Group 1

Bolted Joint Rig Test Development

Technical Report

Alex Dugé
Ana Erb
Ronald Rolle
Cedric White

ABSTRACT

In order to better understand the threaded joints of a design, an axial fatigue test was performed on a connecting rod and main bearing cap threaded joints. These tests were performed at the National High Magnetic Field Laboratory by using a servo hydraulic MTS fatigue-testing machine. The testing required two separate test setups and two different test fixtures. One of the set ups tests the connecting rod cap – to – connecting rod bolts, the other set up tests the main bearing cap – to – engine block bolts. Both test fixtures had to be designed to withstand extremely large loads and interface with the MTS machine.

The setup for the connecting rod test 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. For the main bearing cap setup, 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. 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.

1.0 BACKGROUND

1.1 Fasteners

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.

Figure 1: Threaded bolts and nuts

Threads

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. Bolts can be made with single, double, or triple threads and threads can be classified as coarse, fine, or extra fine.

A machine screw (bolt) is a threaded metal rod with a head at one end, intended to be screwed into a threaded hole. A thread is a helix shaped slot, which is machined onto a bolt that causes the bolt to advance into the workpiece when rotated. The lead L of the thread is the distance that a mating thread will advance axially with one revolution. And the distance between two threads is the thread pitch. Threads can either be internal (nut) or external (bolt). Internal threads are usually cut with a special tool called a tap, whereas external threads are cut with a lathe or a die.

Stresses in Threads

Theoretically, when a nut engages a thread, all the threads in engagement should share the load. However, inaccuracies in thread spacing, causes virtually all the load to be taken by the first pair of threads. Thus, the conservative approach in calculating thread stresses is to assume the worst case of one thread-pair taking the entire load. Or the other extreme, that all engaged threads share the load equally. Both of these assumptions can be used to calculate estimated thread stresses.

Preload

Whenever a bolt is put in tension due to an applied load, it common practice to preload the joint. That is done by tightened the bolt with sufficient torque to create tensile loads that approaches their proof strength. For dynamically loaded assemblies (fatigue loads) a load of 75% or more of proof strength is commonly used. Assuming that the bolts are suitably sized for the applied loads, these high preloads make it very unlikely that the bolt will break in service if they do not break while being tensioned.

1.2 Fatigue

Fatigue is one of the primary reasons for failure among structural components. The definition of fatigue failure is the weakening or breakdown of materials subjected to repeated stress. All fatigue failures begin at a crack, notch, or other stress concentration area. The crack can either develop over time due to cyclic straining or the crack may have been present since the material was manufactured. Virtually all materials contain discontinuities, ranging from the microscopic to the macroscopic levels, introduced at the manufacturing or fabricating process.

Overview

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 to cause 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. Automobiles, aircraft, compressors, pumps, and turbines, are some examples of equipments, which have components that are subjected to fatigue failures.

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 2 is an illustration showing the different ways in which cracks are initiated.

