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Title: RESULTS OF STICK-SERVO LOOP INSTABILITY AND STUDY

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ABSTRACT

An analysis has been made by MH in accordance with a request made by AVRO at the November coordination meeting.

The results of the REAC study of the "stick buzz" problem in the CF-105 Pitch axis gear down mode are presented. A solution to the problem is given along with other various schemes which were tried. REAC results indicate that a successful solution, one that will eliminate the "stick buzz", yet give an acceptable time response, requires two system modifications:

1. A viscous damper to provide a damping ratio of approximately .5 about the front quadrant (13 lb-ft/rad/sec.)
2. A first order lag in the stick signal to parallel servo with a time constant of .15 seconds.

Introduction

During the simulation of the gear down mode of the CF-105 pitch axis on the flight simulator at AVRO, a "stick buzz" was encountered. The frequency of oscillation varied from 10-25 cps with a peak to peak displacement on the order of .25 inches at the grip.

A short summary of the various approaches tried is given below:

Approach 1.

When the stick oscillation problem was first encountered, it appeared that a solution might be a mass balanced stick. By this is meant a stick whose grip displacement relative to the column displacement would be zero for all motions of the column. To state more simply, the transducer output would only be a function of the force applied by the pilot on the grip. This would mean a varying mass balance to compensate for the apparent mass variations supplied by the pilot's hand. Obviously this is impossible.

When it was finally determined that the Humphrey stick, the one to be used in the CF-105, was not mass balanced, an MH mass balanced stick was tried. The oscillations were reduced, but not stopped.

Approach 2.

At the suggestion of AVRO, the parallel servo was moved to the front quadrant. AVRO assumed that all of the spring rate of the mechanical linkage was due to the spring of the cables with no spring from the front quadrant to the grip. Therefore, by moving the parallel servo to the front quadrant, the spring rate would be omitted from the loop thus eliminating the stick oscillation. MH feels that the above assumption is not valid since the column is not rigid and does undergo actual bending when it is oscillated about the stick pivot. Results observed by simulating the parallel servo at the front quadrant did not indicate that the instability could be eliminated in this manner. In addition, MH is reluctant to agree to mounting the parallel servo at the front quadrant because of the expected phase lag between servo movement and power valve motion. This lag, based on past experience on other aircraft, could deteriorate damper and AFCS system performance.

Approach 3.

AVRO then suggested adding damping to the stick force transducer. Since the stick signal was due to the relative displacement between the grip and column, damping the transducer should lessen the stick signal for a given column movement. MH feels this is not the solution since the trouble was not in the grip but in the column and mechanical linkage combination. Furthermore, it was felt that the natural frequency of the grip was so far removed from the system frequency of oscillation that the damping of the grip was not an important factor.

Discussion

A schematic drawing of the physical system, system block diagram, and REAC simulation are attached as Figures 1 through 6.

Differential equations of the mechanical linkages were simulated rather than transfer functions. Physical system constants were determined by calculations and experimental methods. It is felt that the simulation is quite representative of the physical system except in possibly the following areas. A spring is shown connecting the front quadrant to the control tube. In the actual system they are connected by a solid rod. The spring simulates the spring rate of all the members from the top of the control column to the front quadrant. The grip simulated was a Humphrey grip. An accurate location of the center of gravity of the grip relative to the grip pivot point was not available, however, an estimate based on the physical construction of the grip was used. Accurate values of system frictions in the mechanical linkages were not available. The values used were based on previous experience with similar problems. Torque tube twisting was neglected because of the high spring rates associated with it.

A nebulous area, has been the characteristics of the parallel servo operating on the flight simulator with the mechanical linkages. To date, parallel servo characteristics for the CF-105 have been determined at MH by loading a servo through a spring to a concentrated mass with viscous friction. This is thought to be representative of the actual loads. The characteristics change considerably with different loads as expected. There is some doubt, however, what type of load the control system presents to the servo. As an approximation, a second order simulation with rate limits was used for the parallel servo simulation. MH has assumed that a 5 cycle servo with .7 damping should be quite representative of the servo in the control system.

The output of the parallel servo was connected to the rear quadrant through a spring. This was to simulate the hydraulic spring rate of the oil in the cylinder, lines and trim sensor. Nonlinearities used in the simulation were coulomb friction, rate limits on the parallel servo, and thresholds on the grip. Whenever these were used appropriate notation is indicated.

The system input used was a force on the grip. Without the pilot, the force was a step command on the grip. With the pilot in the loop, the grip force is a complex function of his intended command and of the grip motion due to movement of the column. At low frequencies the grip force is close to the intended input. Above the human-response cutoff, the stick grip is essentially stationary when firmly held. Thus, there are stick signals due to stick column motion beneath the grip. With the pilot in the loop, the grip force can be represented as the difference between the intended force and the force due to the high frequency motions of the column. In the analysis the force due to the high frequency motions of the column is obtained by a high-pass transfer function on grip position. The time constant of the hi-pass is assumed to be T = .3. The gain in the feedback loop is a function of the firmness of the pilot's grip.

With rate limiting on the parallel servo, the stick speed of response is dependent upon the magnitude of the command. During the study, it was assumed that the nominal rate on the parallel servo was 40 deg/sec. Subsequent information indicates the rate can vary from 40 to 80 deg/sec as a result of manufacture and environmental differences. It appears the solution devised will be valid for these increased rate limits. Response times for the same magnitude of command will be faster with the higher rate limits for commands which reach the rate limits.

Figures 7-a,b,c, respectively show the open loop transient response of the linkages to a 20 lb. step force on the grip for the following friction conditions:

1. 1 lb-ft/rad/sec damping at the quadrants and stick pivot,

2. .1 damping ratio for the linkages,

3. .5 lb-ft coulomb friction at the quadrants and stick pivot.

The natural frequency of the free linkages is 10-12 cps. This agrees with experimental test data.

Figure 7-d shows the closed loop transient response to a disturbance of the system with 1 lb-ft/rad/sec damping at the quadrants and stick pivot. The system oscillates at 17 cps with an amplitude of ±.175 inches at the grip. The natural frequency of the parallel servo was 8 cps with a damping ratio of .7.

Figure 7-e shows the closed loop transient response to a 20 lb. step command on the grip of the system with .5 lb-ft coulomb friction at the quadrants and stick pivot. The system oscillates at 20 cps with an amplitude of ±.18 inches at the grip. The natural frequency of the parallel servo was 10 cps. with a damping ratio of .7.

RESULTS AND CONCLUSIONS

Summary of possible solutions evaluated on REAC.

1. Viscous Damping of Linkages

As a first solution to the problem viscous damping of the linkages was considered. Two convenient points to add viscous damping are at the front quadrant and at the grip. It appears that the damping of the grip within reasonable limits is not an important factor for system stability because its natural frequency is higher than the frequency of instability. Viscous damping at the front quadrant alone did not solve the problem. Only when additional viscous damping was added about the stick pivot was the instability corrected. In the physical system, this "fix" would not be feasible because the spring associated with the column is not between the front quadrant and the column bottom as simulated, but it is distributed along the length of the column. Thus, viscous damping could not be added about the stick pivot to obtain damping at the top of the column.

