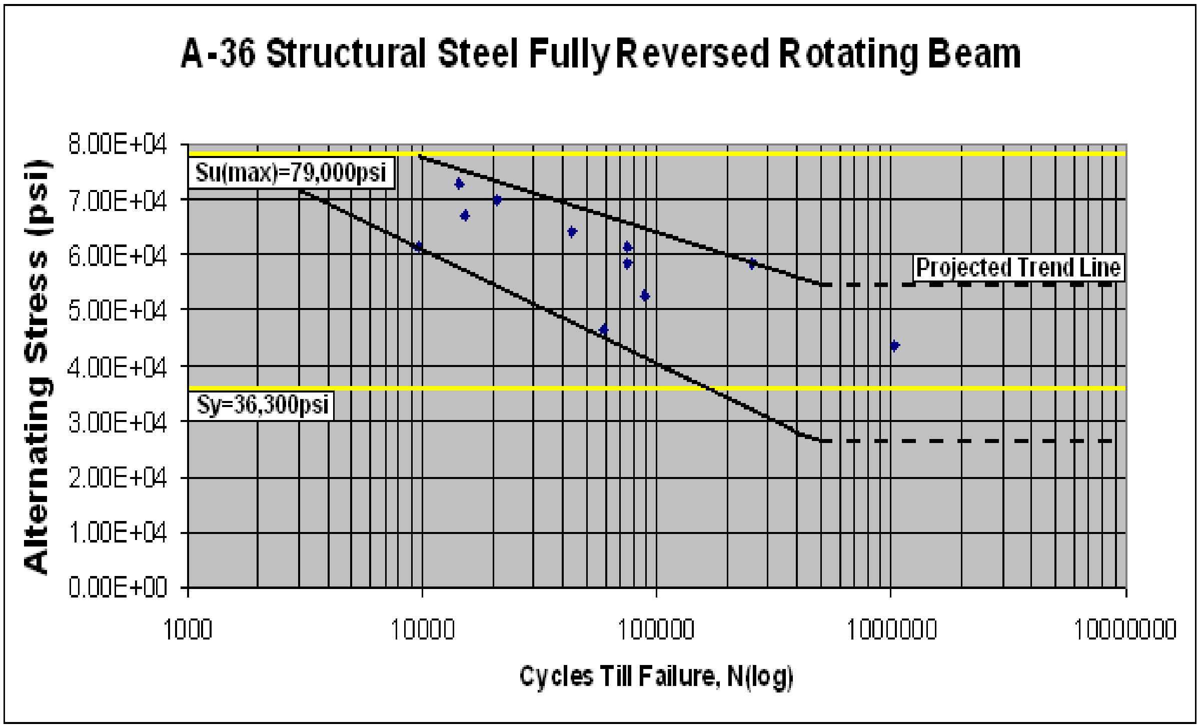

Figure 11. S-N plot for A-36 structural steel

Due to the fact that three specimens are produced from a single rod, the exact yield and ultimate strength can be known for two specimens if so desired. Otherwise, a statistical analysis must be performed in order determine the standard deviation.

Table 3. Data for fatigue testing of 1018 steel

1018 Cold Rolled Steel			
Trial #	**Cycles**	**Weight** (lbs)	**Stress** (psi)
1	2642953	14.102	42240
2	1790516	15.102	45160
3	655050	16.102	48080
4	586024	18.102	53920
5	143000	21.102	62680
6	36623	24.102	71445
7	3332	27.102	80205

