Group 1

Bolted Joint Rig Test Development

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Table of Contents

Abstract

The purpose of this project is to design two test fixtures that will allow a comprehensive test and analysis of the fatigue life of a connecting rod and a main bearing cap threaded joints. Each fixture will interface with the MTS servo-hydraulic fatigue-testing machine, which is accessible at the National High Magnet Field Laboratory (NHMFL). The test fixture should be able to validate Cummins history of varying bolted joint designs. It should aid in the evaluation process for finding design improvements to fatigue life of threaded joints. It should be able to accurately reproduce failure modes that are experienced in real engine components. Typical pre-loading values for the connecting rod and the main bearings threaded joints are approximately 16,000 lbs and 45,000 lbs respectively, which were used to calculate what torque to set the bolts to. After going through a thorough analysis process two final concepts were selected. These two concepts, best satisfied the design objectives, constraints, needs, wants, cost, and expectation specified by the sponsor.

The first concept decided on was for the connecting rod. This called for the two bolts on the side of the connecting rod to be isolated and cut out. The sample was then placed within four wedges within the MTS machine. The second concept was for the main bearing cap. A section of the engine block containing the threaded bolt was isolated and machined down to fit within the wedges. The bolt was then put through a bolt-housing fixture and into the engine block. The top of the housing fixture was made so that an adapter could screw

into it and then into the MTS block. The MTS block was then held on by the MTS wedge grips.

A total of four tests were conducted. The first connecting rod underwent both tension and compression at 3 Hz. It went on for 1.3 million cycles and did not break. The second connecting rod underwent tension only. A wedge failed at around 980,000 cycles. The two main bearing cap tests both failed at the same point, along the engine block. This was not the desired place of failure. The failure was due to the modifications done to the engine block so that it would fit within the MTS machine. The fixture and adapter however did not fail.

Overall, the connecting rod fixture worked adequately and can be improved further with some different material selection for the wedges. Also, minor modifications will have to be made to the main bearing cap fixture in order to improve its performance.

1.0 Introduction

1.1 Project Objective

While most of the load is carried in the first few threads, this is not necessarily where failure occurs. Therefore to properly evaluate the failure mode of the bolts, a test fixture that can be used to evaluate design improvements of the fatigue life belonging to threaded joints had to be designed. There will be two different threaded joints under evaluation for this project. The first is that of the joint between the main bearing cap – to – engine block. This joint can be seen in Figure 1 and 2 below. The second joint in question is within the connecting rods of the engine. This joint can be seen in Figure 3 and 4 below. Parameters to be studied include: bolt boss diameter, thread pitch, thread type, counterbore depth, number of engaged threads, bolt pre-load and alternating load.

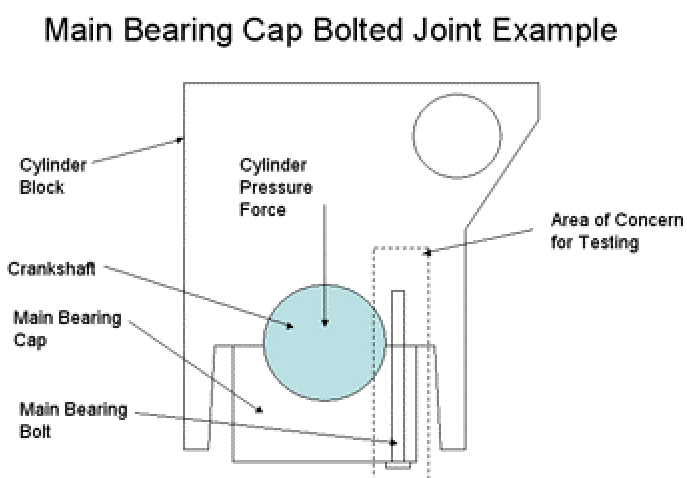

Figure 1: Main Bearing Cap Bolted Joint Example, figure from Cummins.

Figure 2: Photograph of Actual Main Bearing Cap

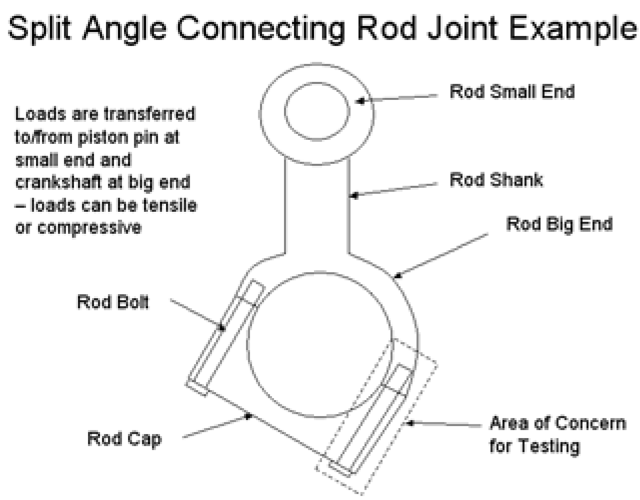

Figure 3: Split Angle Connecting Rod, figure from Cummins.

Figure 4: Photograph of Actual Connecting Rod

1.1.1 Cummins's Needs

Cummins, the sponsor, needs a test fixture and/or set-up that can evaluate current threaded joints on a 15L connecting rod and main bearing cap. These sections will be isolated from the complete engine part. The loads in diesel engines require advanced testing equipment due to a loading of approximately 45,000lbs for the main bearing cap bolts and 16,000lbs for the bolts in the connecting rod. Test facilities of Cummins have an MTS machine that can produce the desired loads; therefore all designs must interface with a standard MTS servohydraulic machine.

Cummins will also need detailed drawings of the fixtures, operation documentation (or manual), and data along with a demonstration that the test will reproduce the failure mode of the real components. The exact design specifications are gone over in more detail within section 3.0.

1.1.2 Industrial Uses

This project is of importance for Cummins and it's role in industry. The final project will help to develop an experiment to test the fatigue failure for the bolts within the connecting rod and that of the main bearing cap. This will in turn be used to better understand where the joints are failing and might lead to a better design.

1.2 Screws and Fasteners

1.2.1 Basic theory, definitions and terms

The success or failure of a design more than often hinges on the proper selection and usage of its fasteners. A fastener is a bolt, nut, or screw designed to hold things together. Fasteners come in a wide variety of shapes and sizes and are found in virtually every machine and mechanism used today. Cummins for example, uses hundreds of fasteners, machine strews or cap strews, on every engine they manufacture. Their engines can be found in tractor-trailer, which we all know, run long hours and occasionally under harsh condition. Thus, their fasteners must go through a scrutiny of test to eliminate the risk of failure. Areas of concern include threads, stresses in threads and preloads.

Threads

A thread is a helix shaped slot, which is machined onto a bolt that causes the bolt to advance into the work piece when rotated. The threads of a bolt may seem to be an insignificant component of a design, but in fact threads are one of the most fascinating. Bolts and threads are manufactured in a variety of shapes, sizes, and materials. Thread forms originally was different in each major manufacturing country until after World War II when the United States, Great Britain and Canada standardized them into what is now the Unified National Standard (UNS) series. A European Standard is also defined by ISO and has the same thread cross-section shape but uses metric dimensions. Figure 5 shows the standardized thread form used in both the UNS and the ISO standard. Both UNS and ISO standards define thread size by the major diameter of an external thread. The thread pitch is defined as the distance between adjacent threads. The crests and roots are made flat in order to reduce stress concentration from that of a sharp corner.

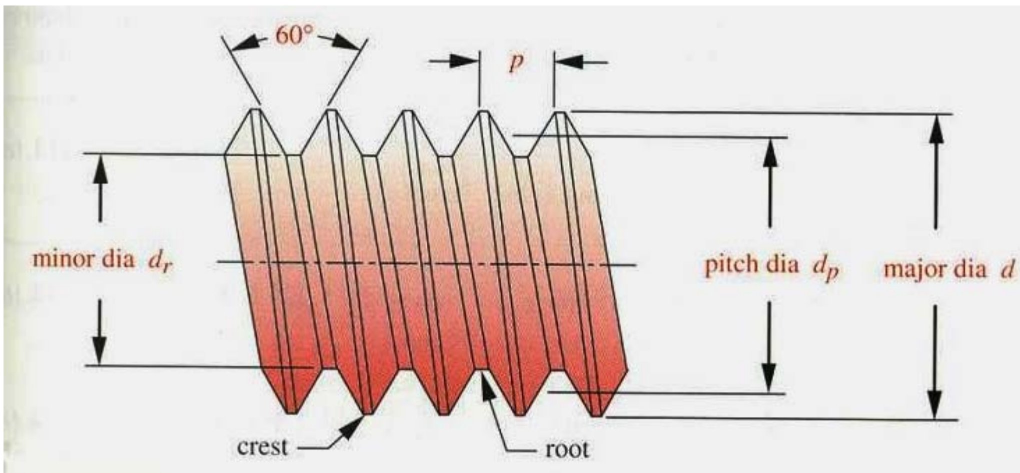

Figure 5: Unified National and ISO standard thread form

All Standard threads are right handed by default. A right-handed thread will advance the screw into the work piece when turned clockwise. A thread is specified with a code that defines its series, diameter, pitch and class of fit. The pitches of UNS threads are defined commonly as the number of threads per inch, while metric (ISO) thread pitches are defined by the pitch dimension in mm. Below are examples of how UNS and ISO threads are specified.

$$\tfrac{1}{4} - 20 \text{ UNC} - 2\text{A} \quad \text{(UNS)}$$

$$M8 \times 1.25 \quad \text{(ISO)}$$

1.2.2 Stresses in Threads

When a nut engages a thread, theoretically all the threads engaged should share the load. However, inaccuracies in threads spacing cause nearly all the load to be taken by the first pair of threads. So, the conventional approach is to assume the worst case where one thread pair carries the entire load or to assume that all the engaged threads share the load equally. The true stress will lie between these two extremes, but most likely closer to the one-thread assumption. Power screws and fasteners for high-load applications are usually made of high-strength steel and are often hardened.

Axial Stresses

A power screw can see axial load of either tension or compression. If a threaded bolt is subjected to pure tensile loading, its strength is defined by the average of the minor and pitch diameters. A tensile-stress area A_t is then defined as

$$A_t = \frac{\pi}{4}\left(\frac{d_p + d_r}{2}\right)^2 \tag{1}$$

Where for ISO metric standard screw threads:

$$d_p = d - 0.649519 \cdot p \qquad d_r = d - 1.226869 \cdot p \tag{2}$$

And for UNS threads

$$d_p = d - 0.649519/N \qquad\qquad d_r = d - 1.2299038/N \tag{3}$$

d = outside diameter, p = pitch in mm, N = number of threads per in

So to calculate the stress in a threaded bolt due to pure axial tensile load F the following equation is used:

$$\sigma_t = \frac{F}{A_t} \tag{4}$$

For screws loaded in compression, the possibility of column buckling must also be investigated.

Shear Stress

One possible shear-failure mode involves stripping of the threads either out of the nut of off the screw. If the nut material is weaker, it may strip its threads at its major diameter. If the screw material is weaker, it may strip its threads at its minor diameter. If both are of equal strength, the assembly may strip along the pitch diameter. If we express the shear area in terms of the number of threads in engagement, an assumption can be made as to what degree of load sharing is appropriate.

The stripping shear area for one screw thread is the area of the cylinder of its minor diameter d_r.

$$A_s = \pi \cdot d_r \cdot w_i \cdot p \tag{5}$$

Where p is the thread pitch and w_i is a factor that defines the percentage of the pitch occupied by metal at the minor diameter.

The shear stress for thread stripping is then found from the following equation

$$\tau_s = \frac{F}{A_s} \tag{6}$$

Torsional Stress

When a screw is tightened on a nut, a torsional stress can be developed in the screw. The torque is depended of the friction at the screw-nut interface. If the screw is well lubricated, less of the applied torque is transmitted to the screw and more is absorbed between the nut and the clamped surface. To accommodate the worse case of high thread friction, the equation for torsional stress in a round section is used.

$$\tau = \frac{T \cdot r}{J} = \frac{16T}{\pi \cdot d_r^3} \tag{7}$$

d_r is the minor diameter.

1.2.3 Loading

The amount of preload is an important characteristic in bolts. The most accurate method of controlling the amount of preload on a bolt is to have both ends of the bolt be accessible. Then the amount of bolt elongation can be directly measured with a micrometer, or an electronic length gage and the bolt stretched to a length consistent with the desired preload based on equation 8 below:

$$\delta = \frac{F \cdot l}{A \cdot E} \tag{8}$$

Where F is the force, l is bolt length, A is the cross-sectional area and E is the Young's Modulus of the material. Ultrasonic transducers are sometimes used to measure change in bolt length when tightened, and these only need access to the head end versus both bolt ends. This method is exceedingly accurate and costly. An ultrasonic gage can run up to $24,500. Due to constraints both of these methods are not as useful in high-production or field-service situations, since the require time, care, precision instruments, and skilled personnel.

A more convenient but less accurate method measure or controls the torque applied to the head of a cap screw. A torque wrench gives a readout on a dial of the amount of torque applied. Torque wrenches are generally considered to give an error in preload of up to ±30%. If great care is taken and the threads are lubricated, this error can perhaps be halved, but it is still large. Pneumatic impact wrenches can be set to a particular torque level at which they stop turning. These give more consistent results than a manual torque wrench and are preferred.

The torque necessary to develop a particular preload can be calculated from equation 9:

$$T_i = F_i \cdot \frac{d_p}{2} \cdot \frac{(\mu + \tan\lambda \cdot \cos\alpha)}{(\cos\alpha - \mu \cdot \tan\lambda)} + F_i \cdot \frac{(1 + 1.5)d}{4} \cdot \mu_c \tag{9}$$

Factor out the force and bolt diameter to get:

$$T_i = K_i \cdot F_i \cdot d \tag{10}$$

Where:

$$K_i = \left[0.5 \cdot \frac{(\mu + \tan\lambda \cdot \cos\alpha)}{(\cos\alpha - \mu \tan\lambda)} + 0.625\,\mu_c \right] \tag{11}$$

K_i is called the torque coefficient.

