A MECHANISM FOR CONTROLLING LARGE-SCALE
FATIGUE TESTING MACHINES

TECHNICAL REPORT
to the
OFFICE OF NAVAL RESEARCH
Contract N6091-71, Task Order V
Project NR-031-182

By
W. J. Hall
and
G. K. Sinnamon

DEPARTMENT OF CIVIL ENGINEERING
UNIVERSITY OF ILLINOIS
URBANA, ILLINOIS
A MECHANISM FOR CONTROLLING LARGE-SCALE
FATIGUE TESTING MACHINES

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W. J. Hall and G. K. Sinnamon

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A MECHANISM FOR CONTROLLING LARGE-SCALE
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I. INTRODUCTION

A. General Discussion

With the advent of light weight structural elements and increased speeds, the fatigue life of materials has become of great importance. However, progress in fatigue design studies has been retarded by the lack of suitable theories and by the difficulty of relating the basic fatigue data of materials to their uses in service. In other words, the S-N curve and endurance limit, which are derived under constant load testing conditions, may not be directly applicable to service conditions where the members are subjected to a more or less random pattern of load variations. Recently, in the search for an explanation, there has been increasing interest in the manner in which damage accumulates in the progress of a fatigue test. A number of hypotheses have been suggested to account for the cumulative damaging effects which are produced when a material is subjected to varying amplitudes of stress. The ultimate aim of these hypotheses is to arrive at a theory which will enable the designer to estimate the endurance life of a material under any stress-cycle pattern.

To substantiate a proposed theory it is advisable to run physical tests. Up to the present time most fatigue tests have been performed under conditions of constant maximum and minimum load during the application of the cycles of stress. Thus, to prove (or perhaps disprove) any cumulative damage theory, and to simulate more closely actual service conditions in fatigue tests, there is a critical need for a fatigue testing machine which can vary the maximum and minimum load according to a predetermined pattern.
A diagram of the 50,000-lb. lever-type fatigue testing machine, used extensively at the University of Illinois for a cumulative damage program, is shown in Fig. 1. This machine, which has a lever arm ratio of approximately 10 to 1, was designed to operate with a constant displacement. If in the course of a test a change in load is desired, it is necessary to stop the machine and adjust manually the turnbuckle and the eccentric to obtain the desired load. This procedure, one of trial and error, is tedious and time consuming.

Therefore, the purpose of the investigation described in this report was to design and fabricate a control mechanism which would replace or modify the turnbuckle and variable eccentric of a 50,000-lb. machine and which would be capable of functioning automatically with the machine in operation. In the course of the investigation a number of mechanisms were studied. These are described in Section II. The completed control mechanism consists of a geared nut assembly to replace the turnbuckle and a two-armed linkage mechanism which is used to modify the displacement provided by the eccentric. Future plans call for the installation of electronic controls, which, with the aid of a programming device, will make it possible to automatically subject the fatigue specimen to a predetermined systematic variation of loads over a period of time. In this manner it is anticipated that the fatigue test conditions can be made to approximate more closely the actual service conditions and thereby help to give a further insight into the fatigue phenomena.

B. Acknowledgment

This report is based in part on a dissertation presented for a degree of Master of Science in the Department of Civil Engineering, prepared by W. J. Hall,
under the supervision of W. H. Munse, Research Assistant Professor of Civil Engineering, and G. K. Sinnamon, Research Associate in Civil Engineering, and under the general direction of N. M. Newmark, Research Professor of Structural Engineering. This work was performed as part of a program on "Fundamental Criteria for the Selection of Structural Metals" sponsored by the Office of Naval Research as Task V of Contract N6or1-71, Project Designation NR-031-182, in the Engineering Experiment Station of the University of Illinois.
II. MACHINE DESIGN

A. Basic Design Requirements

The basic requirement was to design a control mechanism (or combination of control mechanisms) for the 50,000-lb. lever-type fatigue testing machine which would be capable of varying both the maximum and the minimum load while the fatigue testing machine was in operation. This could be accomplished by a single mechanism which would change the maximum or minimum load separately or together, or by a combination of two mechanisms which would change the range and the mean load independently. The latter arrangement was adopted in the completed design.

