A 1 kHz SERVOHYDRAULIC FATIGUE TESTING SYSTEM

Jill M. Morgan[1] and Walter W. Milligan[2]

1) MTS Systems Corporation
14000 Technology Drive, Eden Prairie, MN 55344

2) Department of Metallurgical and Materials Engineering
Michigan Technological University, Houghton, MI 49931

Abstract

High cycle fatigue testing to long lives requires high frequency equipment. For example, a 10^9 cycle test takes over seven months in a traditional 50 Hz servohydraulic testing machine, but only 11 days at 1,000 Hz. In response to these needs, and in order to study frequency and temperature effects in the vibrational loading regime, a servohydraulic system capable of load-controlled testing at 1,000 Hz and high temperature has been developed. This paper describes modifications which are necessary to the load frame and servovalve design, the control algorithms, and the calibration procedures. It also describes an approach to measuring crack opening displacement at high temperature and 1,000 Hz, and some preliminary results.

Published in proceedings of the conference “High Cycle Fatigue of Structural Materials” edited by W.O. Soboyejo and T.S. Srivatsan, TMS, Warrendale PA, 1997, pp. 305-312.

Introduction

Significant interest has developed recently in high cycle fatigue testing of aircraft engine structural materials. This has been driven by the US Air Force, which is in search of new methodologies for life prediction and design against high cycle fatigue failure. Typically, fatigue problems emanate from a damage site, such as a compressor blade airfoil which has been damaged by a bird strike or an ingested hard particle, or a dovetail face which has been damaged by fretting. Fatigue failure occurs when these damaged sites lead to crack propagation, driven by vibrational stresses, and usually at high mean stresses but low stress amplitudes. Since the vibrational frequencies are high and the stress amplitudes are low, real data is scarce. Lifetimes of up to 10^9 cycles are of interest. These lifetimes are difficult to come by in standard testing machines. As an example, a fatigue test conducted to 10^9 cycles at 50 Hz in a traditional servohydraulic testing machine would take over 7 months (231 days) to complete. Few laboratories can afford to tie up a servohydraulic machine for 7 months in order to obtain just one data point, and the likelihood of completing a 7 month test without a mechanical failure or power outage is very low.

In response to these needs, and in order to study frequency effects at high frequencies, new types of machines are being developed. Several different design philosophies are being explored, but documentation is lacking since the efforts are all recently initiated. One approach being developed at Southwest Research Institute is to set up a resonance condition using piezoelectric drivers [1]. Frequencies near 2 kHz can be achieved, but frequency is limited to the resonance point and is a function of the specimen and grips. At Wright Patterson Air Force Base, two approaches are being pursued [2]: in the first approach, electrodynamic “shakers” are used in conjunction with pneumatic chambers (which apply the mean stress); in the second approach, prototypes are being developed which utilize a magnetostrictive material as an actuator. The “shakers” run at 350 - 600 Hz, depending on the size of the machine, the grips and specimen. The magnetostrictive actuator machine runs at about 2 kHz. Finally, in the approach described in this paper, servohydraulic machines are being modified to run at frequencies up to 1 kHz. Each of these design approaches has advantages and disadvantages when compared to the alternatives. For each design, the durability remains to be established.

This paper describes a servohydraulic machine, capable of 1 kHz, which has been developed by MTS Systems Corporation. The system will be integrated for high temperature fatigue testing and crack growth testing at Michigan Technological University. Crack opening displacement will be measured by a capacitive displacement probe which is capable of measuring displacement in the kHz frequency range and at temperatures up to 1000°C. At the time of this paper, the machine had been constructed and had successfully accomplished several demanding room-temperature test sequences.

High Frequency Servohydraulic Fatigue Testing

Evolution of the Present System

MTS developed expertise in high frequency testing due to a market need in the elastomeric materials community. Elastomers utilized for vibration damping materials (such as automobile engine mounting fixtures) must be characterized for damping behavior over a range of frequencies. Typically, these “elastomer systems” perform short duration bursts of varying load levels, while sweeping through a wide range of frequencies, from 20 to 1,000 Hz. These systems are designed to keep load frame resonance frequencies out of the full range of frequency sweeping. To avoid resonance, a large seismic-mass base and short columns are used. These frames are clearly different in physical appearance from the more traditional MTS 250 kN (50 kip) systems which are generally used for metals testing.

The first kHz-regime machine designed by MTS specifically for high cycle fatigue testing was based on the elastomer system. The changes to the system were subtle. Dual, as opposed to single, high frequency servovalves were used to achieve higher flow rates resulting in actuator displacements of +/-0.065 mm at 1,000 Hz. Although the frequencies utilized by the elastomer researchers are high, the duration of each test is generally short, so little information is currently available about the expected durability of these systems in an HCF environment.

More recently, the traditional 250 kN frame was fitted with a high performance, high flow "voice coil" servovalve. (Servovalve issues are discussed in the next section.) The goal behind this new combination of servovalve and frame type was to design a low cost, high frequency system capable of achieving higher actuator displacements in the vicinity of 1,000 Hz. To meet these requirements, the traditional 250 kN frame was redesigned. The height of the base plate was reduced to limit deflection and increase the stiffness of the frame, thus minimizing vibration effects. The column height was also reduced to eliminate any high frequency "tuning fork" effects. A tie bar across the top of the columns was added for stability. The crosshead was modified to tighten tolerances, increase stiffness and achieve better alignment. The actuator was coated with a wear resistant ceramic coating, to prevent potential wear of the actuator (rod banding). With these changes, the system is capable of approximately +/-0.1 mm of displacement and dynamic loads of around +/-20 kN at 1,000 Hz.

A note regarding the resonance frequencies of the system: the design intention of the present system was to develop a discrete high frequency system for metals applications. Unlike the elastomer rigs, the modified 250 kN system was not designed for resonance-free operation over a full range of testing frequencies. Frame resonant frequencies do exist. They are variable and are dependent upon the particular system configuration, in particular crosshead height and grip mass. Once the test configuration is determined, a frequency sweep can be performed to determine the resonance-free areas of operation. In the MTU system, with the axial grips installed, resonance points exist at approximately 630 and 930 Hz.

Servovalve Issues

The most critical design decision in the present system was the choice of the appropriate servovalve. Two goals were agreed upon: first, to increase the flow rate and response speed to obtain large deflections at 1,000 Hz; and second, to prevent possible premature fatigue failure of the servovalve. In machines that can run 10^9 cycles in 11 days, fatigue failure of the testing machine is a real concern.

A cross section of a traditional servovalve is shown in Figure 1. In the first stage, a torque motor is used to bend the so-called "flexure tube" and move the "nozzle flapper." The subsequent pressure change due to the movement of the flapper controls spool position and ultimately actuator displacement. The expected life of traditional servovalves is "greater than a billion cycles", but the concern was that the flexure tube might fail after only one or two long-term HCF tests. Further, the flow capacity of the nozzle flapper valve is limited. Therefore, it was decided that an alternative approach would be pursued.

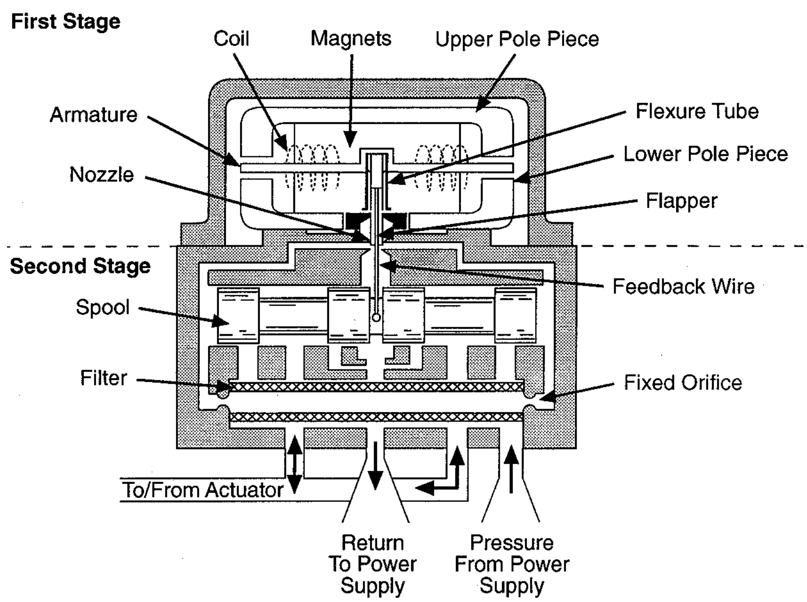

Figure 1. Cross section of a traditional servovalve (MTS model 252). The flapper and flexure tube may be subject to fatigue failure.

The approach used in the current system was a "voice coil" servovalve, shown schematically in Figure 2. The voice coil valve uses an electrodynamic pilot stage, driven in the same way as an audio speaker. The voice coil moves a spool in the pilot stage back and forth, and this spool acts exactly like the spool in a traditional servovalve. Hydraulic fluid from the pilot stage is used to drive the spool in the main stage servovalve. Essentially, the main stage acts as a hydraulic amplifier of the pilot stage. There is no torque tube or flapper, and the only parts which are mechanically stressed are the coil springs in the pilot stage.

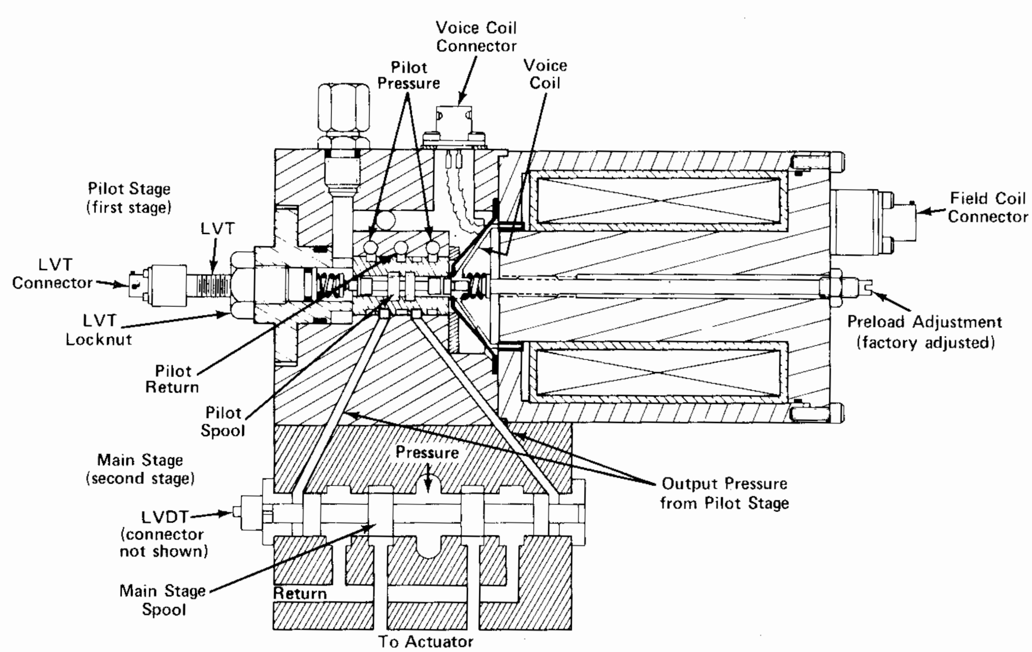

Figure 2. Cross section of a voice coil servovalve (MTS model 257). The only mechanically-stressed parts are the coil springs in the pilot stage.

In addition to the expected increase in durability, the voice coil valve has superior high frequency response and higher flow rates, characteristics which are both crucial to achieving larger actuator displacements at very high frequencies. Yet further consideration was given to the expected modes of failure. Failure of the flapper valve is likely to occur by rupturing of the internal flexure, the aftermath being a high pressure fine spray of oil in the test area. When the voice coil valve fails, however, there is no external rupturing; the oil is self-contained. The failure would most likely be observed as an error detected in the outer-loop control circuit, as the programmed load end levels would not be maintained.

Acceleration Compensation of the Load Cell

Acceleration effects due to moving mass of the load train are significant at high frequencies and must be compensated for in the control loop. Actual acceleration measurements are discussed in the “preliminary results” section of this paper. The inaccuracy in the load signal, if the load cell is not compensated for acceleration effects, has been determined to be approximately 25% at 1,000 Hz with the axial grips. There are two methods of compensating for acceleration, each of which utilizes an accelerometer positioned on the active elements of the load cell.

The first method is quick and simple. The system is run at frequency in displacement control with only the grip masses attached. With no specimen inserted and the load train open, the acceleration effects can be seen as load output from the system load cell. The affiliated load cell acceleration signal is then conditioned and inverted. The user then gradually sums this inverted signal into the system load cell feedback signal via potentiometer adjustment or a digital slide bar, depending on the controller being used. This is done until no further load output is monitored. This method brings the load values to within 8% of actual values.

The second and preferred method is essentially a dynamic load verification test with a strain gaged specimen closing the load train. The specimen strains are first statically calibrated as a function of load using the system load cell. An initial test is then performed at actual test conditions. The difference between the system load cell signal and that from the measured specimen strain is monitored. Via the same procedure as in the first method, the acceleration compensation circuit is then adjusted until the difference between the two load signals is nulled.

The Controller

The digital “TestStar” controller which runs traditional servohydraulic systems also runs the 1 kHz system; the only modifications are the acceleration compensation boards, and the use of an adaptive control scheme which yields steady-state load control at 1,000 Hz. The digital controller utilizes a patented phase amplitude compensator (PAC) to detect and correct amplitude roll-off and phase lag in sinusoidal command waveforms. An analog controller version, based on the MTS 458 system, is also under development.

Grips, Fixtures, and Loading Strategies

One advantage of the servohydraulic approach to high frequency fatigue testing is that very few modifications to standard fatigue lab procedures are necessary. The MTU system has axial grips that are capable of fully reversed loading and high temperature testing with induction heating. Traditional pin-and-clevis grips for compact tension specimens should work as well. There are only two constraints on grips and fixtures: first, the mass of the grips should be minimized for optimum system response; and second, the length of the load train should be short, to minimize load frame motion and potential resonance. The axial grips add about one kilogram of mass to the actuator, and it would not be advisable to use grips that are much bulkier than these.

Much of the FCP research in this program will be accomplished in 4-point bending. The reasons for this are: the experimental setup is simple and easy to align; the grips and fixtures weigh less than 400 grams; and it is possible to conduct Mode II and Mixed Mode I/II testing by using an asymmetric 4-point bending geometry [3].

Measurement of Crack Length in FCP Testing at 1,000 Hz and High Temperature

A major experimental challenge at high temperatures and high frequencies is measurement of crack length. Because of the high temperatures, techniques like back-face strain gage and surface-mounted metal foils that measure remaining ligament length are not possible. Because of the high frequencies and temperatures, traditional crack opening displacement (COD) gages will not work. Measuring compliance from actuator displacement and load may be possible, but this technique may lack the necessary sensitivity, and it may be necessary to slow the test down (or stop it completely) to accurately measure the compliance. This is not desirable in high temperature testing, because slowing down or stopping the test will influence time-dependent and rate-dependent processes, and therefore the crack growth rates.

It is highly desirable to measure crack length dynamically and with as little lag time as possible. Two techniques appear to be promising. The first is the potential drop technique [4,5], and the second is the use of a capacitive COD probe. Neither of these have been demonstrated at 1,000 Hz, and the technological challenges associated with developing them are not trivial. Both will require significant development and testing, and it is not known at this time what degree of success will be attainable. In this research program, the use of the non-contact capacitive COD probe will be explored first. The system is described below.

Capacitec, Inc [6] manufactures non-contact displacement probes capable of temperatures up to 1,000^{o}C and frequencies up to 5 kHz. Depending on the probe diameter, the absolute accuracy can be as small as 0.25 μm. The probes measure the gap between the tip of the probe and any metal plate which is connected to ground. The unit provides a 0-10 V DC signal that can be calibrated to displacement. It is important to note that a “wide-band” amplifier must be specially ordered from the manufacturer to apply the capacitive approach to 1 kHz frequencies [6]. A schematic of the intended configuration in 4-point bending is shown in Figure 3. The probes can also be used with compact tension specimens, or as simple axial extensometers. A major challenge will be developing mounting procedures for the probes that do not shake off during 1,000 Hz loading.

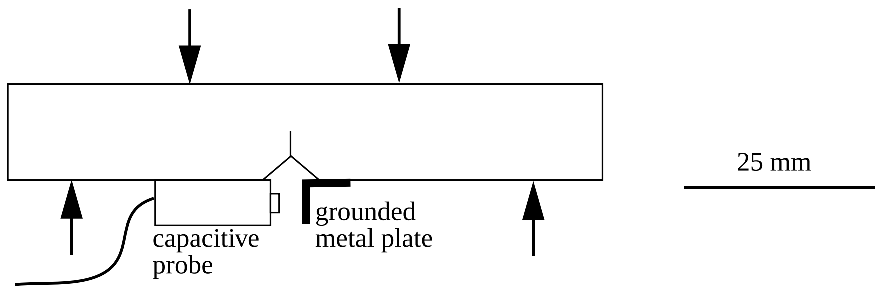

Figure 3. Schematic diagram of the 4-point bend geometry and the capacitive COD probe.

During fatigue testing, a dedicated PC with a 200 kHz, 16 bit data acquisition card will be used to acquire the DC waveform corresponding to COD. Analysis software and calibration data will be used to calculate crack length from COD and provide a 0-10 V DC "crack length" signal to the digital controller which is running the fatigue test. In this way, K-controlled fatigue testing at 1,000 Hz and 1,000^{o}C should be possible.

