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The Design of a Six Specimen Tensile Fatigue Machine

Charles Mark Percival
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THE DESIGN OF A SIX SPECIMEN
TENSILE FATIGUE MACHINE

A Thesis
Presented to the
Department of Mechanical Engineering
Brigham Young University

In Partial Fulfillment
of the Requirements for the Degree
Master of Science

by
Charles Mark Percival
August, 1961
This thesis, by Charles Mark Percival, is accepted in its present form by the Department of Mechanical Engineering of Brigham Young University as satisfying the thesis requirement for the degree of Master of Science.

Typed by Carolyn P. Lloyd
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CHAPTER I
INTRODUCTION

Although metal fatigue has been recognized for many years and during that time many persons have tried to find the explanation for it, there is still not a complete theory concerning this phenomenon. Metal fatigue is one of the major problems the modern designer must face. If the design techniques are to progress as other technological advances are made, it is evident that it will involve a better understanding of metal fatigue.

I. THE PROBLEM

The purpose of this study was to develop a high force, tensile fatigue machine capable of testing large statistical groups of specimens. The machine represents a major addition to facilities at Brigham Young University for the study of metal fatigue.

II. DEFINITIONS OF TERMS USED

Metal fatigue. Metal fatigue is the failure of a machine part or test specimen under the action of repeated stresses. The maximum values of these stresses may vary from a value slightly less than the ultimate strength of the material to values well below the yield strength of the material. The distinguishing characteristic of a fatigue failure is that the stress is usually repeated many times before failure occurs.
Fatigue machine. A fatigue machine is a machine capable of applying cyclic loads or stresses to a specimen for testing purposes. These loads may be simply tension, compression, bending, torsion or a combination of any of these loads.

Ultimate strength. The ultimate strength of a material is that stress which applied to the material under static conditions would rupture the material.

Yield strength. The yield strength of a material is that stress above which permanent deformation of the material occurs.

Endurance limit. The endurance limit of a material is that stress which the material can endure for an infinite number of cycles without failure.

III. SURVEY OF METAL FATIGUE STUDIES

One of the first to carry out tests of materials under repeated loads was a German Engineer, Wilhelm August Julius Albert. Albert (1), in 1829, built a machine to test mine hoisting chain with repeated loading. Interest in metal fatigue increased when the use of wrought iron and steel became common in construction of the early railroads. In England, a Commission was appointed in 1847 to study the suitability of iron for railway structures.

During the middle and latter portions of the nineteenth century, a German, August Wohler (2), published several reports and papers concerning the fatigue problem. Wohler conducted experiments on many types of materials, and as a result of his work, put forward what became known as Wohler's law. Wohler initiated the practice of testing specimens instead of using full scale tests, and his work has served as an
example for many later investigations.

Since Wohler's time, over 5,000 papers on the subject of fatigue have been published. (3) Interest in fatigue was greatly increased by the two world wars and the advent of the airplane. It was estimated in 1950 that as high as 80 per cent of current mechanical failures in service were due to metal fatigue. (4) From this, it is seen that greater effort must be put forth to achieve a better understanding of this problem. The literature survey failed to reveal any information directly applicable to this project.

IV. DESIGN DECISIONS

The ultimate goal of this work was to produce an operational machine with a minimum amount of rework and design changes. This, coupled with the facts that only one of these machines was to be fabricated, and that many of the ideas for the machine were untried, suggested that the machine should be overdesigned in those areas where its proper operation would not be affected. It was felt that it was simpler and more economical to have the parts made too large than to risk failures in critical areas. For those parts where size was an important factor, effort was made to keep the working stresses well below a critical level. Wherever possible the joints and connections were designed to take the loads with compression or shear forces in the members. These types of joints are far less susceptible to fatigue failure.

It should be stated here, however, that certain features on the machine were incorporated after initial fabrication to correct poor characteristics observed in the initial phases of shake-down operation. The drawings presented with this paper include these changes. Discussion
of these changes will be made in those sections concerning the parts involved.

V. GENERAL DESCRIPTION OF FATIGUE MACHINE

The fatigue machine is built around a six cylinder automotive engine. The head was removed and compression springs attached to the top surface of the pistons. The engine is driven through the crank shaft, and the reciprocating action of each of the pistons is used to compress a spring against one end of a lever arm. The action of the spring applies a force to that end of the lever arm. The arm is pivoted about a fulcrum and the specimen attached to the arm on the opposite side of the fulcrum such that the moment caused by the spring is resisted by a tensile force in the specimen. Thus, the force of the spring is transmitted through the lever arm to the specimen.

The machine will test six specimens simultaneously, one for each cylinder, at approximately 350 cycles of stress per minute. The arrangement is such that the maximum load applied to the specimens can be varied and the machine is capable of loading some of the specimens at one load level while the other specimens can be tested at another load level. Instrumentation is included to measure the time a particular specimen has been stressed and to monitor the force applied to the specimen.

The fatigue machine is divided into four assemblies: (1) table assembly, (2) lever arm assembly, (3) top plate, and (4) instrumentation. (See Figure 1 and Figure 2.) These assemblies and their individual components will be discussed in the following chapters.
Figure 1. Fatigue machine with instrumentation.
Figure 2. Fatigue machine showing the following assemblies: (1) table assembly, (2) lever arm assembly, and (3) top plate.
CHAPTER II

TABLE ASSEMBLY

The table assembly is composed of three main components: (1) the engine, (2) the table, and (3) the drive unit. This assembly serves as the base for the machine and provides the power and necessary reciprocating action for the operation of the lever arm assembly.