The torque coefficient K_i is dependent on the friction coefficient μ_c between the head and the surface as well as the thread friction coefficientμ. If we assume a friction coefficient of 0.15 for both of the previously mentioned locations and calculate the torque coefficients K_i for all standard Unified National Standard coarse and fine threads, the value of K_i varies very little over the entire range of thread sizes. Thus the tightening torque T_i needed to obtain a desired preload force F_i in lubricated threads can be approximated as

$$T_i = 0.21 \cdot F_i \cdot d \tag{12}$$

1.3 Overview of Metal Fatigue

It has been recognized since 1830 that a metal subjected to a repetitive, fluctuating or vibration stress will fail at a stress much lower than that required causing fracture on a single application of load. This type of failure is called fatigue failure. It is often stated that fatigue failures account to 90% of all service failures due to mechanical loading. This type of failure usually occurs after a substantial period of time and without any previous indications during services. Below are some examples of equipments, which have components that are subjected to fatigue failures.

- Automobiles
- Aircrafts
- Compressors
- Pumps
- Turbines, etc.

Fatigue results in a brittle-appearing fracture with no gross deformations at the fracture. On a macroscopic scale, the fracture surface is usually normal to the direction of the principal stress. To recognize whether a component has failed due to fatigue loading, here are some of the characteristics of the appearance of the facture surface. It usually shows:

- Smooth region, due to the rubbing action as the crack propagated through the section or,
- A rough region, where the member has failed in a ductile manner when the cross section was no longer able to carry the load.

On a microscopic level, frequently the progress of the fracture is indicated by a series of rings or “beach marks” progressing inward from the point of initiation of the failure. Failure usually occurs at a point of stress concentration such as a sharp corner or notch or at a metallurgical stress concentration like an inclusion. Figure 6 is an illustration showing the different ways in which cracks are initiated.

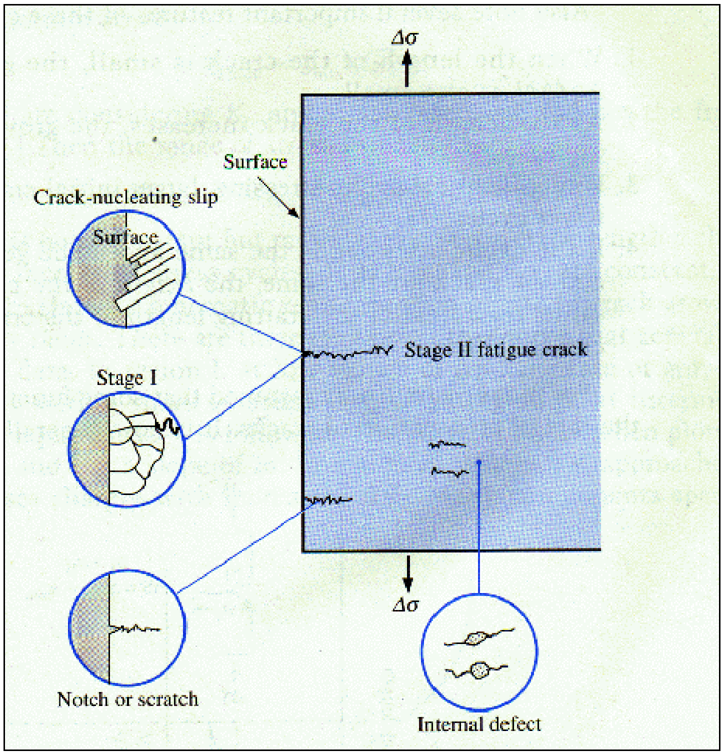

Figure 6: Deformation of crack propagation due to fatigue

Cracks, the initial step to fatigue failure, can be initiated from a series of events; from surface scratches caused by handling of the material to work hardening. From this point the crack propagates as a result of continuously applied stress. The next stage is failure, which comes soon after the crack is initiated, when the material is unable to withstand the applied stress.

1.3.1 Stress Cycles

There are three common ways in which stresses may be applied to a component: they can be applied through axial loading (tensile or compressive), torsional loading, bending loading or a combination of the three. Figure 7 gives a visual illustration of the different modes.

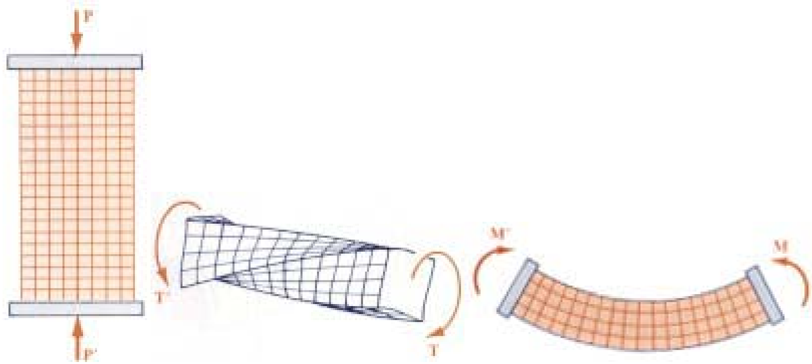

Figure 7: Loading Cases, axial, torsion, and bending

There are also three stress cycles with which loads may be applied to a component. The **reversed stress cycle**, shown in Figure 8a is the simplest stress cycle. This type of stress cycle has amplitude that is symmetric about the x-axis. The maximum and minimum stresses are equal in magnitude, but are opposite in sign. The most common type of cycle found in engineering applications is the **repeated stress cycle** (8b) where the maximum stress (σ_{max}) and minimum stress (σ_{min}) are asymmetric (the curve is a sine wave) not equal and opposite. A final type of cycle mode is where stress and frequency vary randomly. An example of this would be automobile shocks, where the frequency magnitude of imperfections in the road will produce varying minimum and maximum stresses.

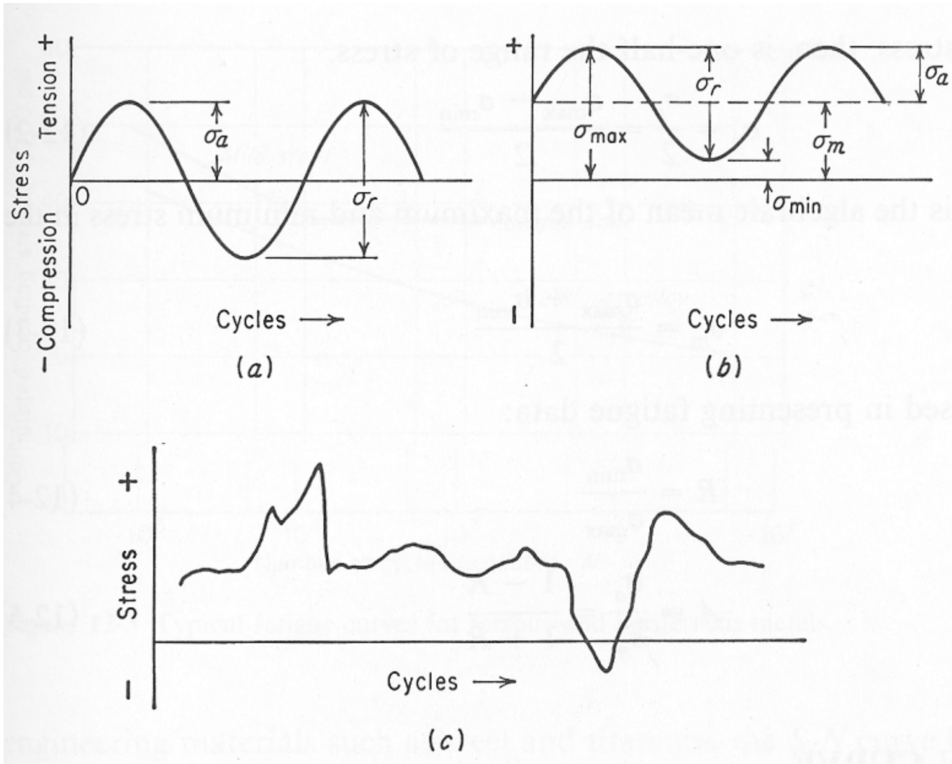

Figure 8: reverse cycle, repeated cycle and random stress cycle

A fluctuating stress cycle is considered to be made of two components, a mean stress component (σ_m) and an alternating stress component (σ_a). The range of stress, which is as the algebraic difference between the maximum and the minimum stress is also considered and is given by equation 13.

$$\sigma_r = \sigma_{max} - \sigma_{min} \tag{13}$$

The mean stress, which is the algebraic mean of the maximum and the minimum stress in the cycle is given by,

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} \tag{14}$$

The alternating stress is

$$\sigma_a = \frac{\sigma_r}{2} \tag{15}$$

The following two ratios are used in representing fatigue data.

$$R = \frac{\sigma_{\min}}{\sigma_{\max}} \qquad\qquad A = \frac{\sigma_a}{\sigma_m}$$

Stress Ratio — Amplitude Ratio

The most common method of presenting engineering fatigue data is by means of the S-N curve, a plot of stress S against the number of cycles to failure N as demonstrated in Figure 9.

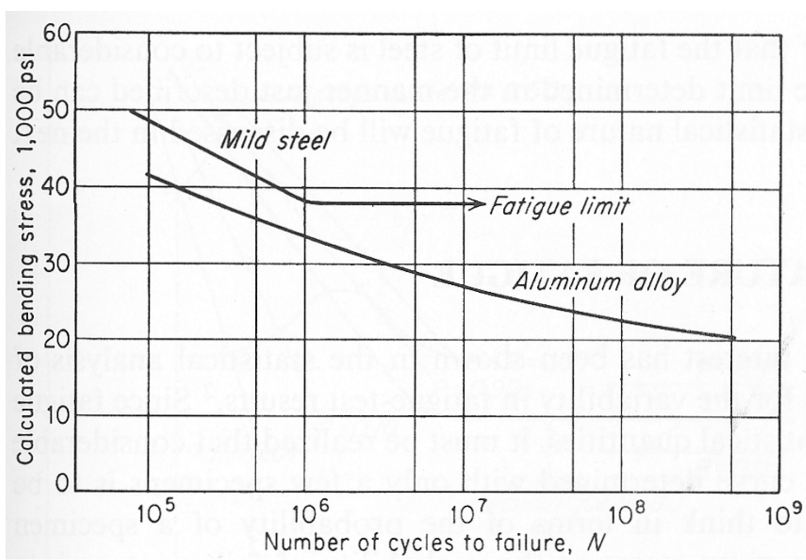

Figure 9*:* Typical S-N curve for Ferrous and Non-Ferrous materials

The value of the stress that is plotted can be any of the stresses listed above and the N-axis is almost always logarithmic. Most determinations of the fatigue properties of a material have been made in complete reverse bending because the mean stress is zero. For a few important engineering materials such as steel and titanium, the S-N curve becomes horizontal at a certain limiting stress. Below this limiting stress, which is called the **fatigue limit** or **endurance limit**, it is assumed that the material will endure an infinite number of cycles and will not fail. However, most non-ferrous materials such as aluminum, magnesium and copper alloys have an S-N curve that slopes downward with increasing number of cycles. These materials do not have a fatigue limit because the curve never becomes horizontal as you increase the number of cycles. So for such materials, it is common to arbitrarily choose the number of cycles, 10^7 for example, in order to characterize the fatigue properties of the material. The Basquin Equation sometimes characterizes the S-N curve in the high cycle region.

$$N\sigma_a^p = C \tag{16}$$

Where σ_a is the stress amplitude and p and N are empirical constants.

1.3.2 Fatigue Testing

Unlike some areas of mechanical testing, many testing devices and specimen designs have been developed for fatigue testing. Fatigue testing machines are defined by several classifications.

1. The controlled test parameters, i.e. the load, deflection, strain, twist, torque etc.
2. The design characteristics of the machine used to conduct the test

3. Operating characteristic of the machine i.e. electromechanical servohydraulic, electromagnetic etc.

Testing machines may be universal-type machines that are capable of conducting several types of loading depending on the fixture used. All fatigue-testing machines consist of the same basic components. They have a **load strain**, which consists of the load frame, gripping devices, test specimen, and loading system. The **control system** is another basic component, which initiate and maintain the controlled test parameter. They also terminate the test at a predefined status, i.e. failure, load drop, extension, or deflection limit. The control of time varying deflection or displacement can be obtained in mechanical systems by cam or hydraulically through a piston limited by stops.

1.3.3 Servohydraulic Systems

Servo hydraulic systems offer optimum control, monitoring, and versatility in fatigue testing. These can be obtained because component systems can be upgraded as required. A hydraulic actuator typically is used to apply the load in axial fatigue testing. The axial fatigue-testing machine subjects the specimen to a uniform stress or strain through its cross section. Such machine can be used for both high-cycle and low-cycle fatigue testing. A wide variety of grips, particularly self-aligning types, are available for those machines. Figure 10 shows a variety of grip designs that can be used for axial fatigue testing.

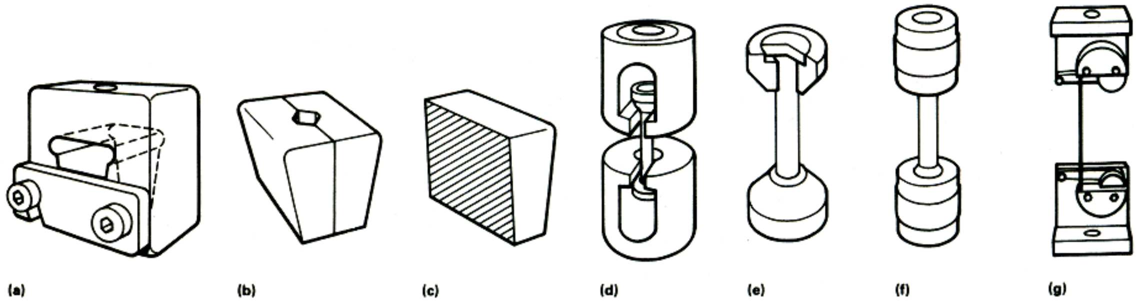

Figure10: Grip designs for axial fatigue testing

(a) Standard grip body for wedge-type grips. (b) V-grips for rounds for use in standard grip body. (c) Flat grips for specimens for use in standard grip body. (d) Universal open-front holders. (e) Adapters for special samples (screws, bolts, studs…) for use with universal open-front holders. (f) Holders for threaded samples. (g) Snubbed-type wire grips for flexible wire or cable.