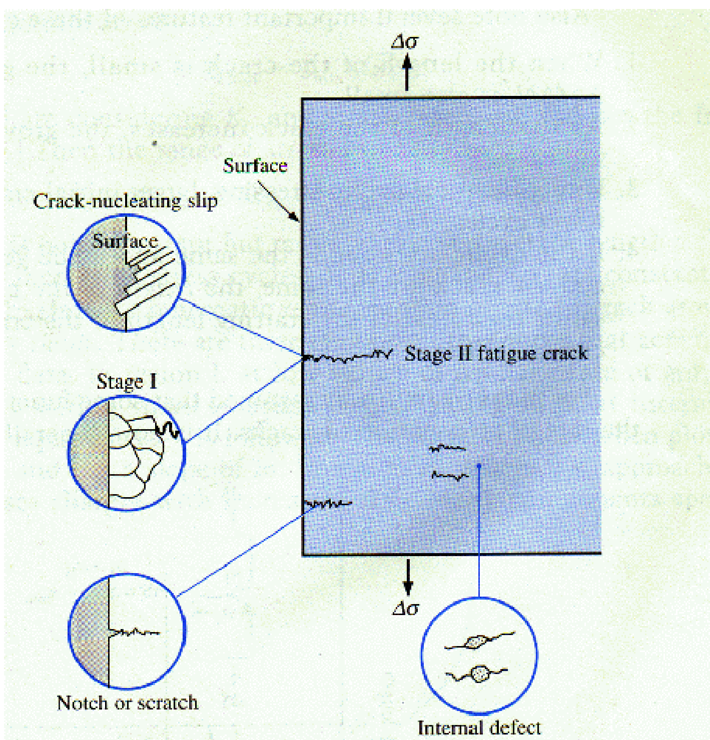

Figure 2: Deformation of crack propagation due to fatigue

Failure Mode

In general there are three stages in fatigue failure, crack initiation, crack propagation, and failure. Stage one, crack initiation, is where preexisting voids or inclusions serve as stress raisers and start a crack. 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. This stage is of short duration mainly in brittle materials. Stage two, crack propagation, is where the crack spreads across the material as a result of continuously applied stress. Crack propagation normally consume the entire life of the parts. The third stage, failure, is where the material is unable to withstand the applied stress and instantaneously fails. This is caused by unstable crack growth in the material.

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 3 gives a visual illustration of the different modes.

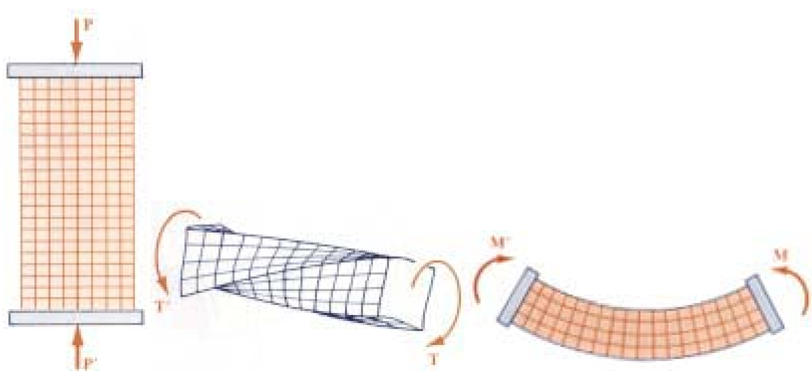

Figure 3: Loading Cases, axial, torsion, and bending

There are also three stress cycles with which loads may be applied to a component. The reversed stress cycle, shown in Figure 4a 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 (4b) 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.