2. Integrate CSS; Sum Parallel Servo Position with CSS.

An effort was made to use the integrator that is in the CSS loop in the gear-up mode. Instead of summing normal acceleration with stick signal, parallel servo position would be summed with the stick signal. It was felt that the integration would solve the instability by attenuating the frequencies at which the instability was occurring. This configuration solved the problem of instability, however, was unacceptable due to poor response to a step command force on the grip. With the normal gain between the transducer displacement and the rear quadrant position the integration rate was too slow to allow the parallel servo to operate at its maximum rate limits. By increasing the loop gain the response was faster, however, the system went unstable before the rate limits on the parallel servo were reached. Figures 8-a,b,c, show typical REAC traces for this configuration with different forward loop gains. Because of the slow responses, this solution was unacceptable.

3. Move the Parallel Servo to the Front Quadrant.

Moving the parallel servo to the front quadrant did not solve the problem. The system with 1 lb-ft/rad/sec damping at the quadrants and stick pivot oscillated at 14 cps with an amplitude of ±.12 inches at the grip. It appears that the lower frequency of oscillation is due to the lower natural frequency of the column spring-mass system. Two objections to the front quadrant location are:

1. Damping of the column cannot be easily accomplished,
2. A lag between parallel servo movement and valve motion is introduced because of cable stretch.

4. First Order Lag in CSS to Parallel Servo.

Another solution that was tried was the addition of a first order lag between the stick force signal and the parallel servo to attenuate the frequencies at which the system was unstable. Frictions used in evaluating this solution were 1 lb-ft/rad/sec at the front and rear quadrants and at the column pivot. Time constants on the lag were varied from .1 to 1 sec. It appears that a lag which will provide system stability will not give an acceptable time response. Figures 9-a,b,c, show the effects of various lags with the frictions of 1 lb-ft/rad/sec used for a 5 lb. step command using and 8 cps parallel servo with .7 damping. At this point it was realized that this value of friction was rather pessimistic. AVRO analysts had used a damping ratio of .1 for the linkages. Figure 10-a shows the results of using such a damping ratio for the mechanical linkages, a .15 sec. lag and a 5 cps parallel servo with .7 damping. It is felt by the author that a damping ratio of .1 for the mechanical linkages is an optimistic value. Further discussions indicated that a value of .5 lb-ft of coulomb friction about both quadrants and the column pivot is quite representative of the actual system. This value of friction was used in determining the final system fix.

5. Viscous Damping and First Order Lag in CSS to Parallel Servo.

The solution which seemed to provide the best performance was a combination of a first order lag on the stick signal and viscous damping at the front quadrant. With coulomb friction values of .5 lb-ft at the quadrants and at the stick pivot the values chosen, based on REAC results, were: (1) The lag between the stick force signal and the parallel servo has a time constant of .15 sec., (2) The viscous damping is chosen to give the front quadrant a damping ratio of .5. This is equivalent to 13 lb-ft/rad/sec at the front quadrant. The nominal solution is based on the use of a 5 cycle parallel servo with .7 damping. Figures 10-b,c, respectively, show the nominal case without and with the pilot loop in. The system is stable for ±50 percent variations in the lag time constant, the viscous damping force about the front quadrant, the loop gain, and parallel servo damping singly or all at once. The above statement is valid with and without the pilot loop in the system over a parallel servo natural frequency range of 2-10 cps. The system is stable for the above gain variations singly for a parallel servo natural frequency range 11-15 cps. With rate limits on the parallel servo the response times to a step command are dependent upon the magnitude of the command. For the nominal case with 40 deg/sec rate limits, the time to 90°/o of the final value for a 20 lb. force is .8 sec. This 20 lb. force commands 18 degrees of rear quadrant. It appears that this response time will be adequate.

Effects of Grip Threshold

For step force commands the problem is not affected by the grip threshold. For a free grip, however, the threshold has an effect on the amplitude and frequency of oscillation of the limit cycle. With the thresholds used, ±2 lbs at the grip, the frequency and amplitude were slightly lower than without the threshold.

Effects of Grip Damping

It appears that the damping of the grip within limits is not an important factor in the stability of the system. The grip damping ratio was varied from .1 to 1 with no apparent effect on system stability. The natural frequency of the grip is above the frequency at which the system oscillates. On the traces this shows up as a higher frequency mode superimposed on the lower frequency of the system. A damping ratio of .5 for the grip was used throughout.

Effect of Parallel Servo Natural Frequency.

Increasing the natural frequency of the parallel servo has a detremental effect on system stability. As the natural frequency of the parallel servo is increased, the system frequency increases and the system damping decreases. Figures 7-d, 11-a,b,c, illustrate this for a 20 lb step command on the grip. In all of the above runs .7 damping of the parallel servo and .5 lb-ft of coulomb friction at the quadrant and stick pivot were used. The system with the 2 cps parallel servo is stable whereas the systems with the 5 cps -15 cps parallel servos are unstable. The stick signal is not lagged in the above runs.

Effect of Pilot Loop

Figure 4a shows a rearranged system block diagram for clarity. In the rearranged diagram the hi-passed pilot loop appears as two blocks. In block (1) the hi-pass has the effect of adding lag to the system. It appears that sufficient lag is added to the system to have a destabilizing effect on the system at the natural frequencies which were used for the parallel servo. Block (2) is shown as hi-pass outer loop feedback. This seems to have a stabilizing effect on the system at "buzz" frequencies.

Proposed Solution and Action

As a result of this study, MH proposed: (a) the addition of a first order lag between the stick signal output and the parallel servo with a .15 second time constant and, (b) the addition of viscous damping at the front quadrant to provide 13 lb-ft/rad/sec at that point.

In order to verify and implement the proposed solution, MH proposes the following steps:

(1) AVRO and MH should evaluate the proposed solution on the B-1 rig to:

(a) Verify the acceptability of the solution,

(b) Evaluate the effect on the gear up mode,

(c) Evaluate the effect on the manual mode.

(2) MH should proceed with the design of a non-mechanical filter to provide a lag of .15 seconds, concurrently with step (1).

(3) Upon completion of items (1) and (2) a plan should be evolved for the incorporation of the "fix" into all MH-64 systems.

Note: If need be, the viscous damper could be switched out on the manual mode or the time lag could be switched out on the gear-up mode.