Servohydraulic fatigue machines are particularly well suited for providing the control capabilities required for fatigue testing. Extreme demands for sensitivity, resolution, stability and reliability are impose by fatigue evaluations. Displacements may have to be controlled for days to within a few microns and forces can range from 100 kN to just a few Newtons. This wide range of performance can be obtained with servomechanism in general and in particular, with the modular concept of servohydraulic systems.

Many commercially manufactured units are available for each component in a typical servohydraulic-testing machine. Moreover, complete systems in which all of the components are properly integrated and specifically designed to meet a particular testing specifications are available. So it is good for people involved with the selection and use of servo systems to know the basic functions of each component. Figure 11 is a block diagram of the components and how they are put together.

1. **The Programmer** supplies the command signal to the system, which is generally an analog of the desired behavior of the controlled parameter.
2. **The Servo-Controller** makes most of the adjustments necessary to optimize the performance of the system.
3. **The Servo-Valve** controls the volume and direction of flow of hydraulic fluid between the hydraulic power supply and the hydraulic ram.
4. **Hydraulic rams** or actuators furnish the forces and displacements required by the testing system.
5. **The Load Cell.** The strain gage load cell is the most widely used force-measuring and feedback device in fatigue machines.
6. **Load Frame.** In a fatigue machine, the load frame supplies the reaction forces to the specimen and to the housing of the ram.
7. **Specimens.** The specimen is part of the servo-loop, and its requirements of force and deflection affect total system performance. So, its design should be such that all unnecessary elastic deflections are eliminated.

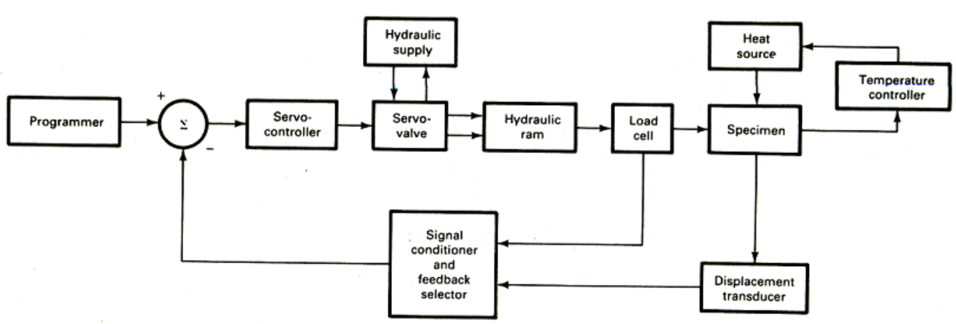

Figure 11: Block diagram of components of a servohydraulic testing system

1.3.4 Controls

All the major components can be controlled by either, the load unit, cross head lift, and or pressure supply controllers. Figure 12 shows the different components of the MTS machine. The load unit controls the vertical position of the lower wedge grip. The bottom wedge grip has about a six inches stroke. The crosshead lift controls the vertical position of the top wedge grip. To allow for extra clearance, the bolts would have to be loosened prior to hydraulically raising or lowering the crosshead. The pressure supply controls the lateral positions of the wedges, which are inside the wedge grips.

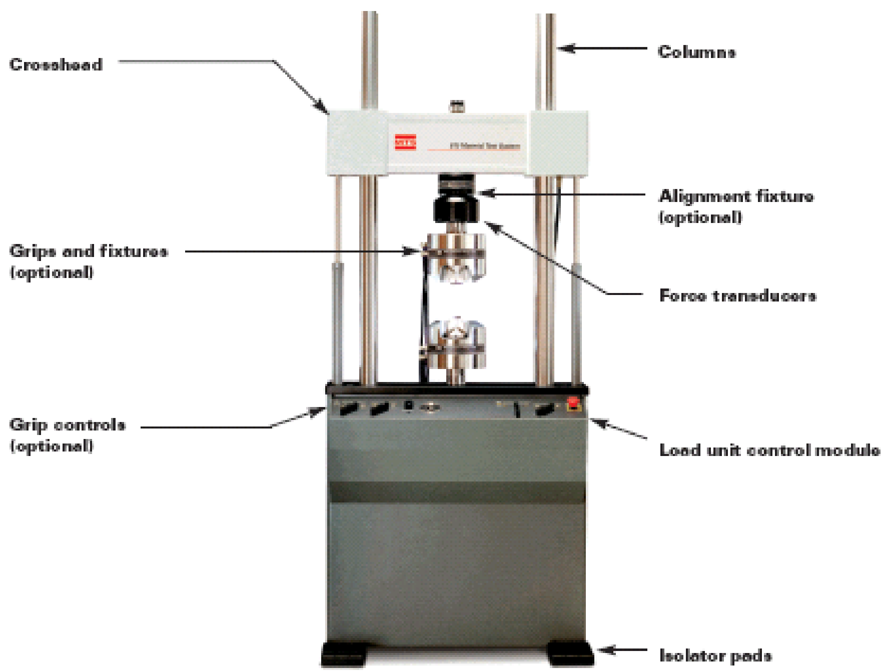

Figure 12: MTS servohydraulic machine and its components.

2.0 Project Planning

2.1 WBS

WBS stands for work breakdown schedule. The main purpose of this type of schedule is to breakdown the various categories needed to finish the project. What is needed in order to accomplish each of the categories should be listed directly bellow them. This type of schedule helps to list out the major tasks that need to be completed. The WBS that was prepared and used for this project was broken up into 7 major categories listed below:

1. Gather requirements
2. Background research
3. Concept generation
4. Interface with MTS servohydraulic machine
5. Determining ways of measurement
6. Design of tests
7. Machining / manufacturing

The actual WBS that was used can be found within Appendix A.

2.2 Schedule

A schedule with major milestones, deliverable due dates, and with proposed meeting times was put together using Microsoft Projects. This schedule extends through until the end of the spring 2005 semester. The finalized schedule with all the details can be found in Appendix B.

2.3 Project Procedures

The purpose for the project procedures document, which can be found in Appendix C, is to list out all of the things that will be utilized in order to accomplish all of the tasks needed to finish the project. These might include rules for meetings, contact information, task delegation, and technical issues. The sections within the project procedures document are broken down into as much detail as to not leave any confusion as to how things will get accomplished.

3.0 Design Specifications

3.1 Customer Needs

As previously stated in section 1.0, Cummins is in need of a test fixture to test two different threaded bolt joints within one of their diesel engines. The sections in question will be isolated for testing. A list of the customer demands along with a description and a ranking of importance (10 being the highest on a scale of 1 to 10) can be seen in Table 1 below.

Table 1: Demand List from Sponsor and Importance

Demands	Requirement	Importance
Demand	Two different set ups, one that will test the main bearing cap and another that will test the connecting rod joints.	10
Demand	Test fixtures should reproduce the failure mode of real components	10
Demand	Design will interface with a fatigue test rig, (e.g. MTS Servohydraulic machine)	10
Demand	Load on the main bearing Cap be 45,000 lbs	10
Demand	Load on the connecting rod be 16,000 lbs	10
Demand	Bolt boss diameter, thread pitch, thread type, counterbore depth, number of engaged threads, bolt preload, and alternating load should be studied	10
Demand	Estimated cost of Hardware (samples will be provided)	8

Along with the two fixtures, Cummins needs certain elements of the actual threaded bolts to be studied. This includes bolt boss diameter, thread pitch, thread type, counterbore depth, number of engaged threads, bolt pre-load, and alternating loads.

3.2 Specifications

Cummins wants the results obtained to reproduce the way in which the real components fail or have been failing. In order to do this the test will be carried out on the MTS servohydraulic machine located within the National High Magnetic Field Laboratory in Tallahassee, Florida. This machine will be able to archive various high loads in both compression and tension that will be needed to run the fatigue tests.

Cummins has requested tat there be a pre-load amount per cap screw of 16,000 lbs for the connecting rod and 45,000lb for the main bearing cap. Along with the load requirements, Cummins would like the dimensions of the final fixtures sent to them. In order to do this the actual threaded bolts where sent and their dimensions where taken.

To determine the torque that would be applied to each bolt in order to achieve the required pre-load, the equation below was used:

$$T = k \cdot d \cdot F \tag{17}$$

Where T is the torque applied, k is the torque constant (specific to diameter of bolt), d is the diameter and F is the pre-load value specified by Cummins. The calculations can be found in Appendix D. The final torque amounts calculated can be seen in Table 2 below.

Table 2: Torque Values Needed for Pre-load Requirements.

	Connecting Rod	**Main Bearing Cap**
Pre-load, T (lbs)	16,000	45,000
Diameter, d (mm)	12	18
Torque constant, k	0.21	0.21
Torque, T (ft-lb)	**132**	**533**

4.0 Design Generations and Selection

4.1 Design Generations

There are 10 concepts that where generated, 5 concepts for the main bearing cap and 5 for the connecting rod. Brief descriptions along with a Pro-E drawing of the concepts are shown within this section.

4.1.1 Concepts Generated for the Main Bearing Cap

During the first half of the year, five separate concepts were made for the main bearing cap fixture, seen below in Figure 13A-E. After the final selection was made, seen later in section 4.0, it was found that some modifications had to be made. This resulted in the fixture seen in Figure 13F. Further discussion on the modification made during the spring semester can be found in section 4.3.

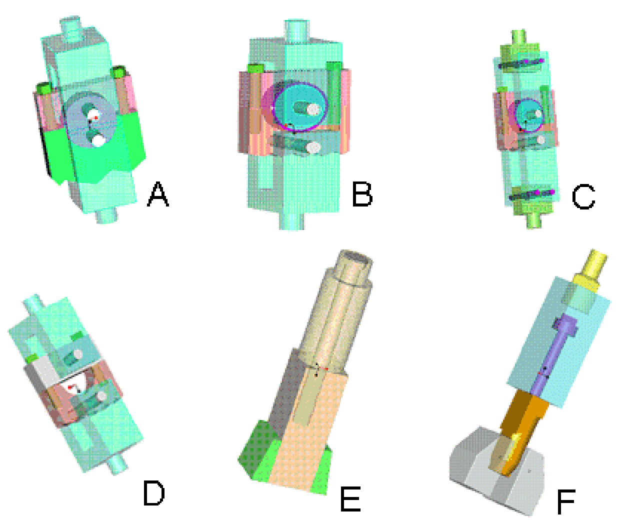

Figure 13: Concept generations for Main Bearing Cap. Brief descriptions of each of the concepts are listed below. Concept F is discussed further in section 4.3.

Concept A: For this design, a section of the engine block has been cut out. A steel disk will be mounted into the crankshafts position. The disk will be cut along its diameter into two pieces. Two holes will then be drilled into the upper and lower halves of the disk. A pin will connect the clevis to the main bearing assembly through the holes drilled.

Concept B: Design B is exactly the same as design A except for the following: one hole will be drilled threw the center of the disk. A second hole will be drilled threw the cut out section of the engine block.

Concept C: This design is exactly the same as design B. The difference is that it will have a three-piece clevis, which is shown in Figure 13 below. The advantages of having a three-piece clevis are that it would be easier to machine an in turn cheaper than the above concepts.

Concept D: A rectangular piece of steel will replace the main bearing cap. Two holes will be drilled into the top of the steel where the main bearing bolts would go. Another hole will be drilled threw the center of the side which will mount to the clevis. A second hole will be drilled threw the cut out section of the engine block. The last hole will be drilled threw the cut out section of the engine block.

Concept E: For this design, a smaller cut out section of the engine block and main bearing cap will be used. The green pieces at the bottom are very slim wedges. The wedges will have to be specially made. The dark brown piece at the bottom is the cut out section of the engine block. This part must be machine/cut into a smooth square or rectangular shape. The light brown part at the top is a cylinder shape piece of steel with threads tapped into the upper outer end. The wedges fit directly into the bottom wedge gripping apparatus of the MTS machine. And the light brown part threads directly into the upper arm of the MTS machine.

Concept F: Similar idea to that of concept E. Adapter was altered to better fit into the MTS machine. Further discussion on changes found in section 4.3.

4.1.2 Concepts Generated for the Connecting Rod

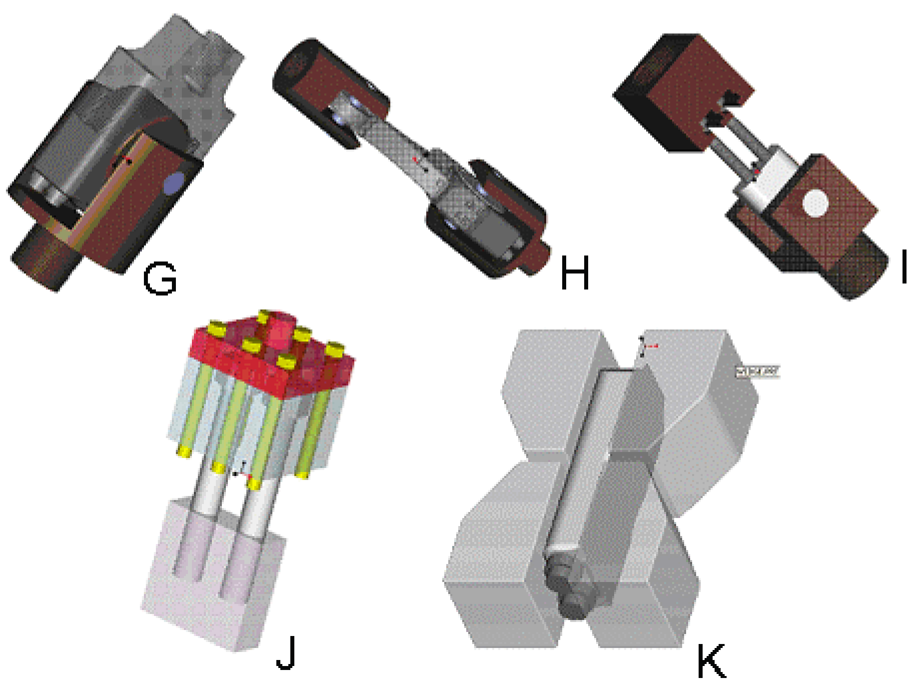

Figure 14: Concept generations for connecting rod. Brief descriptions of each of the concepts are listed below.

Concept G: This concept is for the connecting rod as well. The connecting rod is cut up on this design and a threaded grove is cut into it. This grove will be screwed onto one end of the MTS machine. For this setup all four bolts will be tested along with the cap and to minimize the size of the clevis, a bearing will be used. A pin will go through the bearing and the clevis. The extruded part of the clevis will then be mounted onto the other end of the machine. This clevis was made circular because the engineer that runs the lab told us that it is more economical.