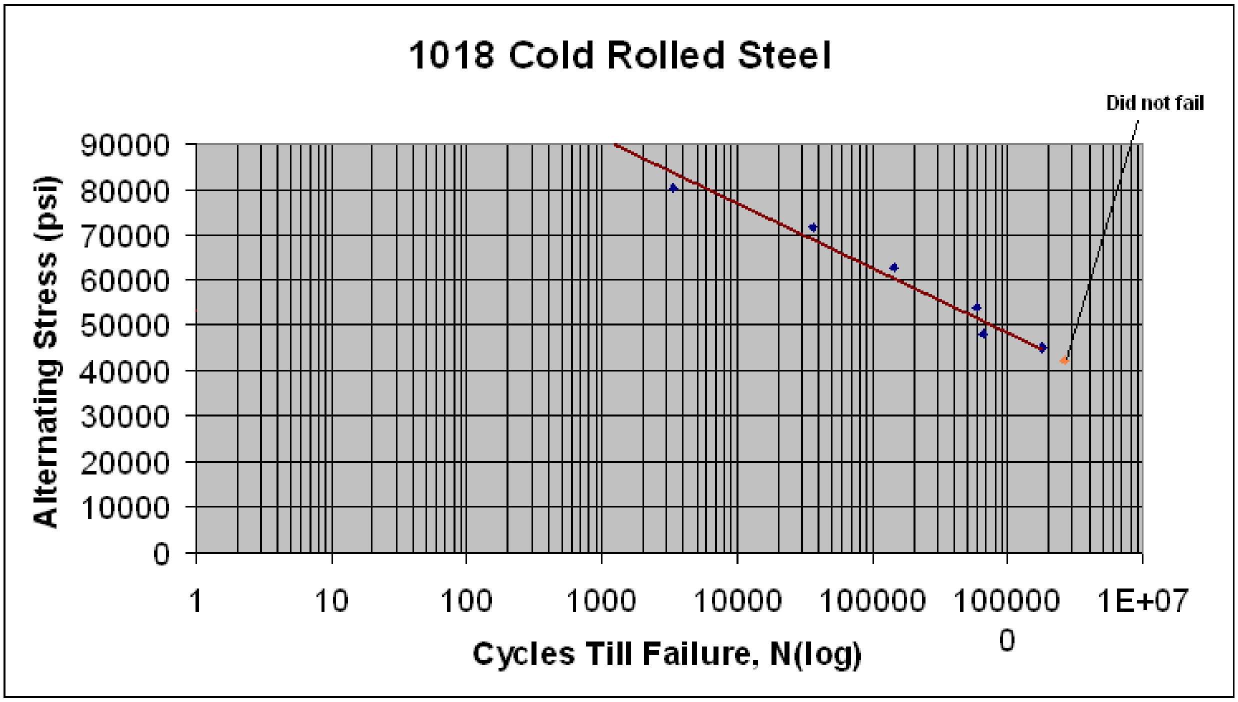

Figure 12. S-N plot for 1018 cold rolled steel

Utilizing the yield strength of 53,700 psi for the 1018 steel, which is most likely significantly lower than the actual yield strength of the specimen tested, one can see that two of the specimen failed below this yield strength.

When compared with textbook plots of this material; in the region between 10^3 and 10^6 cycles, one can see that the data remains rather close to the theoretical trend line. This similarity is expected to continue and is very promising. However, further testing should be conducted in order to draw further valid conclusions.

Figure 13. Actual Tested Specimens of the 1018 cold rolled steel

Confirmation with Theory

The most conservative fatigue theory based on a specimen's ultimate strength is the Goodman Criteria. The equation is expressed as:

$$\sigma_a / S_e + \sigma_m / S_u = 1$$

Where *Se* is the endurance strength, Su is the ultimate strength, and σ_a and σ_m are the alternating and average stress, respectively. The average stress is zero due to nature of rotating beam fatigue testing, and the equation simplifies to:

$$\sigma_a / S_e = 1$$

Taking an approximate endurance stress/strength of 42,200 psi for 1018 cold rolled steel, which is the stress level for the specimen that has yet to fail, any stress level above 42,200 psi will result in a value larger than 1 for σ_a / S_e. This means that any alternating stress level above 42,200 psi will result in failure of the 1018 specimen. This is exactly what can be seen for every value test for the 1018 specimen.

VII– Observations

The following is a listing of the interesting observations made up to this point in the process:

1. All solid specimens with reduced diameter (at center) failed at the Mid-Span,
2. All specimens with discontinuities (near the Mid-Span) failed at the discontinuity,
3. All specimens failed at higher (more conservative) values than expected/listed,
4. Certified specimens fail at values closer to the predicted ones than the non-certified ones,
5. Steel specimens fail at values closer to the predicted ones than the Aluminum samples,
6. Failure of all specimens was abrupt-no warning,
7. The test results may be considered as Precise but certainly not perfectly Accurate.