It might be well at this point to define the terms "mean load" and "range", as they will be used frequently throughout the remainder of this report. The range, \( R \), as illustrated in Fig. 2, is related to the eccentricity or throw of the machine and is the algebraic difference between the maximum and minimum load, whereas the mean load is the algebraic average of the maximum and minimum load. In the basic 50,000-lb. fatigue testing machine shown in Fig. 1, the range is fixed by the amount of eccentricity, and the mean load is adjusted by the turnbuckle; both settings remain constant while the machine is running. In the new control mechanism both the range and mean load can be adjusted while the machine is in operation.

Another primary requirement in the design of the control mechanism was that the loading curve be smooth. Figure 3 shows the theoretical displacement curves of the completed mechanism for nearly maximum and minimum range with a fixed setting of the eccentric. These curves practically coincide with a sine curve and were considered to be satisfactory.
The load requirements for the parts of the control mechanism are not severe. The 50,000-lb. fatigue machine for which the mechanism was designed has a lever arm ratio of 10 to 1. Therefore, the mechanism must be capable of withstanding an axial cyclic load of ± 5000 lb. Nevertheless, it should be realized that in general the parts for a fatigue machine must be carefully designed since the machine must be capable of testing a large number of specimens in fatigue and still not fail in fatigue itself.

B. Mechanisms Considered

In the search for suitable controlling mechanisms, a number of devices were studied. A brief description of these devices with the reasons for their acceptance or rejection follows.

1. Range and Mean Load Control Combined

Linkage Mechanism. The first device considered for use as a control apparatus was the two-armed linkage mechanism, similar to that shown schematically in Fig. 3. This mechanism was developed by Professor Newmark, but no designs had been previously completed. The original plan was to have both the mean load and range controlled by proper orientation of the moving point M. However, a detailed investigation showed the following: (1) To vary the range and mean load (with minimum load held constant) point M followed a circular path oriented approximately the same as the circular path shown in Fig. 3, but with the center at the extreme right position of the travel of point C; (2) To vary the range and mean load (with maximum load held constant) point M also followed a circular path, but the center was at the extreme left position of the travel of point C; (3) To vary the mean load only (the range remaining constant) point M followed a spiral-shaped curve which was oriented approximately at a right angle to the aforementioned circular paths; (4) To
vary the range only (mean load remaining constant) point M followed a nearly circular path with the center close to the mid-point of the travel of point C. As is pointed out later, this feature was utilized in the final design.

Thus, it appeared that for the linkage mechanism to control both the range and mean load simultaneously would require that the point M be equipped for random two dimensional movement. In view of this requirement it appeared advisable to discontinue the development of a control mechanism utilizing a single scheme to control both the range and mean load.

2. Range Control

a. Auxiliary Beam with Sliding Fulcrum. This mechanism would consist of a second short walking beam, called the auxiliary beam, mounted parallel to and beneath the main walking beam. One end of the auxiliary beam would be fastened by a vertical member to point A in Fig. 1, and the other end would be connected by a rod to an eccentric. As considered, the auxiliary beam was fitted with a sliding fulcrum which, when moved, would vary the lever arm ratio of the auxiliary beam. Thus, as the fulcrum was moved, the amount of eccentricity transmitted to the main walking beam would be varied. This mechanism was discarded primarily because of the large number of mechanical fits -- large cumulative mechanical tolerances must be avoided in fatigue machines whenever possible. Secondly, no satisfactory method was found for moving and locking the sliding fulcrum with respect to the auxiliary beam.

b. Planetary Gear Arrangement. This type of range control would vary the eccentricity by means of a gear arrangement mounted within the eccentric itself. The gears would be operated by an electric motor which would rotate with the eccentric and which would be supplied with power through slip rings. The planetary gear arrangement was eliminated from consideration for reasons of expense and the difficulties of fabricating it in the Civil Engineering shop.
c. Hydraulic Mechanism. The possibility of designing a hydraulic control mechanism to vary the eccentricity was considered but never exploited primarily because it would require a completely new set of controls and equipment.

d. Linkage Mechanism. The two-armed linkage mechanism, shown schematically in Fig. 3, is a product of the original linkage mechanism considered for control of both the range and mean load (see this section, part 1), and was selected as the range control mechanism in the completed design. It was selected primarily because its principles of operation appeared satisfactory and secondly because it could, for the most part, be built quickly and economically. The linkage mechanism is described in detail in Section III.