Concept H: This setup tests the whole connecting rod. Again, bearings are put on both holes to minimize the size of the clevises. All four bolts will be tested along with the cap and the connecting rod. The clevises are round because they're more economical.

Concept I: For this design, the bolt boss is cut out of the connecting rod and since the cap is not being tested, it is discarded. The bolts head are tight fitted into groves on the clevis and the clevis has a thread grove that will fit into the MTS servohydraulic machine. On the other end, where the cut out boss is, a hole is place at the center of the piece and a pin is inserted through it and through the other clevis. This clevis has an extruded threaded part that will be mounted on the other end of the MTS machine.

Concept J: The purple piece at the extreme bottom is the connecting rod that has been cut into a rectangular sample. This piece will sit directly into the lower are of the MTS machines wedge grip apparatus. The red piece is the upper portion of the clevis that will have a square shaped base with a cylinder protruding up, with threads on it. The threaded part will thread directly into the upper arm of the MTS machine. The base will have 6 holes drilled threw it which will connect the upper and lower parts of the clevis. The light blue piece is the lower portion of the clevis and will also have a square shape with 6 hole drilled threw its base, just like the upper clevis. However, two more holes will be drill into the lower clevis that will house the connecting rod bolts.

Concept K: In this design concept the threaded joints are cut out on each side from the entire connecting rod. This allows for one-half of the load that the entire connecting rod normally sees, also two samples can be taken out of one connecting rod. The individual halves are fixed into the MTS grips by wedges that are placed on each side and at the top and bottom.

4.2 Factors for Selection Process

There are a total of 10 concepts that were generated. In order to narrow the concepts down to the final two various factors were evaluated for each of the concepts. These factors where then weighted according to their importance of the final design selection.

The first of the factors is **cost**. It would be ideal to have limitless money in order to make the best fixture possible, but that is not the case nor is it realistic. The fixture needs to be affordable and able to be reproduced at a fair price. For that reason cost is weighted at a 20% (out of 100%).

The manufacturability of the fixture is dependent on cost. If a fixture is hard to manufacture then it will in turn cost more. This is why manufacturability is the next factor. It is weighted at 15%.

The fixture should be reliable and not break down before the bolt reaches it's fatigue limit. As long as the correct dimension calculations are made and the proper material(s) are chosen then the fixture should be rather reliable. This is why **reliability** got a weighted value of 15% as well.

The fixture should also perform the task at hand without any problems. The joints should be able to fit within the fixture as demonstrated in the concept generations and should not alter the actual joints themselves. The fixture should also perform the test in the manner in which Cummins has asked for. Performance becomes a factor and is weighted at 20%.

The last and most heavily weighted **factor is customer satisfaction**. One of the ultimate goals of this project is to make sure that Cummins is happy with the final design so they can use it for their testing. This category houses a little bit of all the categories stated above and is weighted at 30%.

4.3 Design Selection

A design matrix was utilized in order to make the proper selection. Each of the concepts described above were evaluated according the factors described above. Each concept is ranked from 1-10. (1 being the worst, and 10 being the best) The ranking is then multiplied by the weighted values below and then added up to show the final total. The concept with the highest total will show what the final design concept should be. Table 3 below shows the final outcome of the design matrix.

Table 3: Design selection matrix

Factors	Cost	Man.	Rel.	Perf.	Cust. Sat.	Total
Weight	20%	15%	15%	20%	30%	
Concept						
A	0.8	0.75	1.05	0.04	2.1	4.74
B	0.8	0.75	1.05	1.4	2.1	6.1
C	1	0.9	1.05	1.4	2.1	6.45
D	0.8	0.6	1.05	1.4	1.5	5.35
E	1.6	1.35	0.75	1.4	2.1	**7.2**
F	1.6	1.35	0.75	1.4	2.1	**7.2**
G	1	0.6	0.6	1.4	0.6	4.2
H	0.8	0.45	1.2	0.6	2.1	5.15
I	1	1.2	0.75	1.6	2.1	6.65
J	1.2	1.2	0.75	1.4	2.1	6.65
K	1.8	1.05	0.75	1.4	2.1	**7.1**

The three cells shaded in and in bold in the design matrix above show the concepts that could be considered as the final fixtures. Concept E is for the main bearing cap and concept K is for the connecting rod. The majority of the fixtures all seem to have similar values for that of performance and reliability. The biggest discrepancies could be seen in the rankings for cost. Some of the fixtures are large and require both a lot of machining and money to make. Both concept E and K appear to be the fixtures that will get the job done as well as being cost efficient. Concept F is the concept (E) that was altered after the initial decisions were made. Concept F is the final concept that was decided on for the connecting rod. A better view of the final design selection with a description can be found in Appendix E.

4.4 Spring Modifications

4.4.1 Modifications for Main Bearing Cap

The initial fall design seen in Figure 13E was chosen as the final design for the main bearing cap test. However, it had to be modified because the bolt head was too big and would not fit into the MTS Block. An adapter was then used in order to connect the fixture to the MTS block. The modified set up is shown in Figure 13F. New engineering drawings had to be made and they are included in Appendix F. The engineering drawing for the MTS block is also available in Appendix F. The one used for testing was provided by the NHMFL.

The engine block sample from Cummins also had to be modified in order to fit the wedges we designed. The engineering drawings for this are in Appendix F. The connecting rod resizing modifications for testing are also included in Appendix F.

4.4.2 Material Selection Modifications

At the end of the fall semester steel 4340 was chosen as the material in which the fixtures and wedges would be made out of. Holmes Machine Shop informed us that it would take much longer to use that material as well as being more expensive. After some deliberation, steel 4140 was chosen as the new final material selection. Steel 4140 has a hardness of 311HB, which is harder than the samples sent by Cummins. Using steel 4140 would also save us time and money because Holmes Machine Shop would provide it.

4.4.3 Wedge Finish Modifications

It was fist suggested by Bob Walsh at the NHMFL to use a double-cut file grip finish on the wedges. This finish can be seen below in Figure 15. Once at Holmes Machine Shop it was seen that having this type of finish would be both money and time inefficient. Holmes Machine Shop provided us with an alternative pyramid finish, seen in Figure 16. This finish was sufficient for our testing needs.

Figure 15: Double-Cut File Grip Finish

Figure 16: Pyramid Finish

4.5 Cost Analysis

Table 4: Cost Analysis for machining at Holmes Machine Shop.

QTY	Description	Unit Price	Total Price
4	Wedges w/pins, 4140 HT steel	$250	$1,000
1	Modify connecting rod	$200	$200
1	Cylinder for fixture	$125	$125
1	Square housing for fixture	$300	$300

Total: $1,625.00

The cost analysis given to us by Holmes Machine Shop can be seen above (Table 4). The final price to get everything machined along with sample modifications totaled to $1,625.

The engine block parts that needed modifying were all machined at the FAMU/FSU College of Engineering machine shop at no cost.

5.0 Final Design and Machining

5.1 Final Design

The design drawings for the final two fixtures where put into solid works, where engineering drawings were made. These drawings and calculations for the wedges can be found in Appendix F and G respectively. The final dimensions of the fixtures are included. These engineering drawings will then be sent to the Holmes Machine Shop, where they will manufacture the fixtures so that testing can begin. The material of steel 4140 was chosen for the final fixtures, which will be provided by Holmes Machine Shop and is included within the final price.

5.2 Machining

As mentioned above, the fixtures were sent to Holmes Machine Shop in Bonifay, Florida where they were machined to what was specified. The actual specimens had to be cut down and altered so as to ensure a fit into the MTS machine. Holmes Machine Shop machined one of the main bearing cap samples down to fit the MTS. The FSU/FAMU College of engineering machine shop machined the two connecting rods and one other engine block sample.

5.2.1 Connecting Rod Machining: Final Product

The final wedges can be seen below in Figure 17. A total of four wedges were made, only two are shown below. Figure 18 shows the final connecting rod sample that was machined down in order to fit within the MTS machine.

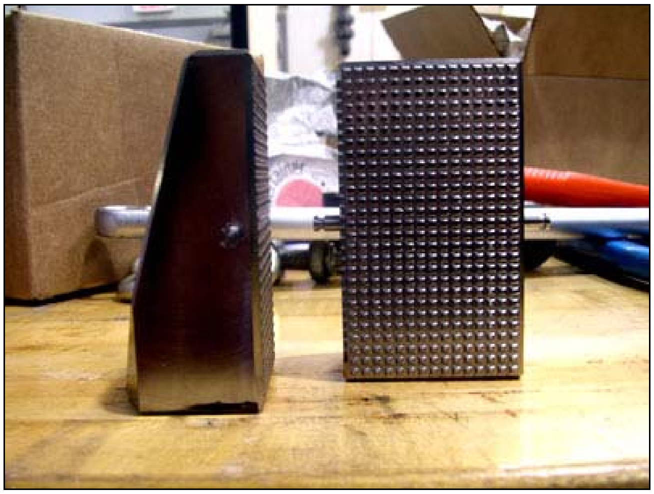

Figure 16: Two of four wedges machined by Holmes Machine Shop.

Figure 17: One of two connecting rod samples ready for testing.

5.2.2 Main Bearing Cap Machining: Final Product

Figure 19: Main bearing cap fixture

Figure 20: MTS adapter for main bearing cap

Figure 21: Engine block samples ready for testing, within MTS.

6.0 Testing

6.1 Preparing The Samples

Prior to testing, the bolts had to be pre-loaded by setting the torque to the amounts discussed in section 3.2: 130 ft-lbs for the connecting rod and 533 ft-lbs for the main bearing cap. There was a torque wrench capable of torquing the connecting rod bolts to the appropriate amount within the testing lab at the NHMFL. For the main bearing cap bolts however, an outside source was looked into because a bigger torque wrench was needed. Florida Rock had a wrench available that was capable of applying over 533 ft-lbs torque to the bolt. The bolt was only torqued to 330 ft-lbs. This was due to the fact that a higher torque was too much of a human strain and increased the risk of our limited samples being damaged.

Figure 22: Florida rock applying the 330ft-lb torque to the main bearing cap bolt.

6.2 Test Plan

Before any testing was conducted a test plan was put together. This includes the procedures needed to carry out the test, safety precautions, and some brief background information on the testing apparatuses. The test plan can be found in Appendix H.

6.3 Tests Setups

A total of four tests were done. Two tests were done for the connecting rod and two for the main bearing cap. The testing was set up in the manner seen below in Table 5. All the data was acquired by computer software integrated with the MTS in the testing lab.

Table 5: Test Requirements

	Load Amplitude (lbs)	**Compression**	**Tension**
Connecting Rod #1	16,000	X	X
Connecting Rod #2	16,000 24,000*		X
Main Bearing Cap #1	45,000	X	X
Main Bearing Cap #2	22,500		X

* The initial load of 16,000 lbs was increased to 24,000 lbs after 210,000 cycles.

7.0 Results

7.1 Results From Spring Testing

One of the objectives was to design fixtures to test the fatigue life of threaded joints. Two joints in question are that of the connecting rod cap-to-connecting rod and that of the main bearing cap-to-engine block. A total of four tests were performed and some very interesting results were found. The following table shows the amount of cycles as to when the testing was stopped, either because of time or failure.

Table 6: Results Table with Number of Cycles Conducted

	Number of Cycles	**Failure**
Connecting Rod #1	1.3 million	No
Connecting Rod #2	980,389	No – sample Yes – wedge (1)
Main Bearing Cap #1	1,000	Yes
Main Bearing Cap #2	10,000	Yes

The first test was done on the connecting rod sample labeled REBD, shown in Figure 22. The two bolts that camped this sample together were both torqued to 65 ft-lb. A tension load of 8000 lbs was applied to the sample then released. Then a compression load of 8000 lbs was applied to the sample then released. This fatigue cycle went on for approximately one thousand cycles, until the test sampled slipped out of the wedges. After this happened the force exerted by the wedges was increased from 3000 psi to 6000 psi. Then the testing restarted and the wedges held up until approximately 1.3 million cycles at which time the sample slipped out again. Therefore, do to the continuing slippage of the test sample the testing was ended.

Figure 22: First connecting rod sample, REBD

The second connecting rod labeled JB shown in Figure 23 was tested a little differently. The bolts on the sample were torqued to 130 ft-lb, and the testing was done in tension only. Initially a 16,000 lb tension load was applied to the test sample, but at approximately 200,000 cycles the applied load was increase to 24,000 lb. Then at approximately 1 million cycles, one of the four wedges failed. The gripping pattern on the face of one of the wedges was stripped away, for that reason the second test was ended. The pattern failed because the finish on the face was not sharp and hard enough to penetrate the sample. Some recommendations are listed below as to how this problem can be fixed.

Figure 23: Second connecting rod sample, JB

The third test was performed on the main bearing cap-to-engine block joint. The bolt was torqued to 300 ft-lb. A 22,500 lb tension load was applied then release. Then a 22,500 lb compression load was applied then released. The fatigue cycle went on for approximately 1,000 cycles, when suddenly the engine block sample failed. Therefore, the third test was ended.

The fourth tests were also done on the main bearing cap joint. The bolt in this test was torqued to 330 ft lb. This test was performed in tension only and the applied load was 22,500 lbs. This test ran for 10,000 cycles at which time the engine block sampled failed. Both the main bearing cap setups failed in the same manner; Figure 24 below shows the failed engine block sample.

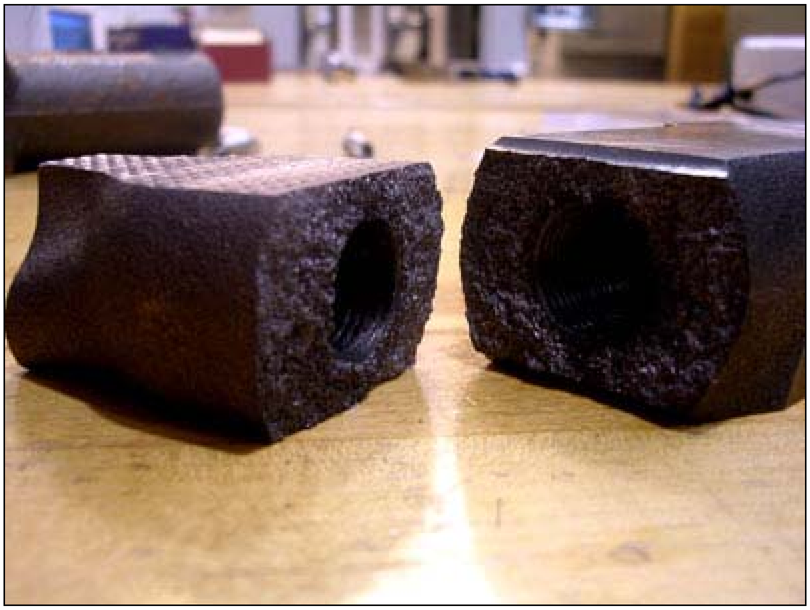

Figure 24: Main bearing cap test, engine block failure.