I. THE ENGINE

As mentioned earlier, the reciprocating action necessary for the operation of the lever arm assembly is achieved by driving a six cylinder automotive engine through the crank. The choice of an automotive engine and, in particular, the type having six cylinders in line, was made for several reasons:

1. The fatigue machine has no moving parts except those necessary to obtain reciprocating action. The engine contains its own lubrication system, thus eliminating lubrication problems.

2. Automotive engines are designed to resist fatigue failures for an infinite number of cycles. Replacement parts are readily available. The loads on the top of the piston, which are representative of the loads in the connecting rods, crank shaft, and bearings, are not as high in the present application as they are in the actual operation of the engine.

The mean effective pressure for this type of engine is approximately 150 psi which represents an average force of 1,500 pounds on the
top of the piston. Maximum firing pressure is approximately 600 psi which represents a maximum force of 6,000 pounds. The largest springs which will be used give a maximum of 2,000 pounds force when deflected 3.5 inches. Thus, the maximum force exerted by the spring is one-third of the maximum design load for the engine. The average force exerted by the spring is 1,000 pounds, which is two-thirds of the force caused by the average operating pressure. This means that life as represented by number of operating cycles of the engine used in the fatigue machine should be as long as the life of the engine in actual operation.

3. The six cylinder automotive engine has what is referred to as a balanced crank shaft. This means that the connecting rods from the pistons attach to the crank shaft at 120° intervals with each other. Thus, the moment about the center of the crank shaft is zero, except for friction and the unbalance due to variation in spring constants, if springs having the same spring constant are attached to one of the two sets of three pistons whose connecting rods attach to the crank shaft at 120° intervals. This fact greatly reduces the peak power requirement for the fatigue machine and eliminates the need for a large fly wheel. Thus, the only power required is that necessary to overcome the moment due to friction.

The engine is a 1954 Ford, 6 cylinders in-line type. This engine has a 3.625 inch bore and 3.5 inch stroke. The engine had been used previously, but it has been overhauled and the head, valves, and associated components removed prior to being installed in the fatigue machine.
II. THE TABLE

The table serves as a base to which the other components of the fatigue machine attach. The center to center distance for the pistons is 4.25 inches. This space limitation requires that the lever arm assemblies be alternately placed on one side and then the other. Thus, the table was built using the centerline of the engine as a centerline for the table. The working drawings for the table are shown in Figure 3 and Figure 4. The table was fabricated from materials heavy enough to ensure rigidity. The weight of the machine is such, and the vibrations are so small, that it is not necessary to bolt the machine down.

III. THE DRIVE UNIT

The maximum power necessary to drive the fatigue machine was estimated to be less than 5 horsepower. A 5 horsepower electric motor was chosen because it was readily available. The electric motor is rated at 440 volts and 6.95 amps. It operates on a 3 phase electrical supply and turns at 1,160 rpm. A belt drive was chosen because of ease of installation and simplicity of maintainence. The power is transmitted to the engine using two "A" type belts. A 5 inch adjustable sheave is attached to the electric motor and a 20 inch pulley is attached to the clutch end of the engine crank shaft. The pitch diameter of the sheave can be adjusted by changing the width of the belt grooves. The speed reduction from this arrangement is approximately 4 to 1 and it is possible to make adjustments in the speed of the fatigue machine.
Figure 3. Fatigue machine, table details.
Figure 4. Fatigue machine, table assembly.
CHAPTER III

LEVER ARM ASSEMBLY AND TOP PLATE

The lever arm assembly is that portion of the fatigue machine which multiplies the spring force produced by the reciprocating action of the pistons and applies the resulting tensile force to the specimen. As shown in Figure 5, the lever arm assembly can be divided into five parts or groups of parts: (1) springs and spring cups, (2) lever arm, (3) pedestal, (4) flexures, and (5) flexure mounts.

I. SPRINGS AND SPRING CUPS

Springs. The force initially applied to the end of the lever arm is developed by compressing a helical spring with an engine piston. From the mechanical arrangement of the machine, it is seen that the load in the specimen is directly proportional to the force applied by the spring. Thus, one of the methods used to vary the load in the specimen is to change the spring used to apply the force. To give a wide range of loads, four spring sizes were chosen. These springs, supplied by Rockwell-Standard Corp., Logansport, Indiana, were specially designed to give 500, 1,000, 1,500, and 2,000 pounds force when deflected 3.5 inches. Since the pistons have a fixed displacement of 3.5 inches, each spring is compressed this distance, and thus, the maximum spring force is fixed when a particular spring is installed. All of the springs have the same outside diameter but the lengths are not the same. A list of spring specifications is included in Appendix A.
Figure 5. Lever arm assembly showing the following parts: (1) spring and spring cups, (2) lever arm, (3) pedestal, (4) flexures, and (5) flexure mounts.
Spring cups. The attachment of the spring to the piston and to the end of the lever arm is accomplished through the use of spring cups. The arm spring cup is attached to the arm with one large bolt to allow easy removal for changing springs. A flat was milled on the end of the arm and an arm plate rigidly installed to provide a true bearing surface for the spring cup. Each spring cup is attached to a piston by grinding the top of the piston flat to mate with the base of the cup, and securing the two parts with four bolts.

The cups have a one-inch lip which extends over the outside diameter of the spring. The springs are secured in the cups by a small amount of initial compressive load. Due to the difference in length of the springs, it was necessary to provide spacer blocks which are inserted between the arm plate and the arm spring cup when the shorter springs are used. The spring cups, arm plate, and spacer blocks are shown in Figure 8.