3. Mean Load Control

a. Hinged Platform. A hinged platform was considered in connection with the auxiliary beam range control (this section, part 2a.) This arrangement consisted of a platform upon which the auxiliary beam and accessories, including the eccentric, were mounted. The platform would be hinged at the eccentric end and raised or lowered at the other end by means of a screw mechanism or some other device. Although this was a suitable method for varying the mean stress, it was discarded with the auxiliary beam mechanism since it involved a revamping of the entire machine base.

b. Hydraulic Mechanism. The possibility of designing a hydraulic device to control the mean load was considered but never developed for the same reason as given previously in this section, part 2c.

c. Geared Nut Assembly. The geared nut unit is essentially a turnbuckle similar in operation to the turnbuckle shown in Fig. 1. However, in the geared nut assembly a stationary threaded stud projects vertically from the dynamometer. A nut, rotated by a circumferential worm gear, moves along this threaded stud. This system essentially changes the length of the eccentric rod and thereby the mean load. This mechanism was accepted in the completed design for use with
the linkage mechanism primarily because it appeared to function suitably, and secondly because it could, with the exception of the gears, be rapidly and economically made. A detailed description of this assembly is presented in Section III.
III. DESCRIPTION OF THE COMPLETED CONTROL MECHANISM

A. Geared Nut Assembly -- Mean Load Control

The geared nut assembly derives its name from the fact that it consists of a threaded sleeve (actually a long nut) which is rotated by a worm gear mounted on its circumference. The geared nut moves along a stationary threaded stud projecting vertically from the dynamometer. This action results in a change in overall length of the assembly, which in turn varies the mean load.

The geared nut assembly, a compact box-shaped device hanging directly beneath the end of the walking beam, is shown in its completed form in Fig. 4.

The +5000 lb. vertical cyclic thrust on this geared nut is carried by two Rollway T-22 roller-type thrust bearings, one mounted on the top and one mounted on the bottom of the geared nut. These thrust bearings are obscured in the photographs by the grease retainers. Any transverse thrust caused by oscillatory movement of the mechanism is carried by two Oilite (powdered metal) sleeve bearings mounted in the top and bottom support plates.

The vertical stud has 6 threads per inch and the circumferential worm gear has 120 teeth (single acting). Consequently, the sensitivity of the geared nut assembly amounts to $\frac{1}{720}$ or 0.00139 inch vertical movement of the screw per revolution of the worm. In its present form the assembly is being hand operated. Eventually, the installation of an electric motor will allow the system to be automatically controlled. If this motor operates at 1750 rpm, with a 16 to 1 gear speed reducer, the system will provide an overall change in length of 1 1/2 inches in 10 minutes. Obviously, the time limit on the vertical travel of the screw can be altered by changing the gear reduction ratio or motor speed. When the machine is in operation the geared nut assembly undergoes an oscillatory motion. Therefore, it will be necessary to incorporate a flexible shaft, made
by utilizing two Hookes joints at right angles, and a slide between the assembly and the speed reduction unit.

E. Mechanical Linkage Mechanism -- Range Control

The linkage mechanism controls the effective vertical throw of the machine and thereby the range of load. As is shown in Fig. 3, schematically, the short connection arm "b" is attached at one end to an eccentric with a fixed amount of eccentricity. The mechanism was designed on the basis of a one-inch fixed eccentricity, though the amount may be varied between zero and one inch if desired. The longer connecting arm "a" is pivoted at its base (point M) on a pin which is supported by two quadrants. These two quadrants make it possible to move point M along a circular path as indicated by the dotted lines. From point C, the pinned connection of the two linkage arms, there is a vertical connection (not shown in Fig. 3) to the dynamometer. As the eccentric revolves, it imparts a cyclic motion to arm "b" and consequently to point C. The amount of effective vertical motion imparted through point C to the vertical assembly is controlled by the position of point M along its circular arc. The linkage mechanism operates in such a manner that when point M is moved the range is changed, but the mean load is not affected. Figure 3 shows the theoretical cyclic motion of point C for two positions of point M. These curves differ only minutely from sine curves.