The objective to test the fatigue life of threaded joints had been completed. Based on the tests performed and the data collected, there was no deformation in the threaded joints of the connecting rod or main bearing cap threaded joints. However, more thorough tests are recommended in order to get more detailed results.

Both of the main bearing cap specimens had to be shaved down in a way that introduced some localized stressed concentration along the sides of the sample. This is the location where both of the samples failed. The area where the specimens failed was not the anticipated area of failure. Due to the time constraint, no further tests was carried out but below are some recommendations pointing out ways to improve the designs.

7.2 Future Improvements

There are various improvements to the designs that can be done in order to run better tests. The first would be to make the wedges of stronger steel. This was not feasible in our situation due to money and time, but should be done in the future. Two materials to take into consideration for the wedges are high carbon steels and tool steel. The harder the material is, the better grip the wedges will have on the specimen. This is because the wedges will actually indent themselves into the specimen, rather than just clamping down on them.

Another improvement would be to make the wedges a bit narrower. This will allow larger engine block samples to be tested. The increase in the sample size will help reduce the likelihood of the engine block failing rather than the bolts. To allow for even better grip, the diamond cut finish is recommended for the wedges. Also, the gripping faces on the engine block specimen should be made flat in order to reduce localized stress concentration.

8.0 Conclusion

The expected results were obtained for the most part. The anticipated cycles per test was set to about 500,000 cycles and the tests for the connecting rod went for much longer. Due to money constraints only four sets of wedges were manufactured and six were needed. The design for the main bearing cap bolt had to be modified over and over again in order to get the samples to fit within those wedges. Those extra modifications led to the failure of the sample in an unexpected area. Instead of the bolts failing, the engine block sample failed because it was shaved to fit within the wedges. The sample failed due to stress concentration area that was introduced to it. If money was not an issue, the wedges for the main bearing cap set up would have been designed differently. They would have been slimmer and made of a harder material. Time was also working against us so we had to settle for a lower grade steel for the wedges and a less intricate finish. With the harder wedges, the normal applied pressure would not have to be as high because the wedges would have indented the specimens. This is why we recommend using harder material for the wedges.

The setups would have worked if time and money allowed us to make the recommended modifications. 500,000 cycles would not have been enough to cause failure in the bolts. The tests for the connecting rod samples ran for about a million cycles and did not cause the bolts to fail. The main bearing cap bolts did not fail either. To better simulate the failure mode of the real component, the tests would have to run for much longer period of time and the setup would have to be modified accordingly. That way, better results would have been obtained.

9.0 Acknowledgements

We would like to take the time to thank the following people and companies for their support and help on the project:

- Cummins
    - Dave Parsons
    - Bob Tickle
- Dr. Luongo
- Dr. Kalu
- NHMFL
    - Bob Walsh
    - Chika Okoro
- Holmes Machine Shop
- Florida Rock

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Callister, William D. Jr. Material Science and Engineering: An Introduction. New York: John Wiley & Sons Inc, 2003.

Kalu, Peter. Fatigue of Metals. PowerPoint Presentation, Fall 2003.

Norton, Robert L. Machine Design: An Integrated Approach. Upper Saddle River, NJ: Prentice Hall, 2000.

http://www.sv.vt.edu/classes/MSE2094_NoteBook/97ClassProj/anal/kelly/fatigue.html

Appendix A

The following chart is the actual **WBS** that was developed during the fall 2005 semester.

Final Product: Bolted Joint Rig Test Development

- **1.0 Gather Requirements**
- **2.0 Background Research**
  - 2.1 Standards of testing
  - 2.2 Availability of equipment
- **3.0 Concept Generation**
  - 2.1 Infrared Hydrocarbon Flame Detectors
  - 3.2 Receive critique on initial design
  - 3.3 Take feedback and incorporate into design
- **4.0 Inter-face with MTS Servohydraulic Machine**
  - 4.1 Dimensions and orientations of samples to be tested
  - 4.2 Learn programming of MTS machine
  - 4.3 Design wedges/grips for MTS
- **5.0 Determining Ways of Measurement**
  - 5.1 Gauges
  - 5.2 Bolt boss diameter
  - 5.3 Thread pitch
  - 5.4 Counter bore depth
  - 5.6 Bolt pre-load
  - 6.1 Equate actual conditions to lab testing
  - 6.2 Program/database for collecting data
- **7.0 Machining/Manufacturing**
  - 7.1 Integrate samples (bolts, engine blocks) with wedges/grips for MTS

Appendix B

The following pages within Appendix B contain the schedule developed in the fall 2005 semester. The schedule includes all of the major milestone dates included within section 2.2, as well as any other important due dates. The schedule is a work in progress and dates are subject to change as different obstacles arise.

Appendix C

The following contains the project procedure developed during the fall 2005 semester. It goes into great detail as to how work will be divided, contact information, hardware and software to be used, things to research, and how the different tasks will be divided amongst the group.

- **Contact Information**

Group Members	Email:	Phone Number
Alex Duge	avd4555@fsu.edu	305 725 3951
Ana Erb	ame5918@fsu.edu	786 229 5183
Ronald Rolle	rlr03c@fsu.edu	850 513 0931
Cedric White	crw5759@fsu.edu	850 445 7699
Cummins Liaisons		
Bob Tickel	bob.g.tickle@cummins.com	812 377 5363
Dave Parsons	dave.h.parsons@cummins.com	812 377 5645
Grad Student Help		
Jeffrey Cooper	jic0558@fsu.edu	-
NHMFL Help		
Chika Okoro	mokoro@magnet.fsu.edu	-

- **Keeping In Contact**
  - Cedric White is in charge of contacting the Cummins liaisons and setting up conference calls.
  - All four-group members will be present during conference calls with Cummins liaisons.
  - Conference calls are to be held bi-weekly or when needed.
  - The main mode of contact between the group members will be email and phone calls to set up meetings; meetings will be held in person.
- **Keeping Organized**
  - Every group member keeps their own lab notebook
    - Work/research they have done
    - Notes taken at meetings
    - Individual task lists
    - Causes each member in charge of their tasks
  - Web site up and running early on in the project
    - Main use as a storage place for files to be easily found by group members
    - Updated often keeping members up to date
    - Under construction/being modified up until end project
  - File naming / standards for deliverables
    - File_name_mmddyy.(extension)

        - Footnotes
            - File_name_mmddyy
            - On bottom of every page
- **Work Division**
    - Each member will present one of the 10-minute presentations. To be decided before each presentation.
    - Each group member will take his/her turn in physically writing the deliverable (once discussed within group).
    - Every deliverable is discussed ahead of time, not left up to one person to decide.
    - Cedric White is in charge of keeping contact with Cummins.
    - Ana Erb is in charge of keeping in contact with NHMFL members whom have agreed to help us with mechanical testing.
    - All other aspects are decided as they come along.
    - All tasks are discussed in a group previous to completion.
    - Research tasks will be discussed and divided evenly; deadline and meeting time will be set up to discuss findings.
- **Research Needed Prior to Designing**
    - ASM &/or ASTM standards for fatigue testing on MTS
        - Specifically for threaded bolt sample
    - Grade A 10.9 cast iron
        - Material properties
        - Machine-ability
    - Engine block
        - Set up
        - Where the parts are in question
            - Set up meeting at auto store.
        - Diagrams sent by Cummins
    - Definitions and dimensions (dimensions can be figured out after preliminary designs)
        - Bolt boss diameter
        - Thread pitch
        - Counterbore depth
        - Number of engaged threads
        - Bolt pre-load
        - Alternating load
        - Dimensions of final fixture allowed within MTS machine
- **Software Needed**
    - Pro-E
        - Final design
        - Final design engineering drawings
    - MathCAD for any needed calculations
    - MS Office
        - Report/deliverables (MS Word)
        - Scheduling (MS Projects)
        - Presentations (MS PowerPoint)

        - Charts/data analysis (MS Excel)
    - HTML editor for web site
- **Sample(s) Needed For Testing**
    - Cummins will provide samples
        - Find out number of samples that will be provided
        - Figure out room for error with given number of samples
    - Research as to where Cummins gets their material(s)
        - Research costs of ordering extra samples (in case of it being needed)
- **Hardware Needed**
    - MTS machine
        - Used to run the fatigue tests after rig is built.
        - Contact: Chika Okoro.
        - Need training on MTS machine.
    - Machine shop elements (if applicable after material research)
        - Need to machine cast iron samples for the fixture.
            - Not sure if parts can be machined in machine shop here at the COE.
            - If so, need to schedule time & discuss costs with machine shop ASAP.
        - Need to fit together rig/fixture.
- **Outside Machining Help**
    - If cast iron samples cannot be machined within COE
    - Research
        - Where it can be machined
        - Cost

Appendix D

Connecting Rod

$$d := \begin{pmatrix} 11.81 \\ 11.76 \\ 11.75 \\ 11.79 \\ 11.76 \\ 11.76 \\ 11.71 \end{pmatrix} mm \qquad d_{avg} := \frac{\sum_{i=0}^{6} d_i}{7} \qquad d_{avg} = 11.763mm$$

Page 901. Table 14-7, Class Number 12.9,then its minimum Proof Strength is 970 MPa

$$S_p := 970000000Pa$$

$$\frac{92.07 - x}{92.07 - 61.20} = \frac{12 - 11.763}{12 - 10} \text{ solve}, x \rightarrow 88.411905000000000001$$

Page 882. Table 14-2, major diameter 11.763 mm (by interpolation), then its Tensile Stress Area is 88.41 mm^2.

$$A_t := 88.41mm^2$$

Preload tension $F_i := 16000lbf$

Tightening Torque

$T_i := .21 \cdot F_i \cdot d_{avg}$ $T_i = 175.808N \cdot m$ $T_i = 129.669ft \cdot lbf$ Torque force on one bolt

Bearing Cap

$$d := \begin{pmatrix} 17.81 \\ 17.85 \\ 17.92 \\ 17.88 \\ 17.85 \\ 17.83 \\ 17.79 \\ 17.82 \\ 17.84 \\ 17.80 \end{pmatrix} mm \qquad d_{avg} := \frac{\sum_{i=0}^{9} d_i}{10} \qquad d_{avg} = 17.839 mm$$

Page 900. Table 14-7, Grade Number 8, then its minimum Proof Strength is 120 kpsi

$$S_{p1} := 120000 psi$$

$$\frac{216.23 - x}{216.23 - 167.25} = \frac{18 - 17.839}{18 - 16} \text{ solve}, x \rightarrow 212.28711000000000000$$

Page 882. Table 14-1, major diameter 17.839mm (by interpolation), then its Tensile Stress Area is 212.29 mm^2.

$$A_t := 212.29 mm^2$$

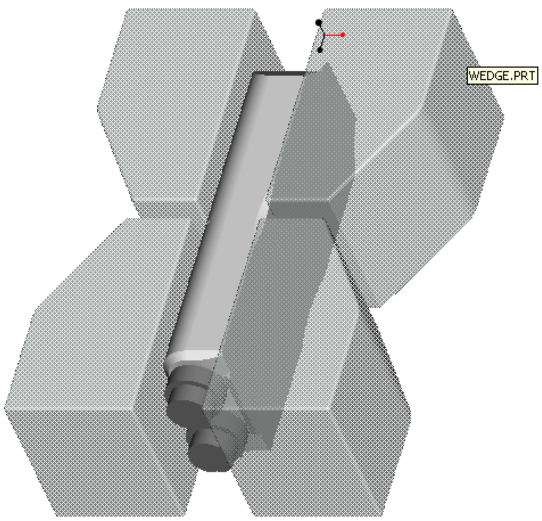
if preload is 45000lbf

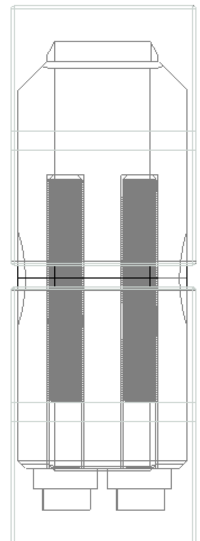
$F_i := 45000 lbf \quad T_i := .21 \cdot F_i \cdot d_{avg} \quad T_i = 749.875 N \cdot m \quad T_i = 553.079 ft \cdot lbf$ load across one bolt

Torque neccessary for torquing the connecting rod bolts and main bearing cap bolts are 129.67 ft-lbf and 553.08 ft-lbf respectively.