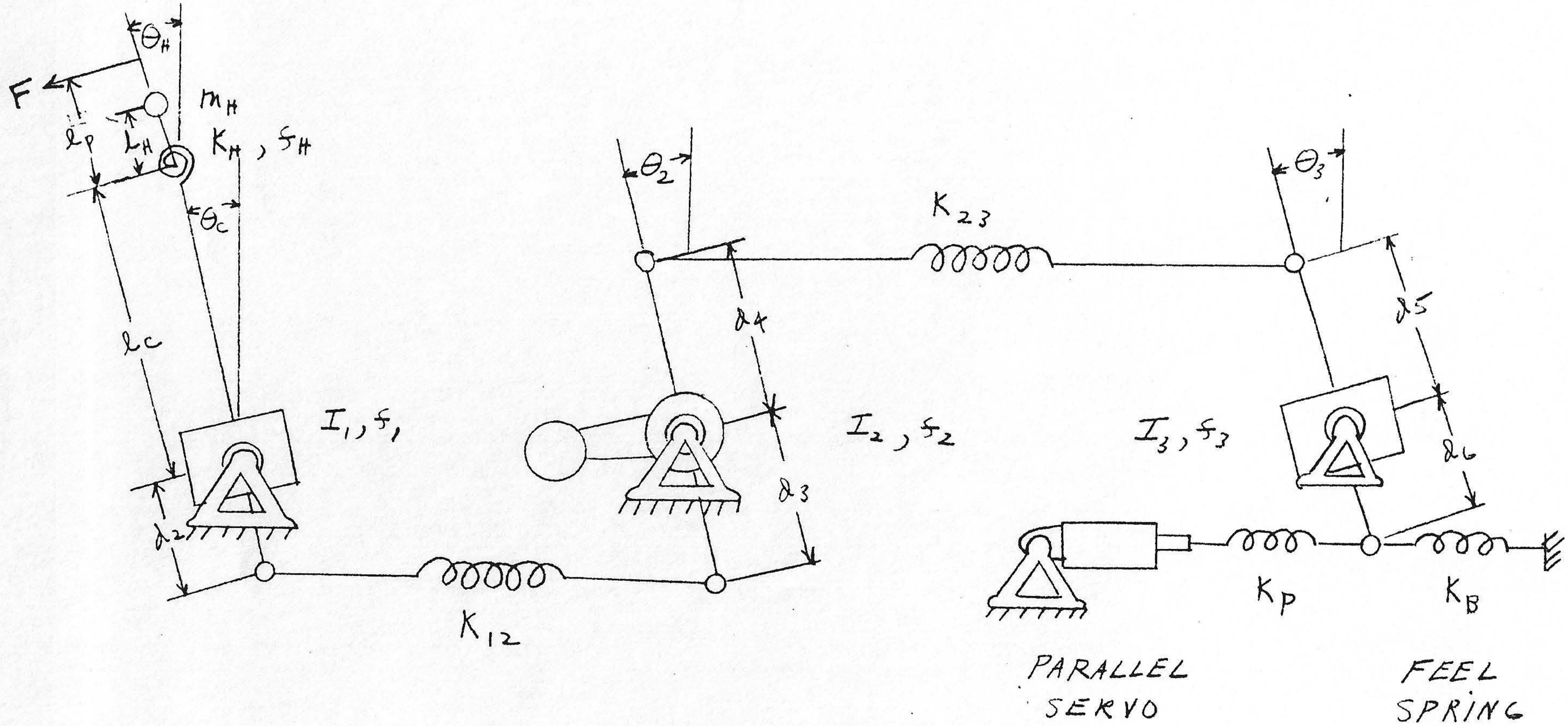


Fig. 1 FORWARD LINKAGE

Mechanical Linkage Equations

$$\ddot{\theta}_H = \frac{-\dot{\theta}_H f_H - \theta_H[-m_H g l_H + K_H] - \ddot{\theta}_c m_H l_H l_c + \theta_c K_H + F l_p}{m_H l_H^2}$$

$$\ddot{\theta}_c = \frac{-\dot{\theta}_c f_1 - \frac{\dot{\theta}_c}{|\dot{\theta}_c|} f_{1col} - \theta_c[\delta_2^2 K_{12} + K_H] + \delta_2 \delta_3 K_{12} \theta_2 + K_H \theta_H}{I_1}$$

$$\ddot{\theta}_2 = \frac{-\dot{\theta}_2 f_2 - \frac{\dot{\theta}_2}{|\dot{\theta}_2|} f_{2col} - \theta_2[\delta_3^2 K_{12} + \delta_4^2 K_{23}] + \delta_3 \delta_2 K_{12} \theta_c + \delta_4 \delta_5 K_{23} \theta_3}{I_2}$$

$$\ddot{\theta}_3 = \frac{-\dot{\theta}_3 f_3 - \frac{\dot{\theta}_3}{|\dot{\theta}_3|} f_{3col} - \theta_3[\delta_5^2 K_{23} + d_6^2 K_p + \delta_6^2 K_B] + \delta_4 \delta_5 K_{23} \theta_2 + \delta_6^2 K_p \theta_p}{I_3}$$

θ_H grip angular motion
θ_c column angular motion
θ_2 front quadrant angular motion
θ_3 rear quadrant angular motion
θ_p parallel servo angular motion

Fig. 2

Fig. 3 Values Used

ζ_1	1 lb-ft/rad/sec	ζ_{1col}	.5 lb-ft
ζ_2	1 "	ζ_{2col}	"
ζ_3	1 "	ζ_{3col}	"

I_1 .368 slugs-ft^2

I_2 .068 slugs-ft^2

I_3 .12 slugs-ft^2

m_H .0465 slugs.

K_H 232 lb-ft/rad

K_{12} 10,000 lbs/ft

K_{23} 6,600 lbs/ft

K_p 24,000 lbs/ft

K_B 900 lbs/ft

l_H .29 ft

l_p .278 ft

l_c 2.0 "