VIII– Recommendations

The following recommendations may be made at this stage of the task:

1. Avoid the use of small diameter sections as the variation in results may become quite troublesome-[do not use sections with a (effective) diameter of less than 1/8" for Steel and less than 1/4" for Aluminum],
2. Work with specimens that have a Length to Diameter Ratio of: $12 \leq L / D_{Effective} \leq 25$,
3. In reduction of the diameter of the specimen in the mid-span; to avoid Stress Concentration, do not use "fillets"(even with maximum possible radius),
4. Select motors that provide a reasonable combination of power and RPM,
5. If you choose to work with aluminum, don't set it as the base metal for the experiment,
6. Try to obtain materials with certification as this may save you a great deal of time,
7. If possible, run a complete tensile test on a sample of the bars used for the specimens,
8. If possible, cut "all" of the specimen (in one set of tests) from the "same" bar,
9. To examine potential variations (perhaps due to set-up), double/triple the number of tests for a "single" level of stress,
10. Exercise the safety precautions in this experiment to the full extent,
11. Obtain the most updated results and recommended procedures from the authors,
12. Share your findings and Alternative Solutions with the authors so that they may share them with other interested parties.

IX – Total Cost of *EFTM*

A complete list of the materials and the components for construction of the proposed EFTM is presented in *Appendix (E)*. In addition, we have taken extra care in recording the machining and assembly times for the creation of the apparatus. The total (conservatively) estimated cost of $2,500 is certainly an attractive figure. It goes without saying that several hundreds of hours have been dedicated by the collaborating students in the design and fabrication of the tester.

X – A Short Assessment

The authors used the audience in two of the sections of the Mechanical Design and Analysis course at TCNJ for their short assessment. The object for this exercise was to measure if the observations and the running of the tests on the 1018 CR Steel specimens made a noticeable difference in the better understanding of the fatigue phenomenon. A Laboratory handout has been created for running the experiment. Please see *Appendix (F).*

Ordinarily, a *Rating and Assessment* form is handed out for this type of activity at TCNJ. This form is included in *Appendix (G). Tables (H-1) through (H-3)* [placed in *Appendix (H)*] provide detailed summaries of the results for three of the (more measurable) questions on the project's assessment form.

Nearly all participants state that they would incorporate an activity of this nature should they get the opportunity to teach a similar course. The assessment results clearly reflect on the fact that there is (nearly perfect) consensus that the project is a balanced activity that is highly valued by the members of the fifteen (15) groups. They also shared their thoughts on how their exposure to the testing process and completion of the exercise has influenced their much better understanding and appreciation of these important criteria in "failure prevention".

XI – Summary and Conclusions

An affordable and a fully functional educational version of the R. R. Moore Fatigue Testing apparatus that produces dependable results is proposed for national adaptation. Junior and senior Engineering students have collaborated in the design and fabrication of the apparatus. The time factor for conducting fatigue testing in an educational environment has been incorporated in the design process. The process for the design of the apparatus, its subsystems, and the features of components are disclosed in details. A complete list of the materials and the components for construction of the proposed Educational Fatigue Testing Machine (EFTM) is provided. Full details of two sets of tests conducted on two different materials are presented. A sample Laboratory Handout is enclosed for examining the potential of the unit for conducting meaningful experiments. Summary of a short/preliminary assessment reflecting on the positive educational outcomes due to the use of this apparatus is shared with the engineering community. It is believed that in comparison with the commercially available counterparts of the *EFTM*, an alternative solution is offered that may prove feasible for implementation. This approach is beneficial for all parties involved including; the researching/collaborating student(s), underclassmen who would benefit from such experiments, and the enthusiastic instructors/

laboratory coordinators who may be fighting with budgetary issues. The only remaining obstacle is the better understanding of why the experimentally obtained number of cycles are conservatively higher than the (theoretically) predicted ones. So, further examination of the text book models/equations and search for ascertaining materials that do not suffer from a large standard deviation (from the expected mean) should continue. However, although we have not yet achieved an impressive level of "accuracy", we can clearly conclude that the results are certainly "precise".

Acknowledgements

The authors express special thanks to Alexander Michalchuk (former department senior technician and machinist) and Joseph Zanetti (current Machine Shop Supervisor) for their continuous support and dedication to the project.

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Appendix A: Summary of the Fatigue Failure Theories

Modified Endurance Limit

The level of stress at which a member will never fail, no matter how many cycles of stress it experiences, is known as the endurance limit, S'e. This limit is often determined by the "knee" that appears on the S-N diagram. Towards the bottom of the S-N diagram, less than 1 million cycles, a conservative estimate for design purposes can be set at one half of the ultimate strength (Su) for that particular material, even though it varies between 0.45 and 0.6 times the ultimate strength.