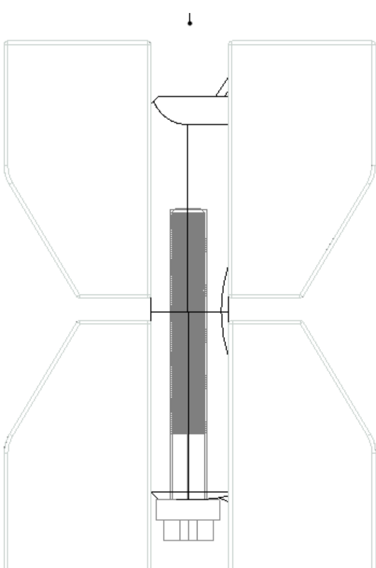
Appendix E

Connecting Rod Final Design:



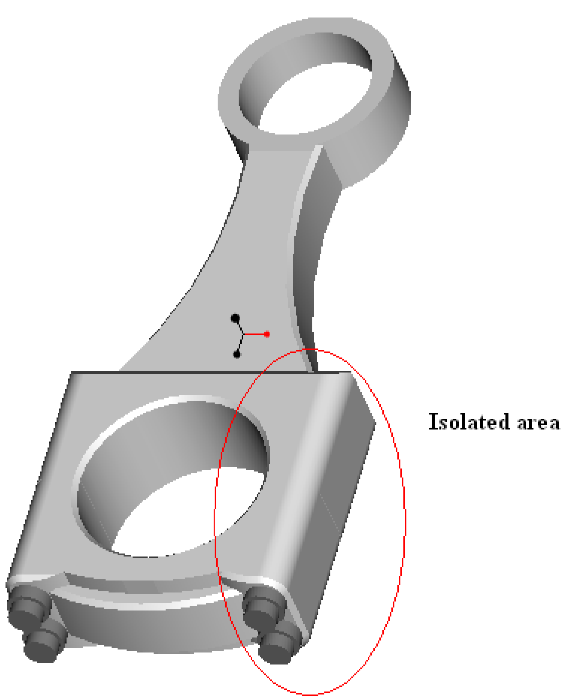


Side View

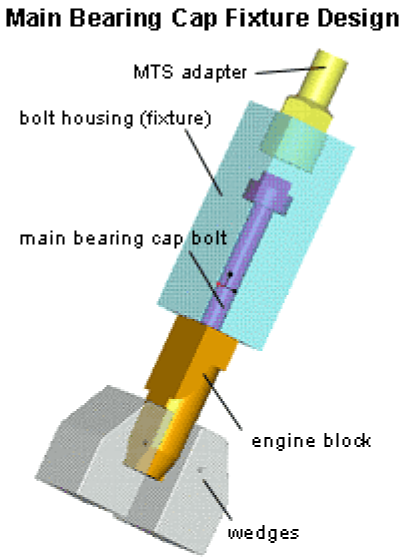


In this design concept the threaded joints are cut out on each side from the entire connecting rod. This allows for one-half of the load that the entire connecting rod normally sees, also two samples can be taken out of one connecting rod. The individual halves are fixed into the MTS grips by wedges that are placed on each side and at the top and bottom.

Front View

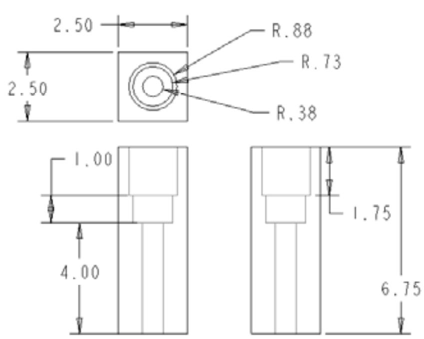


Main Bearing Cap Final Design:

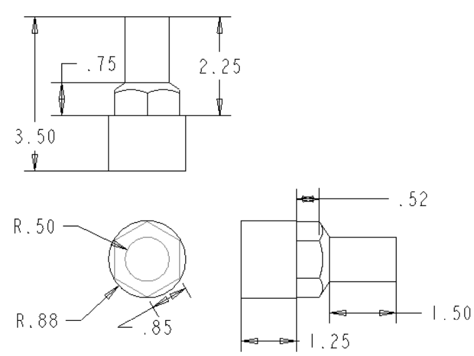


The adapter would be placed into the MTS block. That block would be installed into the MTS wedge grips. The bolt is torques into the engine block, within the fixture. The fixture will simulate the main bearing cap and provide similar tension and compression that would be found in the engine.

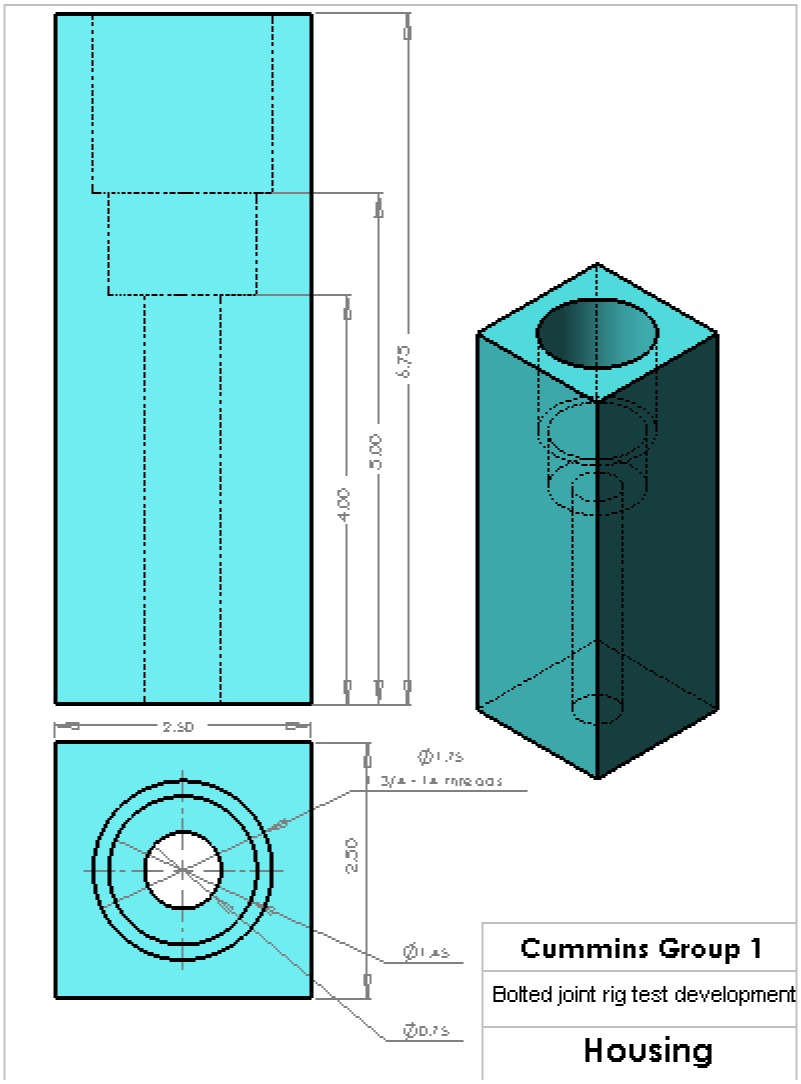
Test Fixture:

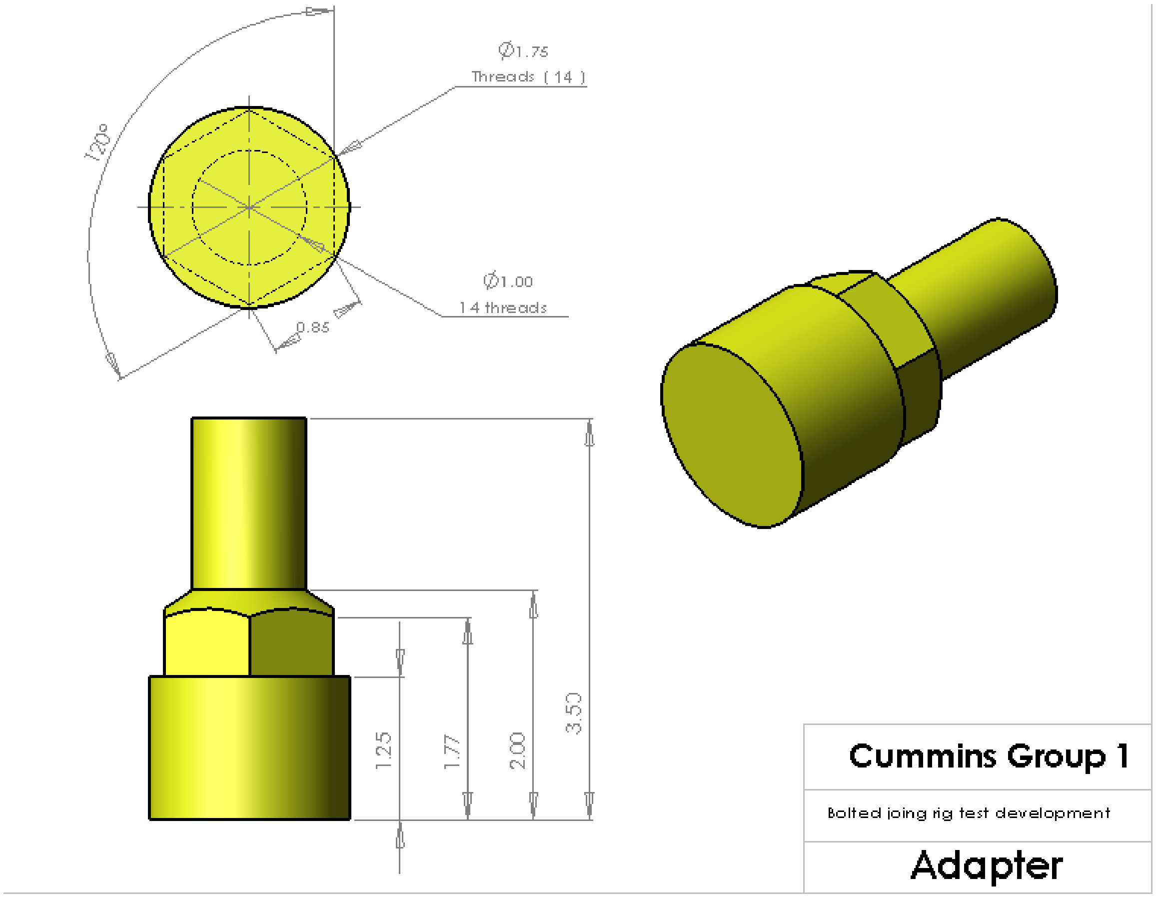


Adapter:

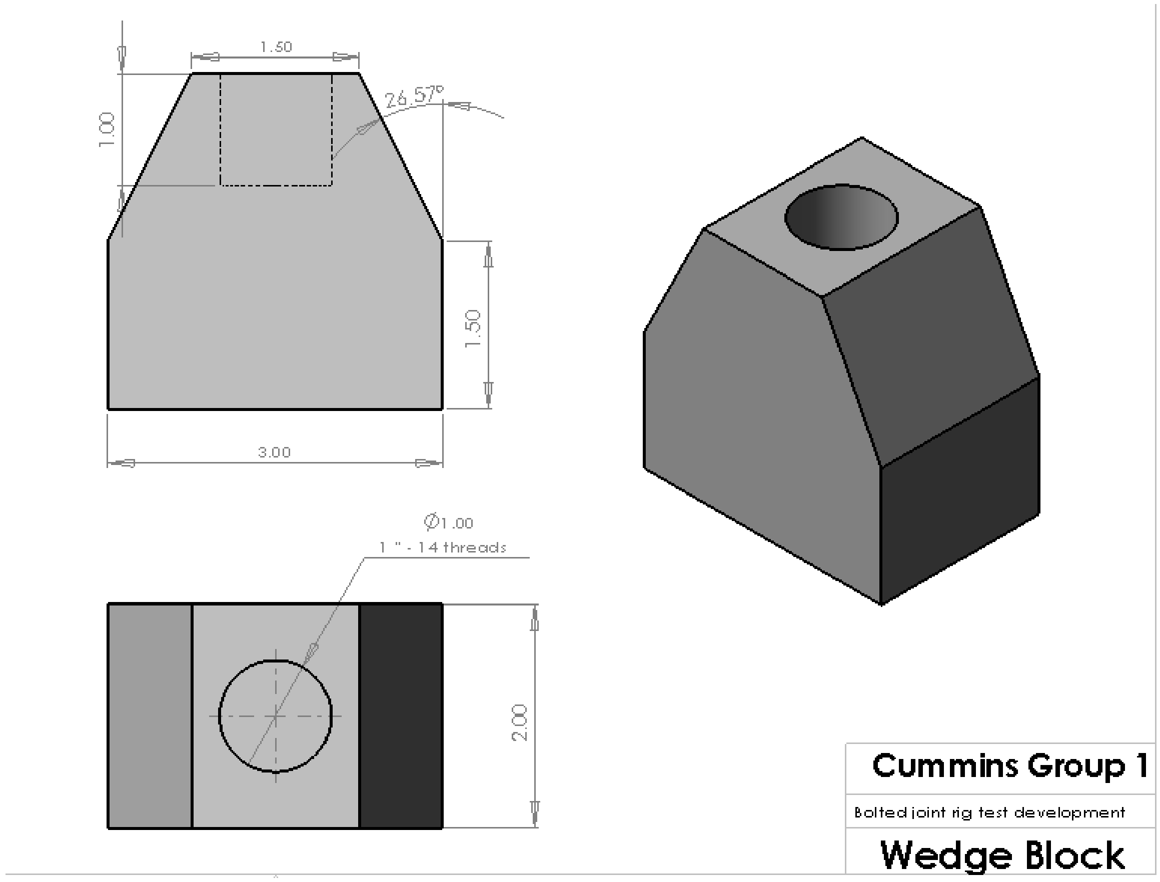


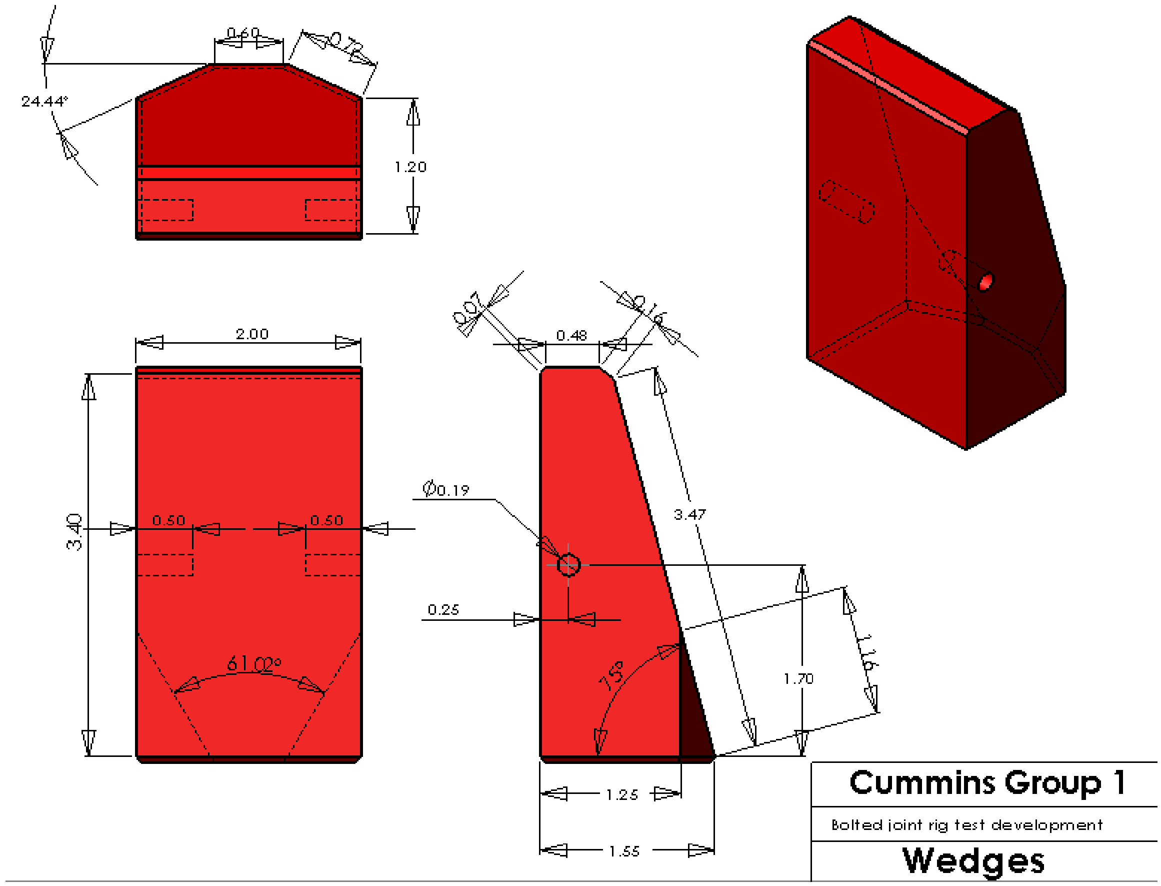
Appendix F

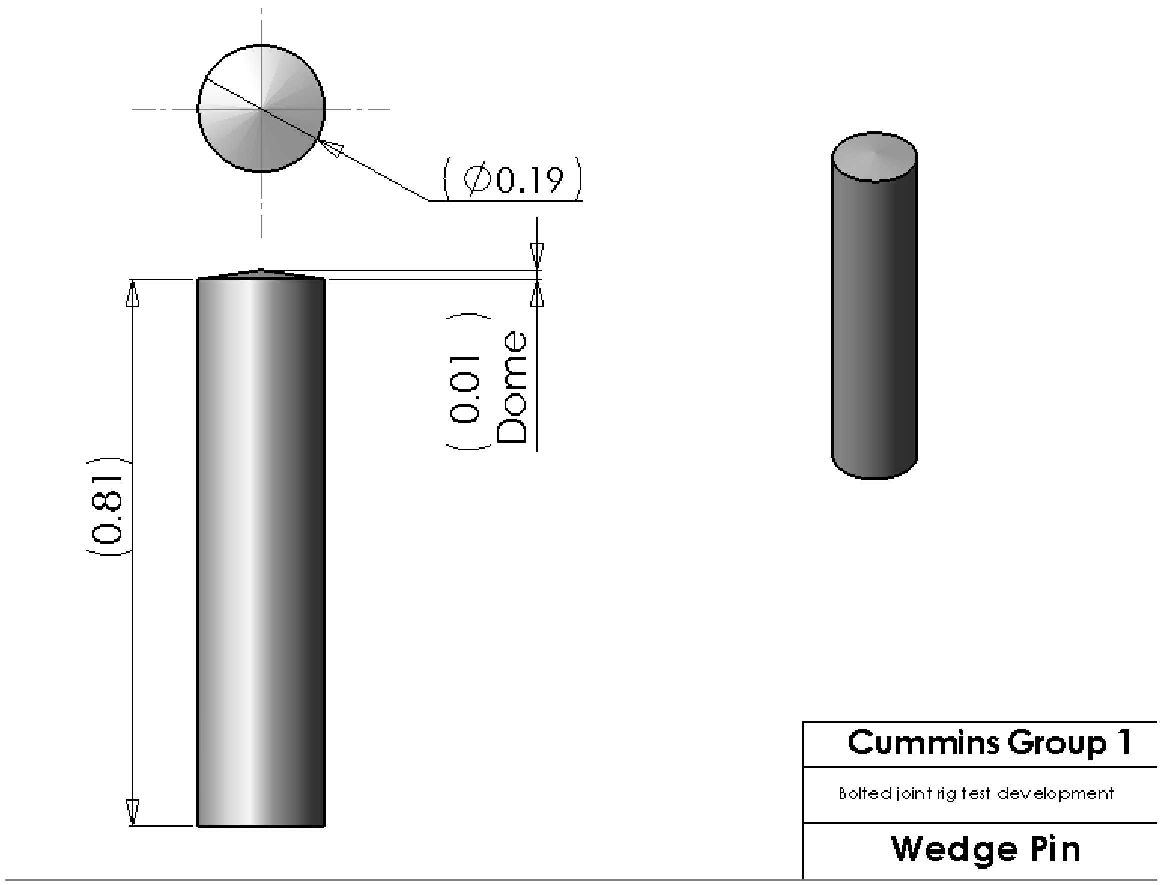

6.75
5.00
4.00
2.50
2.50
Ø1.75
3/4 - 14 Threads
Ø1.45
Ø0.75
Cummins Group 1
Bolted joint rig test development
Housing



Ø1.75
Threads (14)
120°
Ø1.00
14 threads
0.85
1.25
1.77
2.00
3.50
Cummins Group 1
Bolted joing rig test development
Adapter


1.50
26.57°
1.00
1.50
3.00
Ø1.00
1 " - 14 threads
2.00
Cummins Group 1
Bolted joint rig test development
Wedge Block

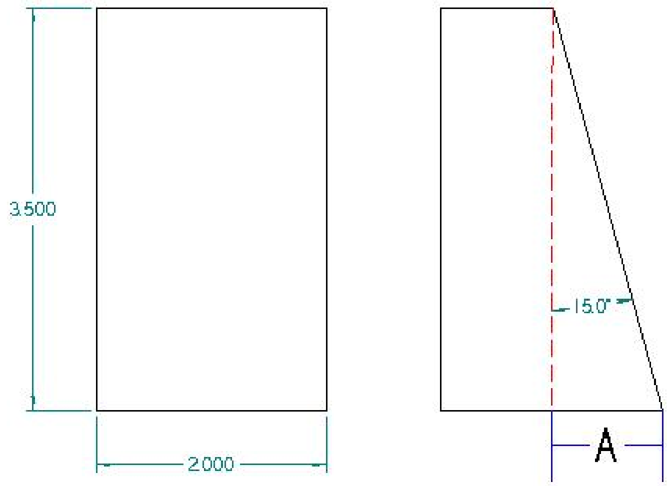

0.60
0.72
24.44°
1.20
2.00
3.40
0.50
0.50
61.02°
0.07
0.48
0.16
⌀0.19
3.47
0.25
75°
1.16
1.70
1.25
1.55
Cummins Group 1
Bolted joint rig test development
Wedges


(Ø0.19)
(0.01) Dome
(0.81)
Cummins Group 1
Bolted joint rig test development
Wedge Pin

Appendix G
Wedge Grip Design

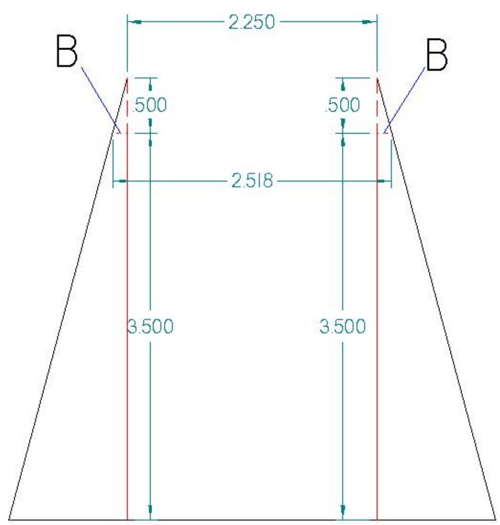
The opening size on the wedge grips needed to be measured to calculate the dimensions of the wedges that would fit securely during testing of the samples. The placement location for the wedges was in the shape of a trapezoid. While the wedge grips were in place the only measurements that could be taken were the opening (2.25 inches), the angle of the sides (-75 and -115 degrees from horizontal) and the height of the trapezoid in the lowest and highest position of the piston was 4 inches and 3 inches respectively.

The height of the wedge was chosen to be 3.5 inches so that the specimen can be gripped at either 0.5 inches below or above the wedge grip dependent on the piston being either in its lowest or highest position. The wedges were given a width of 2 inches to fit into a channel that has been manufactured onto the wedge grips. With these two proportions and the angle of the wedge sizes, further calculations were made to get the necessary dimensions for manufacturing.



$$\tan(15deg) = \frac{A}{3.5in} \qquad A := 3.5in \cdot \tan(15deg) \qquad A = 0.938in$$

The clearance from wall to wall inside the wedge grip at a height of 3.5 inches (in reference from the bottom wedge grip when the piston is at its lowest position) was found so that the largest wedge and largest opening could be known.



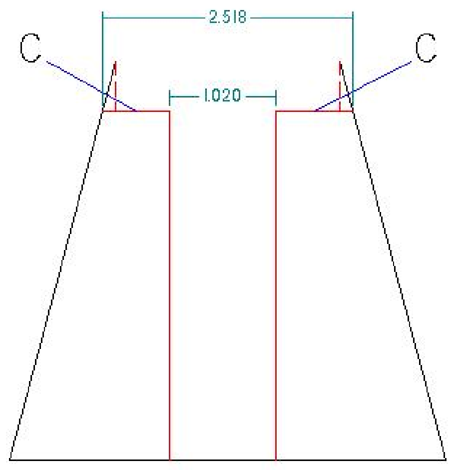
clearance inbetween bottom position (refers to lower grip)

$$\tan(15deg) = \frac{B}{0.5in} \qquad B := 0.5in \cdot \tan(15deg) \qquad B = 0.134in$$

$$2.25in + 2 \cdot B = 2.518in$$

Max width is 2.518 inches at 3.5 inches (height of wedge) from bottom position

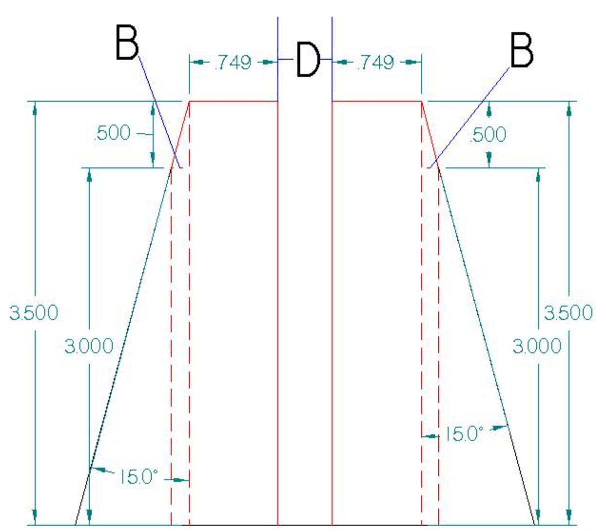
The largest specimen to be tested (main bearing cap) was going to be 1.02 inches, so this amount was subtracted from the total clearance (found above). The remainder was the allowable width of the wedge grips.



wedges in bottom position with opening for largest sized specimen (1.02 inches) that will be tested

$2.518\text{in} - 1.02\text{in} = 1.498\text{in}$ $\qquad \dfrac{1.498\text{in}}{2} = 0.749\text{in}$ $\qquad C := 0.75\text{in}$ Round to C=0.75 inches for top thickness of wedges

Now that the wedge grips have a total width from the addition of A, B and C; the clearance for the testing specimen can be calculate when the piston is in its top position.



clearance inbetween wedges in the top position (refers to lower grip)

$2.25\text{in} - 2 \cdot C - 2 \cdot B = 0.482\text{in}$ $\qquad$ $D := 0.482\text{in}$

These calculations provide the dimensions necessary for wedges to accommodate the connecting rod and main bearing cap that will be tested.

Appendix H

Test Plan:

1. Project Background and Information

The purpose of this project is to develop two separate test fixtures that will be used to conduct fatigue tests. The two test fixtures will be testing threaded bolt joints of a connecting rod and a main bearing cap. The fixtures should simulate fatigue the stresses these two joints undergo throughout their time in use. Along with this, the fixtures should work with the material testing system (MTS) machine.

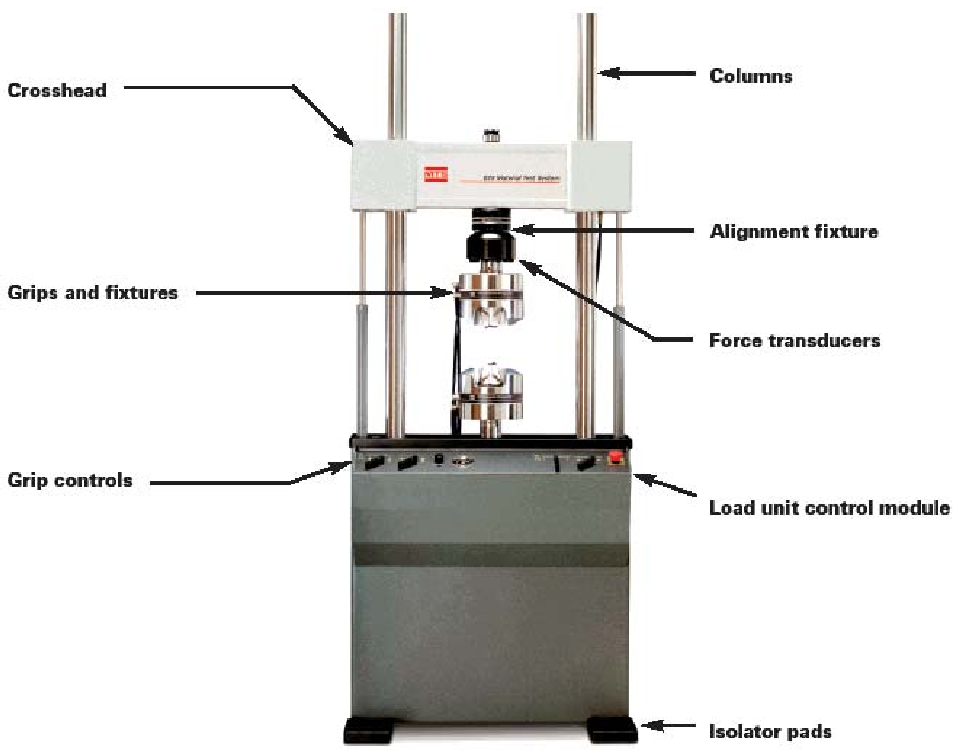
After much work, two final test fixtures were made for the fatigue tests. The connecting rod sample fixture consisted of four wedges that would fit into the MTS wedge grip device. The main bearing cap testing rig consisted of a bolt housing and an adapter that would fit into an MTS block. That MTS block would then fit into the MTS wedge grip.

The National High Magnetic Field Laboratory allowed the group to use one of the MTS machines. Time had to be scheduled in advance so as to prevent conflict with anyone else.

2. Recourses used

The MTS Machine:

The MTS is a servo-hydraulic machine. It is used for material testing. Below is a figure that shows the machine and its components.



The MTS machine (above) is mainly used to carry out fatigue tests. It can operate in tension, compression, or both. There are other uses for the machine depending on the test that is to be run. The load unit control module on the MTS used was off to the side. The software used to control the MTS is called Test Star Controls Machine.

We simply want to test our fixtures to make sure that they operate efficiently.

Test Ware:

The data acquisition software that was used is called Test Ware. It records single cycle data. We chose to use this software because it was already integrated with the MTS machine. Also, we were only concerned with single cycle data. The main concern was that the fixtures work.

Selecting the single cycle option set up the tests. Edit the ramp value (amplitude of force) to the desired value. This all depends on weather you want to run a test in compression, tension, or both. Then, run a single cycle to make sure everything is set up properly. The settings needed for our particular tests are rather simple. This is because the main focus of the tests is to check the fixtures. Only a simple fatigue test with cycles run, stroke of MTS, and constant force was needed.

3. Procedures

(step by step instructions for our test)

1. Torque bolts for testing.
2. If working with the main bearing cap, attach adapter to fixture. Then put the adapter into the MTS block. The MTS block should already be attached to the MTS wedge grip on top.
3. If working with the connecting rod, simply place wedges within MTS wedge grips, top and bottom. Do the same (bottom only) for main bearing cap.
4. Prepare the data software.
5. Turn on the pump using the Load Unit Cell (LUC) on the control panel.
6. Turn the hydraulic power supply on, this opens the valves in the hydraulic service manifold (HSM).
7. MTS can now be controlled either manually or through stroke control.
8. Let the oils warm up, about 15 minutes.
9. Ready to now put fixture in:
    a. Main bearing cap

i. Make sure the wedge grips are open to their maximum for the main bearing cap, controlled with valves.
ii. Bring wedges up to main bearing cap.
iii. Apply pressure to both upper and lower grips to clamp the wedges.

b. Connecting rod

i. Place sample into lower (or upper) wedges.
ii. Apply pressure so that the wedges clamp into the sample.
iii. Bring the sample up to the top wedges and apply pressure to top.

10. Go into Test Ware software and run the test cycle.
11. If everything is ok, switch to function generator and start testing.

4. Safety

It is important that the MTS machine not be used without supervision or someone else in the room. Make sure that all the pieces are properly aligned. Not doing so may cause failure to the alignment fixture, wedges, wedge grips, and other components of the MTS machine. Do not place your hand within the wedge grips once the MTS has been turned on.

Make sure to go over the safety precautions specific to the MTS being used.

5. Test Data

The data gathered from the tests were not valuable to the tests. All it shows is where the sample might have elongated or slipped from the fixtures. The main focus was to ensure the test fixtures worked.

Appendix I

Engineering Standards

Name	IRON,CAST GRAY (CONTROLLED CARBON)	Engineering Standard Number
Identifier	MATERIALS SPECIFICATION (IRON,GRAY)	40060

1. Scope

This specification covers requirements for an alloyed cast gray iron for applications where strength and machinability are significant.

2. Applicable Documents

Applicable documents listed below may be obtained from the respective organizations listed in CES 10054, Standards Organizations Addresses.

a. ASTM A 247, Evaluating the Microstructure of Graphite in Iron Castings
b. ASTM E 8, Tension Testing of Metallic Materials
c. ASTM E 10, Brinell Hardness of Metallic Materials
d. ASTM E 351, Chemical Analysis of Cast Iron --All Types
e. ASTM E 562, Determining Volume Fraction by Systematic Manual Point Count
f. CES 10054, Standards Organizations Addresses
g. CES 10056, Glossary
h. CES 16062, Visual Defects of Castings

3. Definitions

There are no unique definitions in this standard. Terms used in this standard that have a general definition for usage in Cummins Engineering Standards are defined in CES 10056, Glossary.

4. Specification

4.1. Significance and Use

This specification covers requirements for an alloyed cast gray iron for large casting (cylinder block) applications where tensile strength, fatigue strength, and machinability are significant considerations.

4.2. Chemical Composition

4.2.1. Chemical composition requirements shall be in accordance with Table 1: Chemical Composition on page 2, when analyzed in accordance with ASTM E 351. The acceptance limits are for single lot tests at the frequency as agreed upon between manufacturer and purchaser and documented in a plan.

Table 1: Chemical Composition

Element	Weight Percent
Carbon	3.10 - 3.35
Silicon	1.7 - 2.3
Chromium	0.20 - 0.40
Copper	0.60 - 0.90
Molybdenum	0.15 maximum
Manganese	0.35 - 0.80
Phosphorus	0.08 maximum
Sulphur	0.15 maximum

4.2.2. Carbon equivalent shall be 4.05 percent maximum (%C + 1/3%Si).

4.3. Physical Properties

There are no physical property requirements.

4.4. Mechanical Properties

4.4.1. Hardness shall be 187 to 241 HB in the location(s) noted on the engineering drawing, when determined in accordance with ASTM E 10. A maximum hardness of 255 HB is permitted in other machined areas.

4.4.2. Hardness tests shall also be performed on tensile bars cut from castings and the hardness shall be 187 to 241 HB, when determined in accordance with ASTM E 10.

4.4.3. Tensile strength of bars cut from castings shall be determined in accordance with ASTM E 8. Location(s) of test specimen(s) shall be in accordance with the engineering drawing which will be in areas indicative of critical strength requirements. Lot size, quantity of tests, lot acceptance criteria, production audit, and data analysis procedures shall be as agreed upon between the manufacturer and purchaser and documented in a plan.

4.4.4. General Tensile Strength (See Appendix A: Tensile Strength on page 5).

Tensile bar location(s) as indicated on the drawing shall meet the following tensile strength requirements:

4.4.4.1. The one-sided 90 percent confidence limit for the mean tensile strength shall be 33,500 psi (231 MPa) minimum, based on sample size of 10.

4.4.4.2. The one-sided 90 percent confidence limit for the tensile strength of 75 percent of the population shall be 32,000 psi (221 MPa) minimum based on sample size of 10.

4.4.5. Supplemental Tensile Strength (See Appendix A: Tensile Strength on page 5).

When specified on the engineering drawing (in an indicated area other than for the general tensile strength requirement), the following additional tensile strength requirements shall apply:

4.4.5.1. The one-sided 90 percent confidence limit for the mean tensile strength shall meet or exceed the tensile value specified on the engineering drawing based on sample size of 10.

4.4.5.2. The one-sided 90 percent confidence limit for the tensile strength of 75 percent of the population shall meet or exceed the tensile value specified on the engineering drawing based on sample size of 10.

4.5. Microstructure

4.5.1. Graphite

4.5.1.1. Maximum permitted dimension of flake graphite shall be in accordance with Size Class 2 requirement specified in ASTM A 247.

4.5.1.2. Measurement of flake graphite by visual comparison with the applicable graphite size chart in accordance with ASTM A 247 is permitted.

4.5.1.3. The referee method for determining flake graphite size shall be performed by using a calibrated ocular scale or photomicrograph, and measuring end to end length of flake graphite.

4.5.2. Matrix

4.5.2.1. Matrix shall be predominantly pearlite.

4.5.2.2. Ferrite shall be 5 percent maximum, when determined in accordance with ASTM E 562.

4.5.2.3. A 0.060 inch (1.5 mm) ferrite rim is permitted on as cast surfaces.

4.5.2.4. Free or massive carbide and/or steadite shall not exceed 2 percent total in the casting, when determined in accordance with ASTM E 562.

4.5.2.5. To limit the presence of chill, massive carbide or steadite shall not exceed 5 percent in 200X field within regions to be machined, when determined in accordance with ASTM E 562.

4.6. Additional Requirements

4.6.1. Castings shall be supplied in the stress relieved condition as specified on the engineering drawing.

4.6.2. Welding of castings is not permitted except as documented in an approved salvage procedure.

4.6.3. Injurious defects are limited as described in CES 16062.

4.6.4. A consistent inoculation practice is necessary to minimize variability in mechanical properties. Inoculation practice shall be determined by the supplier and documented in a plan approved by the purchaser. Proper inoculation serves to improve machinability by refining the microstructure, which results in higher tensile properties relative to hardness and in a finer, more uniform distribution of intercellular carbides and steadite, in addition to reducing the chilling tendency.

Name	IRON,CAST GRAY (CONTROLLED CARBON)	Engineering Standard Number
Identifier	MATERIALS SPECIFICATION (IRON,GRAY)	40060

Appendix A: Tensile Strength

Tensile Strength Calculations for Normal Populations

Using the mean and standard deviations of the 10 samples:

a. one-sided 90% confidence limit for the mean tensile strength
 = mean - (0.4373 x standard deviation)

b. one-sided 90% confidence limit for minimum tensile strength of 75% of the population
 = mean - (1.257 x standard deviation)

Appendix J

Holmes Tool and Engineeri 850 547 9329 p.1

ATTN (CEDRIC)

350

AVERAGE MECHANICAL PROPERTIES OF VARIOUS STEELS COMMONLY ORDERED TO CHEMICAL SPECIFICATIONS

A.I.S.I. No.	Condition	1000 lbs. PSI Tensile Strength	Yield Point	% Elong. 2"	% Red. Area	Hardness Brinell	Rockwell	% Machin-ability	Heat Treatment Quenched From	Tempered At	Medium
C 1045	Hot Rolled	82	45	16	40	163	C18	55	....	....	....
	Cold Drawn	91	77	12	35	179	C15	55	....	....	....
C 1095	Hot Rolled	139	84	97	10	293	C31	....	....	....	....
	Heat Treated	165	109	10	42	311	C33	....	1457°	1000°	Water
B 1112	Cold Drawn	78	63	23	35	157	B89	100	....	....	....
C 1117	Hot Rolled	62	34	27	47	131	B73	84	....	....	....
	Cold Drawn	69	59	21	52	137	B87	90	....	....	....
C 1137	Hot Rolled	67	54	20	47	192	B92	70	....	....	....
	Cold Drawn	98	82	16	35	199	C16	70	....	....	....
	Heat Treated	122	98	17	51	248	C24	....	1550°	1000°	Water
		110	71	21	56	229	C21	....	1550°	1100°	Water
		108	69	20	52	223	B97	....	1550°	1200°	Water
		108	78	21	58	223	B97	....	1575°	1000°	Oil
		101	69	24	60	207	B95	....	1575°	1100°	Oil
		98	69	24	61	197	B94	....	1575°	1200°	Oil
C 1141	Hot Rolled	94	51	15	35	212	B95	65	....	....	....
	Cold Drawn	105	88	10	30	223	C19	70	....	....	....
C1144	Cold Drawn	125	100	12	34	205	....	83	....	....	
A311 Class B											
C 12L14	Cold Drawn	78	60	10	35	163	B86	160	....	....	....
C 1215	Cold Drawn	78	60	10	55	163	B86	136	....	....	....
4130	H.R. Annealed	84	53	28	54	183	B90	65	....	....	....
	C.D. Annealed	97	87	21	57	201	B95	70	....	....	....
	Heat Treated	151	138	18	54	321	C34	....	1575°	900°	Water
		140	130	19	52	293	C31	....	1575°	1000°	Water
		128	113	21	58	262	C27	....	1575°	1100°	Water
4140	H.R. Annealed	89	62	26	57	187	B91	57	....	....	....
	C.D. Annealed	102	90	18	50	223	C19	66	....	....	....
	Heat Treated	153	131	16	45	311	C32	....	1550°	1000°	Oil
		140	135	20	50	285	C30	....	1550°	1100°	Oil
		118	103	21	60	269	C28	....	1550°	1200°	Oil
4150	H.R. H.T.	165	138	18	51	302	C32	....	1525°	1100°	Oil
4340	H.R. Annealed	101	69	21	45	207	C15	45	....	....	....
	C.D. Annealed	114	96	15	44	223	C21	55	....	....	....
	Heat Treated	173	166	14	46	352	C38	....	1475°	1000°	Oil
		165	150	17	54	331	C35	....	1475°	1100°	Oil
		139	126	20	60	277	C29	....	1475°	1200°	Oil
4615	Hot Rolled	84	60	28	64	179	B89	48	....	....	....
	Cold Drawn	98	82	18	59	197	B93	67	....	....	....
4620	Hot Rolled	86	63	23	63	167	C10	48	....	....	....
	Cold Drawn	99	84	21	58	207	B95	67			
6150	H. R. Annealed	91	58	22	53	183	B90	59			
	Heat Treated	156	132	15	44	302	C32		1550°	1000°	Oil
8620	Hot Rolled	92	64	25	58	192	B92	58	....	....	....
	Cold Drawn	104	79	21	49	212	B96	66	....	....	....
8642	H.R. Annealed	100	62	24	55	197	B93	58	....	....	....
	C.D. Annealed	106	89	17	59	229	C21	66	....	....	....
	Heat Treated	149	129	16	45	302	C32	....	1550°	1000°	Oil

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Appendix K

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Ship to attn:
Receiving

Qty	Description	Unit Price	TOTAL
4	WEDGES W/ PINS 4140 HT	$250.00	$1,000.00
1	MODIFY CONNECTING ROD	$200.00	$200.00
1	CYLINDER FOR FIXTURE	$125.00	$125.00
1	SQ HOUSING FOR FIXTURE	$300.00	$300.00
		TOTAL	$1,625.00

Delivery: Two (2) weeks, ARO.

Terms: 1% 10, Net 30

NOTE: *These prices are effective for one (1) month from this date. If placing order after one (1) month, please call for pricing update.*

Tim Steverson 2/2/05
Tim Steverson, Machine Shop Supervisor Date