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MINIATURE LINEAR-TO-ROTARY MOTION ACTUATOR

Michael R. Sorokach, Jr.*

ABSTRACT

A miniature hydraulic actuation system capable of converting linear actuator motion to control surface rotary motion has been designed for application to active controls on dynamic wind tunnel models. Due to space constraints and the torque requirements of an oscillating control surface at frequencies up to 50 Hertz, a new actuation system was developed to meet research objectives. This new actuation system was designed and developed to overcome the output torque limitations and fluid loss/sealing difficulties associated with an existing vane type actuator. Static control surface deflections and dynamic control surface oscillations through a given angle are provided by the actuation system. The actuator design has been incorporated into a transonic flutter model with an active trailing edge flap and two active spoilers. The model is scheduled for testing in the LaRC 16 Foot Transonic Dynamics Tunnel during Summer 1993. This paper will discuss the actuation system, its design, development difficulties, test results, and application to aerospace vehicles.

DESIGN CRITERIA

The actuation system had to meet the following requirements in order to accomplish research objectives:

- Actuation system operating range from 0-50 Hertz for control surface deflections of 0 to ± 15 degrees.
- A maximum of ± 0.3 degrees rotary motion play in the system over the frequency range.
- Mechanically efficient design.
- Compact design due to space restrictions.
- Maximum actuator operating pressure of 1000 psi.
- Reliable and simple to maintain.

* NASA Langley Research Center, Hampton, VA

- ♦ Conform to the design guidelines and safety requirements of the Wind Tunnel Model Systems Criteria (NASA Langley LHB 1710.15).

MECHANISM DESCRIPTION

The actuation system consists of a linear, piston driven, hydraulic actuator with direct load measurement instrumentation and a linear-to-rotary motion conversion insert. Figure 1 is a schematic of the actuator assembly identifying its components. The actuator consists of its body, piston, two end cap subassemblies (composed of the end cap, an air bleeding screw, fitting, and supply line) for enclosing the system, a drive link subassembly (consisting of the drive link, bearing, journal, and retaining ring), forward and aft drive link constraints which are used to restrict the drive link to linear motion, o-rings for sealing the system, and attachment screws. Two miniature pressure transducers capable of monitoring servo loads are incorporated into the end caps. Figure 2 shows the actuator, both assembled and disassembled, and its components. The actuator is a single-degree-of-freedom mechanism allowing drive link translation along the piston axis.

Figure 3 is a schematic showing the manner in which the linear motion of the piston/drive link arrangement is produced. The cutaway sections shown on the actuator assembly are located along its centerline. A servovalve is connected to the actuator by means of two supply lines, each of which is brazed to an end cap, and the necessary fittings. The end caps channel the flow of hydraulic fluid into the left and right side chambers of the actuator body. The chambers are round holes bored in-line through the actuator body to accommodate the piston. Piston length regulates the amount of linear travel for the drive link. The actuator body is relieved between the two chambers to allow for drive link translation. The amount of relief between the two chambers on the actuator body is larger than the combined chamber clearance between the end of the piston and end cap on both sides. As a result, the piston will bottom out on the end caps when full actuator throw is encountered. This prevents damage to the drive link from occurring since it cannot make hard contact with the sidewalls of the actuator body relief at maximum travel. An o-ring on both ends of the piston is used to seal each chamber at the actuator body relief/chamber interfaces.

To initiate drive link motion, the system is first pressurized to 1000 psi. The servovalve then pulsates a small volume of hydraulic fluid into each chamber at alternating time intervals. The alternating flow of fluid between the two chambers causes the piston to translate in a reciprocating manner. The drive link follows the same motion as the piston since it is attached to the piston. The desired frequency and displacement of drive link oscillation is controlled by a closed loop feedback system which adjusts the time interval and mass flow rate of fluid pulsation into each chamber.

The linear, reciprocating motion of the actuator is converted to rotary motion for control surface actuation through a linear-to-rotary motion conversion insert. This insert has a high precision, helical trough machined in it and the insert is incorporated into the actuated control surface as part of its leading edge. Figure 4 is a view showing the actuator assembly and the linear-to-rotary conversion insert attached to the model trailing edge control surface. The control surface is mounted to the wing on two needle roller bearings. Nylon shims, located at the flap/wing interface, are used as a bearing to restrict axial free play of the control surface. As the drive link assembly reciprocates linearly, the helical trough on the insert converts the actuator linear motion into rotary motion of the control surface about its hinge line. The frequency of drive link reciprocation is equal to the rate of control surface oscillation. This actuation system is kinematically equivalent to a reverse cam mechanism, i.e., linear motion is converted to rotary motion.

The bearing which is attached to the drive link rides inside the helical trough on the insert during actuation. The bearing is fabricated to a very close running fit (.0001-.0002 inch clearance) with the helical trough to reduce the free play between the two components. Any excessive free play between the bearing and helical trough will introduce undesirable rotary free play into the system. The bearing is fabricated from graphitic molybdenum-alloy tool steel (ASTM A681, Type 06) and is tempered to Rc 61-64. The linear-to-rotary conversion insert is fabricated from PH 13-8 Mo stainless steel, heat treated to Rc 43-45. As a result, any high points on the contacting surfaces of the helical trough will wear to form a continuous helix upon initial operation of the actuator. This "wearing-in" of the helical trough will reduce the rotary play and help to sustain the rotary play in the system to a constant value. The combination of high hardness and the inherent lubricating properties of the bearing help to reduce its wear.

Constraining the rotation of the drive link about the piston axis (see Figure 1) is another factor which reduces rotary play in the actuation system. Restricting the drive link to linear motion is accomplished by reducing the cumulative tolerance buildup between the drive link and the contacting surfaces which prohibit any drive link rotation (forward and aft drive link constraints and the upper surface of the actuator body relief). These mating surfaces actually make contact with the drive link. A nickel alloy plating (NEDOX*) is applied to all surfaces having relative motion to improve their surface hardness (Rc 63-67) and lubricity, thus decreasing galling and wear. The internal bore in the actuator body which houses the piston is also plated to help decrease o-ring wear, thus prolonging seal life. Since the piston o-rings exhibit inherent compressibility, they are the only components of the actuator assembly which introduce rotary play into the system. The other location where rotary play occurs in the system is at the interface between the bearing and helical trough. The

* General Magnaplate Corporation, Linden, New Jersey

cumulative rotary play in the actuation system was measured during testing and determined to be ± 0.3 degree.

Actuator loads are monitored by the means of two miniature pressure transducers. Each transducer is located on an end cap and monitors the pressure load on its respective chamber (see Figure 2). The hinge line torque output of the actuation system can be calculated from these pressure load measurements. Any pressure fluctuation in the chambers can be determined and compensated for by the servovalve during operation.

DESIGN OVERVIEW

The actuation system is designed to conform to the requirements of the Wind Tunnel Model Systems Criteria (NASA Langley LHB 1710.15). This document addresses various design guidelines and safety requirements for models and model systems which are to be tested at the Langley Research Center facilities.

The weakest link in the actuation system is the journal which supports the bearing on the drive link subassembly (see Figure 1). Journal failure during wind tunnel testing would result in no damage to the facility since the journal is not a structural attachment for the control surface. The control surface is hinged to the wing body and would remain attached to the wing in the event of journal failure. The journal is fabricated from 18 Ni 300 grade maraging steel. Under normal wind tunnel testing operating conditions (20 Hertz oscillation for control surface deflections of ± 5 degrees) the worst case stress, including stress concentration factors, results in a safety factor of 23 on the material ultimate strength. For the upper limit operating conditions (50 Hertz oscillation for control surface deflections of ± 15 degrees), the worst case stress results in a safety factor of 1.2 on material ultimate strength. The upper limit operating conditions of the actuation system will not be exercised during wind tunnel testing. However, the actuation system may be operated to obtain short throw, high frequency control surface deflections (± 5 degrees at 40-50 Hertz) during testing. The worst case stress for this scenario results in a safety factor of 3.7 on the material ultimate strength.

Figure 5 shows the results of a fatigue analysis performed on the journal for normal wind tunnel testing operating conditions and upper limit operating conditions. This analysis conforms to the requirements of NASA Langley LHB 1710.15 which assumes a strength reduction factor of 0.5 to account for surface finish, reliability, etc. The analysis is therefore conservative. As stated previously, the upper limit operating conditions will not be applied during wind tunnel testing. For short throw, high frequency control surface deflections (± 5 degrees at 40-50 Hertz), journal life is increased by a factor of 406 as compared with journal life at the upper limit operating conditions.

The actuator body, end caps, and forward and aft drive link constraints are fabricated from 15-5 PH stainless steel, heat treated to Rc 35-38. To eliminate any difficulties associated with galling, the piston and drive link are fabricated from PH 13-8 Mo stainless steel, heat treated to Rc 43-45. As discussed previously, all contacting surfaces with relative motion are nickel alloy plated to decrease wear. All fasteners are 180 ksi minimum tensile strength and are preloaded to manufacturer's specified torque.

The transonic flutter model for which the actuator design has been incorporated is shown in Figure 6. Cover plates and the upper spoiler are removed to show the access compartments which house the actuators and instrumentation. The model is equipped with an active trailing edge flap and two active spoilers. Fifty-eight pressure transducers are installed chord wise at 60 percent model span (mid-span on the control surfaces) for dynamic pressure distribution measurement. Additionally, 17 pressure transducers are installed at 40 percent model span, slightly inboard of the control surfaces, to measure any unsteady pressure distributions due to control surface actuation. Accelerometers are mounted on the wing panel and control surfaces to measure model displacements and control surface dynamic response during actuation. Rotary potentiometers are used to monitor control surface angular displacements.

Figure 7 is a detail of the actuators showing their locations and method of control surface actuation. As discussed previously, the flap actuator converts its linear motion to control surface rotary motion through the leading edge insert on the flap. The spoiler actuator consists of two servos incorporated into a common body and is similar in design to the flap actuator. The upper and lower spoilers are independently actuated and static deflections or dynamic control surface oscillations about a given angle are provided by the actuator. The chord wise, linear motion of the spoiler actuator is converted into upper and lower spoiler rotary motion through the control arm shown on section B-B in figure 7. Maximum spoiler deployment is 45 degrees from the undeflected position as shown in figure 7. Rotary play in the spoiler actuator is not a major concern due to geometry and since control surface deflections are parallel to the plane of actuation. The spoilers are fabricated from unidirectional graphite tape to increase stiffness and reduce their mass moment of inertia. The wing panel and trailing edge control surface are manufactured from 7075-T651 aluminum alloy. The wing panel and trailing edge control surface are also designed and fabricated to minimize their mass moment of inertia without sacrificing structural integrity.

TESTING AND EVALUATION

Prior to the fabrication of the production actuator (shown in figures 1-4), a prototype actuator, similar to the production configuration, was designed and fabricated for validation testing and evaluation. Figure 8 is a schematic of the actuator bench test setup. The bench test setup consists of the testbed, actuator assembly, flap simulator, rotary potentiometer, bellows coupling, servovalve, supply lines, and the necessary mounting hardware. The production actuation configuration is shown with the bench test setup in figure 8. The setup also includes a differential pressure transducer for monitoring the difference in actuator chamber pressure and can be used if so desired. The rotary potentiometer is used to measure angular displacements and is connected to the generic control surface by a bellows coupling to compensate for any misalignment. The flap simulator is designed so its mass moment of inertia is equivalent to the mass moment of inertia for the trailing edge control surface on the model. The inertia of the flap simulator is easily modified by either adding or removing material as necessary. Figure 9 shows the bench test setup with the prototype actuator which is similar in design to the production configuration.

Testing and evaluation of the prototype actuator has revealed its capability of short throw, high frequency control surface oscillations (± 5 degrees at 40-50 Hertz) and long throw, low frequency oscillations (± 15 degrees at 5-10 Hertz). Control surface deflections were found to be accurate within 0.3 degree over the frequency range. As a result, the rotary play in the actuation system fell within the acceptable margin. It was determined that the actuation system was capable of meeting research objectives. Prior to wind tunnel testing, the production configuration will be tested and evaluated in the same manner as the prototype system.

LESSONS LEARNED

Four significant problems were resolved while the actuation system was undergoing testing and evaluation. A needle roller bearing was originally used to transfer the motion of the drive link to the helical trough on the insert for control surface actuation. Excessive rotary play was detected in the system and it was determined that the needle roller bearing was the primary source. This excessive rotary play was eliminated by replacing the roller bearing with the solid, precision ground bearing fabricated from graphitic molybdenum-alloy tool steel. In addition to eliminating a significant source of rotary play in the system, wear in the helical trough on the insert was reduced by the inherent lubricating capability of the bearing material.

Another origin of rotary play was introduced into the actuation system from the manner in which bearing and linear-to-rotary conversion insert manufacturing was carried out. Initially the insert was fabricated and then the bearing was hand fitted to the insert. It was determined that the curvature of the helical trough was not a perfect helix. As a result, the trough had some high and low spots on it (.002 inch maximum deviation). Since the bearing could only be manufactured to a specific diameter, the fit between the bearing and helical trough varied over the length of the trough. This varying fit between the two components resulted in a varying amount of rotary play in the system. To meet research objectives, the rotary play in the system needs to be a constant value.

The method of bearing and insert fabrication was reversed to eliminate the variable amount of rotary play. A range of bearings in increments of .0001 inch above and below nominal helical trough height were manufactured. The helical trough will then be fitted to the bearing by progressively changing the bearing diameter until the trough is "worn-in" to fit a specific size bearing over its length. That bearing size will then be used on the actuator. This lapping operation will help to sustain the rotary play to a constant value and at the same time reduce the amount of rotary play due to the bearing/trough fit.

Wear between mating surfaces with relative motion due to rubbing contact was initially a problem. This wear was relieved by coating the components with the nickel alloy plating as previously discussed. The plating greatly improved surface lubricity and hardness, thus decreasing wear and prolonging component life.

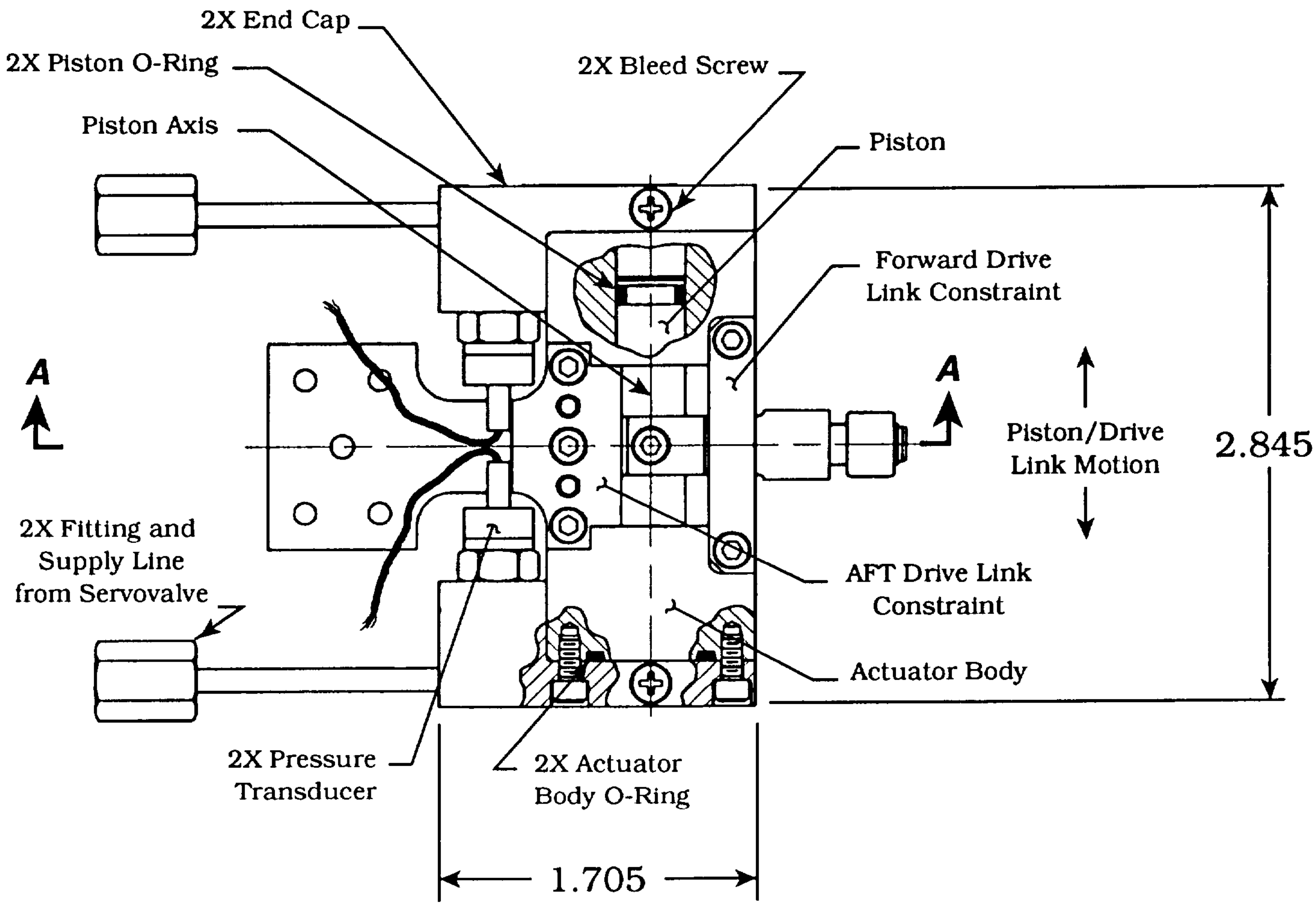
Piston o-ring wear led to increased free play between components, decreased o-ring life, and fluid leakage. An o-ring material more appropriate for high frequency reciprocating motion was selected to remedy these problems. Coating the internal bore on the actuator body with the nickel alloy plating also helped to decrease o-ring wear. Rotary play was reduced since the new o-ring material was harder and therefore less compressible than the originally selected material.

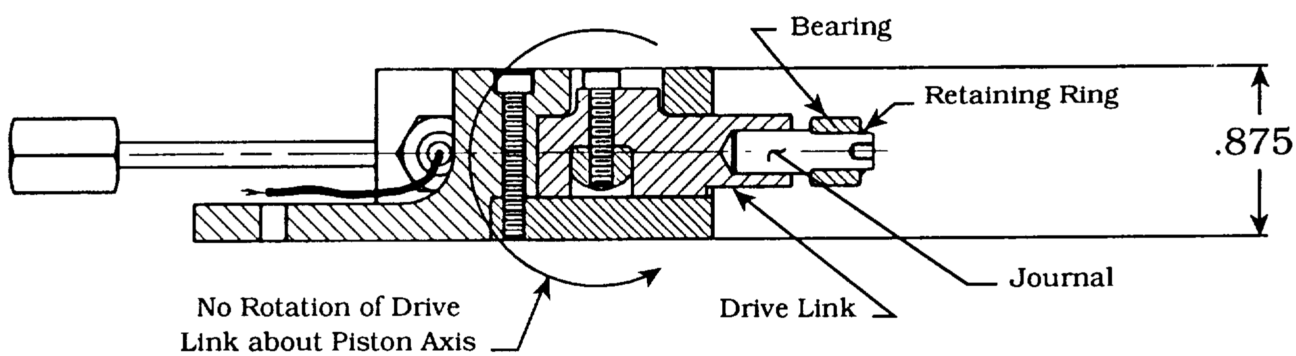
APPLICATION TO AEROSPACE VEHICLES

The primary function of this actuation system is provide flutter suppression capabilities for dynamic wind tunnel models. The system could be applied to military and commercial aircraft which require a flutter suppressing control surface actuation system. Compactness and mechanical efficiency are two merits of this actuation system when space constraints and torque requirements are design parameters which need to be addressed.

Another application of the system is it can be used as a rotary actuator in the traditional sense. Static control surface deflections for any aircraft configuration can be provided for by the actuation system. For applications where airfoil thickness is very thin, the compact design of this actuation system is advantageous. For example, the High Speed Civil Transport has airfoil thicknesses as low as two percent chord length and could be a possible candidate for the system because of its compact design. The quick response of the actuator to its input (the servovalve) is another trait of the system which could be applied to aerospace vehicles.

For applications where rotary play in the control surface is not a concern ($\geq \pm 1$ degree), manufacture of the actuation system is less difficult and less expensive. The reason is manufacturing tolerances do not have to be held as tight as those for applications requiring minimal rotary play (± 0.2 - 0.3 degree). The manufacturing cost is inversely proportional to the amount of tolerable rotary play in the actuation system.





Section A - A

Figure 1. Actuator Assembly

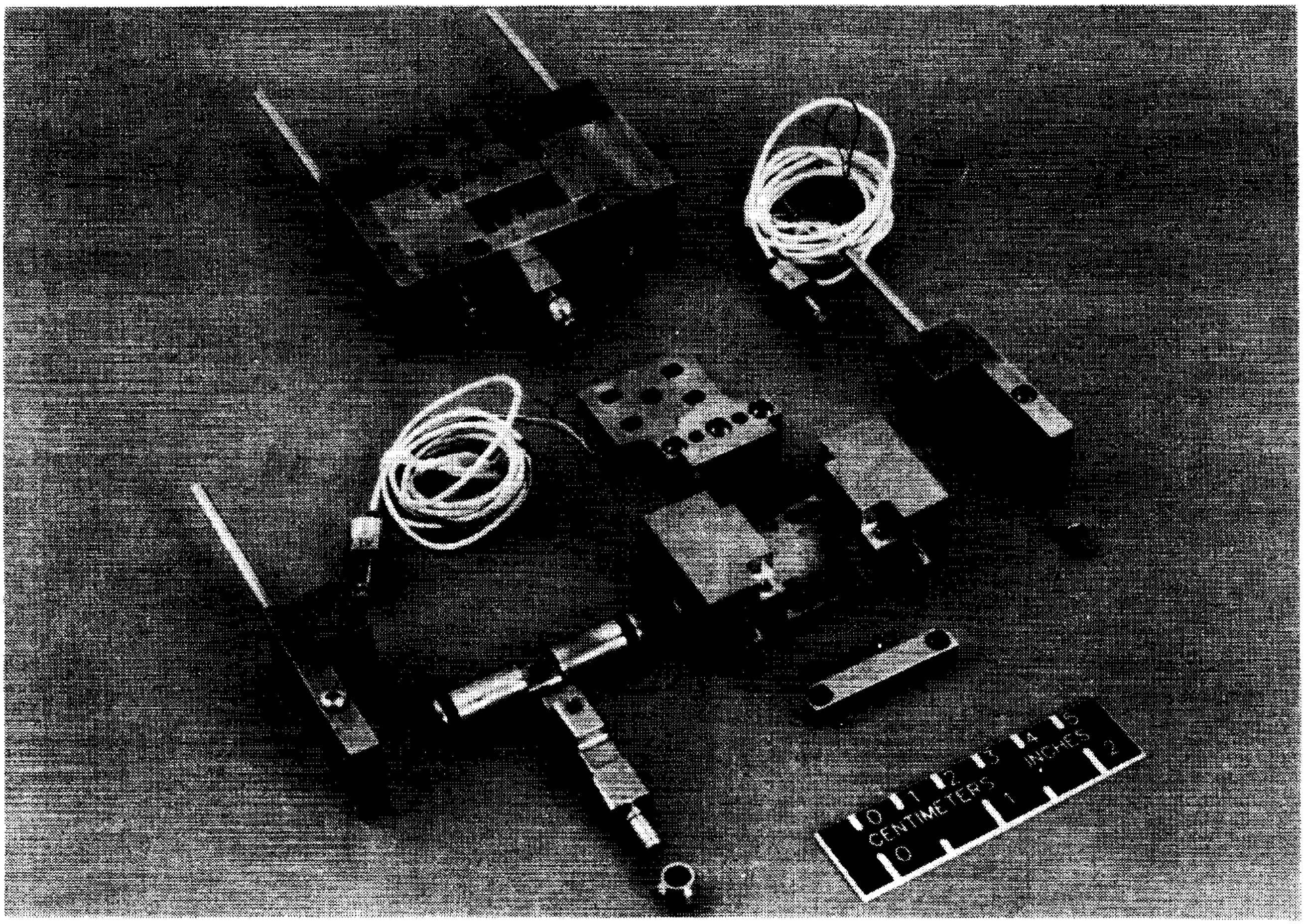


Figure 2. Actuator Assembly and Components

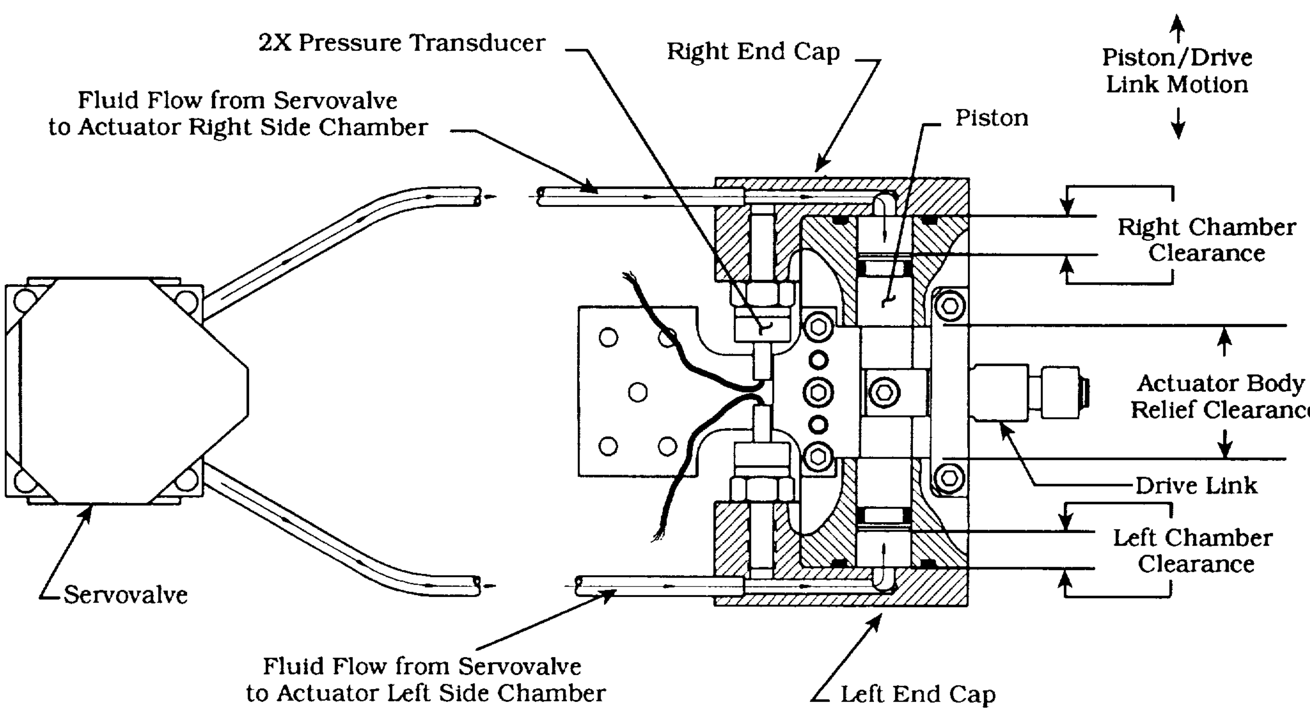


Figure 3. Servovalve/Actuator Operating Schematic

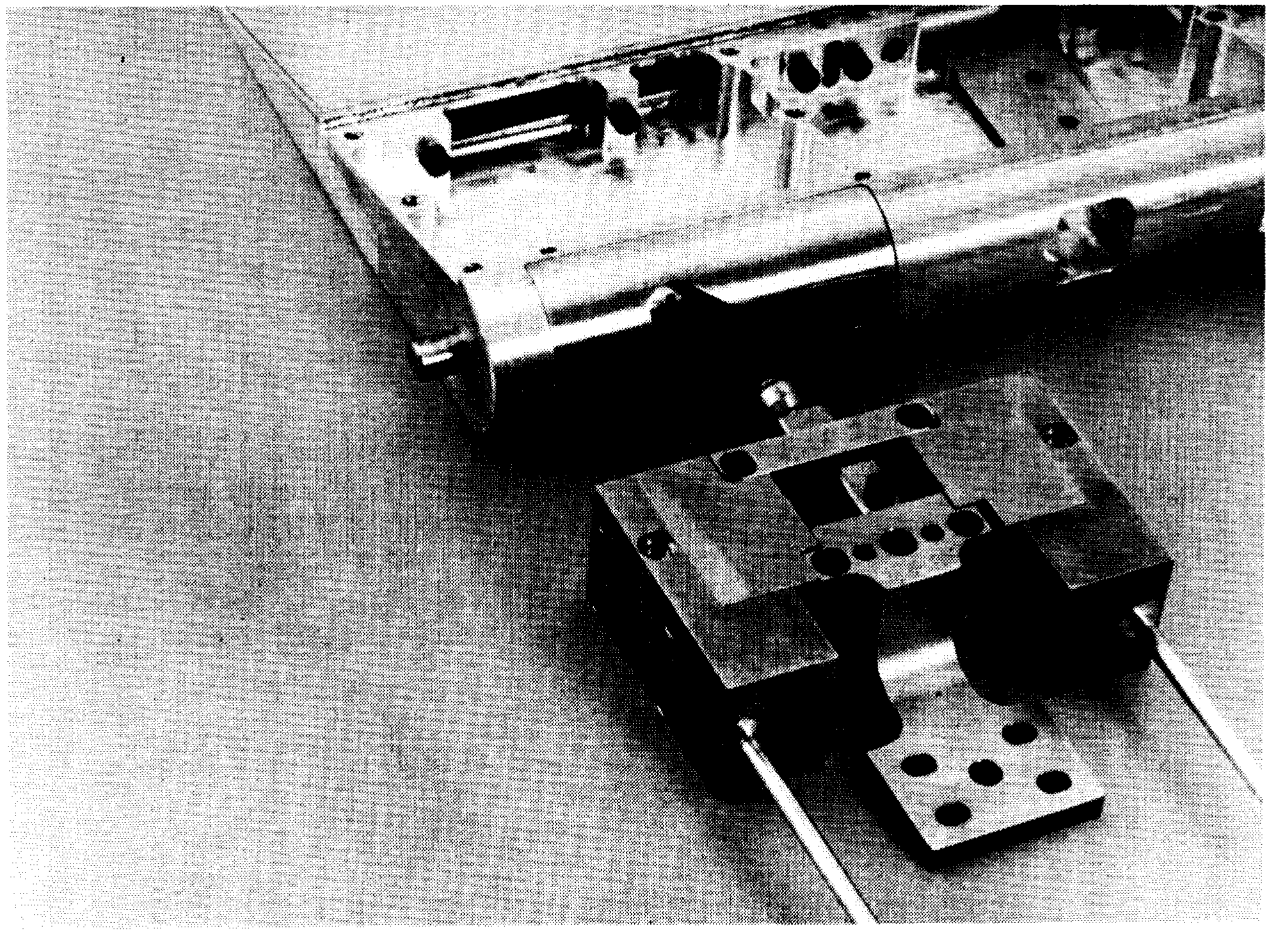
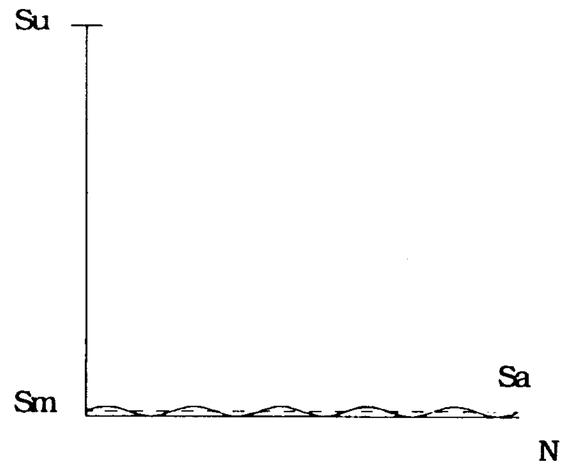


Figure 4. Actuator/Linear-To-Rotary Motion Conversion Insert Detail

Journal Fatigue Analysis for Normal Operating Conditions

Yield Stress	psi	(Sy)	=	270000.
Ultimate Stress	psi	(Su)	=	280000.
Endurance Limit	psi	(Se)	=	140000.
Mean Stress	psi	(Sm)	=	3468.
Stress Amplitude	psi	(Sa)	=	3468.
Stress Concentration Factor			=	1.75
Adj. Endurance Limit		(Se')	=	40000.
Fatigue Limit	psi	(Sf)	=	3511.
Infinite Life	cycles	(Ne)	=	10000000.
Fatigue Life	cycles	(Nf)	=	376788474



Operating Conditions: 20 Hertz for control surface deflections of ± 5°

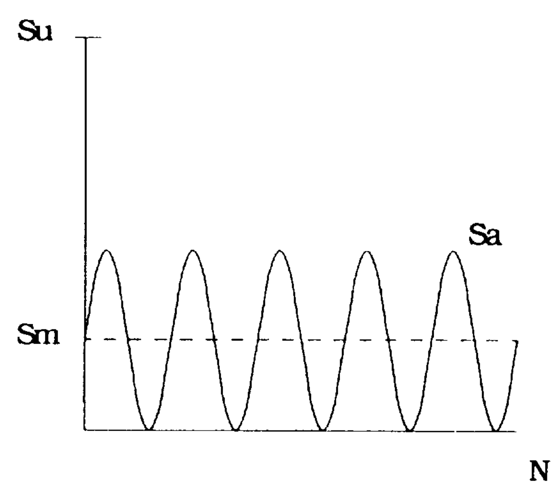
T = Journal Life for 20 Hertz Operating Frequency @ ± 5°

$$T = Nf/w = \left[\frac{376788474 \text{ cycles}}{20 \text{ cycles per sec}}\right]\left[\frac{1 \text{ hr}}{3600 \text{ sec}}\right]$$

T = 5233 hrs

Journal Fatigue Analysis for Upper Limit Operating Conditions

Yield Stress	psi	(Sy)	=	270000.
Ultimate Stress	psi	(Su)	=	280000.
Endurance Limit	psi	(Se)	=	140000.
Mean Stress	psi	(Sm)	=	64382.
Stress Amplitude	psi	(Sa)	=	64382.
Stress Concentration Factor			=	1.75
Adj. Endurance Limit		(Se')	=	40000.
Fatigue Limit	psi	(Sf)	=	83606.
Infinite Life	cycles	(Ne)	=	10000000.
Fatigue Life	cycles	(Nf)	=	11474.



Operating Conditions: 50 Hertz for control surface deflections of ± 15°

T = Journal Life for 50 Hertz Operating Frequency @ ± 15°

$$T = Nf/w = \left[\frac{11474 \text{ cycles}}{50 \text{ cycles per sec}}\right]\left[\frac{1 \text{ min}}{60 \text{ sec}}\right]$$

$T \cong 4$ min

Figure 5. Journal Fatigue Analysis

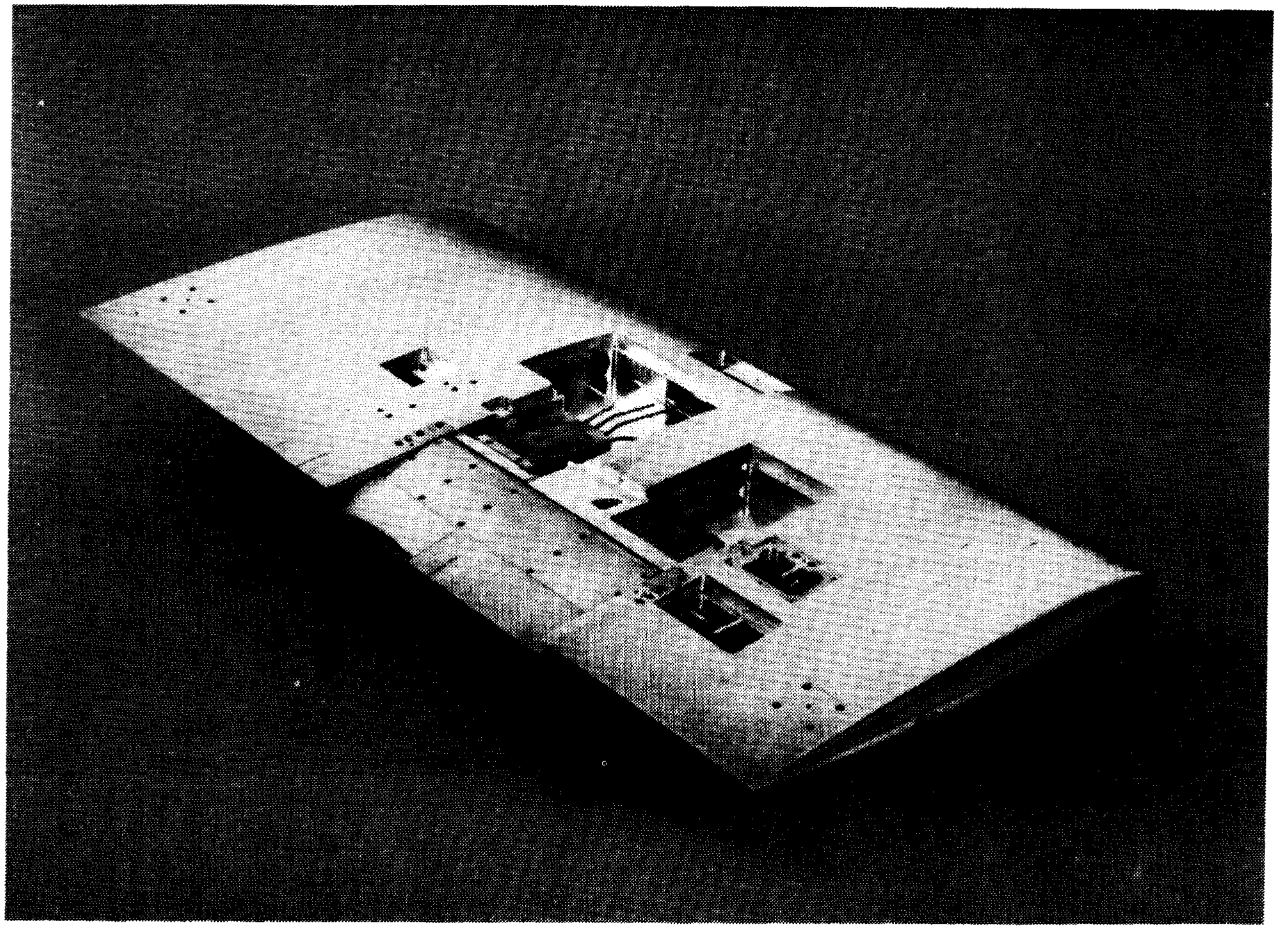


Figure 6. Model Assembly

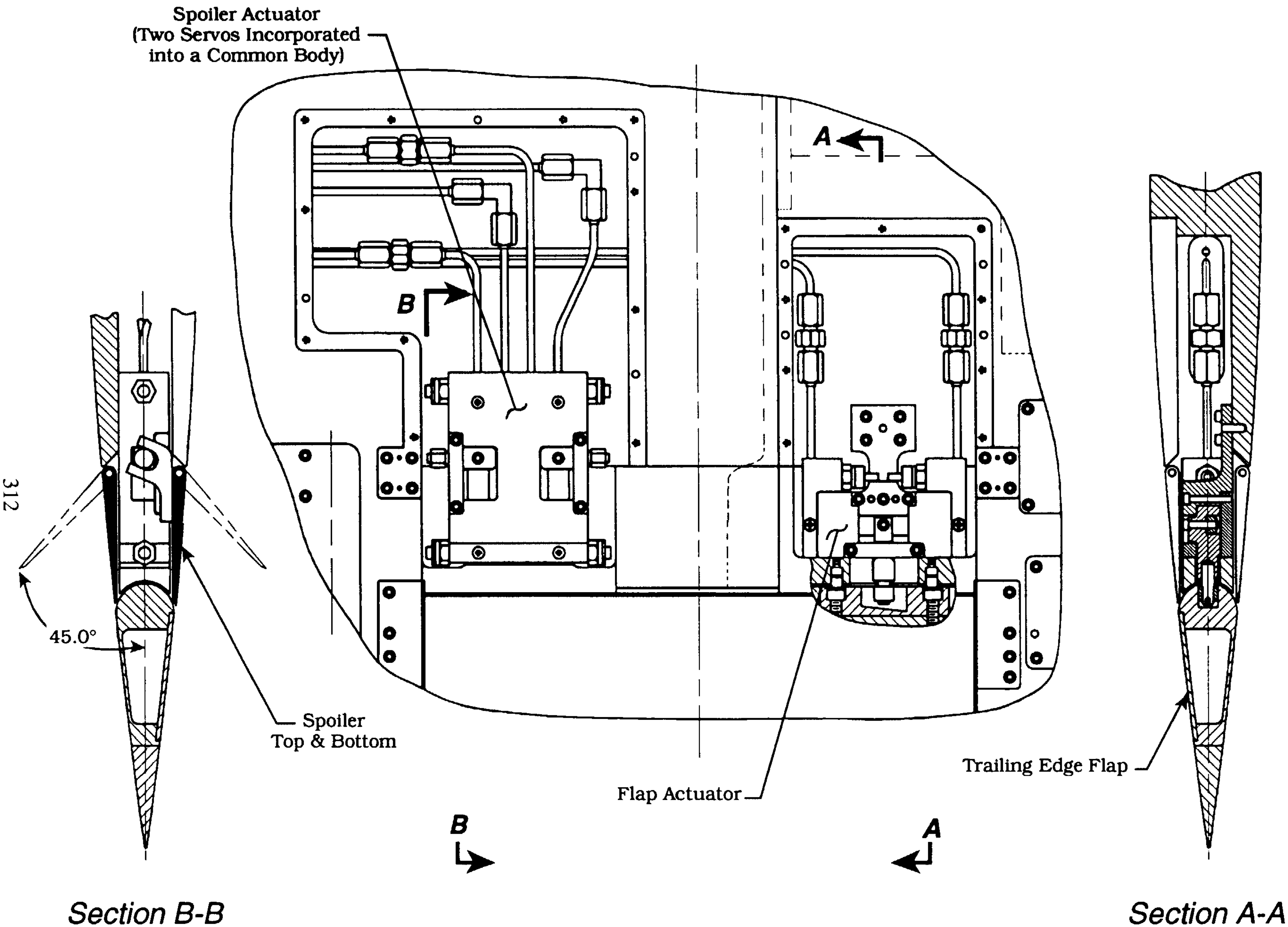