Installation. The springs are installed in the assembled fatigue machine by rotating the crank shaft by hand until the piston to be worked on is in the bottom dead center position. The spring is set in the piston spring cup, the arm spring cup installed on the top and the spacer block and bolt installed. When the larger springs are installed, they must be initially compressed about one inch. The spring is compressed in a press and secured with clamps provided. A compressed spring with clamps is shown in Figure 6. The clamps are wired in place as a safety precaution. The spring can then be inserted in the piston spring cup and the arm spring cup installed. The crank shaft is rotated until the spring is compressed slightly beyond the initial compression for
Figure 6. Various springs and compressed spring with spring clamp installed.
installation and the wire and clamps removed. Springs are removed by reversing the installation procedure.

II. LEVER ARM

The lever arm is the portion of the assembly which transmits the moment caused by the force of the spring acting about the fulcrum and produces the force on the specimen. It consists of a steel bar, 3 inches in diameter, machined to assembly with the other parts. The details for the lever arm is shown in Figure 7, Detail 5. The maximum bending stress on the arm and the maximum deflection of the arm were calculated and found to be below critical values. The calculations are given in Appendix B.

The specimen attaches to the arm by inserting through one of two holes in the specimen end and securing with a nut. These two holes represent a means of adjusting the force in the specimen. The farther the specimen is from the fulcrum, the less the force required to counteract the moment caused by the spring. Thus, the specimen, if inserted in the hole closer to the fulcrum, will be subjected to a higher force. Calculations were made of the force exerted on the specimen and on the flexures when the specimen is in either of the two specimen holes and any of the four springs are used. The tabulated results are included in Appendix C.

III. PEDESTAL

Each pedestal is composed of a pedestal flexure mount and two pedestal sides. (See Figure 7, Details 7 and 8.) At the top, the pedestal sides are joined with a pedestal flexure mount, as shown in
Figure 9. The pedestal flexure mount has two specimen holes drilled in it which, on assembly, line up with the specimen holes in the lever arm. Thus, the specimen is attached to ground at the pedestal and stressed by the action of the lever arm. The pedestal also serves as a grounding device for the fulcrum. This means the moment caused the spring must be taken by the pedestal and by the joint between the pedestal sides and flexure mount. These parts were first positioned and joined by two half-inch bolts, after which four 3/4 inch holes were drilled and line reamed in each side and precision steel dowels inserted. Calculations included in Appendix D verify that this joint is sufficiently strong. The height of the pedestal was raised using pedestal space blocks such that the largest spring can be installed without the use of spring spacer blocks. The installation of springs is discussed in Section I. The pedestal attached to the table with eight 1/2 inch bolts.

When the machine was first assembled, each pedestal was secured only to the table. No other connections were made between the lever arm assemblies. When the machine was running, the pedestals experienced large deflections each time the spring applied force to the end of the lever arm. This deflection was such that the top of the pedestal moved in a direction away from the center of the table while the base of the pedestal remained fixed to the table. An attempt was made to reduce the deflection by typing together all pedestals on each side with a continuous tie bar across the top. It was felt that since high loads were not applied to the three pedestals on the same side at the same time, that those pedestals in the low end of the load cycle would help resist the deflection of the pedestal with the high load. Upon operation it was
observed that the deflections were reduced considerably but were still present to the extent that they were undesirable.

Since the deflection of all of the pedestals was away from the center of the table, it was then decided to tie the three pedestals on one side to the three pedestals on the other side. Thus, the forces causing the deflections would work against each other. A top plate was installed which tied the pedestals on the two sides together and, as an added margin, the plate was tied to the table. The installation of the top plate reduced the deflection of the pedestals to essentially zero. The top plate is discussed in Section VI.

IV. FLEXURES

The fulcrum about which the lever arm pivots is composed of two sets of flexures; one on each side of the lever arm. Each set of flexures consists of a vertical and horizontal flexure. (See Figure 7, Detail 6.) The flexure ends are rigidly fastened to the arm and the pedestal flexure mounts and the flexures cross at their mid-points making right angles with each other. (See Figure 9.) These flexures are thin strips of metal designed to transmit very little force parallel to the direction of the small dimension. The flexures are installed such that horizontal forces act in direction of the thin dimension of the vertical flexures and vertical forces act in the direction of the thin dimension of the horizontal flexures. Thus, the flexures offer very little resistance to the transfer of moments about an axis through the point where the flexures cross. Flexures accurately specify a pivot or fulcrum and at the same time have no moving parts which can wear out as the ball bearing or knife edge type fulcrums have.
Figure 7. Fatigue machine, lever arm details, sheet 1 of 2 sheets.
Figure 8. Fatigue machine, lever arm details, sheet 2 of 2 sheets.
Figure 9. Fatigue machine, lever arm assembly.
By putting a set of flexures on each side of the arm such that each set crosses at a point coincident with the center of the lever arm, an axis of rotation for the lever arm is specified. This axis is a line through the points where the flexures cross. Care was taken that all force applied to the arm acted in a direction perpendicular to that axis.

Since most of the loads applied to the arm are in the vertical direction, the vertical flexures were made heavier than the horizontal flexure. The horizontal flexures are mainly for positioning purposes and thus, the loads in them which may be either tensile or compressive, are relatively low. The loads in the vertical flexures are high but due to the mechanical arrangement of the machine, they are always compressive.

The flexures are fabricated from 17-4-PH Stainless Steel which is a precipitation hardening steel. The flexures were hardened to Rockwell C 45, which corresponds to an ultimate strength of 200,000 psi. The maximum stress in the flexures occurs when the 2,000 pound springs are used and is approximately 88,000 psi compressive. This stress is below the ultimate strength of the material and the buckling stress for the part, since stresses in the flexures are compressive, there should be no problem with fatigue failures of the flexures.