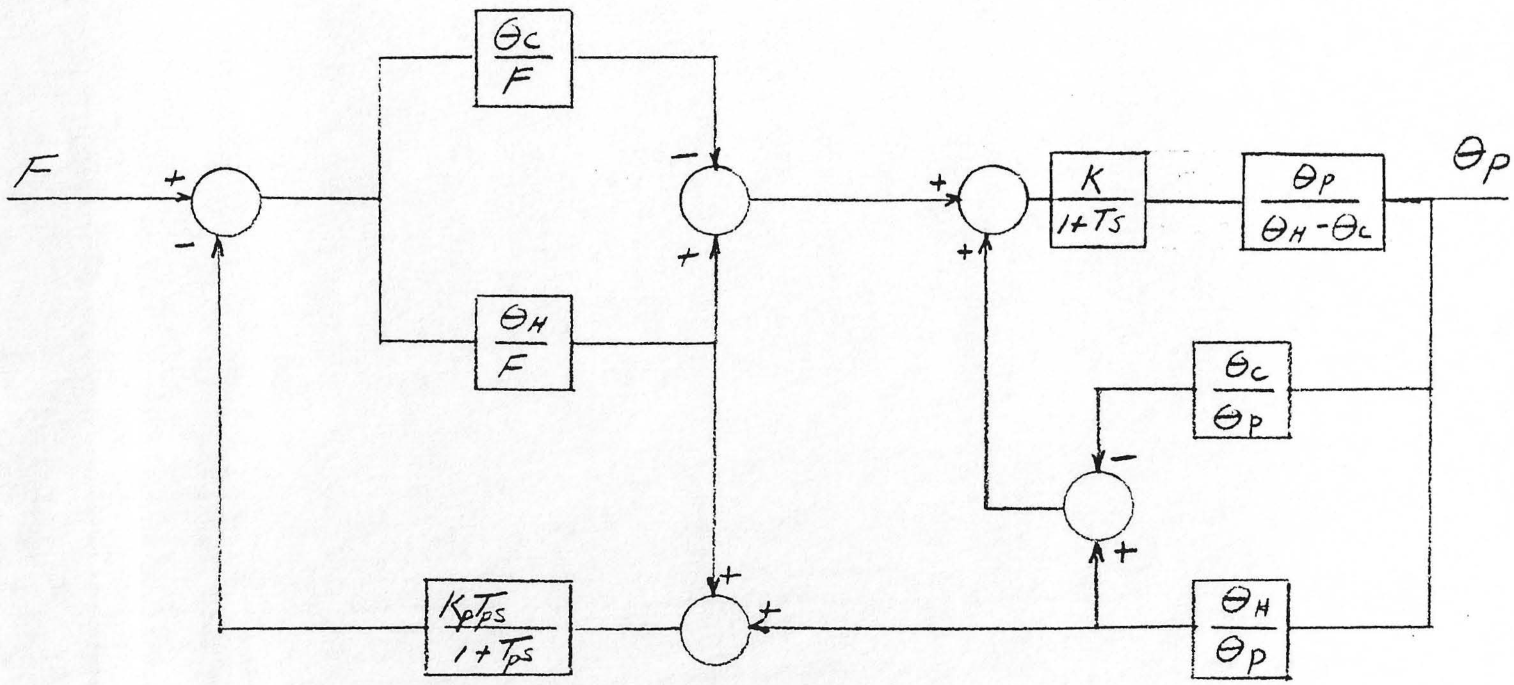
d_2 .692 "

d_3 .292 "

d_4 .5 "

d_5 .496 "

d_6 .250 "



K = GAIN OF $\frac{\text{Parallel servo displacement}}{\text{Transducer displacement}} \sim 15$ for nominal CASE.

Fig. 4 SYSTEM BLOCK DIAGRAM

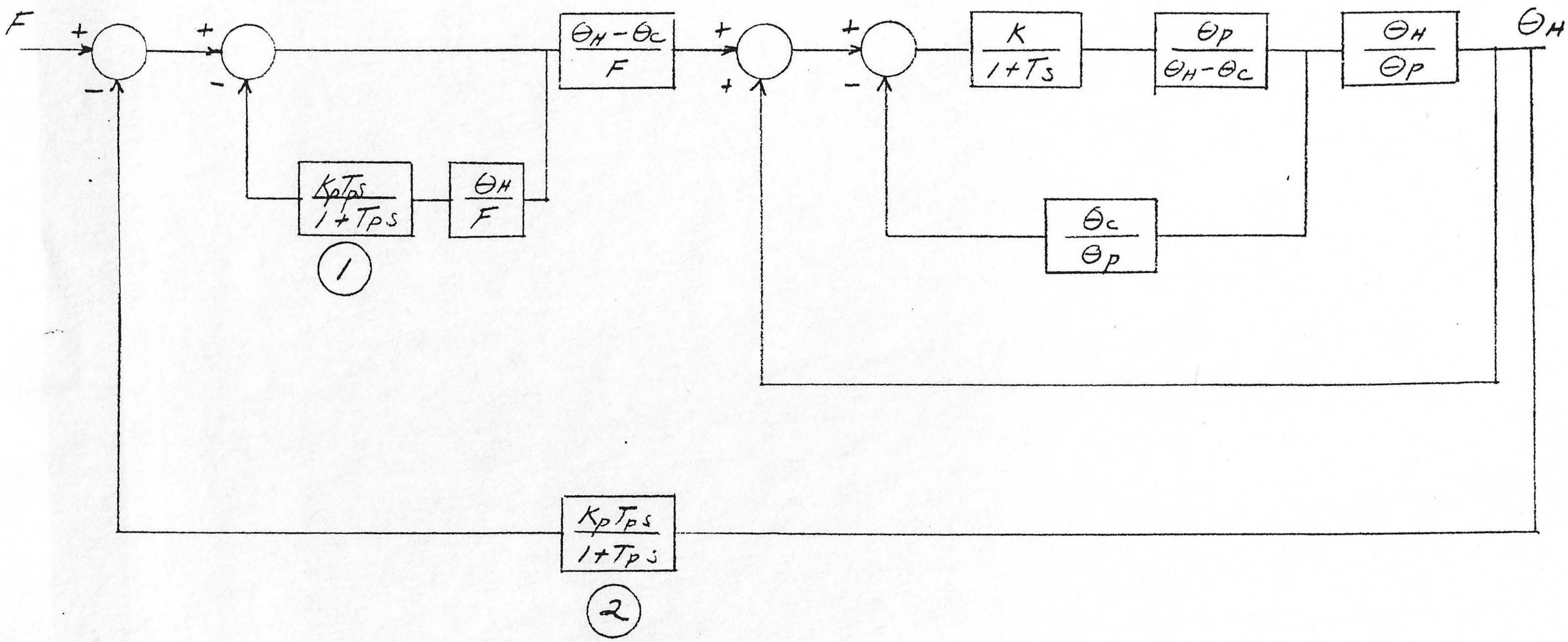
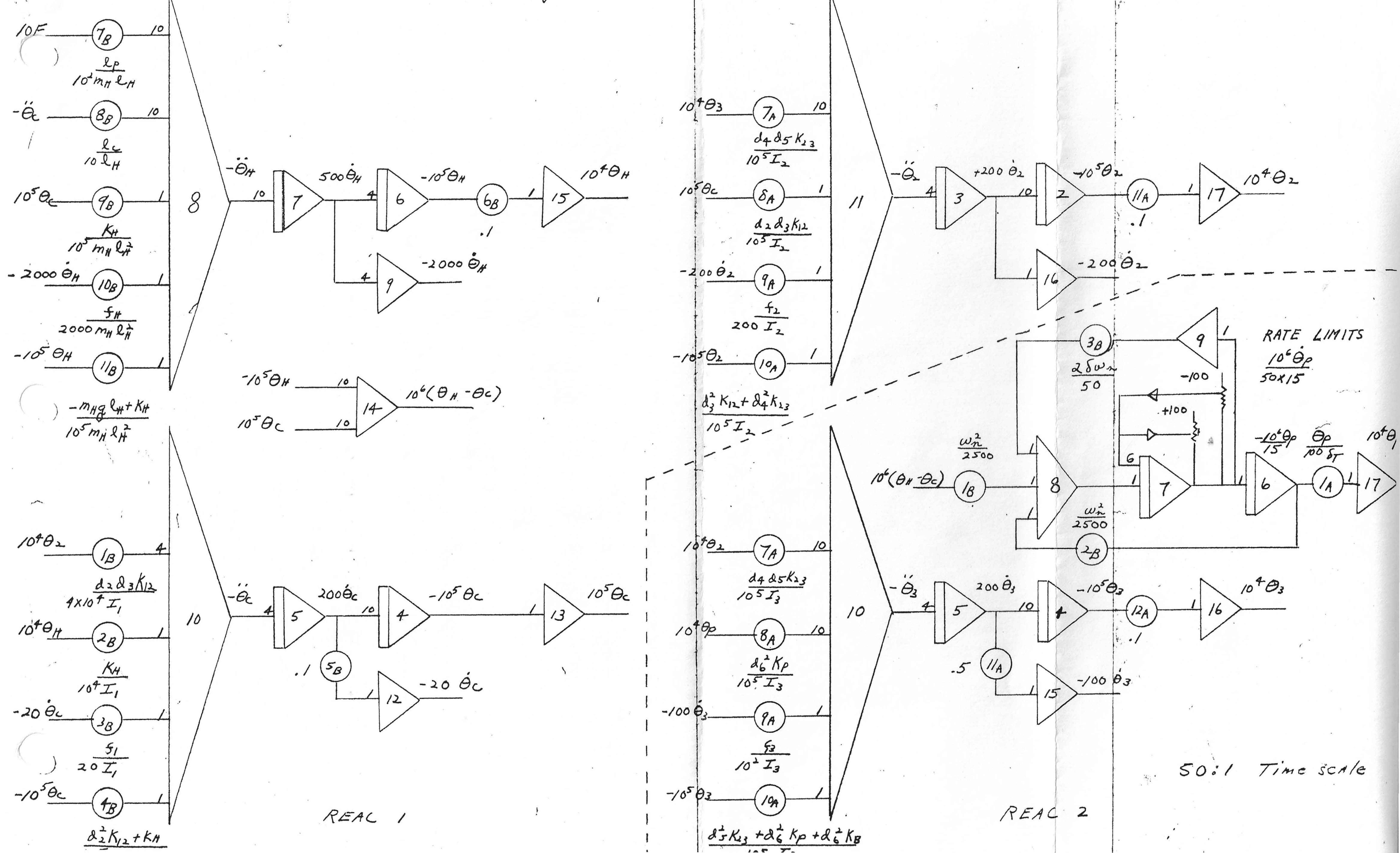
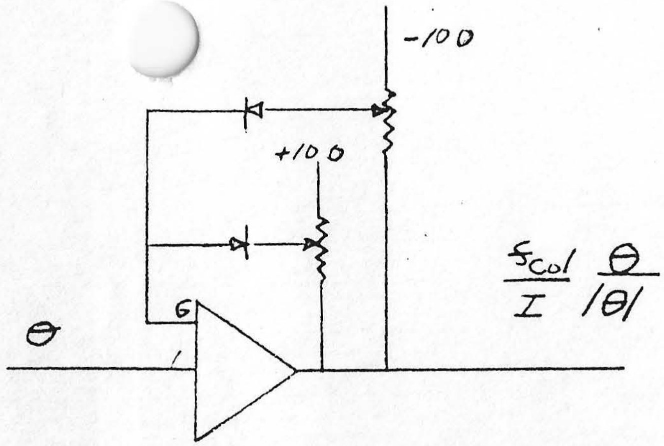


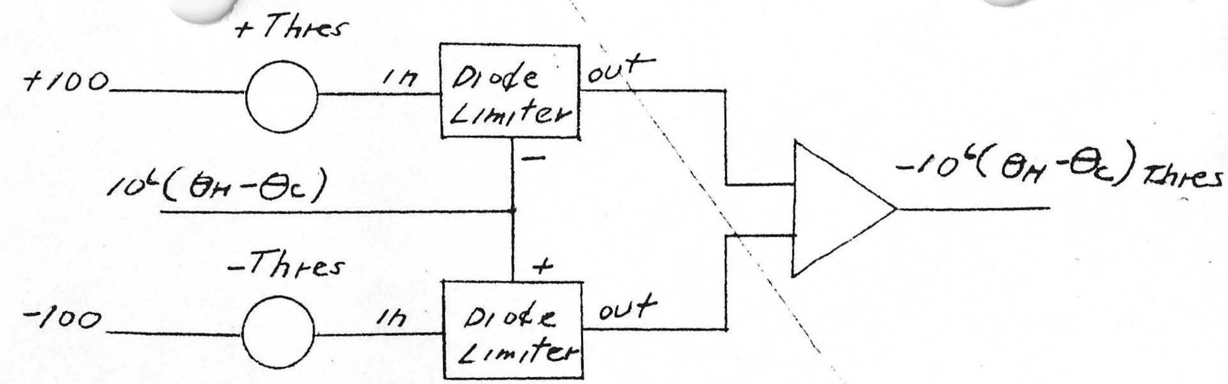
Fig. 4a. REARRANGED SYSTEM BLOCK DIAGRAM

Fig. 5

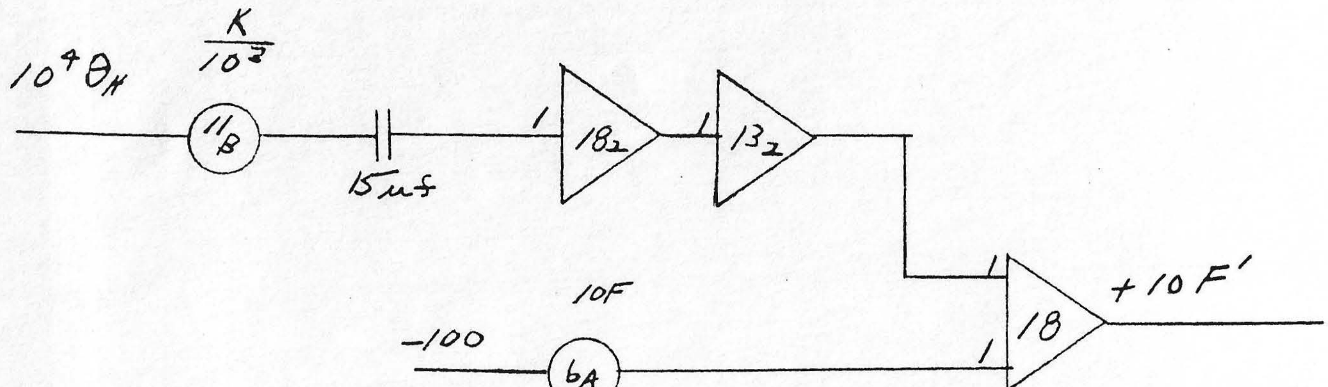




Columb Friction Simulation



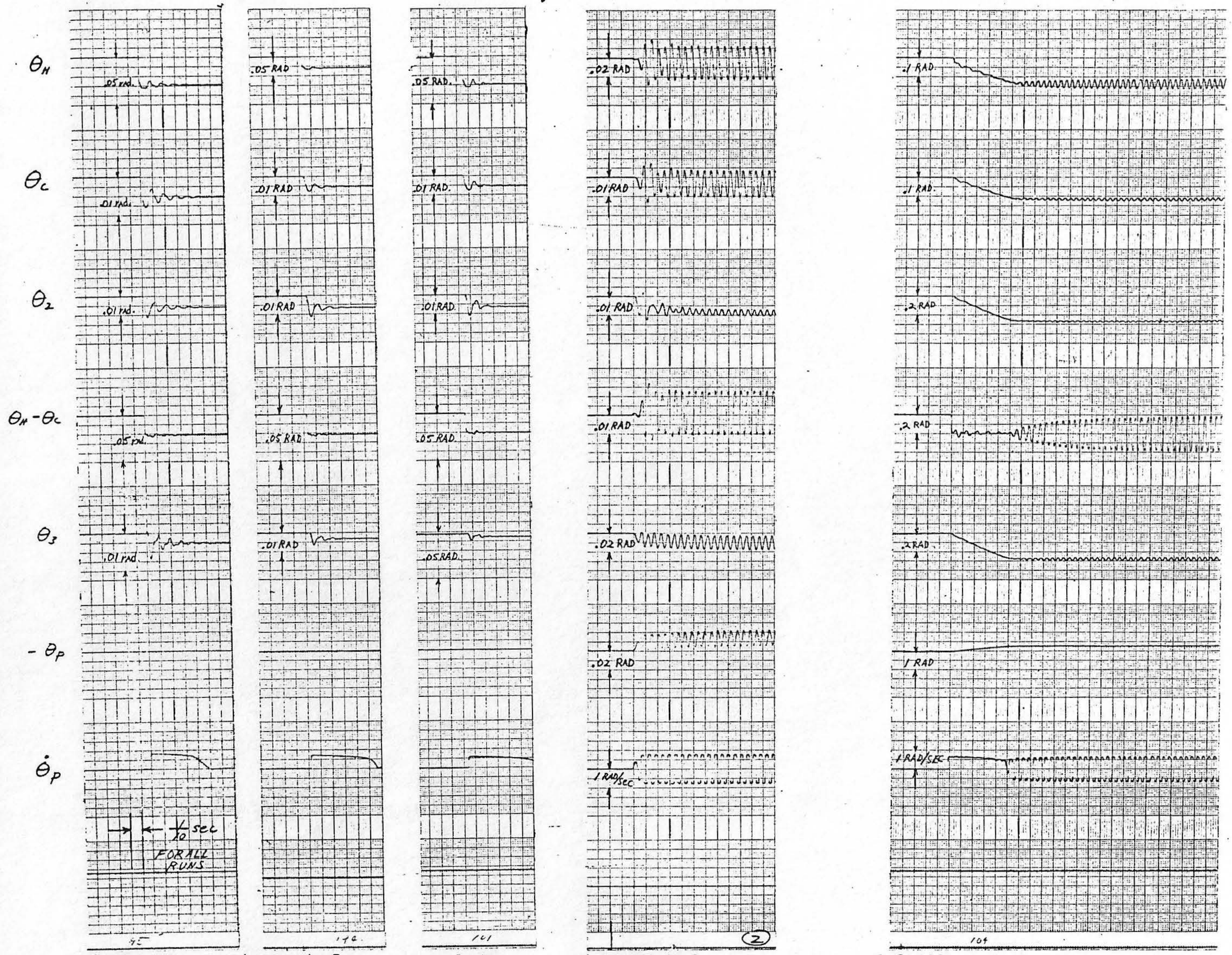
Grip Threshold ± 2 lbs



Pilot Loop

Fig. 6 Nonlinearity simulation

Figure 7.



a. OPEN LOOP
20# FORCE
1 lb-ft/RAD/SEC

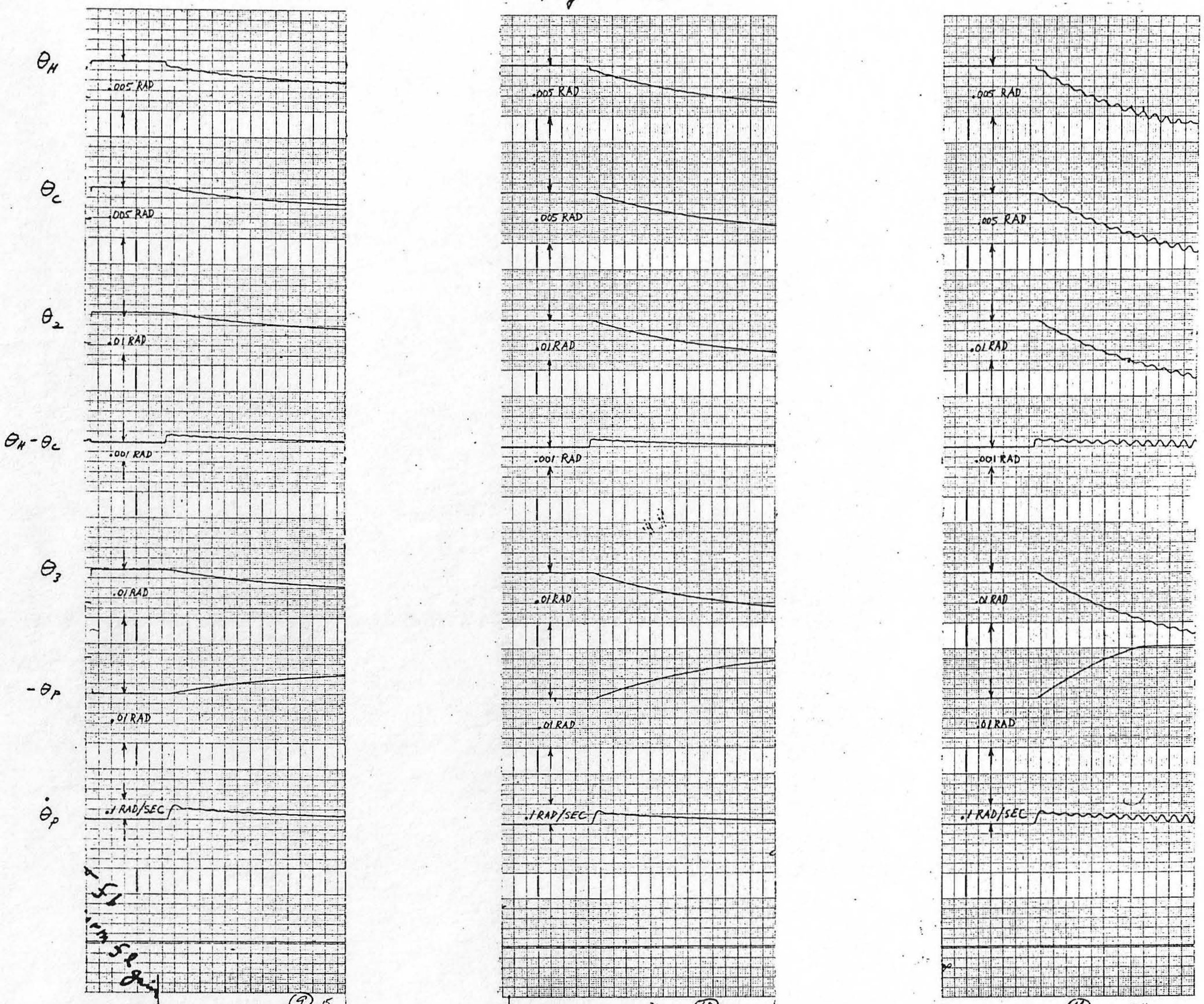
b. OPEN LOOP
20# FORCE
.1 DAMPING

c. OPEN LOOP
20# FORCE
.5 lb-ft Coulomb

d. CLOSED LOOP
NO LAG
1 lb-ft/RAD/SEC
8 cps PS

e. CLOSED LOOP NO LAG
20# Force
.5 lb-ft Coulomb
10 cps PS
2# Theshold

Figure 8.