The mechanism is shown in its completed form in Fig. 5.

The quadrants, as shown in Fig. 5, are supported at their tops by the A-Frames. The two quadrants are rotated by means of a circular worm gear segment mounted between the quadrants at their base. In its present condition the quadrant gear arrangement is hand operated. Eventually the gear segment will be operated through the medium of a gear speed reducing unit and electric motor, which in turn will be run by an automatic control and programming system.
Oilite sleeve bearings are used throughout the linkage mechanism. These sleeves, backed by a lube oil reservoir, are oil saturated and theoretically become self-lubricating as the bearings become warm. This was primarily an experiment to see if the sleeve bearings would function as well, or better than, roller bearings which has a tendency to Brinell when there is no complete rotation but instead a rocking action.

The quadrants are stabilized at the bottom by brass friction bearings which bear against the outside of the quadrants. These bearings prevent the tendency for a lateral translation of the quadrant bases caused by rotation of the worm gear. Lateral stability at the top of the linkage mechanism is afforded by having the connecting pin in the linkage arm junction bear against the heads of the pins which support the quadrants. This is clearly shown in Fig. 5. The in-line action of these pins produces one of the outstanding features of the linkage mechanism, namely, that a change in the load range by rotating the quadrants produces no appreciable change in the mean load.

C. Fabrication, Assembly, and Operation

One of the compelling reasons for adopting the geared nut and linkage mechanism combination was that it could, with the exception of some gear and bearing parts, be fabricated and maintained in the laboratory shop.

On the whole, tolerances were necessarily close since all slack in the parts manifests itself in the form of a knock when the load range is from tension to compression. Excessive knocking is not desirable because it provides an impact loading in addition to the normal smooth cyclic dynamic loading. It is only fair to say here that one of the major tolerance problems presented itself in the design of the screw and nut for the geared nut assembly. Even though the screw fit is extremely tight when first fabricated, screw slack
which may develop as the threads wear in (on the order of one or two-thousandths of an inch) produces a serious problem. The elimination of this bothersome factor may necessitate, at some future date, a spring or hydraulic loading device for the geared nut assembly.

The A-Frame and channel base assembly now appears to provide adequate stiffness for the machine. However, it was necessary to add several stiffeners not originally contemplated. Notable of these are several diaphragm stiffeners to prevent the eccentric support (transverse I-beam and top plate between two adjacent machines) from rotating excessively during each cycle of operation if the machine.

Assembly of the parts presented no particular problem. The geared nut unit was assembled in the shop and then placed on the machine as a unit. Assembly of the linkage mechanism proceeded in the following order: one quadrant with support pin, lower pin for long connecting arm "a", other quadrant with support pin, linkage arm "b", center pull block (connection to dynamometer) and central pin, and finally the eccentric stud which has both a right and left hand thread. A general view of the control mechanism is shown in Fig. 6.

The ring dynamometer shown in Fig. 4 is a load measuring device incorporating wire resistance strain gages to measure the surface strains and a mechanical gage to measure the change in the inside diameter of the ring. The strains are read by means of a Baldwin SR-4 Strain Indicator. An Oscilloscope is utilized when the machine is operating. Figure 7 shows a view of the control mechanism and the instrument table.

It is contemplated that the load pick-up device for the automatically controlled machine will consist of a magnetic-gage type dynamometer. This is essentially a ring dynamometer with the magnetic pick-up unit mounted in the ring. It is anticipated that the automatic control system will be able to
program the fatigue machine through a predetermined systematic variation of stress cycles.

At present the machine is undergoing extensive testing, with the controls being operated by hand. Every indication to date has been that the mechanism will fulfill all the requirements established for the original design.
Fig. 1 Diagram of the 50,000-lb. Lever-type Fatigue Testing Machine
Key:
M.L. = Mean Load
R. = Range

Fig. 2  Typical Patterns of Load Variation in Fatigue Testing
Fig. 3 Schematic Views of Linkage Mechanism and Shape of Displacement Curve for Two Positions of Connecting Arm "a"
Fig. 4 View of Geared Nut Assembly and Ring Dynamometer

Fig. 5 View of Linkage Mechanism
Fig. 6  General View of Control Mechanism

Fig. 7  View of Control Mechanism and Instrument Table
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