The flexures are attached to the pedestal and to the lever arm through the use of flexure mounts which are discussed in the next section. They are attached to these mounts by securing the flexure in a corner with a bolt inserted diagonally through the corner of the flexure and the flexure mount. Thus, in those joints where the forces
between the flexures and the flexure mounts are high, compressive type joints result.

V. FLEXURE MOUNTS

Flexure mounts are used to attach the flexures to the lever arm and to the ground or pedestal. The method of attachment of the flexures to the mounts was discussed in Section IV. The flexure mount on the pedestal was designed as part of the pedestal while the mount for the arm was separate from the arm and installed on the arm.

Arm flexure mounts. The first design of the arm flexure mounts is shown in Figure 10. This mount was installed with the lever arm extending through the large vertical slot and a bolt through the top of the mount securing the two together. The bottom lip for the vertical flexure failed at its base on two of these mounts when the 2,000 pound spring was used. This represented a force of 11,000 pounds on the lip. It is felt that the failures were due to the high stress concentration factor resulting from the sharp corner factor at that point. The calculated stress at that point without the consideration of the stress concentration factor was 44,000 pounds per square inch. This high stress indicates an error in the original design.

The arm flexure mount block was redesigned as shown in Figure 7, Details 9 and 10. Much more material was included in the flexure lip and the whole piece was made more rigid. The sharp corners at the base of the lips giving rise to the high stress concentration were eliminated by including a large radius and then inserting a corner block to give a bearing corner for the flexure. Calculations showed the stress on the lip is reduced considerably by the change. Additional corrective
Figure 10. Arm flexure mount, first design.
measures were taken by shot-peening the radius and adjoining material. Shot-peening (6) is an operation whereby the surface of a material is bombarded with small steel shot. This procedure covers the surface with a layer of compressive stress a few thousands of an inch thick and acts to protect the surface from the formation of tensile cracks. If the load at which the tensile cracks form can be raised, the fatigue life of a material can be shown to increase.

The new arm flexure mounts were installed by inserting the lever arm through a close fitting hole in the arm flexure mount. The two parts are secured together with two precision steel dowels inserted in holes drilled and reamed in place. The close fit between the flexure mount and the lever create an effective joint to transfer the moment. The dowels are used primarily for positioning.

Pedestal flexure mounts. The pedestal flexure mounts now being used are those originally designed. (See Figure 7, Detail 7.) These flexure mounts also include the sharp corner on the lips similar to the old arm flexure mounts. However, there is more material in the pedestal flexure mounts which prevent failures in the lips. When the arm flexure mounts were redesigned, it was decided to rework the pedestal flexure mounts. The corners of the lips were rounded to relieve the stress concentrations and then the corners and adjacent material were shot-peened.

VI. TOP PLATE

As discussed in Section III, a top plate was added to prevent the deflection of the pedestals. This top plate is a large steel plate 5/8 inches thick having the same external dimensions as the overall
dimensions of the table. The center lines of the plate coincide with the center lines of the table. This top plate attaches to the top of each of the pedestals and the center of the top plate is fastened to the center of the table using two vertical tie down plates. Figures 11 and 12 show the top plate details.

The mechanical arrangement of lever arm assembly is such that when a specimen fails, the lever arm to which it was attached is free to ride up and down on the spring instead of compressing it. Unless this violent oscillation is suppressed it will damage the flexures and possibly other parts of the machine. To prevent this from happening, one and one-half inch holes were drilled and tapped in the top plate over each of the pistons. Bolts are threaded in these holes to a point such that the top of the lever arm during operation of the machine just clears the bottom of the bolt. When the specimen fails, the arm comes to rest against the bottom of the bolt and the force of the spring is transmitted directly to the top plate.
Figure 11. Fatigue machine, top plate details, sheet 1 of 2 sheets.
Figure 12. Fatigue machine, top plate details, sheet 2 of 2 sheets.
CHAPTER IV
INSTRUMENTATION

There were certain features which are included with the machine to measure certain parameters of operation. Included in these items are the timing system, the load measurement system, and the control console.

I. TIMING

The measurement of the number of cycles of stress for a particular specimen is accomplished by measuring the number of cycles of stress per minute and the total time of stress cycling. The number of cycles per minute is found by measuring the rotational speed of the engine with a tachometer. The time of stress cycling is measured using an electric clock and a shut off device that stops the clock when the specimen fails. The shut off block is shown in Figure 13. The shut off device is a block through which the specimen is inserted on installation in the fatigue machine. The specimen holes in the pedestal and tie bar are oversized. Thus, the specimen can be aligned in the machine by correctly positioning the shut off block over the specimen hole.

To the underside of the block are attached light springs which hold the block up off the tie bar when no force is applied to the top of the block. There is also a micro-switch which is normally open in this up position. When the specimen is tightened on installation, the load compresses the springs which seats the block firmly on the tie bar.
Figure 13. Shut off block.

Figure 14. Specimens installed in lever arms.
This closes the micro-switch and when the control switches are turned on, starts the clock. When the specimen fails, the load holding the block down is released and the block springs to the up position. This opens the micro-switch and shuts off the clock. Control switches provided make it possible to turn any or all of the clocks off and on as required. A wiring diagram of the timing system is shown in Figure 15.

II. LOAD MEASUREMENT

The dynamic load on the specimen is measured with electrical, resistance type, strain gages on the upper and lower surfaces of the lever arm at the points of maximum bending stress. The bending stress in the lever arm is directly proportional to the load on the specimen and this stress can be measured with the strain gages. The strain gages were installed in a four active element Wheatstone bridge to measure bending stress. The installation used on the lever arms is shown in Figure 16.