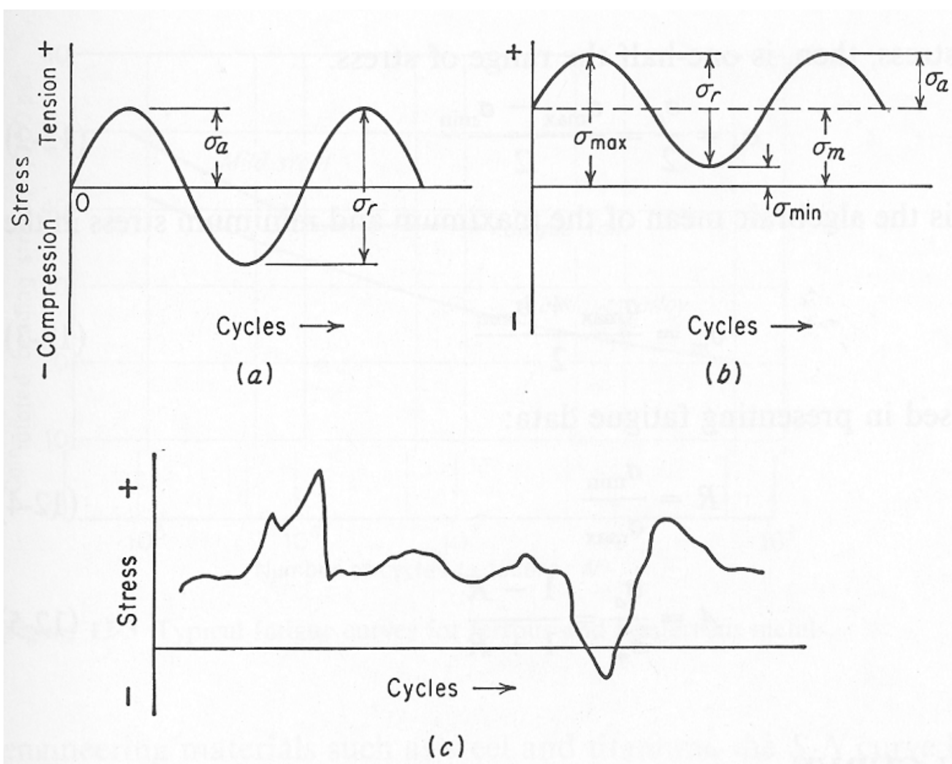

Figure 4: Reverse cycle, repeated cycle and random stress cycle

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 1.

$$\sigma_r = \sigma_{max} - \sigma_{min} \tag{1}$$

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

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

The alternating stress is

$$\sigma_a = \frac{\sigma_r}{2} \tag{3}$$

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 5.

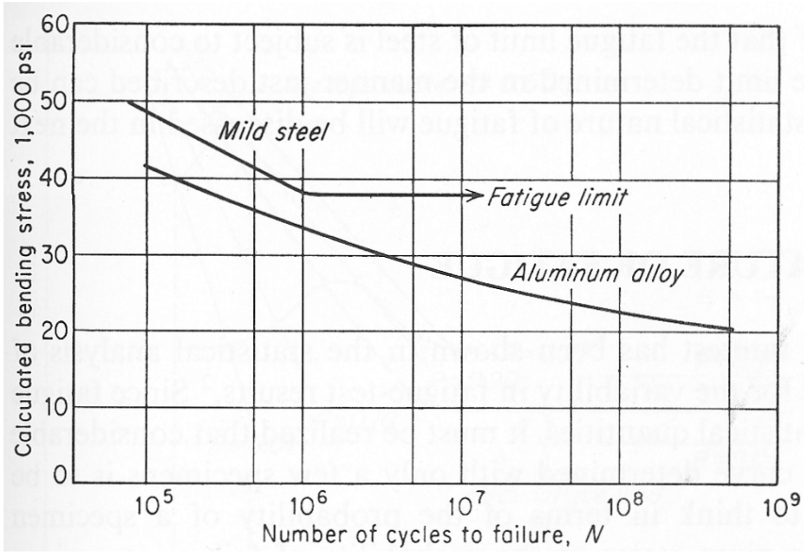

Figure 5: Typical S-N curve for Ferrous and Non-Ferrous materials

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.

$$N\sigma_a^p = C \tag{4}$$

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

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.

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

The material testing system (MTS), model 312-810 is a state of art servohydraulic fatigue tester. In Figure 8 below, all the major components of the MTS can be seen. Each of the two wedge grips weight approximately 180 pounds and can exert a force up to 250 kips. This machine can be controlled by personal computer using TestWare, advance computer software. The MTS is very universal and can be programmed to perform a variety of test, mainly tension and compression. This machine has enormous capabilities and is perfect for advance material testing.

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 6 shows a variety of grip designs that can be used for axial fatigue testing.

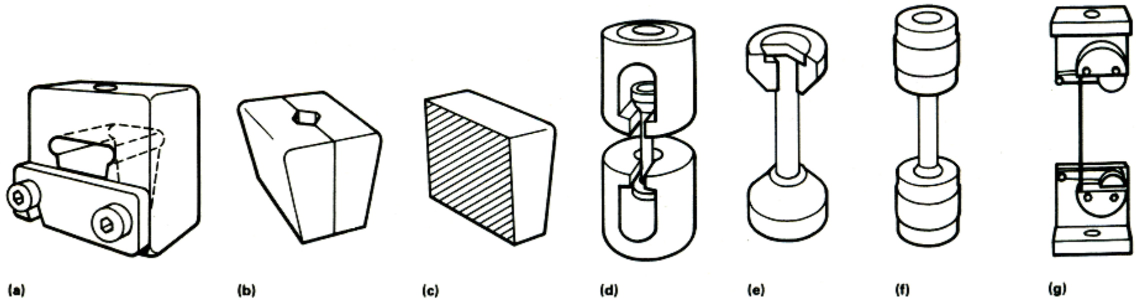

Figure 6: Grip designs for axial fatigue testing
(a) Standard grip body for wedge-type grips. (b) V-grips for rounds for use in standard grip body. (c) Flat grips for specimens for use in standard grip body. (d) Universal open-front holders. (e) Adapters for special samples (screws, bolts, studs…) for 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 7 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.