Preliminary Results

A 1020 steel specimen with the geometry shown in Figure 3 was cycled in 4-point bending at 1,000 Hz for 4 minutes. The test was designed and conducted only as a demonstration of the 4-point bend geometry at 1,000 Hz, and so ASTM procedures for pre-cracking and testing were not followed. The specimen was notched with a 250 μm thick water-cooled abrasive wheel, and then loaded at $\Delta K \approx 10$ MPa$\sqrt{m}$ and R = 0.1. The test ran well, with no apparent fretting under the loading pins. About 2 mm of crack growth was observed in 4 minutes, which is in the expected range for this steel and stress intensity factor [7]. From this test, it appears that bend testing at 1,000 Hz should be achievable. Crack length was not monitored by the capacitive probe, and the development of this technique is currently underway.

A 4140 steel smooth-bar specimen was cycled in the axial grips, between 5 and 55 MPa (100 to 1,000 lbs). This test was intended to demonstrate the feasibility of testing to 10^9 cycles with stable long-term control and durability. The test ran flawlessly for over 8 days, to 840 million cycles. The load waveform at the end of the test was identical to that at the beginning of the test. At 840 million cycles, the machine shut itself down due to a hydraulic system interlock, apparently related to an irregularity in the incoming cooling water pressure. The test was successfully re-started after the interlock was reset, and continued until 970 million cycles, at which time the test room was no longer available and the test was manually aborted.

During this test, accelerometers were mounted in various places on the load frame. The acceleration output on the crosshead and base plate measured 10.5 g's. Those on the upper grip attached directly to the load cell showed 25 g's of acceleration. The largest accelerations were measured on the lower grip mass attached directly to the moving actuator rod. These were found to be as high as 121 g's. As per the earlier discussion regarding acceleration compensation of the load cell, the effect due to the moving mass of the load train are significant. The uncompensated vs compensated load comparisons are very dependent on the specimen spring rate, load and frequency test conditions, and grip mass. For the conditions of this particular test, the difference between uncompensated and compensated values was found to be up to 25%.

After the completion of the long-term test, various components of the load frame were inspected. The actuator rod was examined with a profilometer and microscope. There were no indications of rod banding. The servovalve was taken off and the various step responses and valve specifications were re-verified on the test bench. The springs were also inspected. All conditions were normal. The load cell was disassembled for inspection as well and was found to be fine. This test leads to some confidence that long-term 1,000 Hz testing will be feasible.

Summary

HCF testing to 10^9 cycles at room temperature and 1,000 Hz has been achieved in the current system, with no apparent deterioration. FCP testing in 4-point bending has been demonstrated. Current efforts are aimed at developing a capacitive COD probe which will be capable of measuring crack length at 1,000^{o}C and 1,000 Hz.

Acknowledgments

This work was supported by the MURI on "High Cycle Fatigue", funded at Michigan Technological University by the Air Force Office of Scientific Research, Grant No. F49620-96-1-0478, through a subcontract from the University of California at Berkeley.

References

1. D.L. Davidson, Southwest Research Institute, San Antonio, TX, private communication, 1997.

2. T. Nicholas, Air Force Research Laboratory, Wright-Patterson AFB, OH, private communication, 1997.

3. S. Suresh, C.F. Shih, A. Morrone, and N.P. O'Dowd, "Mixed-Mode Fracture Toughness of Ceramic Materials," J. Am. Ceram. Soc., 73 (1990), 1257-1267.

4. J.K. Donald and J. Ruschau, "Direct Current Potential Difference Fatigue Crack Measurement Techniques", *in* Fatigue Crack Measurements: Techniques and Applications", K.J. Marsh, R.A. Smith and R.O. Ritchie, eds., EMAS, West Midlands UK, (1991), 11-37.

5. R.H. Van Stone and T.L. Richardson, "Potential Drop Monitoring of Cracks in Surface Flawed Specimens", *in* Automated Test Methods for Fracture and Fatigue Crack Growth", ASTM STP 877, W.H. Cullen, R.W. Landgraf, L.R. Kalsand and J.H. Underwood, eds., American Society for Testing and Materials, Philadelphia, PA, (1985), 148-166.

6. Jeffrey R. Peduzzi, Capacitec Inc, Ayer, MA, 01432, Phone (508) 772-6033, private communication, 1997.

7. R.W. Hertzberg, Deformation and Fracture Mechanics of Engineering Materials, 3rd Ed., Wiley, New York (1989), 555.