A Bridge Amplifier and Meter, Model BAM-1, for measuring strain at the gage locations and a Switch and Balance, Model BSG-6, to facilitate the measurement of more than one channel of information were purchased from Ellis Associates, Pelham, New York.

Using the calibrated strain gages on the lever arm it is possible to make dynamic measurements of specimen loads by connecting the output of the Bridge Amplifier to an oscilloscope or oscillograph. The strain gage system on the lever arm is calibrated using a calibration specimen prepared for that purpose. The calibration specimen has a larger cross section area than the specimens which would normally be used in the fatigue machine. (See Figure 17.) Four resistance type strain gages
Figure 15. Wiring diagram for timing system.

Figure 16. Wiring diagram and location of strain gages for load measurement system.
were attached to the specimen at its midpoint and equally spaced around its circumference. The calibration set up is shown in Figure 18. The procedure for calibration and load measurement are included in Appendix G.

III. CONSOLE

The clocks and switches for the timing mechanism and a plug board for the leads from the strain gages were installed on a console. This console shown in Figure 19, provides a central location for the fatigue machine instrumentation.
Figure 17. Calibration specimen.

Figure 18. Calibration setup.
Figure 19. Console showing the following components: (1) timing instrumentation, (2) strain gage plug board, (3) bridge amplifier and meter, and (4) switch and balance.
The purpose of the work discussed in this chapter is to provide information pertaining to the operational characteristics of the fatigue machine. The work involved is divided into three areas: (1) specimen stress field analysis, (2) vibrational analysis, and (3) actual operation.

I. SPECIMEN STRESS FIELD ANALYSIS

The fatigue machine is designed to apply an axial tensile force to a specimen. Due to rotation of the lever arm, deflections and misalignments, bending stresses are induced in the specimen and superimposed on the axial tensile stresses. In order to obtain meaningful results in a tensile test, it is necessary that this bending stress be maintained at a low level. Tests were conducted to determine the magnitude of the bending stress induced by the machine.

To test for bending, a calibration specimen with four active strain gages was installed in the machine. A description of the calibration specimen is included in Appendix G. Measurements were taken of the stress at various locations on the specimen as the load was applied. The load was applied dynamically by operating the machine and statically by turning the machine over by hand. The read out from the various strain gages was observed on an oscilloscope. Since the total resultant axial load due to bending is zero, the average of the stresses in various
locations for each load was assumed to be due to the axial load. The percentage difference of the values at different positions from the average value was taken as a measure of the effect of bending at that point.

Measurements of the stress were taken at intervals as the specimen was rotated in the specimen hole and the same load applied each time. From these trials it was possible to pick the positions of maximum and minimum stresses. These stresses usually occurred 180° apart and on a line coincident with the centerline of the lever arm. The maximum stress usually occurred at the point on the specimen farthest from the fulcrum, which is where the maximum displacement would occur. Representative values of the percentage bending for both the dynamic and static situations are included in Appendix H. From these it can be seen that there is little difference between the two cases. The average percentage bending observed was 10 per cent.

From observations taken under various conditions, the following were observed to affect the amount of bending:

1. **Initial angle between lever arm and horizontal.** The larger the initial angle the greater the bending stresses. Bending moment is applied as the specimen was stressed and the nuts tried to seat on non-parallel surfaces.

2. **Preload.** High preload on the specimen increased the initial angle of the lever arm and thus increased the bending.

3. **Maximum load.** The percentage bending decreased as the maximum load on the specimen increased. The higher loads allowed the specimen to deflect and resist more of the load as tensile stresses.
4. **Specimen hole.** The percentage bending is greater for the inside specimen hole. This hole is closer to the fulcrum, thus, the elongation of the specimen causes a greater rotation of the lever arm.

5. **Specimen alignment.** The alignment of the specimen in the specimen holes was found to affect the bending stress.

The amount of bending was measured on a specimen 0.50 inches in diameter. This is a larger cross section than is proposed for test specimens. Test specimens having a square cross section, 0.30 inches on a side, have been used in the machine and are proposed for future use. The same bending moment acting on both the calibration specimen and the test specimen causes a maximum axial bending stress in the calibration specimen which is three times the bending stress in the square test specimen. Thus, if the bending load on the two different specimens is the same for the same axial stress, the effect of bending in the test specimen should be one-third that observed in the calibration specimen.

It is felt that the effect of bending can be further reduced by determining the best initial lever arm angle and adjusting the dimensions on the lever arm assembly to obtain this angle. The lever arm angle can be controlled by the height of the spring and spring cup portion of the assembly. This can easily be adjusted with shims between the arm plate and arm spring cup. The bending can possibly be reduced by changing the method of attachment of the specimen to the pedestal and lever arm. The resulting method of attachment should be such that the specimen is allowed to align properly without applying bending. A possible solution may be something on the order of a ball and socket or a very stiff spring such as a Belleville spring.
The total axial load on the specimen during actual operation was compared to the calculated values for all springs and specimen hole combinations. The results were seen to compare favorably. The variation between the actual and calculated loads is most likely due to the fact that the springs used vary somewhat from their rated force when compressed.

The trace of the total axial load in the specimen during actual operation was observed. The load is applied approximately sinusoidally with no discontinuities or sudden load impacts.

II. VIBRATIONAL ANALYSIS

A vibrational analysis was made to determine if the natural frequencies of the components of the machine were within the range of the operating frequencies. It was found that all critical components are sufficiently rigid to have natural frequencies far greater than the operating frequency of 5 to 7 cycles per second.

The lever arm assembly is that portion of the machine most likely to vibrate at low frequencies. The lowest natural frequency of this system was calculated, (see Appendix I) and was found to be 250 cps.

These calculations and visual observation indicate that the vibration of the machine or its components will have no effect on its mechanical operation.