In order to determine an acceptable stress level for a particular machine component, the modified endurance limit is to be calculated. The *modified endurance limit (Se)* is expressed as:

$$\mathbf{Se} = C_f\, C_r\, C_s\, C_t\, (1/K_f)\, \mathbf{S'e}$$

Where:

- S'e = endurance limit,
- C_f = surface finish factor,
- C_r = reliability factor,
- C_s = size factor,
- C_t = temperature factor, and
- K_f = fatigue strength concentration factor

Fatigue Failure Theories

Fluctuating loads upon a member that already is subject to a constant level of stress have been shown to significantly affect the fatigue life of that particular member. To cope with such situations, the mean stress and the range, or alternating stress, must be determined.

The mean stress is:

$$\sigma_m = \frac{(\sigma_{max} + \sigma_{min})}{2}$$

And the alternating stress is:

$$\sigma_a = \frac{(\sigma_{max} - \sigma_{min})}{2}$$

Fatigue failure theories have been developed in order to try and predict the level of stress that would cause a machine member to fail. Here the focus has been placed on the Goodman, Gerber, SAE, and the Modified Goodman theories. The Goodman, Gerber, and SAE all utilize the endurance and ultimate strengths (fracture strength for SAE). The Soderberg and Modified Goodman equations rely on the endurance, yield, and ultimate strengths. The mathematical expressions for them may be seen in *Table (A-1)*.

In *Figure (A-1)*, the graphical relationship for the fracture theories is illustrated. The fatigue theories based on yield are represented in *Figure (A-2)*. It can be observed that the most conservative estimate for fatigue failure is the SAE model, while the Soderberg equation is the most conservative yield theory.

Appendix A (Continued)

Table A-1. Theories for the Prediction of Fatigue Failure

Fatigue Failure Theory	**Corresponding Equation**
Goodman	$\sigma_a / S_e + \sigma_m / S_u = 1$
Gerber	$\sigma_a / S_e + \sigma_m^2 / S_u^2 = 1$
SAE	$\sigma_a / S_e + \sigma_m / S_f = 1$
Soderberg	$\sigma_a / S_e + \sigma_m / S_y = 1$
Modified Goodman	$\sigma_a / S_e + \sigma_m / S_u = 1$, [for $\sigma_a / \sigma_m \geq \beta$] $\dfrac{\sigma_a + \sigma_m}{S_y} = 1$, [for $\sigma_a / \sigma_m \leq \beta$] Where the material constant $\boldsymbol{\beta} = S_e(S_u\text{-}S_y) / S_u(S_y\text{-}S_e)$

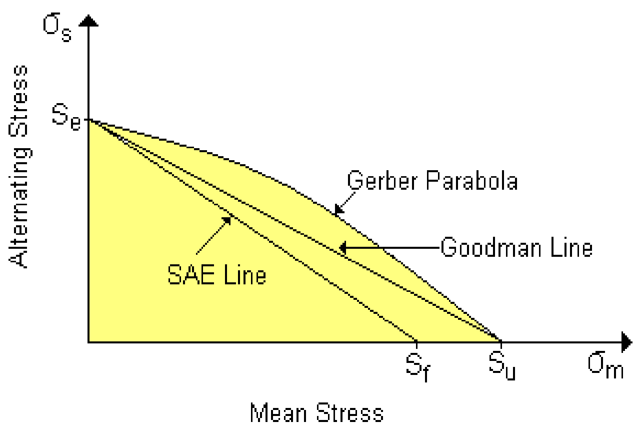


Figure A-1. Graphical Comparison of different fracture theories based on fatigue failure equations

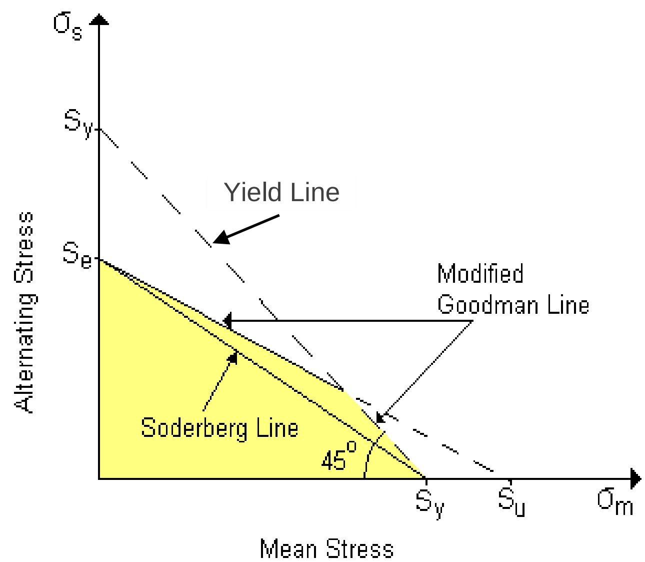


Figure A-2. Graphical comparison of different yield theories based on fatigue failure equations

For a member to be safely utilized, its mean and alternating stress must fall within the shaded areas of the above regions. If a safety factor is utilized, it would have the same trend as the respective equation used, yet be translated in the negative alternating(y) or mean(x) stress direction.

Appendix B: Summary of the Stress calculations

The benefit of the present design is the simplicity of its modeling and ease of understanding by students. The rotating specimen can be modeled as a simply supported beam. The maximum bending stress of a beam can be expressed as

$$\sigma = Mc/I \qquad \text{(B.0)}$$

Simplifying;

$$\sigma = \frac{32\,M}{\pi D^3} \qquad \text{(B.1)}$$

For this model of loading, the bending moment can easily be expressed by:

$$M = \frac{WL}{2} \qquad \text{(B.2)}$$

Where L is the length of the moment arm.
The maximum stress that the beam encounters (derived from shear and moment diagrams) is:

$$\sigma = \frac{16WL}{\pi D^3} \qquad \text{(B.3)}$$

It should be noted that the ends of the beams will be kept straight, and while this will have some effect on the maximum stress experienced by the beam, it may be safely neglected, especially when dealing with fatigue testing.

In order to determine the appropriate loads that need to be applied to the specimen, certain constraints had to be selected. All initial calculations were made for a 0.375 inch 1020 steel rod due to its high strength, high fatigue life, and large moment of inertia relative to other specimen that would be used in such testing. This was done in order to ensure that the machine would be able to handle a myriad of differing materials.

A value for the distance between the two fulcrums had to be set. It was assumed to be 9 inches, allowing 2.5 inches on each of the specimen to be contained within the special holder, and 4 inches of exposed material. The length was selected because it would minimize the size of the machine as well as the total deflection of the specimen.

In order to determine the maximum stress that may be experienced by the beam, a fatigue life of 10,000 cycles was selected. The corresponding level of stress was 375MPA, or 54 KSI. In Table 1.2, the cycles till failure and corresponding values for the status of the beam can be seen.

Table B-1. Properties for a 0.375" diameter 1020 steel specimen

	Cycles till Failure		
	10^6	10^5	10^4
Stress (PSI)	36,600	47,000	54,400
R = Load Required (lbs)	152	195	225
Deformation (in)	0.0793	0.1017	0.117

For the mounting of the carriage, as well as all other members, it was decided that 0.25" bolts would be used if they had an acceptable load bearing capacity.

The shear and bearing stresses on these structural members were determined to be well within their corresponding strengths.

Appendix C: Critical Speed of the Shaft

Because the specimen is performing as a deflected rotating shaft, the critical speed needs to be determined. Despite the fact that the specimen was undergoing destructive testing, it was not desired that the specimen would fail due to dynamic instability. To control vibrations, and for dynamic stability of a deflected shaft, the angular velocity must be either below or considerably exceed the corresponding critical speed. The critical speed of a deflected shaft may be expressed as [5]:

$$N_{cr}= \frac{[(g\Sigma W\delta)/ (\Sigma W\delta^2)]^{1/2}}{2\pi} \qquad \text{(C.1)}$$

Where:

δ = the deformation of the shaft,
W = the applied load, and
g = the gravitational constant

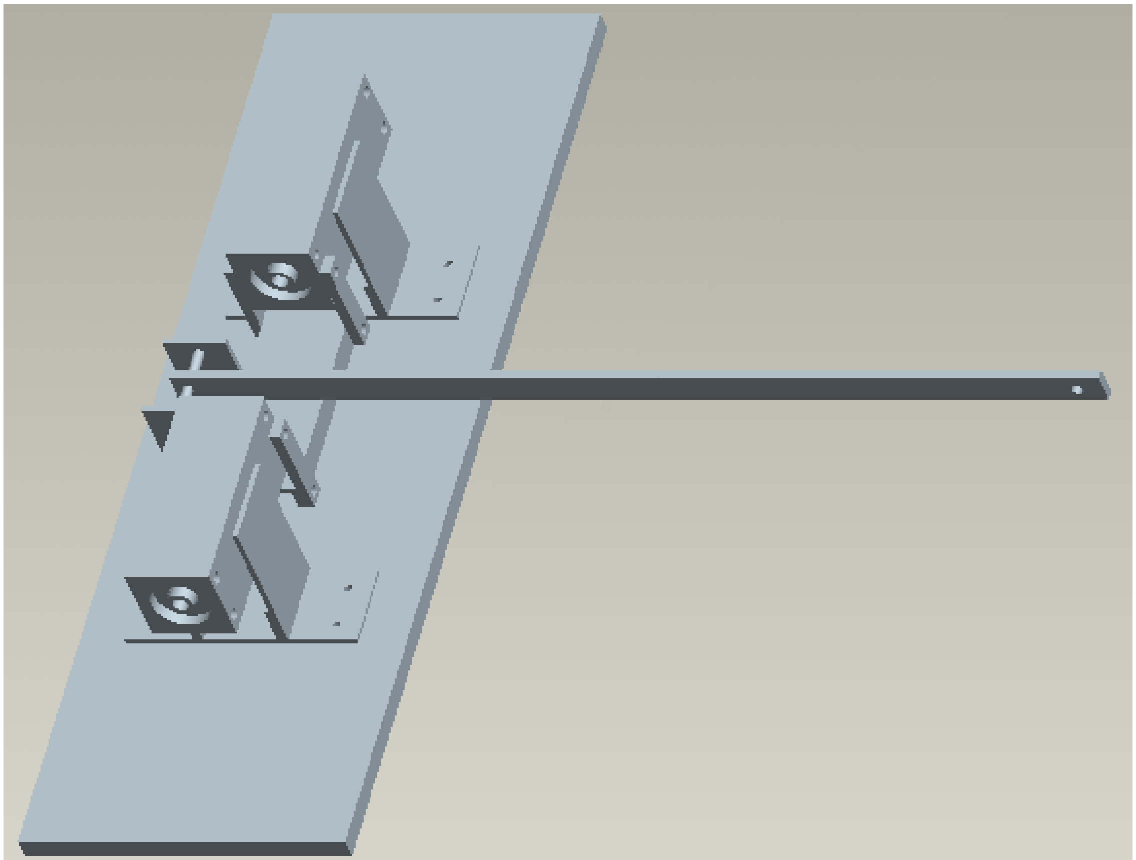
In *Table (C-1)* are listed the different deflections for certain fatigue lives of a 1020 steel specimen and their corresponding critical speed.

Table C-1. Different Critical Speeds for a 0.375" Diameter 1020 Steel Specimen

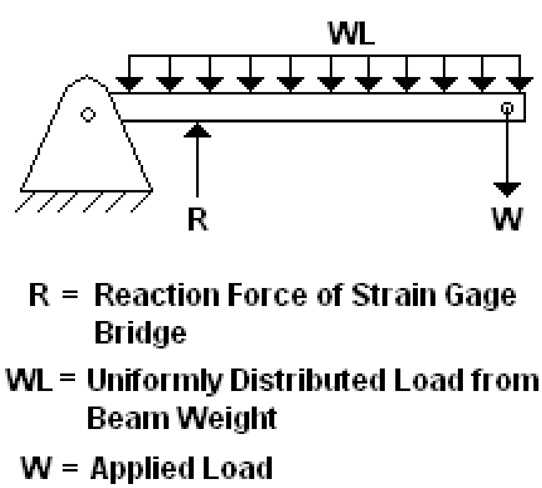
Cycles till Failure	Critical Speed (RPM)
10^6	192
10^5	170
10^4	158

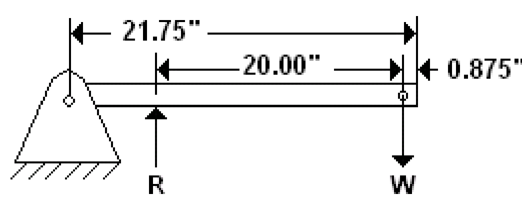
Since failure was desired to occur in under 24 hours, it was determined that the minimum angular velocity needed to be at least 694 revolutions per minute. While this initially seems to be a problem, the shaft can be made dynamically stable by spinning it at a higher rate of rotation than its critical speed. With the low critical speeds listed above, there is no limit aside from the motor output to how fast the specimen can be rotated. By operating at an angular velocity above the critical speed, the experiment may be completed within 24 hours.

Appendix D: Modeling of Lever Arm and Determination of Calibration Equation



Due to the uniformly distributed weight of the lever arm, the load applied to the specimen is more than just ten times the load applied to the end of the lever arm. The modeling of the situation can be seen in the Diagrams below.





Utilizing a density of 0.0362lbs/in^3 for the aluminum lever arm, a beam length of 20 inches, and application of basic statics by taking the moments around the lever arm fulcrum, the applied force to the load bridge is simply expressed as: F= 10*W+3.62(lbs)

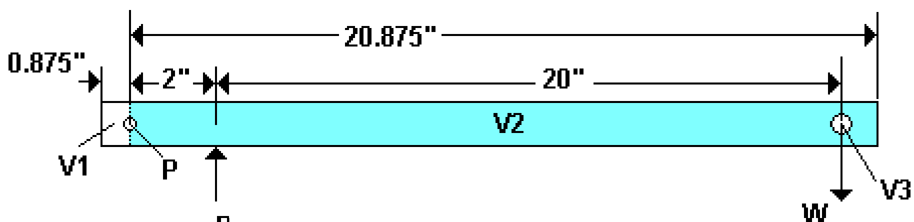
Here F is the applied force at the center of the bridge and W is the weight placed at the end of the lever arm. For a 15 lbs end load, the applied force is 153.62 lbs. When this value is compared to merely multiplying the load by a factor of 10, the percent difference is 2.36%.

With a more refined model of the applied force is used, incorporating the beam overhang from the pivot point, removed material from machining, and the weight of the hook and bolt, the equation becomes: F=10*W +5.797(lbs)

The weight of the load bridge and shackles is 1.23lbs. With this value, a more exact calibration equation for the load applied supported by the specimen holders is:

F=10*W+7.027(lbs)

Due to the sensitivity of fatigue testing to the amount of stress placed on the specimen, determining the exact value of stress present in the specimen is of great importance.



(The Applied Load to the Specimen)

$V1=0.875in^3$
$V2=20.875in^3$
$V3=-\pi d^2/4=0.11in^3$
$\rho=0.0362in^3$
h=hook and bolt weight=0.19lbs

$\Sigma M_p=0= -V1*\rho*(0.875/2)-2*R-V3*\rho*20+20*F+(20.875/2)*\rho V2+20*h$

0.139+2*R+0.7964=20*F+7.887+20*h

R= The Applied Load to the Specimen = 5.79+10*F (Where $F = W$ in the diagram)

Appendix E: Parts List and Breakdown of the cost

Part	Quantity	Price ($)
9″long 2″x4″x0.125″ 6061-T6 Aluminum	1	10.50
0.5″x6″x5″ 6061-T6 Aluminum	2	2x7.50
0.5″x7.75″x23″ 6061-T6 Aluminum	2	2x25
2″x2″x0.125″ 6061-T6 Aluminum Square Stock	2 ft	~40.00
5/8″OD, 3/8″ ID ABEC 1 Bearing	4	4x7.50
7/8″ Diameter SAE 660 Bronze Bearing	1 ft	~40.00
0.25″x0.75″ 6061-T6 Aluminum Flat Bar	2 ft	~10.00
AC Motor	1	140.00
Electrical Counter	1	60.00
Electrical Kill Switch Mechanism	1	40.00
Gear, or Belt Drive System	1	190.00
Test Specimens	1	45/Year
Counter Bracket	1	30.00
Switch Mounting Plate	1	~20
Plexi-glass	1	40
	Total Cost:	*~$765*

1. Overall Cost of the Materials and Components ≤ $ 800
2. Frame on Casters ≤ $ 550
3. Required Machining and Assembly Time:

I - Average Machining: About 16 hours (at $30/hr.)
II - Above Average Machining: About 8 hours (at $60/hr.)
III - Assembly of Frame and Components: About 6-8 hours (at $20/hr.)

Total ≤ $2500

Appendix: F

Laboratory Handout
for
Finite Life Fatigue Experiment

Objectives:
1. To gain better familiarity with Fatigue Failure due to fully reversed bending.
2. To learn how to create and evaluate S-N diagrams.
3. To examine the effects of Discontinuities.
4. To examine the reliability of the Fatigue Failure Theories using test data.

→→→ Safety ←←←
Caution! Please be sure to wear safety goggles while examining the specimen during testing. The beam may behave violently upon failure.

Equipment:
Frame
Fatigue Tester Apparatus

Procedure
1. First, make sure the counter has been set to zero and the power to the motor is off.
2. For a 0.375 specimen filleted down to 0.3125", calculate the required stress level for:
   - 10^6 cycles
   - 2×10^5 cycles
   - 10^5 cycles
   - 5×10^4 cycles
   - 10^4 cycles

3. Run a specimen for each life span at that given stress level.

4. With this data, construct the S-N Diagram and compare it with that created with the theoretically generated one.

5. For 0.375 specimens filleted down to 0.3125" and with the following discontinuities:
   - Notch,
   - Large Hole,
   - Small Hole,

   Run the experiment at the stress level used in Section (2) for the 5×10^4 cycles lifespan.

6. Discuss the effects that these discontinuities may have on the fatigue life of each specimen.
7. Compare the Experimental Results with the Theoretical Results and comment on:

    A - the precision and the accuracy of the data obtained,

    B – what may be the cause of the differences between these results.

Discussion Questions

- What are some of the causes for Fatigue Failure of materials?
- Comment on the statistical nature of fatigue failures.
- Define Endurance Strength and Endurance Limit and Compare them with each other.
- What can be done to minimize the possibility of Fatigue Failure and still conceive a product that is competitive in today's competitive (international and domestic) markets?

Appendix G: Rating and Assessment Form of the Activity

RATING AND ASSESSMENT

1. How many members formed your group? [ ]

2. Indicate the number, duration, and place of EACH of your meetings.
(Use the following TABLE for tabulation)

Meeting #	***DATE***	***DAY***	***TIME***	***PLACE***
Total Time Expended:				

3. How would you rate the time required for completion of this Project?
[Use a √ Mark in the blank box of your choice.]

Too Short	*Short*	*About Right*	*Long*	*Too Long*

4. If you had to do this experiment/activity again, how long would it take the second time? Use the Percentages listed below.

(30-40)%	*(40-50)%*	*(50-60)%*	*(60-70)%*	*(70-80)%*	*(80-90)%*	*Almost The Same*	*Can't Predict*

5. *Would the experience gained in this activity help you optimize your approach the next time you have to deal with a similar task?*

(Use the Rating and the Space provided below)

Highly Unlikely				Definitely
1	2	3	4	5

6. *How would you rate the overall value of this Experiment and Project?*

Lowest				Highest
1	2	3	4	5

7. *If you get to teach a similar course, would you incorporate such an activity in your course? If yes, what changes would you recommend or introduce? (Use the Rating and the Space provided below)*

Highly Unlikely	*Unlikely*	*Probably*	*Very Likely*	*Definitely*
☐	☐	☐	☐	☐
Recommended Changes:				

Appendix H: Tables for the Assessment

Table H-1. Summary of the Results for the *First* Measurable (and relevant) Question on the Project Assessment Form

Question # 1: *How would you rate the time required for completion of this Project?*						
xxxxxxxxxx		Rating				
Section #	# of Groups	Too Short	Short	About Right	Long	Too Long
01	*7*	-	-	5	1	1
02	*8*	-	-	6	2	-
Total	*N = 15*	-	-	11	3	1

Table H-2. Summary of the Results for the *Second* Measurable (and relevant) Question on the Project Assessment Form

Question # 2: *Would the experience gained in this activity help you optimize your approach the next time you have to deal with a similar task?*						
xxxxxxxxxx		Rating				
Section #	# of Groups	Highly Unlikely	Unlikely	Probably	Very Likely	Definitely
01	*7*	-	-	-	4	3
02	*8*	-	-	1	3	4
Total	*N = 15*	-	-	1	7	7

Table H-3. Summary of the Results for the *Third* Measurable (and relevant) Question on the Project Assessment Form

Question # 3: *How would you rate the overall Value of this Experiment and Project?*						
xxxxxxxxxx		Rating				
Section #	# of Groups	Very Low	Low	Medium	High	Very High
01	*7*	-	-	1	4	2
02	*8*	-	-	-	6	2
Total	*N = 15*	-	-	1	10	4