Figure 7. Actuator/Wing Panel Detail

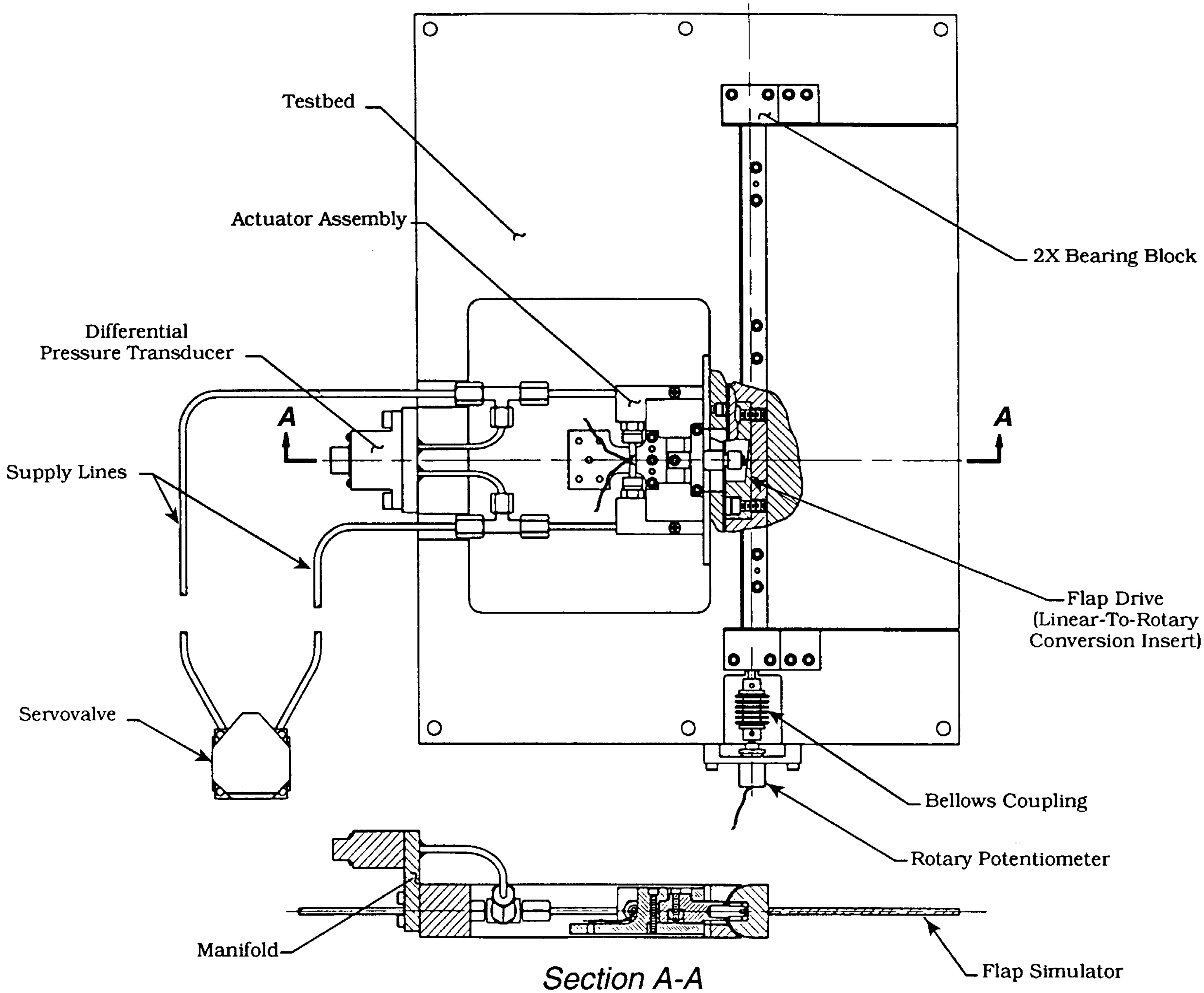


Figure 8. Actuator Bench Test Schematic

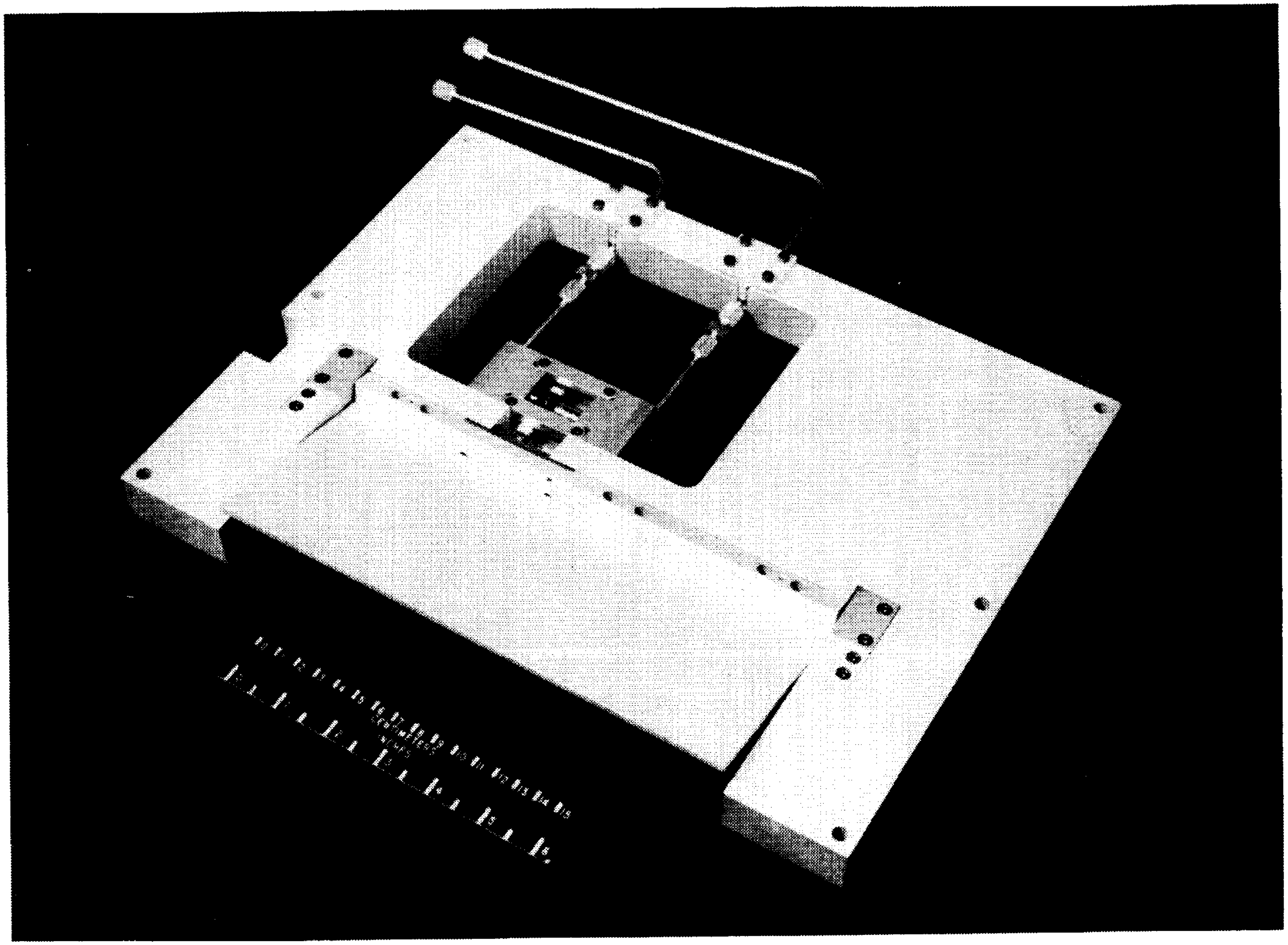


Figure 9. Actuator Bench Test Setup