III. ACTUAL OPERATION

Following its final assembly, the fatigue machine was operated continuously for 300 hours. During operation, visual observation indicated that all components of the machine functioned properly. The machine runs with negligible vibration.
The goal of this work, as described in Chapter I, was achieved. Conclusions and recommendations concerning this study are listed below.

I. CONCLUSIONS

The conclusions which may be drawn from this work are:

1. The mechanical arrangement used for this machine is suited for the operation.

2. The fatigue machine, in its final assembled form, will perform satisfactorily; no future failure of parts is expected.

II. RECOMMENDATIONS

The following recommendations are made concerning the fatigue machine described in this paper and future fatigue machines of this type.

1. The study of bending in the specimen should be continued and a method of specimen attachment to reduce the effect of bending should be devised.

2. In future machines the three pedestals on one side should be machined from one piece of material, possibly a casting. This would reduce machining and increase rigidity.

3. All flexure mounts should have the same type of flexure attachment as in the latest design for the arm flexure mounts.


APPENDIX
APPENDIX A

SPRING SPECIFICATIONS

Approx. 500 lbs. at 3-1/2" deflection -
.375" centerless ground AISI-5160 steel
3-7/16" o.d. +/- 1/32"
8-1/2 coils
7-11/16" free length +/- 1/8"
Rate - 136 to 150-lbs. per inch

Approx. 1000 lbs. at 3-1/2" deflection -
.437" centerless ground AISI-5160 steel
3-7/16" o.d. +/- 1/32"
8-1/2 coils
7-23/32" free length +/- 1/8"
Rate - 272 to 300 lbs. per inch

Approx. 1500 lbs at 3-1/2" deflection -
.500" centerless ground AISI-5160 steel
3-7/16" o.d. +/- 1/32"
9.8 coils
8-27/32" free length +/- 1/8"
Rate - 408 to 452 lbs. per inch

Approx. 2000 lbs. at 3-1/2" deflection -
.531" centerless ground AISI-5160 steel
3-7/16" o.d. +/- 1/32"
9.8 coils
9-1/8" free length +/- 1/8"
Rate - 544 to 600 lbs. per inch

All Springs have the following Specifications:
Magnaglo bar before coiling
Ends closed and ground square
Harden and draw Rc 48 - 52
Press
Shot peen and heat 450° max.
Must have full coverage
100 per cent magnaglo inspection
Must be free from all tool marks, nicks and surface defects
Oil finish
APPENDIX B
CALCULATIONS FOR LEVER ARM

The formulae for the axial bending stress in a beam and for the deflection of a cantilever beam with the load applied at the free end are

\[ S = \frac{M y}{I} \]

and

\[ \Delta = \frac{F L}{3 E I} \]

where

- \( S \) is the bending stress at a particular section
- \( M \) is the moment at that section
- \( y \) is the distance from the neutral axis at which the stress occurs
- \( I \) is the moment of inertia of the section about the neutral axis
- \( \Delta \) is the deflection of the free end
- \( P \) is the load applied at the free end
- \( L \) is the length of the beam
- \( E \) is the modulus of elasticity of the material in the beam.

Using the above formulae and assuming the arm to be a cantilever beam, built in at the specimen-fulcrum end, the maximum axial bending stress was found to be 15,000 psi. The arm is made of tool steel with an ultimate strength of 100,000 psi. The endurance limit of a material is approximately one half the ultimate strength. Thus, the stress in the arm is well below the endurance limit of the material. The bending deflection was found to be 0.044 inches.
APPENDIX C

FORCES IN SPECIMEN AND FLEXURES

The forces on the specimen and the flexures were calculated for the various specimen hole and spring combinations. The actual forces on the specimen were measured after the machine was completed.

<table>
<thead>
<tr>
<th>Spring Size (Pounds)</th>
<th>Specimen Hole</th>
<th>Calculated Flexure Force (Pounds)</th>
<th>Calculated Specimen Force (Pounds)</th>
<th>Measured Specimen Force (Pounds)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2,000</td>
<td>Inside</td>
<td>22,000</td>
<td>19,900</td>
<td>19,000</td>
</tr>
<tr>
<td>2,000</td>
<td>Outside</td>
<td>14,000</td>
<td>12,000</td>
<td>11,500</td>
</tr>
<tr>
<td>1,500</td>
<td>Inside</td>
<td>16,500</td>
<td>15,000</td>
<td>15,400</td>
</tr>
<tr>
<td>1,500</td>
<td>Outside</td>
<td>10,400</td>
<td>8,900</td>
<td>9,100</td>
</tr>
<tr>
<td>1,000</td>
<td>Inside</td>
<td>11,000</td>
<td>10,000</td>
<td>9,400</td>
</tr>
<tr>
<td>1,000</td>
<td>Outside</td>
<td>7,000</td>
<td>6,000</td>
<td>6,000</td>
</tr>
<tr>
<td>500</td>
<td>Inside</td>
<td>5,500</td>
<td>5,000</td>
<td>4,700</td>
</tr>
<tr>
<td>500</td>
<td>Outside</td>
<td>3,500</td>
<td>3,000</td>
<td>2,900</td>
</tr>
</tbody>
</table>
Using the method outlined by Timoshenko and MacCullough (6) the shear stress in the pins between the pedestal sides and the pedestal flexure mount was determined. When the 2,000 pound spring is used, the total moment on the joint is 44,000 inch-pounds. This represents the highest load condition. The load calculations were made using the following assumptions: (1) The joint on one side of the pedestal resists half of the moment. (2) The centroid of the pin pattern is at the center of the half inch bolt. (See Figure 9.) and (3) The bolt takes none of the load. The formulae used are listed below.