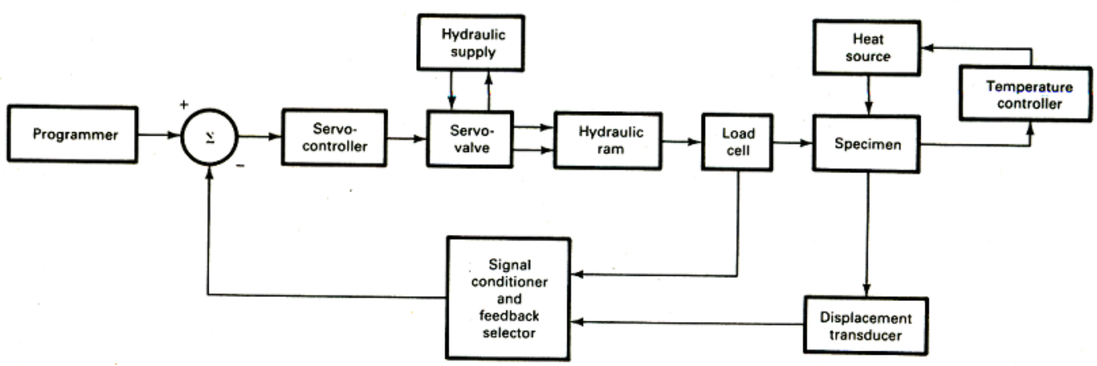

Figure 7: Block diagram of components of a servohydraulic fatigue testing system

Controls

All the major components can be controlled by either, the load unit, cross head lift, and or pressure supply controllers. Figure 8 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.

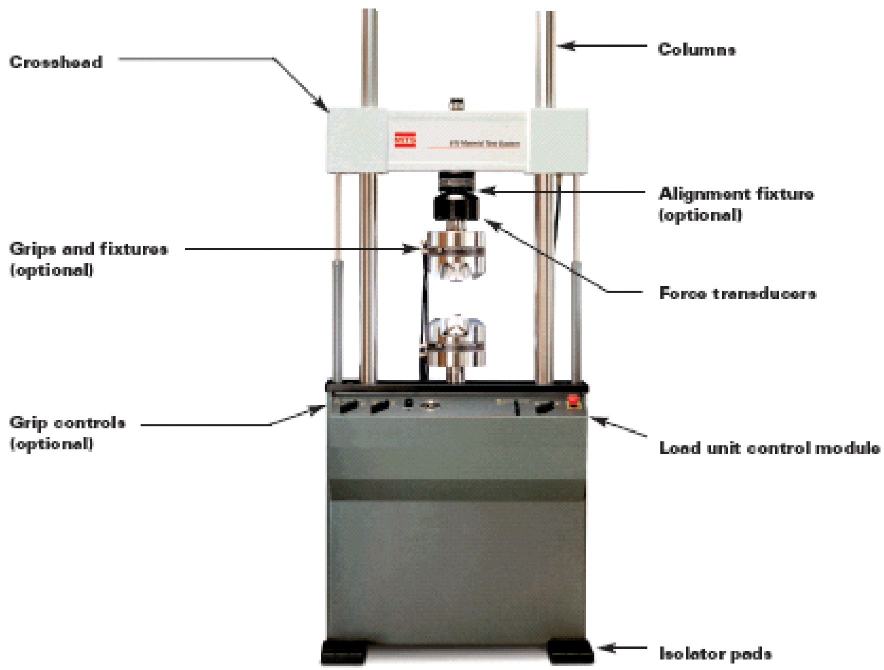

Figure 8: MTS servohydraulic machine and its components.