a. INTEGRATION 8cps PS
LOOP GAIN 15
1 lb-ft/RAD/SEC
.25 # FORCE

b. INTEGRATION 8cps PS
LOOP GAIN 30
1 lb-ft/RAD/SEC
.25 # FORCE

c. INTEGRATION 8cps PS
LOOP GAIN 60
1 lb-ft/RAD/SEC
.25 # FORCE

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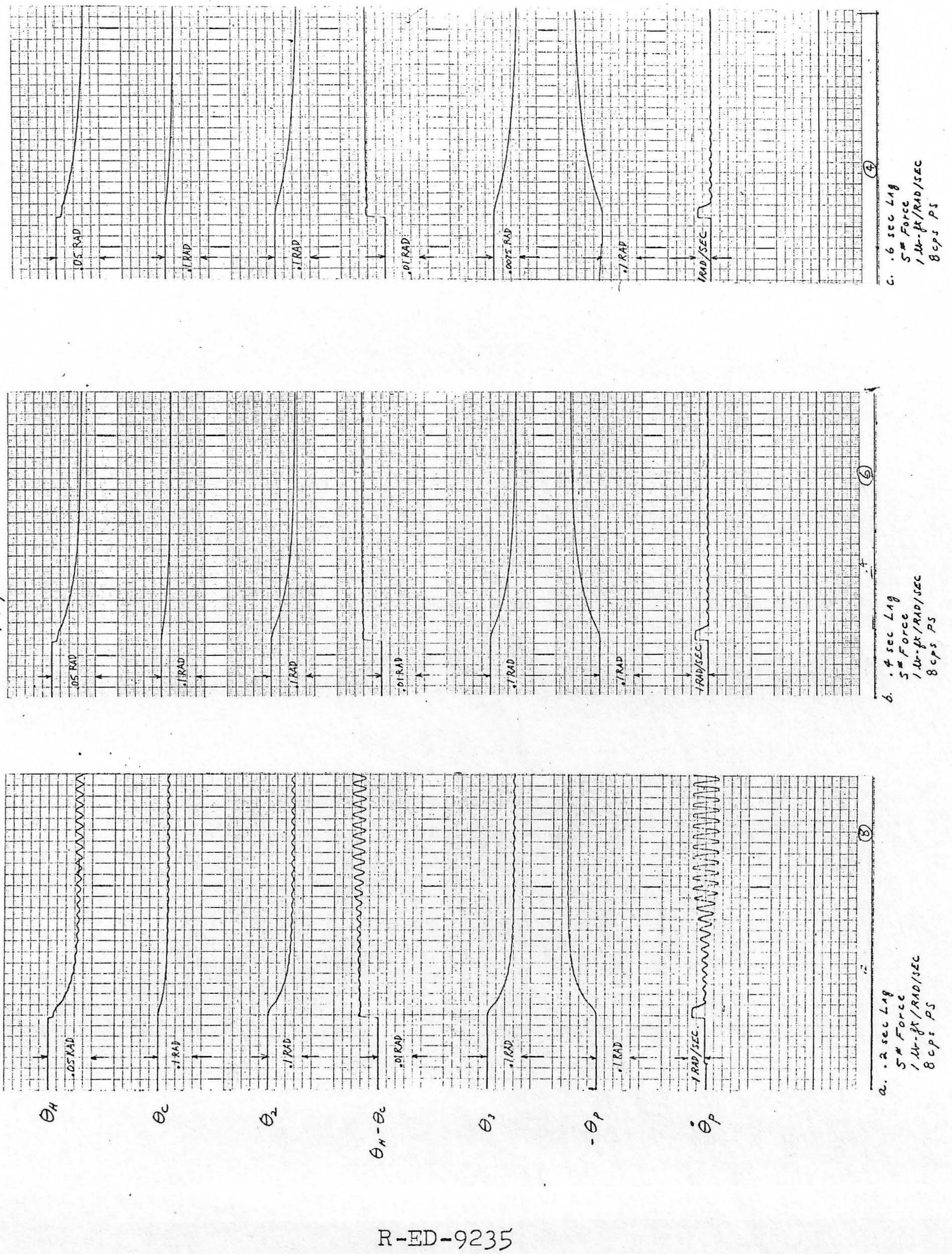
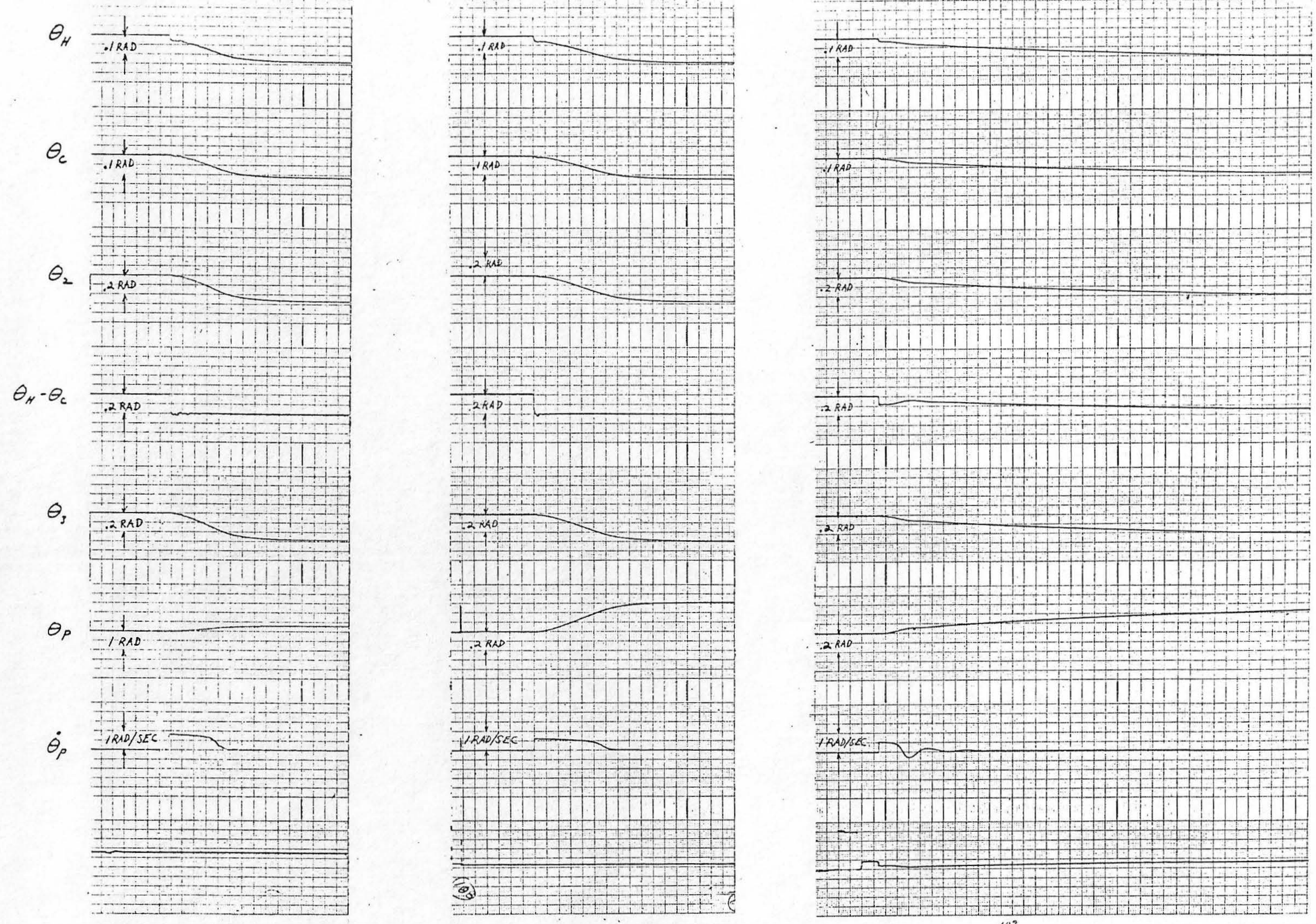
Figure 9,