\[ F' = \frac{P}{n} \]

and

\[ F'' = \frac{MR_n^2}{2R_n} \]

where

- \( F' \) is the force on one pin in the direction of the force causing the moment (i.e., direct shear force)
- \( P \) is the force causing the moment on the joint
- \( n \) is the number of pins in the joint
- \( F'' \) is the force on a pin in a direction perpendicular to a line from the centroid of the pin pattern to the pin
- \( M \) is the moment about the centroid of the joint
- \( R_n \) is the distance from the centroid of the bolt pattern to any pin

The vector addition of \( F' \) and \( F'' \) results in the total force on a particular pin. \((F_n = F'_n + F''_n)\)

\[ F' = \frac{1,000}{4} = 250 \text{ lb for all four pins} \]
Adding vectorially

\[ F_A = 3,800 \text{ lb} \quad F_B = 4,180 \text{ lb} \quad F_C = 3,800 \text{ lb} \quad F_D = 3,700 \text{ lb} \]

The cross sectional area of the 3/4 inch diameter pins used in the joint is 0.44 inches\(^2\). Therefore the maximum shearing stress on the pins used in the pedestal side and flexure mount joint is 9,500 psi. Under dynamic loads the shear stress is less than or equal to the normal stress. The pins used have a minimum ultimate stress of 50,000 psi which means the endurance limit for the pins is more than twice the applied stress.
APPENDIX E

CALIBRATION PROCEDURE AND LOAD MEASUREMENT

1. Calibrate the calibration specimen\(^1\) by comparing the read out from the four strain gages connected in series to a known load applied to the specimen by a tension testing machine.

2. Insert the calibrated specimen in the fatigue machine and complete the required wiring using the four specimen gages in series. Wire both the lever arm bridge and the calibration specimen bridge into the Switch and Balance. Connect the Switch and Balance to the Bridge Amplifier.\(^2\)

3. Balance the calibration specimen bridge and calibrate the read out using the proper constant.

4. Balance the lever arm bridge.

5. Using a hydraulic jack, apply a force to the bottom of the lever arm midway between the flexures and the spring cup end. Observe load on the calibration specimen using the meter on the Bridge Amplifier to prevent overloading of the system.

6. When the desired load is reached, switch to the lever arm bridge and adjust the amplifier gain to achieve the same reading.

7. Remove the load and push the calibration knob on the Bridge

---

\(^1\)The calibration specimen test section is 0.50 inches in diameter by 6 inches long. Four ELH SR-4 strain gages are attached to the middle of the specimen and equally spaced around its circumference. The strain gages attached are type AD-7 with a gage factor of 1.97 \(\pm\) 2 per cent and a gage resistance of 119.5 \(\pm\) 0.3 ohms.

\(^2\)For complete operation instructions refer to instruction manuals for these two instruments.
Amplifier and note the calibration constant for the lever arm bridge.

8. The calibration is repeated for each specimen hole at various load ranges.

9. After the lever arm bridge has been calibrated, the load on a specimen is measured by using the proper calibration constant and measuring the output of the lever arm bridge.
APPENDIX F

SPECIMEN STRESS FIELD DATA

The measurements were taken in the positions shown.

To change the load measurement to a stress, divide the load by the area of the calibration specimen, 0.196 in\(^2\).

![Diagram of specimen and spring end of lever arm]

<table>
<thead>
<tr>
<th>Position</th>
<th>Load in 1,000 lb.</th>
<th>Deviation from average</th>
<th>Percent Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>RUN 1. Dynamic, 2,000 lb spring, Outside specimen hole</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>12.2</td>
<td>+.6</td>
<td>+5</td>
</tr>
<tr>
<td>B</td>
<td>11.5</td>
<td>-.1</td>
<td>-1</td>
</tr>
<tr>
<td>C</td>
<td>10.8</td>
<td>-.8</td>
<td>-7</td>
</tr>
<tr>
<td>D</td>
<td>11.9</td>
<td>+.3</td>
<td>+3</td>
</tr>
<tr>
<td>RUN 2. Static 2,000 lb spring, Outside specimen hole</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>12.1</td>
<td>+.7</td>
<td>+6</td>
</tr>
<tr>
<td>B</td>
<td>11.6</td>
<td>+.2</td>
<td>+2</td>
</tr>
<tr>
<td>C</td>
<td>10.5</td>
<td>-.9</td>
<td>-8</td>
</tr>
<tr>
<td>D</td>
<td>11.4</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>RUN 3. Dynamic, 500 lb spring, Outside specimen hole</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>3.3</td>
<td>+.3</td>
<td>+10</td>
</tr>
<tr>
<td>B</td>
<td>3.2</td>
<td>+.2</td>
<td>+7</td>
</tr>
<tr>
<td>C</td>
<td>2.5</td>
<td>-.5</td>
<td>-17</td>
</tr>
<tr>
<td>D</td>
<td>3.0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>RUN 4. Static, 500 lb spring, Outside specimen hole</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>3.3</td>
<td>+.4</td>
<td>+14</td>
</tr>
<tr>
<td>E</td>
<td>3.0</td>
<td>+.1</td>
<td>+3</td>
</tr>
<tr>
<td>C</td>
<td>2.4</td>
<td>-.5</td>
<td>-17</td>
</tr>
<tr>
<td>D</td>
<td>2.8</td>
<td>-.1</td>
<td>-3</td>
</tr>
</tbody>
</table>
APPENDIX G
VIBRATION CALCULATIONS

The natural frequencies of the lever arm assembly were calculated using an approximation for the system involved. The calculation was made by assuming the lever to be a rigid mass supported on three springs. This system was given two degrees of freedom, the vertical displacement x of the center of gravity of the lever arm and the rotation θ of the lever arm about its center of gravity. The items represented by springs are the specimen, the flexures and the spring attached to the end of the lever arm.