2.0 PROCEDURES

Main Bearing Cap Setup

Refer to Figure 9 below for Set up

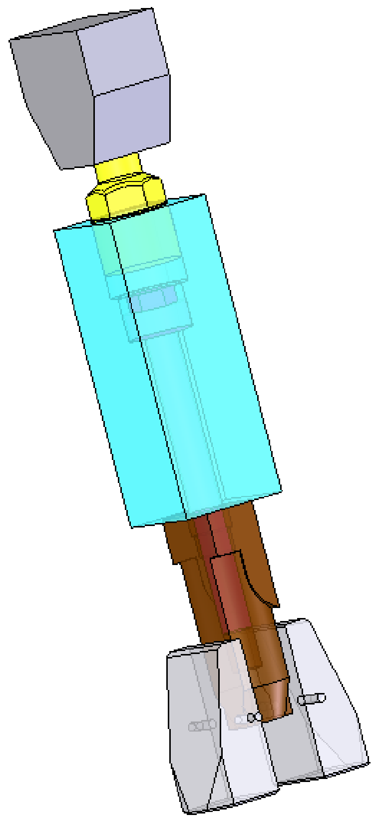

Figure 9: Set up for the main bearing cap fixture.

- Torque bolt to 330 ft-lb
    - This torque is for a 22.5 kips preload
- Screw adapter to fixture
- Screw the other end of the adapter to MTS holding block
- Configure software for single axis fatigue testing
    - Software
        - Function Generator

        - Test-Ware for data collection
        - Test Star controls the machine

- Assign and Calibrate the stroke hydraulically
- Turn on Hydraulic pump control on the Load Unit Control Panel (LUCM)
- Turn on Hydraulic Service Manifold (HSM)
- Switch to manual control
- Let Hydraulics warm up for about 15 minutes
- Release the top Hydraulic wedge grips
- Set the MTS holding block on the top wedge grip and zero out the weight
- Apply pressure (normal) to the MTS Block to tightly hold it in place
    - Pressure – 9000 psi
- Release the bottom Hydraulic wedge grip
- Align the engine block specimen with the bottom wedge grip on the MTS machine
- Manually adjust the actuator to align wedges with the specimen
    - Make sure the Hydraulic pump unit is on stoke control
- Adjust normal pressure on the wedges so they can tightly hold the specimen into place
    - Pressure – 9000 psi
- Switch to computer control
- Open Test Ware
    - Add 22.5 kips for ramp up
    - For ramp down, input zero

- Run a single cycle to test if wedges are holding the specimen
- If the specimen slips, increase the normal pressure.
- If everything is working fine, switch to function generator

➢ Function Generator

- Input 11.25 kips for the mean
- Input 22.5 kips for the Amplitude
- This will fatigue the specimen in tension only.
- Change the frequency to 2 Hz
- Keep the minimum stoke constant

Connecting Rod Setup

Refer to Figure 10 below for Set up

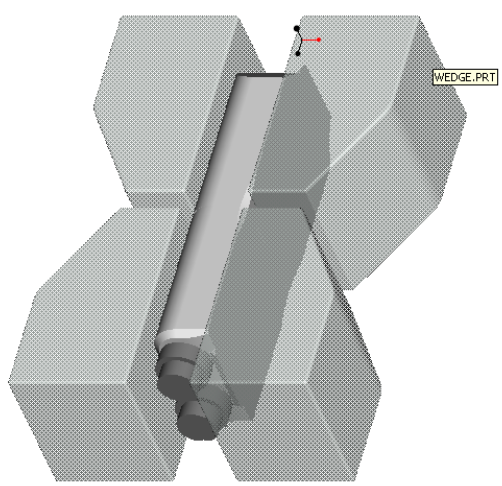

Figure 10: Set up for the connecting rod fixture

- Torque bolt to 130 ft-lb
    - This torque is for a 16 kips pre-load
- Configure software for single axis fatigue testing
    - Software
        - Function Generator
        - Test-Ware for data collection
        - Test Star controls the machine
- Assign and Calibrate the stroke hydraulically
- Turn on Hydraulic pump control on the Load Unit Control Panel (LUCM)
- Turn on Hydraulic Service Manifold (HSM)
- Switch to manual control
- Let Hydraulics warm up for about 15 minutes
- Lower the crosshead for adequate spacing between the wedge-grips
- Release the top and bottom Hydraulic wedge-grips
- Set the wedges inside the top wedge-grip and use the springs to hold them into place
- Place the specimen in between the wedges
- Apply pressure (normal) to the wedges to tightly hold them in place
    - Pressure – 9000 psi
- Set the other two wedges inside the bottom wedge grip on the MTS machine
- Manually adjust the actuator to align wedges with the specimen
    - Make sure the Hydraulic pump unit is on stoke control