θH
θc
θ2
θH - θc
θ3
- θP
θ̇P
.05 RAD
.1 RAD
.1 RAD
.01 RAD
.1 RAD
.1 RAD
1 RAD/SEC.
.0075 RAD
a. .2 sec Lag
5# Force
1 in-#/RAD/SEC
8 cps PS
b. .4 sec Lag
5# Force
1 in-#/RAD/SEC
8 cps PS
c. .6 sec Lag
5# Force
1 in-#/RAD/SEC
8 cps PS
8
6
4

Figure 10.

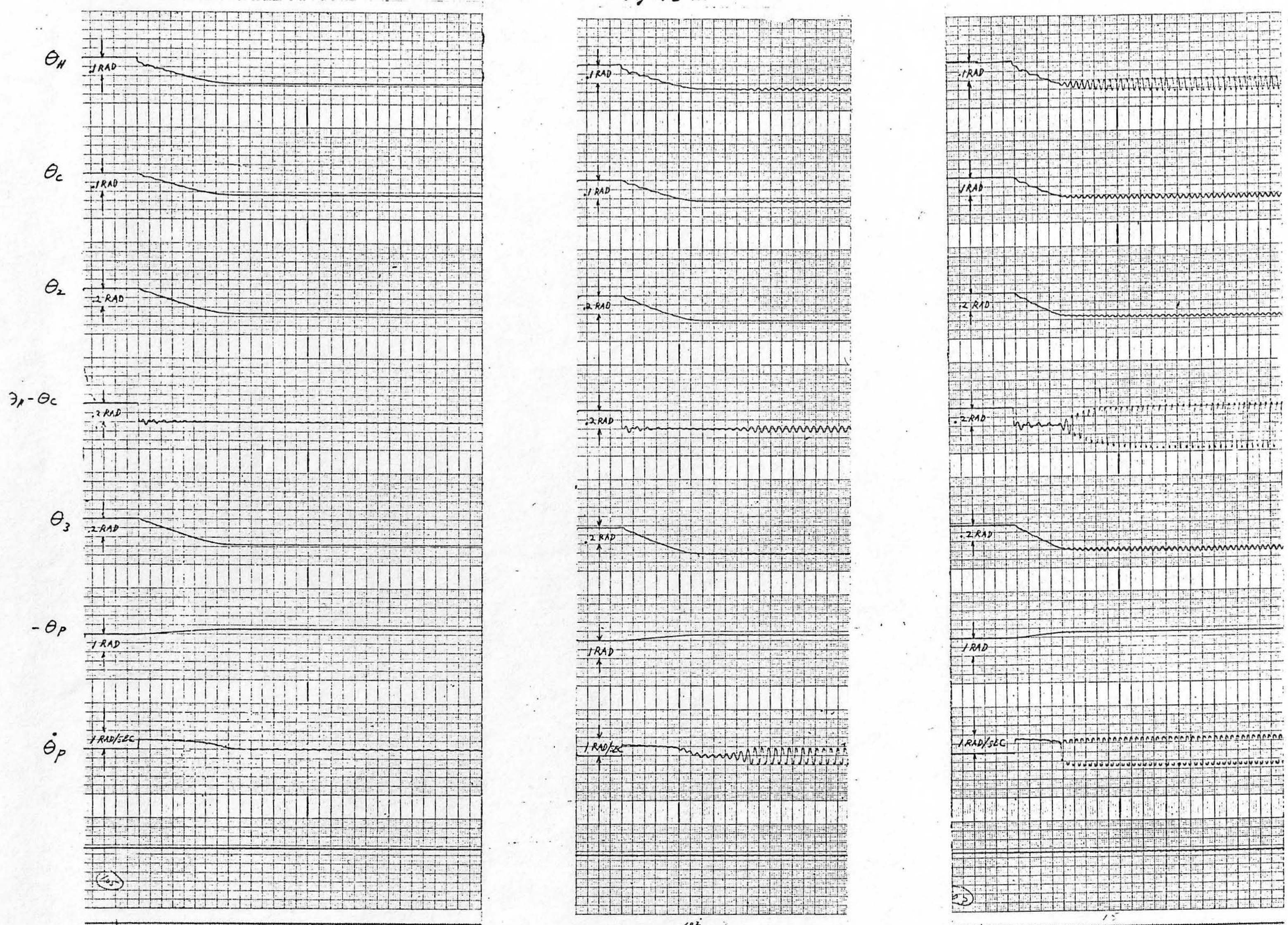


a. .15 sec Lag
20# Force
.1 Damping
2# Threshold
5 cps PS

b. .15 sec Lag, 20# Force
.5 lb-ft coulomb, 2# Threshold
13 lb-ft/RAD/SEC AT FRONT QUAD
5 cps PS

C. Same As b. With Pilot Loop

Figure 11.



a. NO LAG   2 cps PS
.5 lb-ft COULOMB
20# FORCE
2# THRESHOLD

b. SAME AS a. WITH 5 cps PS

c. SAME AS a. WITH 15 cps PS