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USAAVLABS TECHNICAL REPORT 69-77

EVALUATION OF MECHANICAL SERVO-ACTUATOR FOR FLIGHT CONTROL SYSTEMS

By

N. M. Gabriel
C. R. Schreiber

November 1969

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DAAJ02-67-C-0074
CURTISS-WRIGHT CORPORATION
CALDWELL, NEW JERSEY

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USAAVLABS Technical Report 69-77
November 1969

EVALUATION OF MECHANICAL SERVO-ACTUATOR FOR FLIGHT CONTROL SYSTEMS

Final Report

By

N. M. Gabriel
C. R. Schreiber

Prepared by

Curtiss-Wright Corporation
Aerospace Equipment Group
Caldwell, New Jersey

for

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

DEPARTMENT OF THE ARMY
U S ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS VIRGINIA 23604

This report presents the results of the evaluation of an experimental model all-mechanical servo-actuator for helicopter flight control application. The results of the effort indicate that the development of such a system is feasible.

This Command concurs with the findings of the contractor.

SUMMARY

This report discusses the results of a program conducted to design, fabricate, and test an all-mechanical servo-actuator with manual override capability for helicopter flight control application.

This all-mechanical system is a power boost with a mechanical override feature and provides mechanical amplification of signal torques from 0.5 to 1.0 lb-in. to operate output torque loads up to 1,142 lb-in.

The goal of this research program was to develop a mechanical servo-actuator to operate the helicopter swashplate, for cyclic and collective pitch control, with performance characteristics comparable to those of the hydraulic servo-actuator on the XH-51A helicopter.

The test item was subjected to a series of tests, including force thresold, resolution, frequency response, step response, and endurance. The results of this testing were compared with those of similar tests previously conducted on a prototype mechanical actuator as well as the performance characteristics of a hydraulic servo-actuator presently in use on the XH-51A helicopter.

Performance characteristics of the mechanical servo-actuator demonstrate that the actuator has met the requirements of the application when compared to the capabilities of the existing hydraulic power boost system. Specific details of the performance capabilities are discussed.

FOREWORD

This report pertains to work performed by Curtiss-Wright Corporation under U. S. Army Aviation Materiel Laboratories (USAAVLABS) Contract DAAJ02-67-C-0074 (Task 1F162204A13905).

The test article was designed by Mr. John S. Perryman, Chief Project Engineer, and Mr. C. W. Chillson, Technical Director, of the Curtiss-Wright Corporation, Caldwell Facility. The unit was built and assembled in the Experimental Test Laboratories of the Caldwell Facility under the direction of N. M. Gabriel, Senior Engineer in the Development Project Group. This report has been prepared by N. M. Gabriel and C. R. Schreiber of the Curtiss-Wright Corporation as co-authors.

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INTRODUCTION

This program is a continuation of a USAAVLABS-sponsored feasibility investigation into the use of mechanical servo-actuators and mechanical boost systems for helicopter application where mechanical independence of hydraulic fluid provides advantages in reduced vulnerability to enemy firearms damage.

The initial program phase was conducted by the Lockheed-California Company, and a report was issued covering the results[1]. The initial program indicated a need to improve the mechanical actuator in two respects. The first was to reduce the manual signal force level at the actuator from a 10-pound level to under 1 pound. Also, the minimum increment of signal motion to produce an output was judged to be larger than that of an equivalent hydraulic actuator, and a design goal was established to equal this minimum-motion capability.

This program demonstrated that the mechanical system could be designed to achieve the low input force levels and the lower minimum increment of motion of the same order of magnitude as those produced by a reference hydraulic actuator.

The design which has resulted is a small, compact, lightweight mechanical actuator with comparable characteristics to the hydraulic system presently in use.

DISCUSSION

Based on the recommendations discussed in Reference 1, a follow-on program was authorized to permit design improvements of the mechanical actuator and to reduce force threshold and minimum pulse characteristics. This report discusses the results of the program. Several significant improvements have been accomplished in this program and are briefly presented to highlight these achievements.

A total of 13.9 hours of operation was accumulated throughout this test program and induced an equivalent of approximately five times the number of clutch pulses anticipated during 1,000 hours of helicopter flight.

The hardware at the conclusion of the testing was examined and did not show any signs of deterioration other than normal expected wear patterns.

Specific improvements achieved in this program compared to the test results of the prototype unit are as follows:

1. The signal force threshold was reduced from an average torque level of approximately 11 lb-in. to 0.7 lb-in. while operating over the full output torque range to 1,142 lb-in.

2. The response to input command motion was reduced from a dead band of 1.125 degrees to 0.33 degree, with a minimum pulse of 0.03 degree recorded for both the input and the output displacement.

3. Demonstration of the signal threshold "full rate" capabilities showed considerable improvement, since the prototype required 1.125 degrees of input motion, while the new actuator requires only 0.75 degree to attain full output rate. The hydraulic actuator performance was in the order of 2.12 degrees for the same condition.

4. The variable performance criteria and specified design parameters are presented in Table I, Performance Data, for the convenience of direct comparison with specifications, goals, and test unit performance.

TABLE I. PERFORMANCE DATA

Parameter	MIL-H 8501A	Lockheed Specification LR-18027	Curtiss-Wright Initial Prototype	Production XH-51A Hydraulic Actuator	New Design Goals	Curtiss-Wright Development Test Actuator Test Results
Maximum Output Torque (lb-in.)	-	1142	1142	1142	1142	1142
Maximum Output Rate (rpm)	-	15	15	15	15	12*
Signal Threshold						
Minimum Pulse (deg)	-	-	-	-	-	0.03
Dead Band						
(±%)	2.0	1.0	1.5	0.15	0.17	0.44
(±deg)	1.5	0.75	1.125	0.113	0.125	0.33
Full Rate						
(±%)	-	-	1.5	2.83	1.0 & 3.0	1.0 & 3.0
(±deg)	-	-	1.125	2.12	0.75 & 2.2	0.75 & 2.2
Torque (lb-in.)	11.4	5.7	10.0	0.5	0.5 to 1.0	0.70
Total Input & Output Stroke (deg)	-	75	75	75	75	75
Power Input Rate (rpm)	-	6300	6300	6300	6300	6300

*Max Output Rate (See Discussion Section 4)

DESCRIPTION OF TEST ITEM

GENERAL

The mechanical servo-actuator is an all-mechanical power boost system. The power is derived from engine shaft rotation. The low-force mechanical input signal transmitted through linkage directly from pilot control stick motion provides the position signal to the actuator.

The low-force input actuates a variable lead spring clutch which engages the constantly rotating input power drum and drives the output against the applied torque loads. The increments of output rotation are controlled through a mechanical feedback loop that disengages the clutch when the input position command is satisfied. The amount of motion is controlled by the low-force position input. The power transmitted is that required to drive the output torque load at the speed of the input command signal.

Two clutches are used to provide the power boost function in either direction of rotation.

Output position is held against aiding or opposing torque loads by the inherent locking feature of the two spring clutches. The clutches in the deenergized mode act as brakes and hold the desired position by reacting the loads to fixed structure. This anti-load feedback characteristic is typical of Curtiss-Wright mechanical spring clutch servo actuators.

SERVO

When the signal linkage is in the null condition, the spring clutches receive oscillating signals within their allowable dead zone from the action of rotating ramp rings, which produce a high-frequency dither to the energizing tangs of the spring clutch. With two bungee wires in the signal path (each secured to a pivot arm and roller bearing that is in simultaneous contact with both clutch ramp rings), controlled displacement occurs when the unloaded wire is translated to a new position; the second wire deflects until the loading is relieved and follows up due to stored energy.

The ramp rings being unloaded cyclically allow a controlled signal to energize the clutches with very low input forces. Energization of the clutches cannot occur simultaneously since energization is through rollers common to both clutches, where displacement in one direction energizes one clutch and reverse displacement energizes the other clutch.

BYPASS

Should it become necessary to use the manual override, rotation of the input signal arm will automatically disengage the output from the servo clutches and enable the input arm to drive the output arm through direct mechanical coupling, allowing the pilot to manually control the aircraft.

OPERATION

A schematic diagram of the mechanical servo-actuator is presented in Figure 1. Input power to drive the actuator is applied to the input drive shaft (1) at 6,300 rpm. A spur gear train (2) provides a 10:1 reduction to continuously rotate the two drive drums (3) in opposite directions at 630 rpm. Two (2) spring clutch elements are used to provide bi-directional output motion. The spring clutch (4) is pinned to the output shaft (5). Interference fit of the spring clutch with the fixed brake shaft (6) provides a braking action that locks the output to fixed structure. A small clearance exists between the spring clutch and the drive drum to minimize drag of the spring clutch.

Since each spring clutch is energized in a similar manner, the method of energization will be described for one. Only one spring clutch is energized at a time, and a mechanical interlock (not shown) prevents simultaneous energization of both spring clutches.

Motion of the pilot's control stick, transmitted through linkage, results in angular movement of the actuator input signal arm (7). The input signal arm motion is transmitted through the signal linkage (8) to the signal springs (15). The signal linkage is interconnected by ball end fittings to provide universal drive action. This permits the signal motion to be transmitted around corners as dictated by the arrangement of the actuator components. The relationship of the input signal position to the actuator output position is established by a ball (11), driven by the signal linkage (8), operating in a cam slot of the feedback cam (17). The feedback cam is part of the output signal arm (16). The cam slot also permits relative motion between the signal linkage and output arm, thus allowing a signal to be applied while the output is fixed. The signal springs (15) displace the signal sliders (9). The sliders contain slots that cause the cam rollers (12) to pivot about pivot (13). Two cam rollers are used. Pivoting of the cam roller displaces one of the two signal cam plates (10), depending on the direction of the input signal motion. Motion of the cam plate results in axial displacement of a pin (20) and spring bungee (21) to clamp the spring clutch energizing disc (14) to the continuously rotating drive drum (3).

The energizing disc, in turn, being attached to the end of the spring clutch, applies a torque to wind the spring (4) away from the brake shaft (6) into contact with the rotating drive drum (3). Thus the input drive is coupled to the output elements (5).

Since the cam plate is attached to the rotating drive drum, the motion of the cam plate (initiated by the cam roller) is cyclic in nature. Thus the spring clutch will cycle from the braked position to the clutched position and the output elements will respond in a start-stop fashion. As the magnitude of the input signal is increased, the period of time that the output rotates increases until finally, the output will run continuously. Thus the output rate can be varied from zero to a maximum value equivalent to the input drum speed. The output elements (5) are coupled to the output

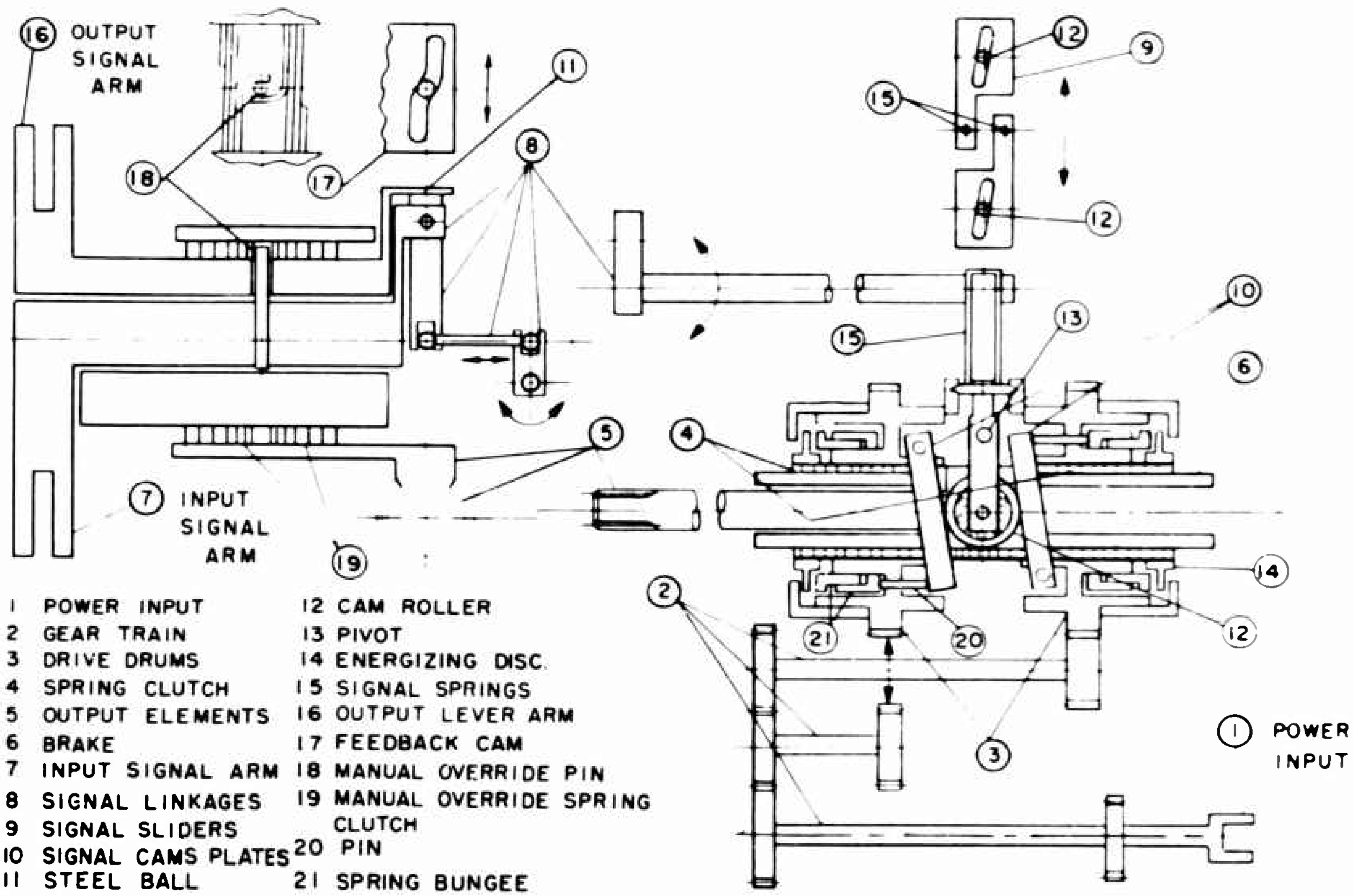

Figure 1. Schematic Diagram - Mechai cal Servo.

signal arm (16) through the interference fit of the manual override spring clutch (19). The spring clutch is disengaged when the feedback cam repositions the ball in the signal linkage to the null position.

In the event the output elements (5) do not respond to an input signal, continued motion of the input signal arm (7) will trigger the manual bypass. The manual override pin (18) carried by the input signal arm contacts the tang on the end of the override spring clutch (19) after a small dead zone is traversed. Two such spring clutches are used to provide bidirectional capability. The pin winds the spring clutch away from contact with the sleeve portion of the output element, disconnecting the servo from the output signal arm. The input signal arm now drives the output signal arm by direct mechanical contact.

A more detailed description of the mechanical servo-actuator may be found in Reference 2.

The photographs in Figures 2, 3 and 4 may be helpful in relating the physical hardware to the functional elements shown schematically in Figure 1 and discussed in the above description.

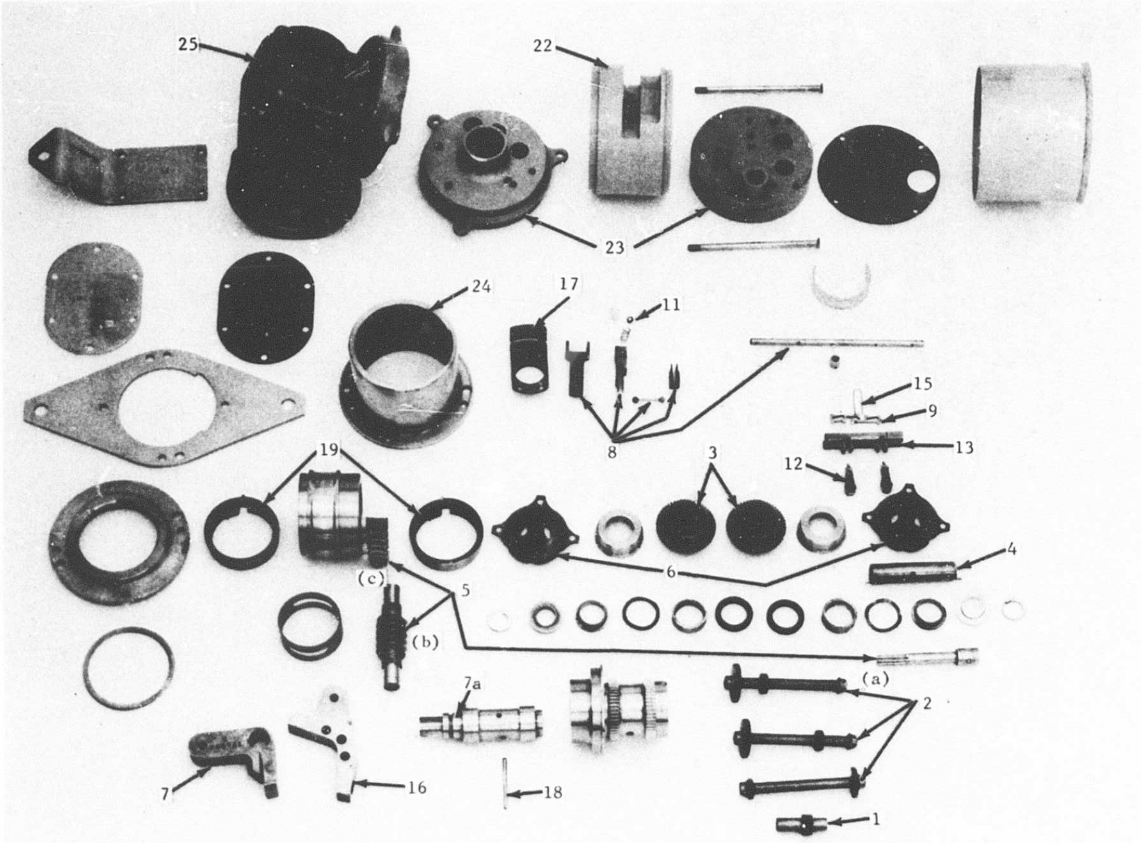

Figure 2. Mechanical Servo - Layout of Detailed Parts.

Mechanical Servo - Layout of Detailed Parts (Parts List).

1 - Input Drive Gear
2 - Gear Train
3 - Drive Drums
4 - Spring Clutch
5 - Output Elements - a) Clutch Output Shaft
b) Worm
c) Worm Gear
6 - Brake
7 - Input Signal Arm - a) Signal Input Shaft
8 - Signal Linkages
9 - Signal Sliders
10 - Signal Cam Plates
11 - Steel Ball
12 - Cam Rollers
13 - Pivot
14 - Energizing Disc
15 - Signal Springs
16 - Output Lever Arm
17 - Feedback Cam
18 - Manual Override Pin
19 - Manual Override Spring Clutch
20 - Pins - Not Shown
21 - Bungee, Spring
22 - Clutch Housing
23 - Clutch Housing End Caps
24 - Manual Bypass Brake Housing
25 - Manual Bvpass Clutch Housing

Figure 2 - Continued

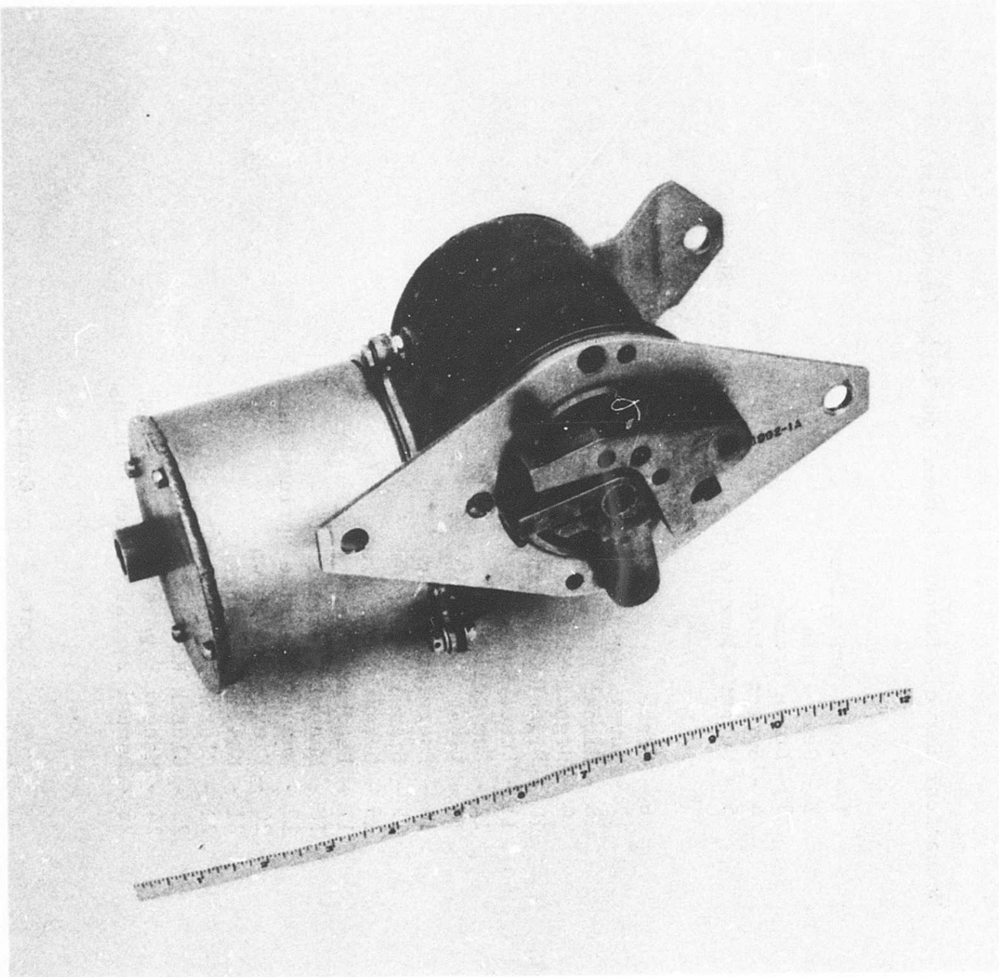

Figure 3. Mechanical Servo - Front View Assembly.

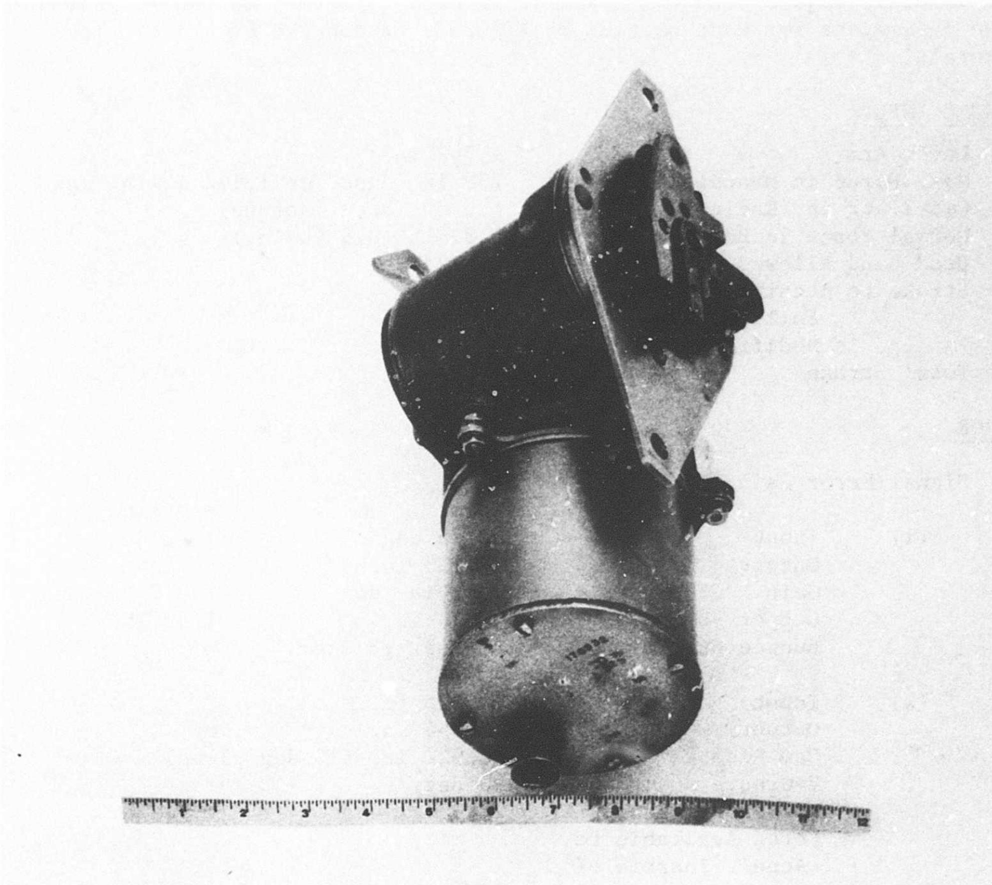

Figure 4. Mechanical Servo - End View Assembly.

DESIGN PARAMETERS

The schematic shown in Figure 5 is presented with the following engineering data, which was used to establish the theoretical design requirements for the mechanical actuator.

The numerals adjacent to the engineering data represent the equivalent design parameters for each section of Figure 5 as denoted by the circled numerals.

SIGNAL INPUT

Lever Arm	1.562 in.
Max. Force in Manual Reversion (at 1.562 in. Radius)	730 lb (based on 1,142 lb-in. max. oper. torque)
Normal Force in Boost Operation	0.33 lb (0.5 lb-in.)
Dead Band Allowed	±0.125 deg
Stroke to Attain	
Full Rate	±0.75 deg
Modifiable to	±2.2 deg
Total Stroke	75 deg

ERROR

Signal/Error Gain 1:1

		Norm.	Modified
(1)	Input	±0.75 deg	±2.2 deg
	Output	±0.125 in.	±0.125 in.
	Gain	0.67 in. deg	0.057 in. deg
	Output Force	0.523 lb	1.53 lb
	Bungee Stroke	Approx. ±6 deg	
(2)	Input	±0.125 in.	
	Output	±0.044 in.	
	Cam Rise	±0.022 in. (10 deg slope)	
	V-Groove	90 deg	
	Rocker Arm Ratio	2.1	
	Force Available to Accel. Inertia of Parts	0.5 to 1.5 lb	
(3)	Output Dither	0.005 in. @ 20 cps	
	Full-Rate Output	0.029 in.	
	Bungee Absorption	0.024 in.	
	Bungee Force	10 lb @ full stroke 7.7 lb @ preload	
	Disc Clutch Torque	0.5 lb-in.	
	Required Clutch Torque	0.5 lb-in.	
	Required Clutch Torque for Main-Spring Energization	0.24 lb-in.	

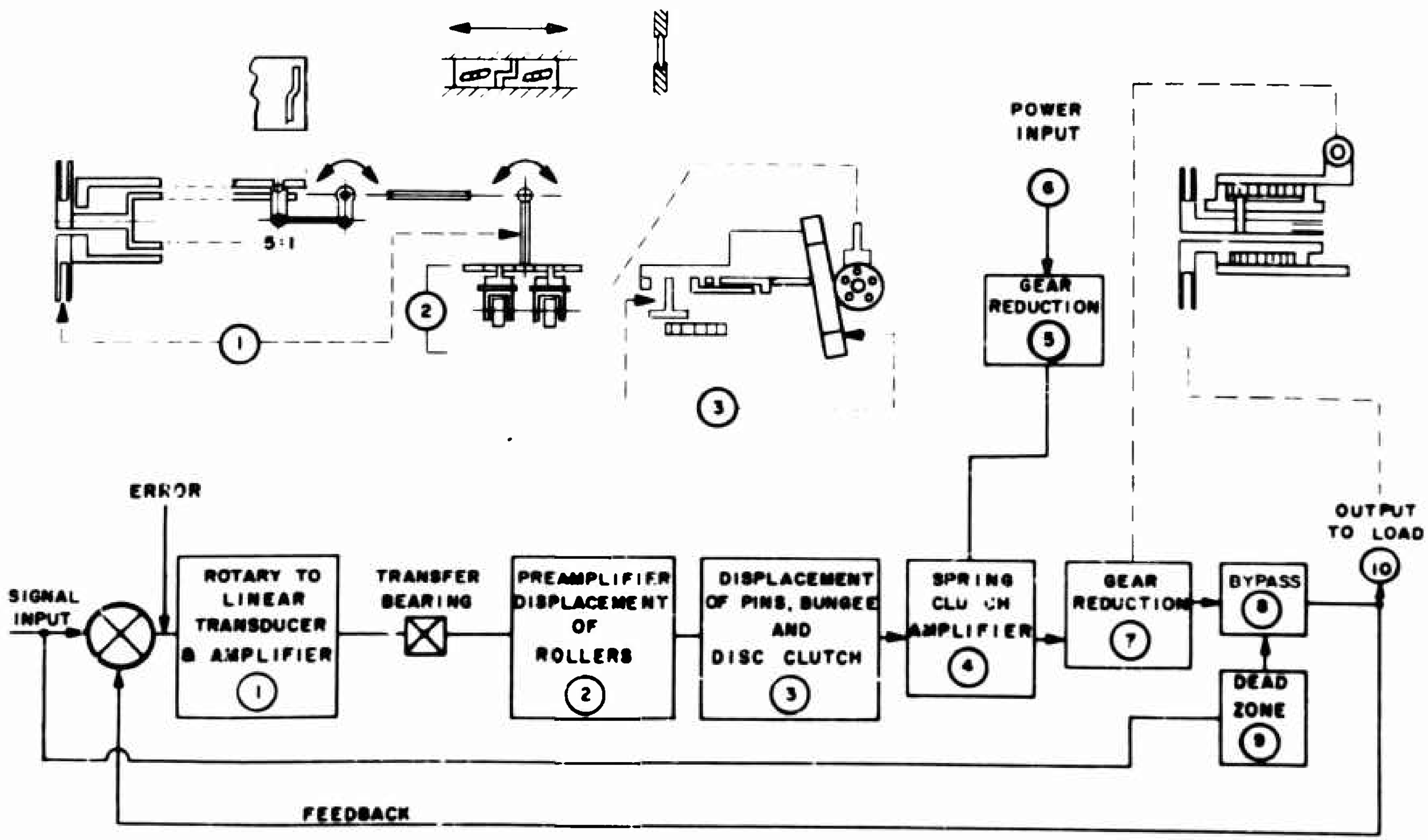

Figure 5. Schematic Diagram - Swashplate Control Boost System.

(4)	Energizing Torque	0.24 lb-in.
	Output Torque, Steady State	34 lb-in.
	Inertia Torque	34 lb-in.
	Total Torque	68 lb-in.
	Design Torque	100 lb-in.
	Clutch Speed	600 rpm
	Clutch Gain	3000:1
	Full Stroke Turns of Output (for 75 deg input)	8.4 Rev
(5) & (6)	Gear Ratio	10:1
	Output to Clutch	630 rpm
	Input From Power	6,300 rpm
	Output Torque	68 lb-in. peak
	Input Torque	7 lb-in. peak
(7)	Gear Reduction	40:1
	Input Peak Torque	34 lb-in.
	Output Peak Torque	1,360 lb-in.
(8) & (9)	Primary Input	
	Torque	1,360 lb-in. (84% eff)
	Speed	15 rpm
	Manual Input	
	Torque	1,142 lb-in.
	Dead Band	Approx. ±6 deg
	Static Torque Capacity	1,693 lb-in.
	Spring Gain	150
	Drag Torque	12 lb-in.
(10)	Max. Oper. Torque	1,142 lb-in.
	Max. Hold. Torque	1,693 lb-in.
	Lever Arm	1.693 in.
	Max. Rate	15 rpm
	Total Stroke	75 deg

OVERALL GAINS

Stroke and Rate	1.1
Torque	2284.1

DESCRIPTION OF TEST FIXTURE AND INSTRUMENTATION

The test fixture originally fabricated by Lockheed-California Company under Contract DA 44-177-AMC-232(T) was provided as Government-furnished property to be used and modified as deemed necessary to perform the required testing.

Modifications to the test fixture permitted the use of a hydraulic actuator controlled by a hydraulic servo valve to provide the required programmed inputs with a hydraulic motor for the power source. An inertia flywheel was designed and fabricated to simulate the rotor inertia and also to prevent speed droop of the hydraulic system, but due to the length of time required to stop the system in the event of a malfunction, it was decided to eliminate its use.

The instrumentation required to perform the necessary testing for this program included:

1. Recordings of input amplitude required for use of one of the dual potentiometers located in the hydraulic servo system, which provided the programmed inputs to the mechanical servo during all automatic operation.

2. During manual operation, a linear variable differential transformer (LVDT) with a range of ± 0.025 inch was used to record input amplitude.

3. Input force was recorded using a 100-pound load cell during the resolution testing and a 1,000-pound load cell for all other tests.

4. A synchro control transmitter used as a differential provided a trace of input minus output when used for the minimum pulse testing. This same transducer, when coupled with the signal of the input linear potentiometer from the hydraulic servo-actuator, provided a trace for recording output position. An instantaneous phase angle was thus achieved on the one recording.

5. A 1,000-pound load cell on the output link arm between the unit and the spring load recorded the force trace for calculating output torque.

6. A Sanborn (Model 850) 8-channel recorder was used to record the above parameters as a function of time.

7. A function generator (Hewlett Packard Model 202A) was provided to program the amplitude and frequency of the input signals.

8. A general radio Strobotac (type 1531-A) was used to monitor power input rpm.

9. During the resolution testing, a manually rotated micrometer type jackscrew provided the incremental motion to the input, while recordings of this motion were obtained using the LVDT described in item 2 above.

10. To monitor actuator body temperatures during the endurance testing, a 0° to 400°F pyrometer (Model 1331) was used.

The entire test rig setup may be observed in Figures 6 and 7; a detailed schematic showing the complete instrumentation setup used in this program is given in Figure 8.

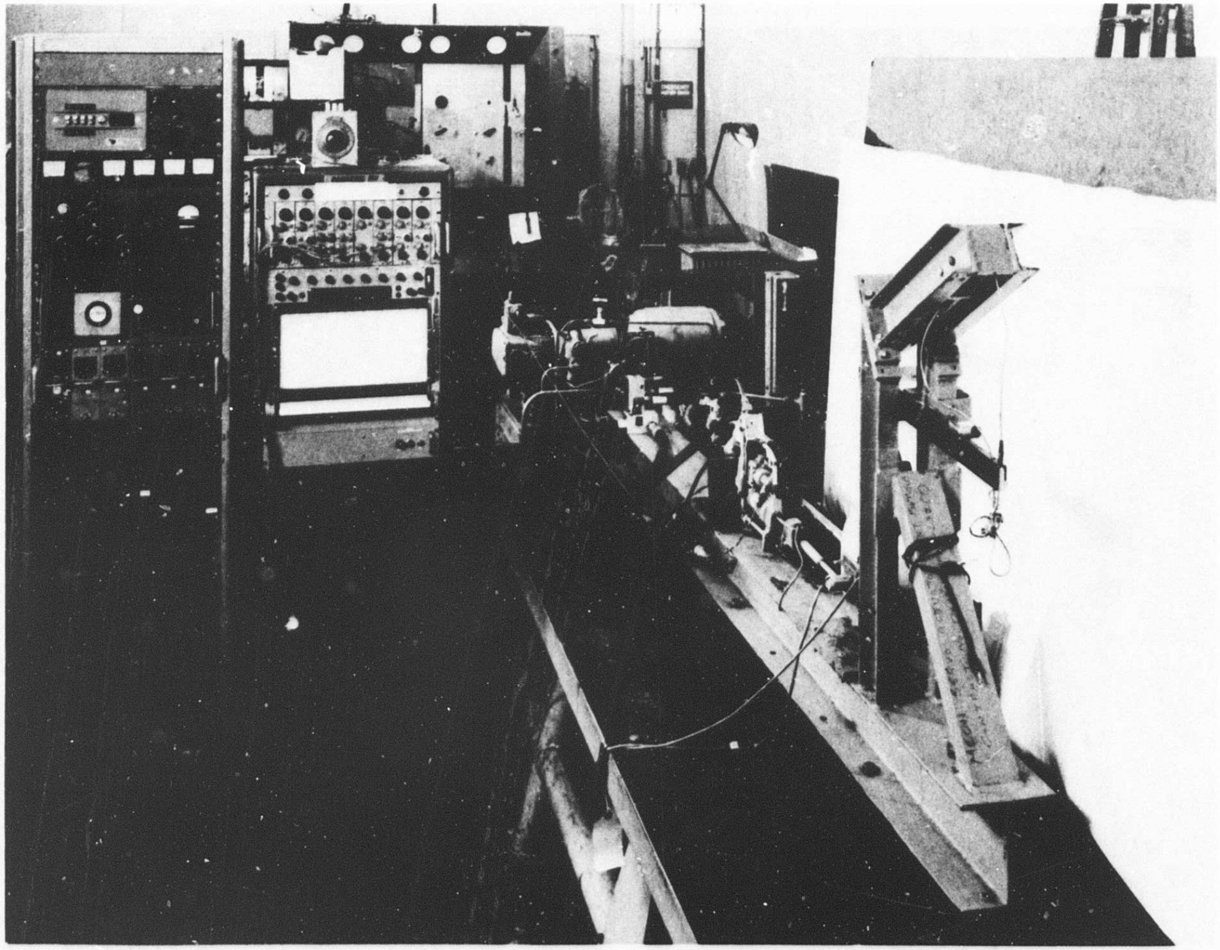

Figure 6. Mechanical Servo - Instrumentation and Test Area.

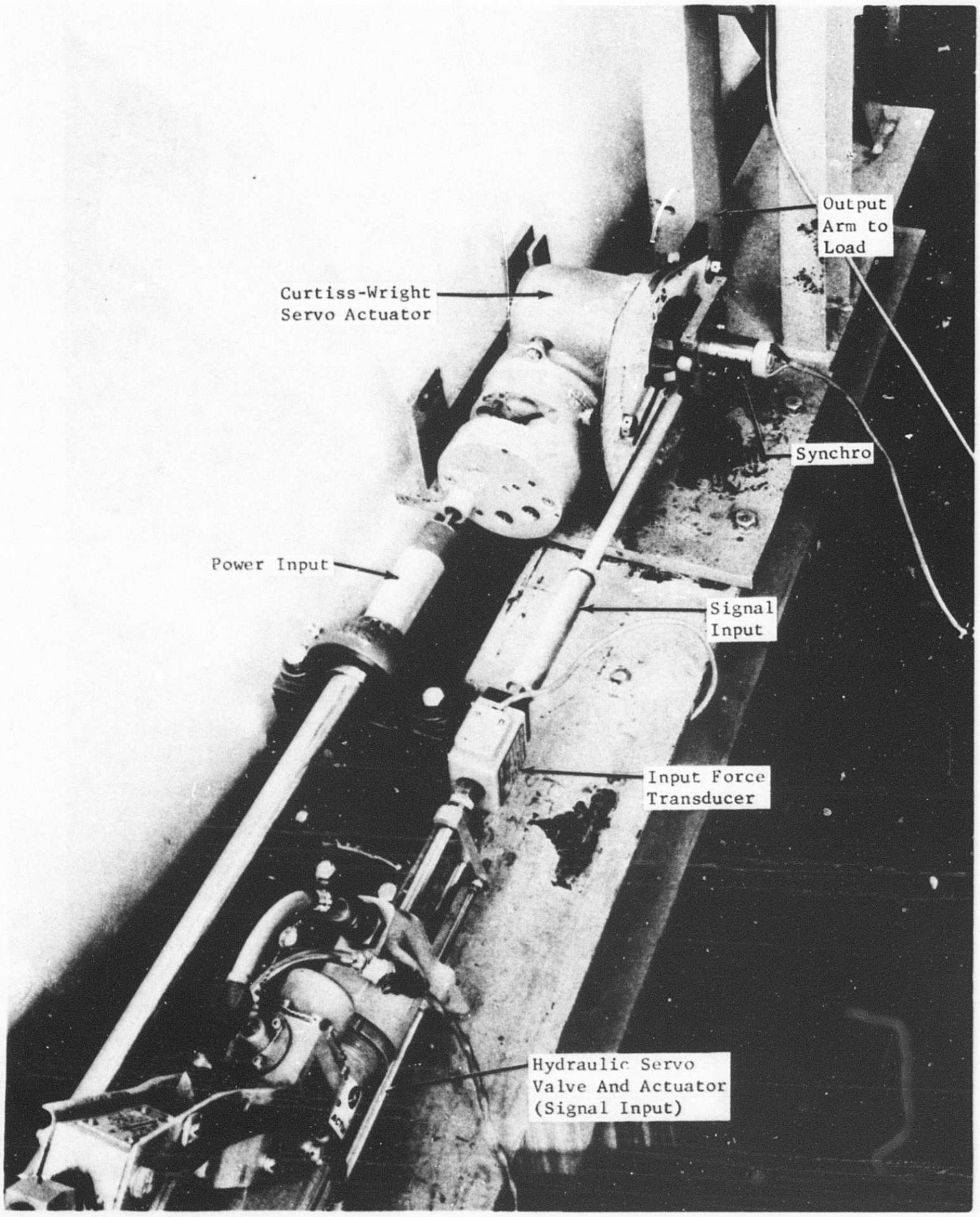

Figure 7. Mechanical Servo - Test Rig With Unit Installed.

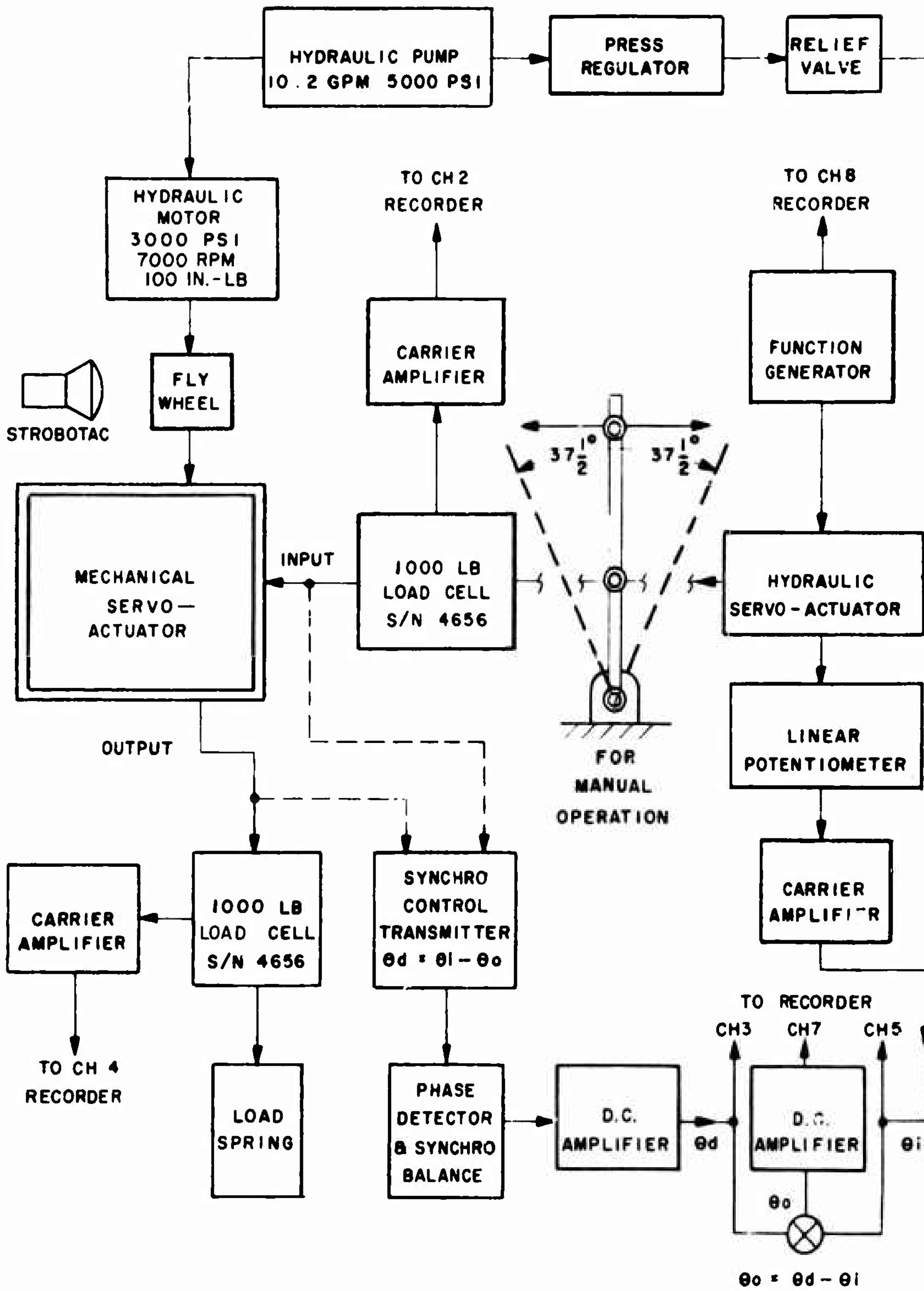

Figure 8. Schematic Diagram - Instrumentation.

TEST PROCEDURE AND RESULTS

The mechanical servo-actuator was mounted in the same load test fixture that was used for the Lockheed evaluation program[1]; modifications to this test fixture are described in the preceding section.

This testing was performed in accordance with the Reference 3 test procedure to demonstrate the operating characteristics of the mechanical servo-actuator under cyclic load conditions. These conditions were more severe than those of the collective mode of operation and provided the criteria for test evaluation.

The maximum spring load of 510 pounds was used throughout this test program so that the unit would be subjected to the most stringent conditions.

The output arm radius of 1.683 inches was used in conjunction with the 9/7 pivot point of the test fixture to provide a maximum torque load of 1,142 lb-in. to the actuator, as shown in Figure 9.

The following tests were performed so that evaluation and comparison to the performance data presented in Table I could be accomplished:

1. Manual Override

   The mechanical override capability was demonstrated by simulating a condition that would exist if the servo became inoperative (drive power off). Input torque was manually applied, transmitting the motion and torque through the actuator bypass to the actuator output. Operation was satisfactory.

   To demonstrate position reset capability, the mechanical override was moved to a new input position, with the servo section not operating. Power was then applied, and the servo demonstrated the ability to reset or index accordingly.

   During testing, it was found that the use of a sector worm gear at the interface between the servo and the bypass does not allow for a cumulative misindex which may occur between these basic elements, eventually allowing the worm gear to disengage from the worm drive. To permit continuation of performance evaluation, the bypass was "locked out", thereby giving assurance that the worm gear would not disengage from the worm drive.

   Future designs would include a full 360 degrees worm gear, eliminating any possibility of misindex or disengagement.

2. Cyclic Load Force Threshold

   Force threshold is defined as the minimum input torque required to cause motion of the output arm of the mechanical servo-actuator.

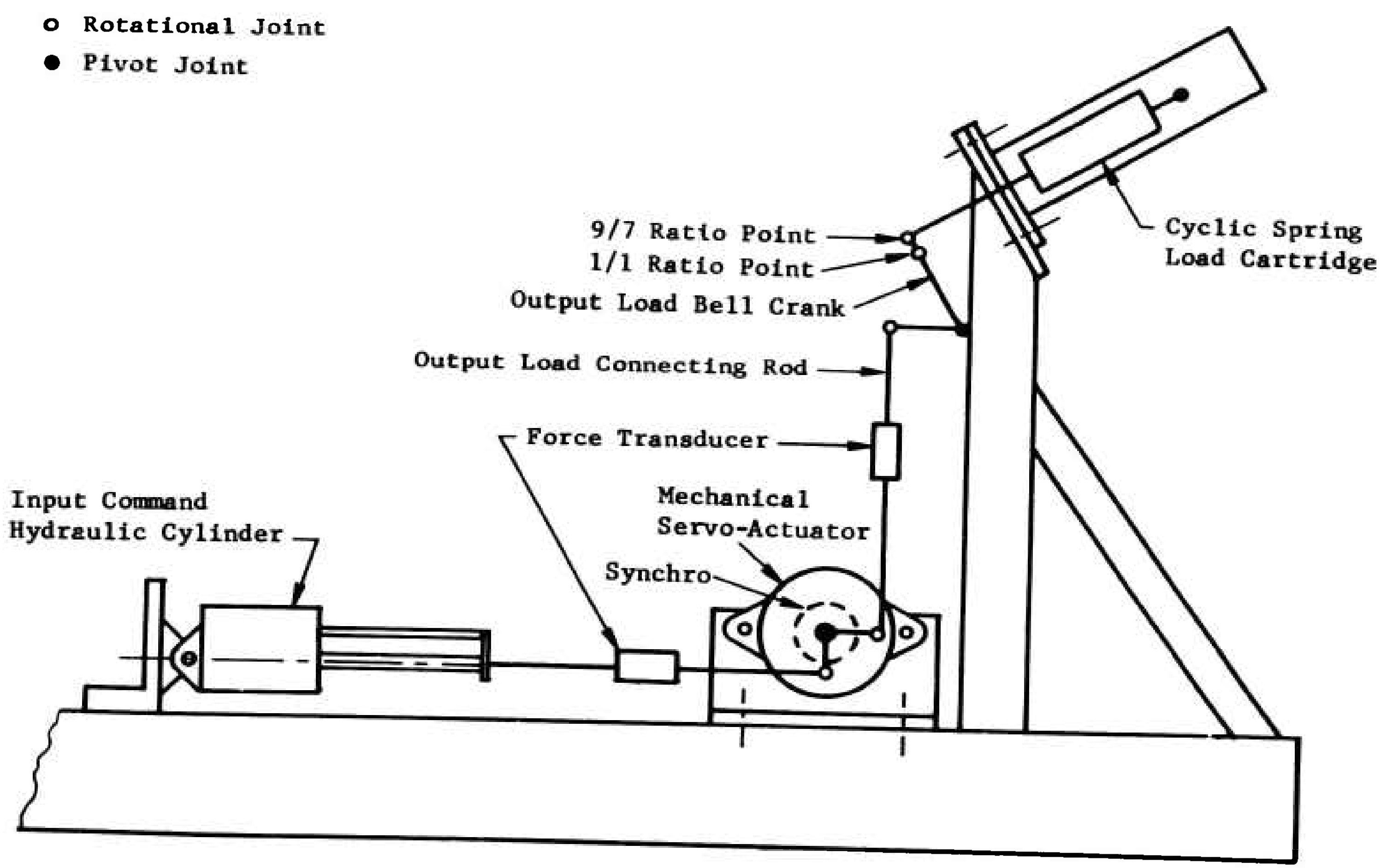

Figure 9. Schematic Diagram - Mechanical Servo-Actuator Installed on Test Fixture.

Testing was performed using a manually rotated micrometer-type screw drive which permitted incremental linear motion to be recorded by a ± 0.025-inch LVDT. The input force was obtained using a 100-pound load cell to an input arm radius of 1.562 inches. With a 510-pound spring load, the output arm was displaced from neutral to 0.25-, 0.50-, and 0.90-inch stroke positions (both CW and CCW) equivalent to ± 10-, 20-, and 36-degree positions of the output arm.

At each of these positions, the force threshold was measured in both CW and CCW directions, and the results are presented in Table II. As may be observed, the maximum input torque required at any position did not exceed 1.3 lb-in., and the equivalent output arm motion varied in steps from a minimum of 0.03 degree to a maximum of 0.75 degree. Normal or average pulses were 0.15 degree to 0.25 degree of output motion. Table III presents the test data for a no-load run.

It was noted that these minimum forces showed a very slight change from a load to a no-load condition, as indicated by the following averages:

a. No-load-force threshold, 0.50 lb-in. for entire range (Table III).

b. 510-pound load-force threshold, 0.90 lb-in. for entire range (Table II).

c. Total average force threshold, 0.70 lb-in. for entire range (Tables II and III).

3. Cyclic Load Resolution Test

Resolution is defined as the minimum input motion required to cause motion of the output arm.

This test was performed simultaneously with the force threshold test, as recorded data permitted monitoring both input position and output position as well as input and output force.

The data presented in Table II show not only incremental steps for resolution but also complete reversals that take dead band into consideration.

Continuous rotation of the micrometer-type screw drive at each output position (neutral ± 10, 20, and 36 degrees) permitted recordings to be taken within the limits of the LVDT travel, permitting several pulses in the same direction as well as complete reversals.

TABLE II. FORCE THRESHOLD AND RESOLUTION TEST RESULTS, 510-POUND SPRING

Output Direction	Output Position (deg)	Applied Force to Input: Dir	Aid/Opp	Torque (lb-in.)	Minimum Input Motion Reqd (deg)	Motion of Output Arm (deg)
Neutral	Zero	CCW	-	1.10	0.18	0.15
		CW	-	0.79	0.65	0.40
		CCW	-	0.50	0.47	0.40
		CCW	-	0.94	0.03	0.03
		CW	-	0.94	0.20	0.20
		CCW	-	0.63	0.22	0.20
Clockwise	10	CCW	A	0.79	0.66	0.68
		CW	O	1.30	0.70	0.20
		CCW	A	0.94	0.80	0.30
		CCW	A	1.10	0.10	0.10
		CW	O	1.10	0.55	0.20
Clockwise	20	CCW	A	0.79	0.45	0.45
		CCW	A	0.94	0.12	0.10
		CW	O	1.10	0.65	0.10
		CW	O	0.50	0.05	0.05
		CW	O	1.10	0.10	0.10
		CCW	A	0.79	0.70	0.30
		CCW	A	0.79	0.17	0.20
		CW	O	0.50	0.76	0.15
		CW	O	0.79	0.25	0.40
Clockwise	36	CCW	A	0.94	0.07	0.10
		CCW	A	0.63	0.15	0.15
		CW	O	0.94	0.33	0.45
Counter-clockwise	10	CCW	O	0.94	0.20	0.05
		CCW	O	0.63	0.07	0.10
		CCW	O	0.47	0.10	0.50
		CW	A	1.30	0.29	0.40
		CW	A	1.10	0.50	0.15
Counter-clockwise	20	CW	A	0.79	0.35	0.15
		CW	A	0.94	0.40	0.50
		CCW	O	0.94	0.40	0.40
		CCW	O	1.10	0.35	0.10
		CW	A	0.94	0.70	0.60

TABLE II - Continued						
Output Direction	Output Position (deg)	Applied Force to Input Dir	Aid/ Opp	Torque (lb-in.)	Minimum Input Motion Reqd (deg)	Motion of Output Arm (deg)
Counter-clockwise	36	CCW	O	0.79	0.15	0.45
		CW	A	1.10	0.30	0.25
		CW	A	1.30	0.15	0.20
		CCW	A	0.94	0.70	0.30
		CCW	A	0.79	0.20	0.40

NOTE: 1) A = Aiding load pulses; O = Opposing load pulses.
2) Dead Zone = ± 0.33 deg.

TABLE III. FORCE THRESHOLD AND RESOLUTION TEST RESULTS, NO LOAD					
Output Direction	Output Position (deg)	Applied Force to Input Direction	Torque (lb-in.)	Minimum Input Motion Reqd (deg)	Motion of Output Arm (deg)
Neutral	Zero	CW	0.16	0.65	0.35
		CW	0.63	0.30	0.15
		CCW	0.32	0.65	0.10
		CCW	0.47	0.15	0.20
		CW	0.32	0.70	0.50
		CW	0.32	0.40	0.30
Clockwise	10	CW	0.32	0.50	0.45
		CW	0.16	0.35	0.20
		CCW	1.10	0.60	0.25
		CW	0.32	0.30	0.60
Clockwise	20	CW	0.16	0.35	0.60
		CW	0.32	0.60	0.60
		CW	0.16	0.10	0.20
		CCW	0.32	0.60	0.15
		CW	0.32	0.85	0.50
		CCW	0.32	0.40	0.25

TABLE III - Continued					
Output Direction	Output Position (deg)	Applied Force to Input: Direction	Applied Force to Input: Torque (lb-in.)	Minimum Input Motion Reqd (deg)	Motion of Output Arm (deg)
Clockwise	36	CCW	0.32	0.35	0.45
		CW	0.63	0.40	0.15
		CCW	0.79	0.35	0.15
Counter-clockwise	10	CW	0.63	0.20	0.10
		CW	0.63	0.55	1.10
		CCW	0.79	0.10	0.10
		CCW	0.63	0.15	0.10
Counter-clockwise	20	CCW	0.63	0.85	0.25
		CW	0.79	0.85	0.10
		CW	0.16	0.10	0.15
		CCW	0.32	0.15	0.20
		CCW	0.32	0.25	0.30
		CW	0.50	0.45	0.05
Counter-clockwise	36	CW	0.79	0.25	0.25
		CCW	1.10	0.55	0.20
		CCW	1.10	0.30	0.30
		CCW	0.79	0.10	0.10
		CCW	0.32	0.20	0.10

4. Cyclic Load - Maximum Rate Test

It was demonstrated that the output rates followed the input rates up to 70 degrees per second. The unit was designed to produce 90 degrees per second with continuous output rotation when used with a dead zone of 0.125 Degree. It was found that the input linkage in the servo had a spring rate which allowed the clutches to cycle in the 0.125-degree dead band.

A large amount of testing and development effort was expended to provide increased stiffness in the linkage system and to eliminate backlash between the input signal and the clutch actuation point. These improvements, within the limitations of this design, resulted in operation with a dead band width of 0.33 degree. This represented a large improvement over the prototype dead band width of 1.125 degrees. While this represents the best dead band that can be achieved with this particular hardware, the test efforts have produced positive design information that can be used to further improve any future mechanical actuator of this type.

The hardware, when adjusted to the 0.33-degree dead zone, limited the stroke range to the cycling cam clutch energizer. This means that as the dead band was increased, it was no longer possible to continuously energize a clutch. The maximum that could be achieved by careful adjustment was clutch "on" for 80 percent of a revolution. This resulted in an output rate of 70 degrees per second because the clutch was "off" for 20 percent of each revolution. To verify the ability of the servo to achieve maximum output rates, the input speed was increased to 8,500 rpm, and recorded data indicated that output rates exceeded the required 15 rpm.

In future designs, the ratios will be maintained on dead zone limitations and maximum stroke lengths to provide for saturation of the clutch signal to produce continuous output rotation.

These design problems resulted in numerous additional test runs in establishing the modifications required.

5. Cyclic Load - Frequency Response

Performance demonstrated during frequency response testing is presented in Figures 10 and 11, where the servo-actuator was cycled sinusoidally at output amplitude of ± 10-percent stroke (± 3.75 degrees) and ± 50-percent stroke (± 18.75 degrees), with the frequency varied from 0.1 cps to 10.0 cps in accordance with the requirements shown in Table IV.

TABLE IV. FREQUENCY RESPONSE TEST REQUIREMENTS

No.	Output Lever Spring Load (lbs)	Input Lever Rotary Motion (±%)	Input Lever Rotary Motion (deg)	Sinusoidal Input Frequencies (cps)
1	510	10	3.75	.1-10
2	510	50	18.75	.1-10

Figure 10 is a plot for the 50-percent stroke (± 18.75 degrees) amplitude and indicates a corner frequency of 0.25 cps. The calculated or predicted curve is also presented as shown, cornering at 1.2 cps.

A similar reduction is indicated in Figure 11 for the ± 10-percent stroke (± 3.75 degrees), where the corner frequency occurs at 0.91 cps compared to a predicted value of 6.0 cps. The corresponding predicted phase lag curves are plotted for both conditions in Figures 10 and 11.

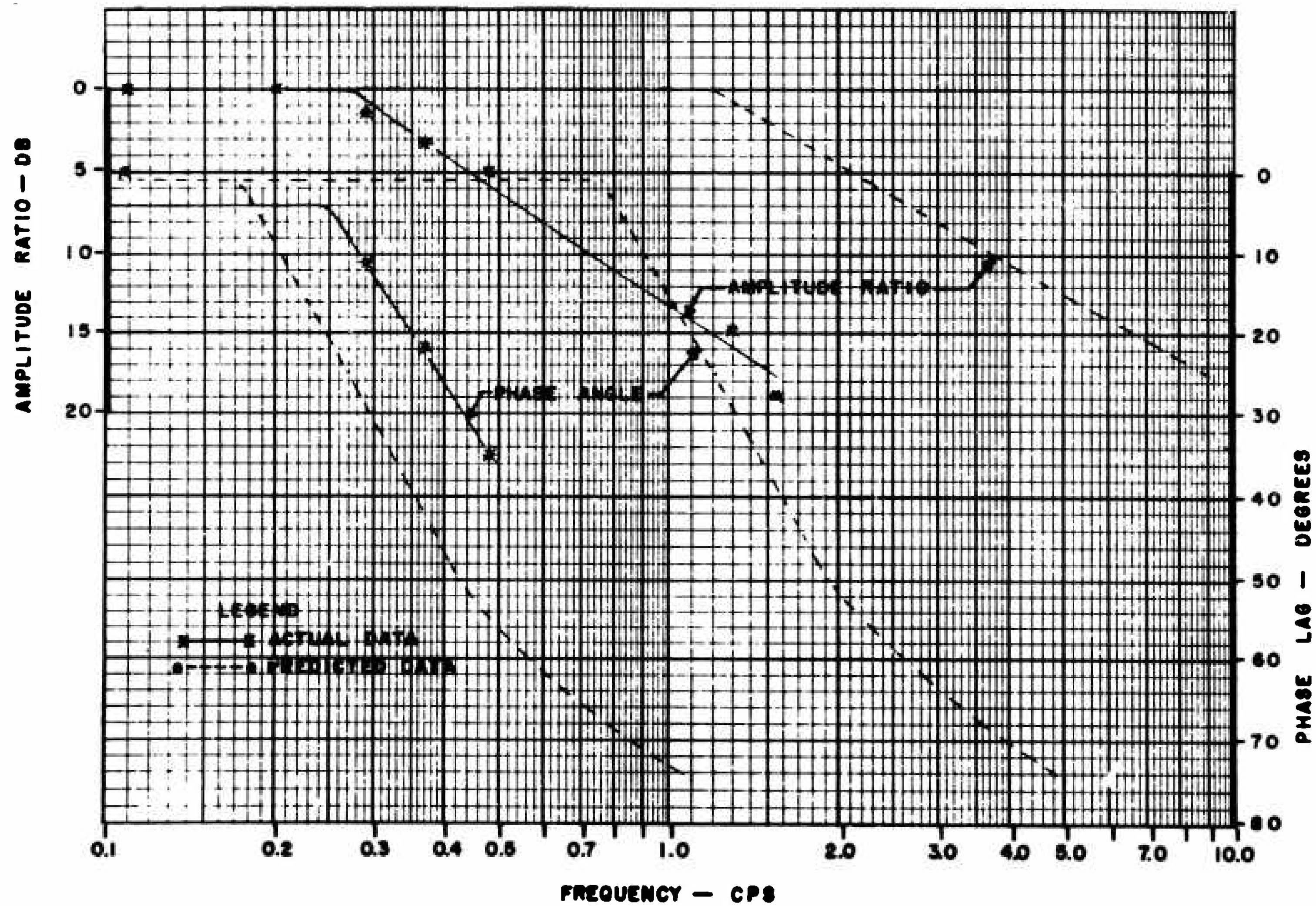

Figure 10. Frequency Response - Cyclic Load, 510-Pound Spring, 50-Percent Stroke.

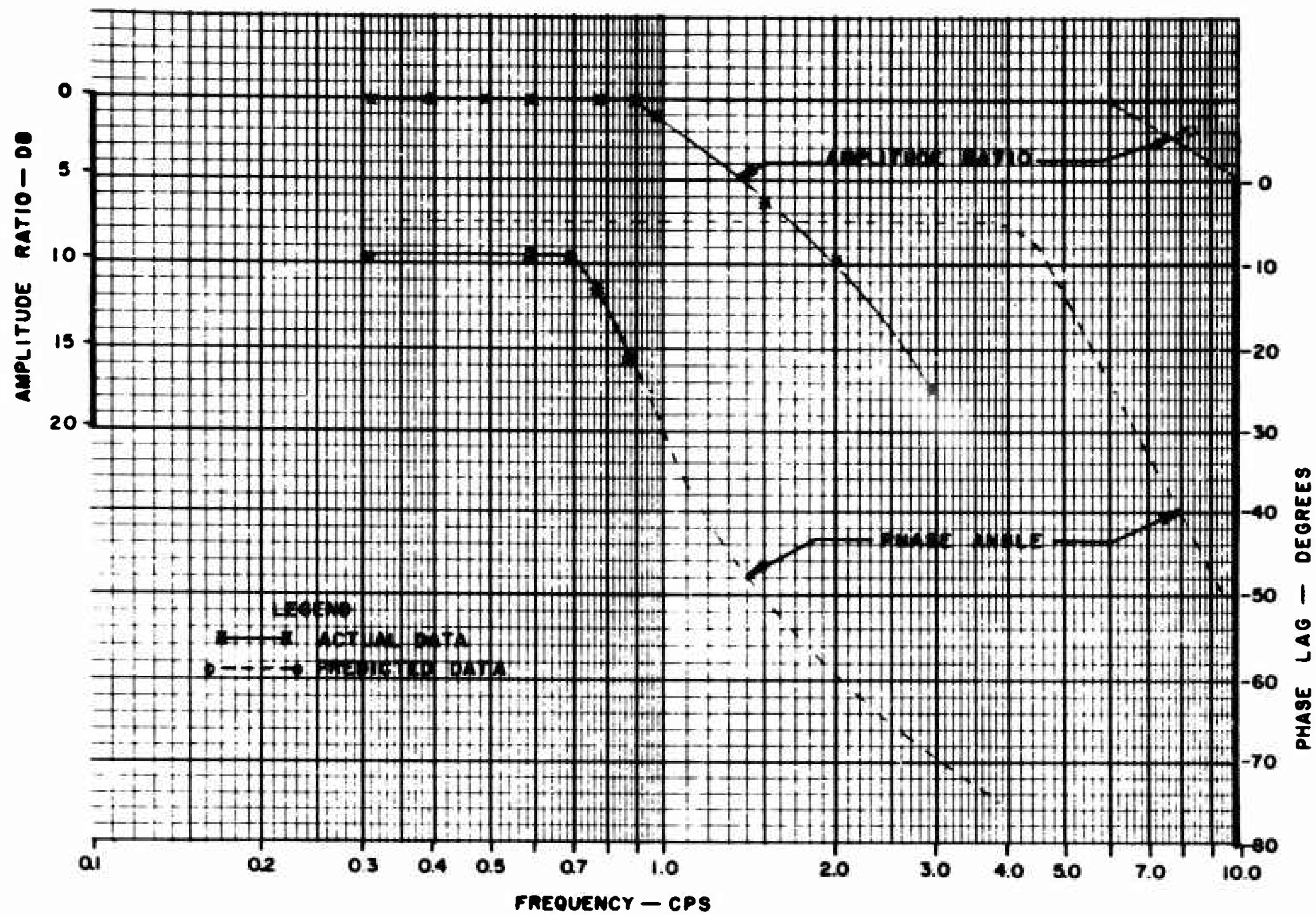

Figure 11. Frequency Response - Cyclic Load, 510-Pound Spring, 10-Percent Stroke.

The general shape and trend of the actual recorded data closely follow the predicted values except for the reduced rates indicated by the lower corner frequencies. The section entitled "Cyclic Load - Maximum Rate Test" describes the limitations observed during this phase of testing.

To obtain the performance characteristics indicated by the predicted curves in Figures 10 and 11, it would be essential to increase the stiffness of the input signal linkage, assuring that a saturated command signal would produce a saturated or solid clutch energization.

CYCLIC LOAD ENDURANCE TEST

A total of 13.9 hours of operation was accumulated during the test program. This demonstrates that the basic clutch and mechanical components would have a life in terms of simulated clutch operation equivalent to almost 5 times the endurance requirements for 1,000 hours of simulated flight.

The endurance test requirements as shown in Table V indicate that a total of 106,000 cycles would be required to simulate 1,000 hours of flight.

The equivalent number of clutch energizations to satisfy this requirement would be in the order of 222,000 clutch pulses, tabulated as follows:

1,000 cycles at 0.1 cps = 7,000 pulses
5,000 cycles at 0.5 cps = 15,000 pulses
100,000 cycles at 5.0 cps = 200,000 pulses

During normal operation, the clutches are cycled at a rate of 1,200 engagements per minute, which is equivalent to an output dither of 20 cps. In 1 hour of running, the clutch would be energized 72,000 times.

The total clutch engagements based on 13 hours 54 minutes would amount to 1,000,000 clutch pulses and would represent the equivalent of almost 5 times the simulat l 1,000 hours of flight.

Sample recordings are presented in Figures 12, 13, and 14 corresponding to the three portions of the endurance test, which are:

1-A ± 100% stroke at 0.1 cps
1-B ± 50% stroke at 0.5 cps
1-C ± 10% stroke at 5.0 cps

To allow for continuous operation during the endurance test, the bypass lock-out feature was maintained so that the worm gear would not be misindexed and disengage from the worm drive. Input amplitudes were adjusted so that output rates would not adversely affect the input load torques. Frequencies were maintained as specified for the particular phase under test so that the equivalent number of load cycles would be obtained.

Endurance test requirements as shown in Table V indicated that testing was to be performed in five equal levels to permit distribution of loads and frequencies over the entire test program.

On two occasions during the endurance testing, the test fixture bypass lock-out pin sheared. The first failure occurred after 3 hours 22 minutes, and the second failure occurred after only 17 minutes of operation. It was decided at that time, due to the possibility of extensive damage occurring, that further testing should be discontinued. The unit was disassembled and carefully examined. The output worm gear that had been disengaged from the worm drive during a previous failure had a chipped portion on the last tooth of the gear. This was the only part in the entire unit that was damaged. All other hardware was found to be in very good condition.

The unit was lubricated and rebuilt with a new lock-out pin so that further testing could be accomplished if desired.

TABLE V. ENDURANCE TEST REQUIREMENTS

Cycling Group No.	No. of Cycles	Output Lever Motion Amplitude (deg)	Frequency (cps)
1-A	200	± 37.5	0.1
1-B	1,000	± 18.75	0.5
1-C	20,000	± 3.75	5.0
2-A	200	± 37.5	0.1
2-B	1,000	± 18.75	0.5
2-C	20,000	± 3.75	5.0
3-A	200	± 37.5	0.1
3-B	1,000	± 18.75	0.5
3-C	20,000	± 3.75	5.0
4-A	200	± 37.5	0.1
4-B	1,000	± 18.75	0.5
4-C	20,000	± 3.75	5.0
5-A	200	± 37.5	0.1
5-B	1,000	± 18.75	0.5
5-C	20,000	± 3.75	5.0

Note: Maximum temperature recorded during continuous endurance operation was 120°F after 3 hours 22 minutes. Phases 1-A, B, and C showed maximum temperature to be 110°F with no shutdown between settings.

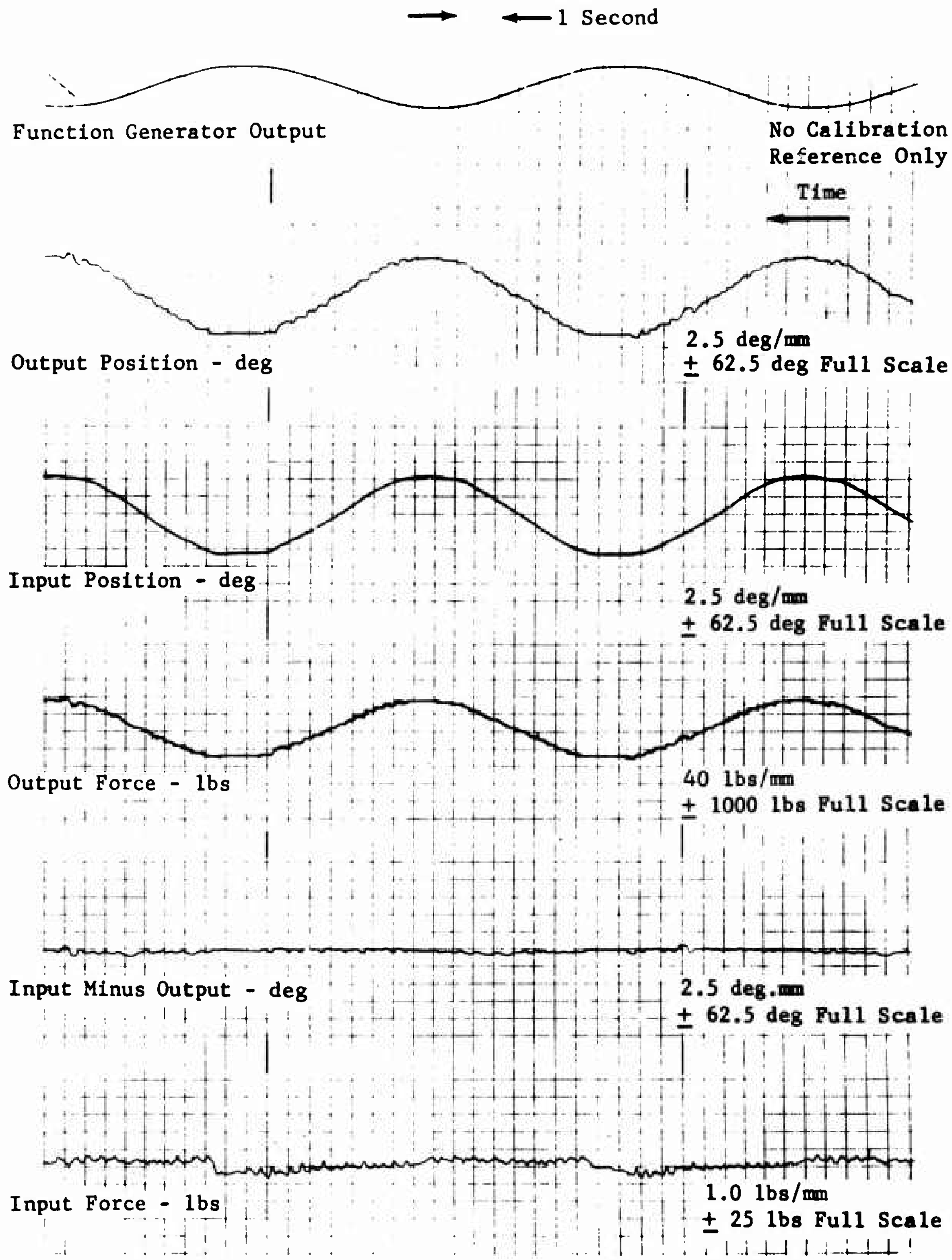

Figure 12. Endurance Cycling - Mechanical Actuator - 510-Pound Spring, 78-Percent Stroke, 0.1 cps.

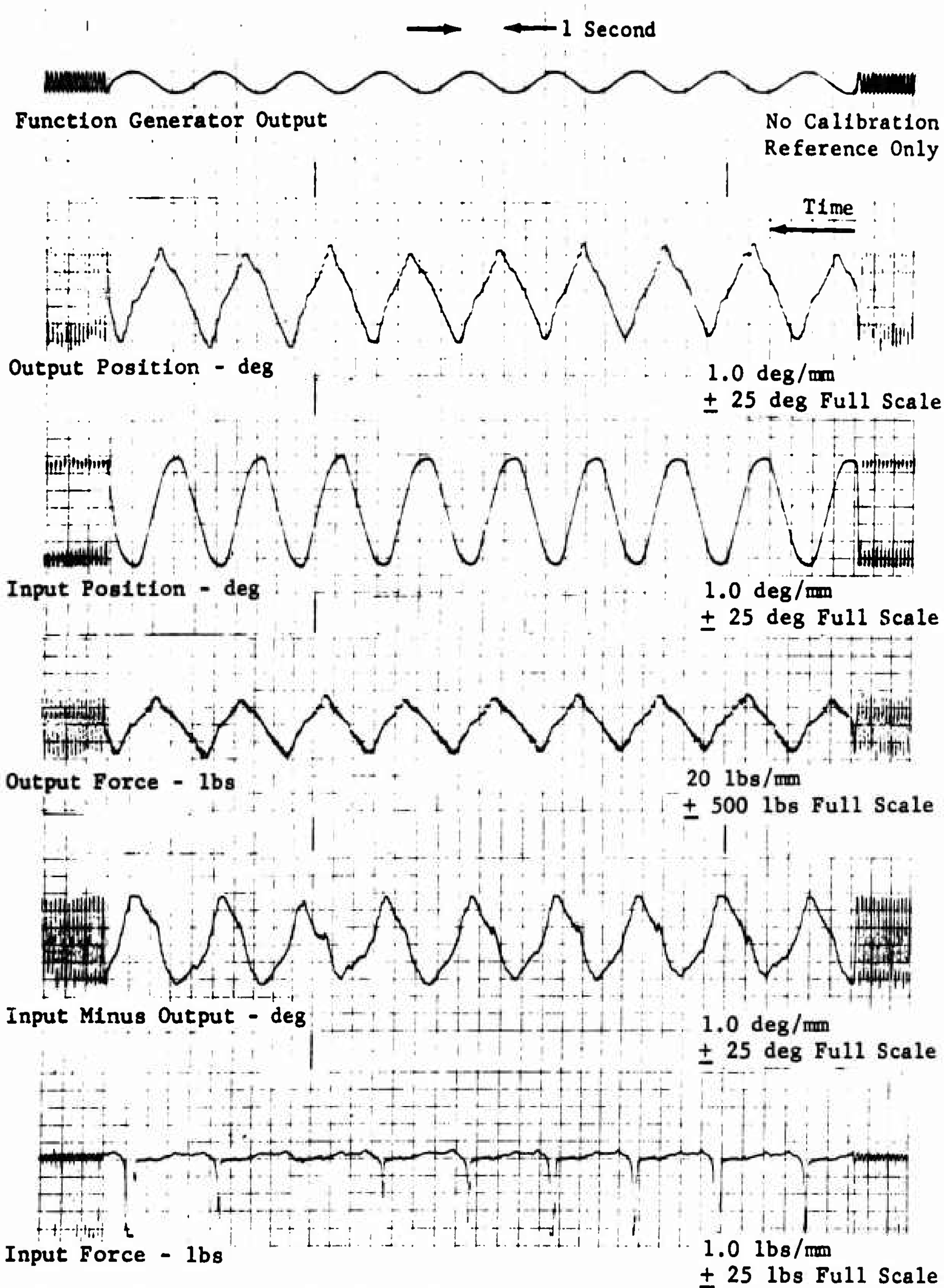

Figure 13. Endurance Cycling - Mechanical Actuator - 510-Pound Spring, 43-Percent Stroke, 0.5 cps.

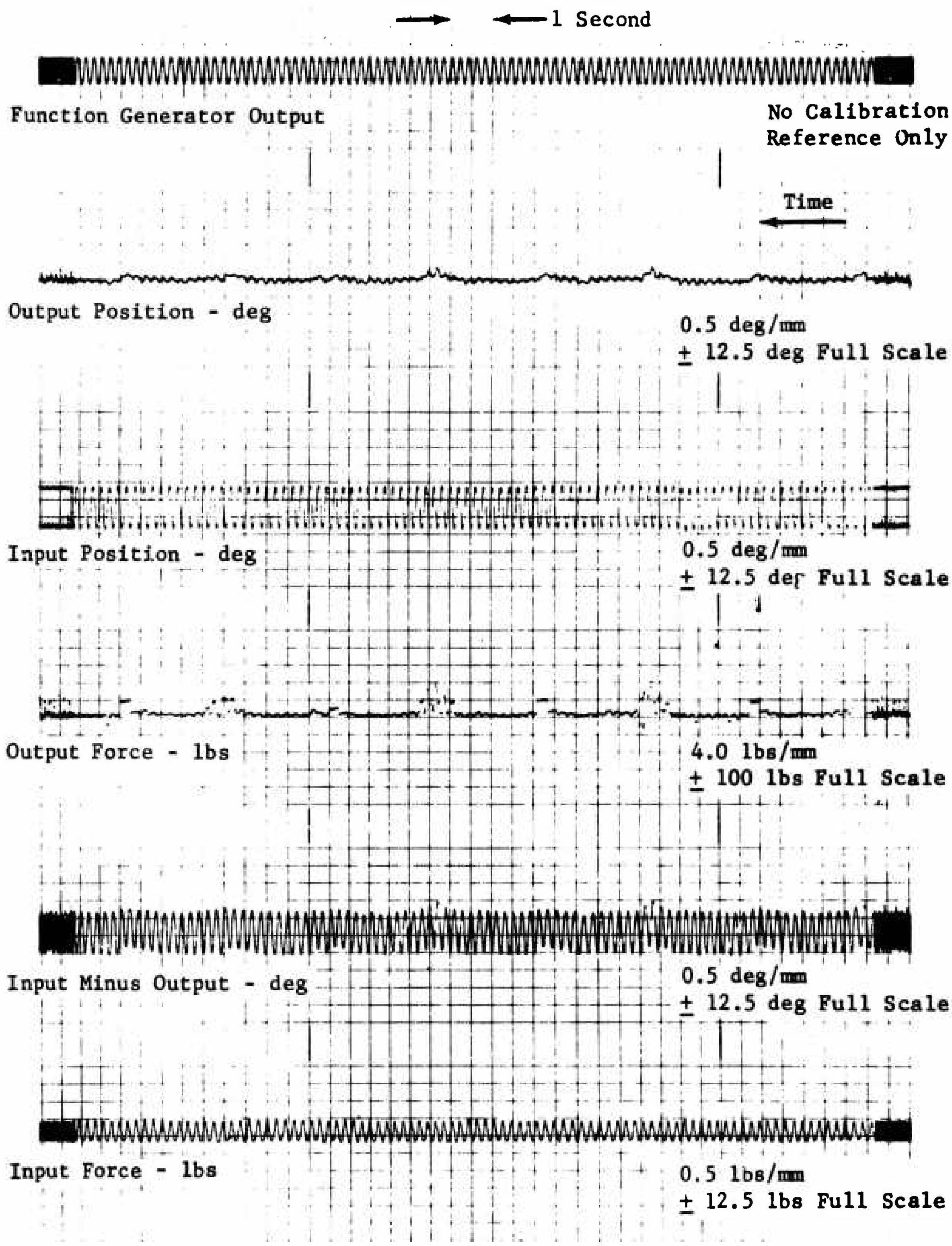

Figure 14. Endurance Cycling - Mechanical Actuator - 510-Pound Spring, 8-Percent Stroke, 5.0 cps.

MAXIMUM OUTPUT OPERATING TORQUE

This requirement was demonstrated by manually controlling the input to the hydraulic servo-actuator so that the mechanical servo-actuator would follow up within rate limitations of the output. Full excursions to ± 37-1/2 degrees indicated the ability of the unit to achieve and withstand full loads of 1,142 lb-in. and may be observed in the representative trace presented in Figure 15.

STEP RESPONSE

Figures 16 and 17 are samples of recordings taken with a 510-pound spring load and square wave input amplitudes of 43-percent stroke for both the clockwise and the counterclockwise clutch operation.

Instantaneous rates calculated for CW and CCW operation are 88 degrees per second and 105 degrees per second, respectively, with average rates considerably less due to insufficient stiffness of the input signal linkage.

SIGNAL THRESHOLD

The signal linkage has been designed to produce a maximum full rate signal at an input arm displacement of 0.75 degree. This feature was satisfactorily demonstrated. Signal full rate capabilities were also demonstrated by the installation of a second flange cam that was fabricated to produce a saturated signal with a 2.2-degree displacement of the input arm and was observed to be satisfactory.

DEAD BAND

By definition, the dead band is the signal arm displacement required from de-energization of one clutch to perceptible output motion of the other clutch.

The existence of oscillation, caused by a comparatively soft input control loop and tight settings of the clutch ramp rings, was sufficient to prevent the analysis of recorded data, even though damping occurred. It was believed that by opening the dead band to 0.33 degree, performance data would be more readily obtainable.

The influence of this increased dead band or; (the mechanical separation of the clutches) was sufficient to prevent the unit from achieving the maximum output rate of 15 rpm (90 degrees per second) at 6,300 rpm power input.

LUBRICATION

A small reservoir of MIL-L-5606 oil was intended in this design to provide for a splash type lubrication over the extended period of time. Lubrication of the system with MIL-L-5606 oil was found to have a noticeable effect on the energization of the clutches in their present configuration. With sufficient force available to produce full rate output as described above, the influence of oil would not be as significant.

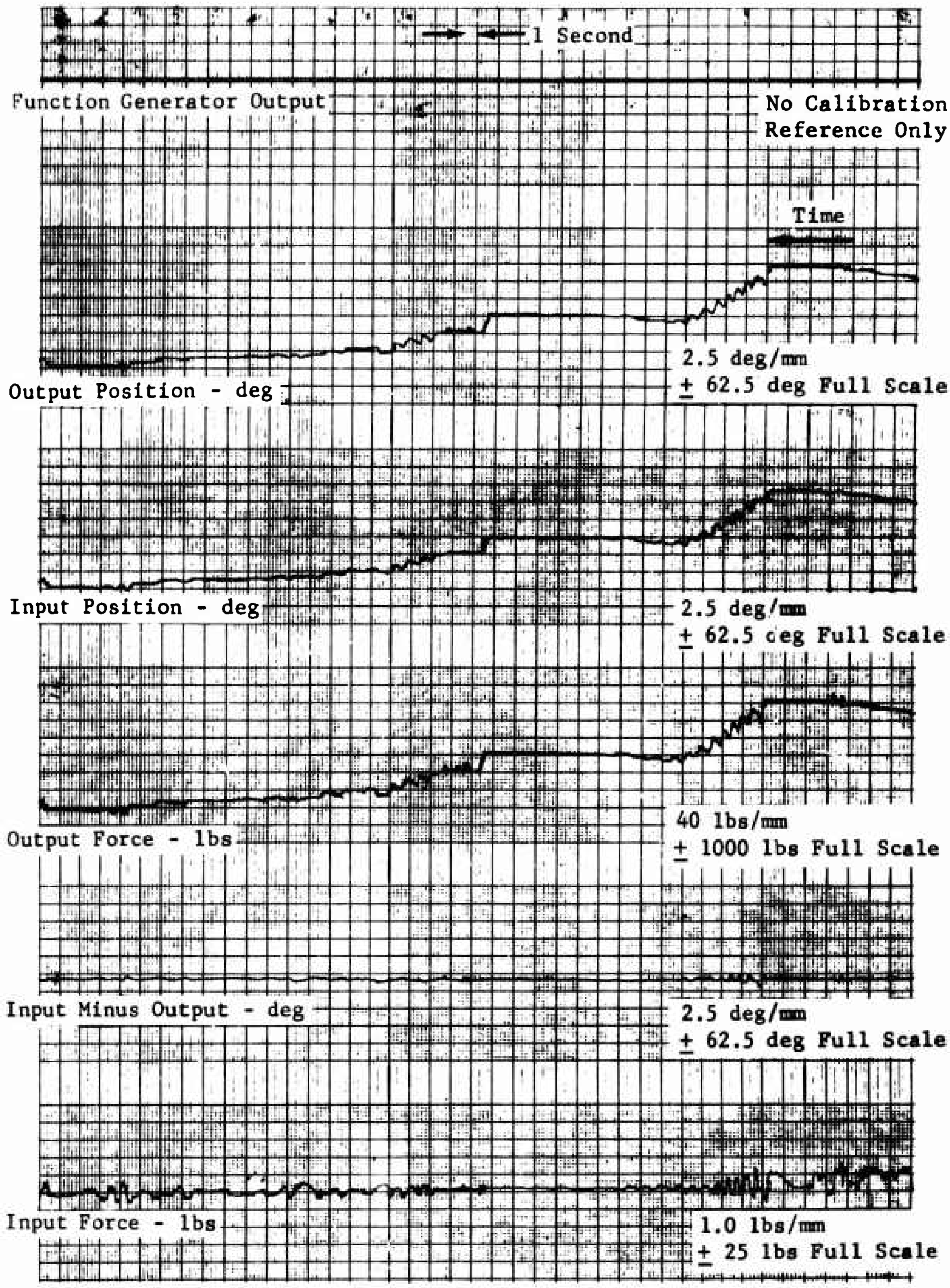

Figure 15. Manual Excursion - Full-Load Test, 510-Pound Spring, 100-Percent Stroke Clockwise and Counterclockwise.

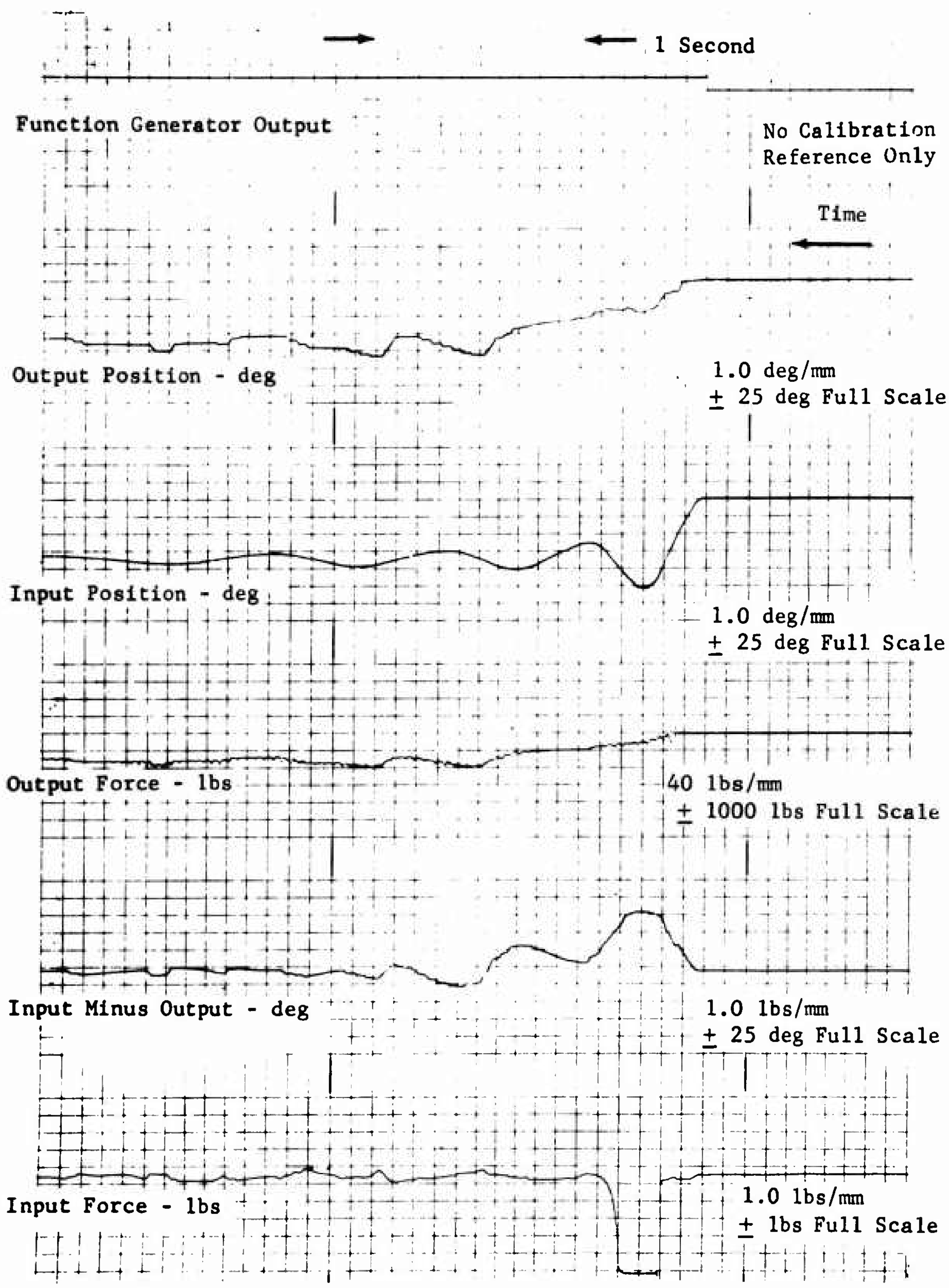

Figure 16. Step Response - Mechanical Actuator - 510-Pound Spring, 43-Percent Stroke Clockwise.

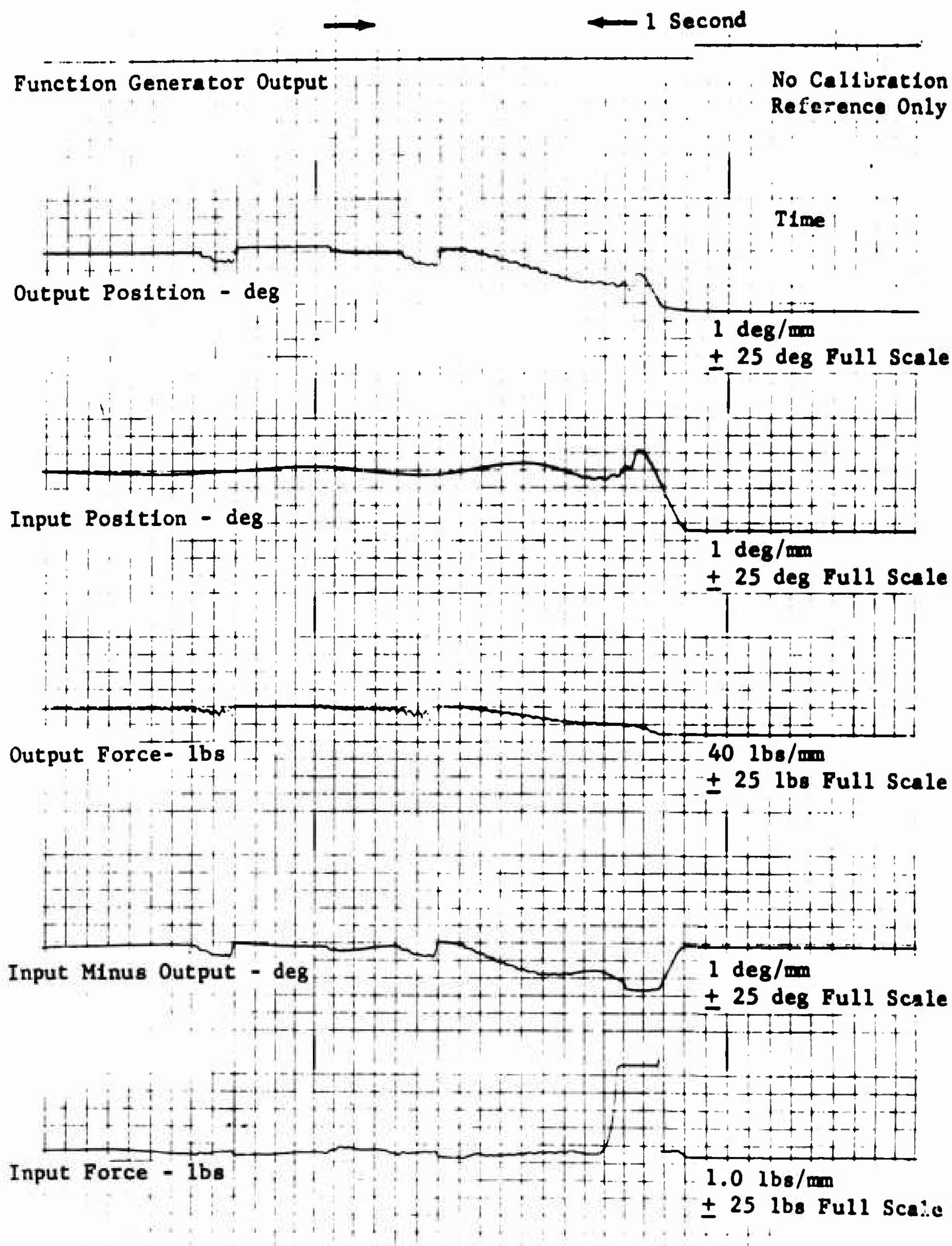

Figure 17. Step Response - Mechanical Actuator - 510-Pound Spring, 43-Percent Stroke Counterclockwise.

Previous experience has shown that there are numerous methods of eliminating this situation; two of these are:

1. Increasing unit pressure on the energizing disc.

2. Fabrication of springs with grooves on the OD to cut through the oil film for a more positive spring engagement. During all preliminary testing, the individual gears, bearings, etc., were lightly lubricated with MIL-L-5606 oil and relubricated when endurance testing was started.

Future designs would permit the incorporation of these features to assure sufficient lubrication over the entire life of these units.

CONCLUSIONS

Results of tests of the Curtiss-Wright mechanical servo-actuator have demonstrated the ability of this power boost system to perform the required cyclic and collective pitch functions of the swashplate control for the Lockheed XH-51A helicopter.

This program has permitted the development of an all-mechanical servo-actuator with capabilities equivalent to or better than those of the hydraulic system specifications presently in use.

The major contributions offered by this program are defined by the following system capabilities:

1. Extremely low input forces required to actuate the system.
2. Signal threshold level producing full rate within 0.75 degree of input motion.
3. Accuracy of positioning by the extremely small minimum pulses obtainable.
4. Ability of this system to handle the required loads and its insensitivity to output load variations.
5. Inherent characteristic of this system to prevent load feedback.

Development of this all-mechanical servo actuation system provides a mechanism that could have tremendous potential in flight control applications where power boost is essential.

The evaluation tests performed by the Lockheed-California Company on the Curtiss-Wright prototype unit indicated that several improvements would be required to enable this unit to comply with the performance characteristics of the hydraulic servo. Specifically, the force threshold and the signal dead band were the two major areas of concern.

The newly designed unit, incorporating all the required improvements, has demonstrated the ability of designing and fabricating an all-mechanical servo-actuator to meet the performance requirements of a hydraulic system.

Achievement of the goals set forth in this program, although satisfactorily accomplished, does not imply that the servo-actuator is ready to be installed in either a whirl tower or a test vehicle for pilot evaluation.

This program has demonstrated a need for additional design modifications to improve the mechanical servo override capability as well as further improvement in stiffness of the signal control loop.

Of necessity, the input signal path would require redesign for two reasons:

1. To permit satisfactory operation of the manual override. A 360-degree output gear must be used in place of the present sector gear. This new output gear will interfere with the present input signal linkage, making its relocation necessary.

2. To increase the stiffness of the input position control loop. This would satisfy the requirement of achieving small step increments consistently.

Evaluation of this mechanical servo was limited to a direct comparison with a hydraulic servo. Future programs should provide system specification so that full advantage may be taken of unique mechanical servo capabilities. The best results will be achieved when the mechanical system is matched to the application rather than compared to fluid system capabilities. Many of the features of the hydraulic system are indigenous to this type of system and are not necessarily specification requirements of the helicopter.

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13. ABSTRACT

This report discusses the results of a program conducted to design, fabricate, and test an all-mechanical servo-actuator with manual override capability for helicopter flight control application.

This all-mechanical system is a power boost with a mechanical override feature and provides mechanical amplification of signal torques from 0.5 to 1.0 lb-in. to operate output torque loads up to 1,142 lb-in.

The goal of this research program was to develop a mechanical servo-actuator to operate the helicopter swashplate, for cyclic and collective pitch control, with performance characteristics comparable to those of the hydraulic servo-actuator on the XH-51A helicopter.

The test item was subjected to a series of tests, including force threshold, resolution, frequency response, step response, and endurance. The results of this testing were compared with similar tests previously conducted on a prototype mechanical actuator as well as the performance characteristics of a hydraulic servo-actuator presently in use on the XH-51A helicopter.

Performance characteristics of the mechanical servo-actuator demonstrate that the actuator has met the requirements of the application when compared to the capabilities of the existing hydraulic power boost system.

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