- Adjust normal pressure on the wedges so they can tightly hold the specimen into place
    - Pressure – 9000 psi
- Switch to computer control
- Open Test Ware
    - Add 16 kips for ramp up
    - For ramp down, input zero
    - Run a single cycle to test if wedges are holding the specimen
    - If the specimen slips, increase the normal pressure.
    - If everything is working fine, switch to function generator
- Function Generator
    - Input 8 kips for the mean
    - Input 16 kips for the Amplitude
    - This will fatigue the specimen in tension only.
    - Change the frequency to 3 Hz
    - Keep the minimum stoke constant

3.0 Results and Discussion

A total of four tests were performed. 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 1.

Table 1: Test Requirements.

	Load Amplitude (kips)	**Compression**	**Tension**
Connecting Rod #1	16	X	X
Connecting Rod #2	16 then 24		X
Main Bearing Cap #1	45	X	X
Main Bearing Cap #2	22.5		X

The following table shows the amount of cycles as to when each test was stopped, either because of time or failure. One of the main problems that arose during testing was slippage. Because the finish on the wedges was not fine enough to properly grip the samples, the samples would slip out of the wedges when the load is applied. Originally the normal pressure was set to 3,000 psi but that was not enough to hold the sample in place. Therefore, the normal pressure was increased to 6,000 psi for the connecting rod and 9,000 psi for the main bearing cap. If the wedges were made out of a harder material this would not have been a problem because they would have been able to indent the specimens and hold them in place with minimal pressure.

Table 2: Results Table With Number of Cycles Conducted

	Number of Cycles	Failure
Connecting Rod #1	1.3 million	No
Connecting Rod #2	~ 1 million	No
Main Bearing Cap #1	1000	Yes
Main Bearing Cap #2	10,000	Yes

Both of the connecting rods did not fail. There was no change noticed to the sample after testing. Connecting rod #1 did slip out of the wedges after about 1 million cycles. The sample was then placed back inside the wedges and the test was continued until about 1.3 million cycles. Connecting rod #2 also slipped out of the wedges. This time however, one of the wedges was damaged.

Both of the main bearing cap specimens had to be shaved down in a way that introduced some localized stressed along the sides of the sample. This is the location in which both of the samples failed. The area when the specimens failed was not the anticipated area of failure. Some modifications will have to be made in order to ensure proper testing. Both of the main bearing cap specimens failed in the same manner and the picture from this failure can be seen in Figure 11 below.

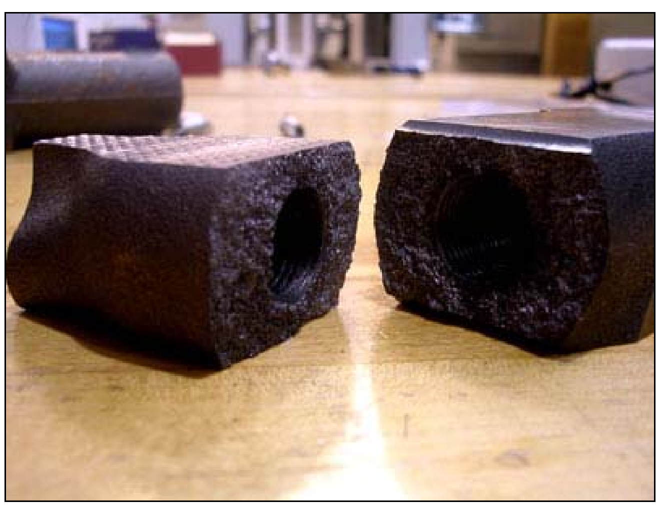

Figure 11: Main bearing cap test, engine block failure.

4.0 Conclusions

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.

5.0 References

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