The approximated system is shown below. The spring constants K for the specimen and the flexures were calculated using the elasticity of these pieces; the spring constant for the spring was already known. The mass M, the location of the center of gravity, and the mass moment of inertia J about the center of gravity for the lever arm, arm flexure mount, and spring cup combined were calculated using the weights and shapes of the pieces.
The radius of gyration \( r \) of the lever arm combination about its center of gravity is \( \frac{J}{M} \), thus \( r^2 \) was found to be 61 in\(^2\).

The equations of motion for translation and rotation are

\[
F_x = M \ a
\]

and

\[
M' \ = \ J \ \gamma
\]

where

- \( F_x \) is a force in the \( X \) direction acting on a body
- \( M \) is the mass of the body
- \( a \) is the acceleration of the body in the \( X \) direction
- \( M' \) is a moment about the center of gravity of a body
- \( J \) is the mass moment of inertia of the body about its center of gravity
- \( \gamma \) is the angular acceleration of the body about its center of gravity

Assuming small oscillations and undamped motion, the equations of motion of the lever arm system are as follows:

1. \( M \ddot{x} - K_1(x - L_1 \theta) - K_2(x - L_2 \theta) - K_3(x + L_3 \theta) \)

and

2. \( J \ddot{\theta} = K_1 L_1(x - L_1 \theta) - K_2 L_2(x - L_2 \theta) - K_3 L_3(x + L_3 \theta) \)

where

\[
\ddot{x} = a = \frac{d^2x}{dt^2} \quad \text{and} \quad \ddot{\theta} = \gamma = \frac{d^2\theta}{dt^2}
\]

Equations 1 and 2 can be reduced rearranging and substitution

3. \( \ddot{x} + AX = B \theta = 0 \)

4. \( \dddot{\theta} + \frac{B}{x^2} x = C \theta = 0 \)

where

5. \( A = \frac{K_1 + K_2 + K_3}{M} \)
6. \[ B = \frac{K_3L_3 - K_2L_2 - K_1L_1}{M} \]
and
7. \[ C = \frac{K_1L_1^2 + K_2L_2^2 + K_3L_3^2}{r^2M} \]

The principal modes of vibration to occur the motions in the X and θ directions must be harmonic at the same frequency, therefore the solutions to the two differential equations are of the form

8. \[ x = X \cos\omega t \]
and

9. \[ \theta = \theta \cos\omega t. \]

Substituting equations 8 and 9 into equations 3 and 4, the following equations are obtained:

10. \[ (A - \omega^2)X + B\theta = 0 \]
and

11. \[ \frac{B}{r^2}X + (C - \omega^2)\theta = 0 \]

Solving for the amplitude ratio \( \frac{X}{\theta} \) in equations 10 and 11 one equation in one unknown \( \omega \) is obtained.

12. \[ \frac{X}{\theta} = -\frac{B}{A - \omega^2} = -\frac{C - \omega^2}{\frac{B}{r^2}} \]

Solving for \( \omega^2 \) in the above equation

13. \[ \omega^2_{1,2} = \frac{1}{2} (A+C) \pm \sqrt{\frac{1}{4} (A - C)^2 + \frac{B^2}{r^2}} \]

Substituting in the numeric values of A, B, and C, the values for \( \omega_1 \) and \( \omega_2 \) were found to be 1,600 and 3,500 radians per second respectively. The frequency in cycles per second is equal to \( \frac{\omega}{2\pi} \). Thus the natural frequencies for the lever are system were found to be 250 cps and 590 cps using a two degree of freedom approximation.
THE DESIGN OF A SIX SPECIMEN
TENSILE FATIGUE MACHINE

An Abstract of
a Thesis
Presented to the
Department of Mechanical Engineering
Brigham Young University

In Partial Fulfillment
of the Requirements for the Degree
Master of Science

by
Charles Mark Percival
August, 1961
ABSTRACT

The purpose of this study was to develop a high force, tensile fatigue machine capable of testing large statistical groups of specimens. The work included the design of the machine and supervision of its fabrication. Load measurement devices were designed and calibrated, and a brief operational analysis of the machine was made.

The system made use of the reciprocating action of an automotive engine when driven through the crank shaft. Attaching a spring holder to the top of each piston made it possible to use the linear displacement of the piston to compress a spring. When the spring is compressed, it applied a force to one end of a lever arm.

The system is arranged to test one specimen for each cylinder of a six cylinder automotive engine. This type of engine was chosen because it has a balanced crank shaft. That is, each of the connecting rods from the pistons are attached to the crank at 120° intervals. Assuming that there is no friction and that all springs have the same spring constant, the moment on the crank due to the spring forces would be zero. Thus, the peak power that would be required to drive the machine is greatly reduced.

Steel flexures are used as a fulcrum for the lever arm. The moment induced by the spring forces is resisted by attaching a specimen to the arm at a point on the opposite side of the fulcrum. This arrangement allows the load level in the specimen to be varied by changing
the spring used to apply the force to the lever arm, and by changing the
distance of the specimen from the fulcrum.

The load on the specimen is directly proportional to the strain
cau sed by the bending load or stress in the lever arm. Thus, measure­
ment of the load in the specimen is accomplished by applying strain gages
to the lever arm. An arrangement is made whereby the time of stress
cycling was measured using electric clocks.

The work also included an operational analysis of the fatigue
machine. This consisted of instrumentation calibration, an analysis of
the stresses induced in the specimen by the machine and a study of the
vibrational characteristics of the machine.

APPROVED: