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D. A. Babcock

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Langley Research Center
Hampton, Virginia 23665-5225

Simulated Dynamic Response of a Servovalve Controlled Hydraulic Actuator

By: D. A. Babcock

Abstract

A general purpose math model of a servovalve controlled hydraulic actuator system is derived. The system consists of a linear actuator with unequal piston areas, a single stage servovalve, a gas charged hydraulic accumulator, and the interconnecting piping. The state equations are integrated using the Advanced Continuous Simulation Language (ACSL) for determining the system's dynamic response characteristics. Using this generalized hydraulic actuator system model, response characteristics were determined for various servovalve commands.

Nomenclature

Variables

A	area
B	fluid bulk modulus
C	viscous damping
DP	differential pressure
f	servovalve bandwidth
Fst	static force
$Fstk$	sticking force
$Fcol$	coulomb friction force
K	pressure loss coefficient
m	fluid mass

M	total mass of piston and load
P	pressure
Q	volumetric flow
S	laplace operator
t	time
V	volume
W_{N}	frequency
x	actuator displacement
y	servovalve spool displacement
γ	ratio specific heats
τ	servovalve time constant
ρ	fluid density
ξ	damping

Subscripts

acc	accumulator
act	actuator
c	actuator cap side
g	gaseous
r	actuator rod side
cmd	command
in	initial condition
l	leakage
lq	liquid
rt	return side
sp	supply side
tot	total

vrt	servovalve return side
vsp	servovalve supply side
v	servovalve
va	servovalve port a
vb	servovalve port b
vc	servovalve cap side
vr	servovalve rod side

1.0 Introduction

Many dynamic systems use linear or rotary hydraulic actuators to generate large forces for obtaining high speed response. To satisfy design requirements for such systems, it is necessary to accurately predict load position, velocity, acceleration and system stability.

A generalized hydraulic actuator system was modeled utilizing the Advanced Continuous Simulation Language (ACSL) computer code to solve the system of nonlinear time-dependent ordinary differential equations. Model parameters are easily changed for purposes of component and system evaluation. Control strategies for particular hardware components may be evaluated by assessing the effect on system performance. For a given operating condition and controller gain setting, ACSL evaluates the poles of the system transfer function, thus providing a method to assess the stability of the system.

Three simulations were performed using typical system parameters; a step command to the servovalve, a steep ramp command, and a shallow ramp command. Results of the simulations are discussed with selected variables plotted.

2.0 Governing Equations

Shown in Figure 1 is a schematic of a typical servovalve controlled hydraulic actuator system. The system model consists of an actuator, single stage servovalve, gas charged accumulator, and interconnecting piping. Presented here is the development of the governing equations for each component. The model of the hydraulic system presented here is an extension of that presented in reference (1) and in addition includes an unequal area piston actuator, Coulomb friction, sticking and pressure losses in the piping system.

2.1 Actuator

The actuator is modeled as a second order spring-mass system with viscous damping, Coulomb friction and sticking. The cap and rod side piston areas are unequal. Piston seal leakage is assumed laminar and given by:

$$Q_l = K_{act}(P_c - P_r) \tag{1}$$

Referring to Figure 1, continuity of mass flow into the cap end of the actuator yeilds:

$$\frac{dm_c}{dt} + Q_l\rho_l = Qvc\rho_c \tag{2}$$

The mass contained in the cap end is given by:

$$m_c = \rho_c V_c \tag{3}$$

Differentiating,

$$\frac{dmc}{dt} = \rho c\frac{dVc}{dt} + Vc\frac{d\rho c}{dt} \tag{4}$$

Fluid bulk modulas is defined as:

$$B \equiv \frac{dP}{d\rho/\rho} \tag{5}$$

Volume rate of change is given as:

$$\frac{dVc}{dt} = Ac\frac{dx}{dt} \tag{6}$$

Substitute (4), (5), and (6) into (2) yeilding:

$$Qvc = \frac{Vc}{B}\frac{dPc}{dt} + Ac\frac{dx}{dt} + Q_l \tag{7}$$

Similarly for the rod end:

$$Qvr = \frac{Vr}{B}\frac{dPr}{dt} - Ar\frac{dx}{dt} - Q_l \tag{8}$$

A force balance on the actuator yields:

$$M\frac{d^2x}{dt^2} = PcAc - PrAr - C\frac{dx}{dt} - (Fst + Fstk + Fcol) \tag{9}$$

At the point of zero actuator velocity, the fluid pressure must overcome the breakout or sticking force, Fstk, before motion is established. At that time a constant force due to Coulomb friction, Fcol, opposes the direction of motion. These non-linear terms have the effect of increasing system damping.

2.2 Servovalve

The servovalve spool actuator is modeled as a first order system with response given as:

$$\tau\frac{dy}{dt} + y = Y_{cmd} \tag{10}$$

with

$$\tau = 1/(2\pi f)$$

Servovalve bandwidth, f, is typically specified by the manufacturer. The oil flow of the servovalve is a function of the square root of differential pressure across the valve and is given as;

$$Qvc = Kv(P_{vsp} - P_{vb})^{1/2} \tag{11}$$

$$Qvr = Kv(P_{va} - P_{vrt})^{1/2} \tag{12}$$

The variable, Kv, is computed from manufacturer ratings of flow for a given differential pressure. For instance, if the servovalve is rated for 40 gpm at a pressure loss of 1000 psi, Kv would be calculated as:

$$Kv = (3.85)(40)/(1000)^{1/2} = 4.87in^4/(sec(lbf)^{1/2}) \tag{13}$$

A servovalve spool may be manufactured with overlap to insure positive shutoff. An overlapped spool results in a deadband about the neutral position and acts to increase system damping. Deadband is modeled with an ACSL command by setting the dependent variable to zero when the independent variable lies between a specified lower and upper bound. (ref. 2)

2.3 Accumulator

The accumulator is charged with gas which is assumed to behave isentropically. Accumulator pressure is shown to be:

$$P_{accin}(Vgin)^{\gamma} = P_{acc}(Vg)^{\gamma} \tag{14}$$

$$Vg = V_{tot} - V_{lq} \tag{15}$$

$$\frac{dV_{lq}}{dt} = Q_{sp} \tag{16}$$

$$V_{lq} = \int Q_{sp}dt \tag{17}$$

$$P_{acc} = P_{accin}[Vg_{in}/(V_{tot} - \int Q_{sp}dt)]^{\gamma} \tag{18}$$

2.4 Piping

Pressure losses in the innerconnection piping is modeled by use of pressure loss coefficients. There are four significant sections of piping; the accumulator to servovalve, servovalve to return, and servovalve to rod and cap ends of the actuator cylinder. These equations are defined as:

$$DP_{sp} = K_{sp}(Q_{sp})^2 \tag{19}$$

$$DP_c = K_c(Q_c)^2 \tag{20}$$

$$DP_r = K_r(Q_r)^2 \tag{21}$$

$$DP_{rt} = K_{rt}(Q_{rt})^2 \tag{22}$$

2.5 Transfer Function

The system of equations for the generalized hydraulic actuator system may be linearized by making the following assumptions.

1. Coulomb friction and sticking are negligible
2. Pressure losses in the piping system are negligible
3. Supply pressure remains constant
4. Zero servovalve overlap

Incorporating these assumptions, a block diagram of the linearized system is shown in Figure 2. The block diagram may be reduced to yeild the system transfer function:

$$\frac{X}{Y_{cmd}} = \frac{BK_v\sqrt{\frac{P_{sp}}{2}}\left(\frac{Ar}{Vr}+\frac{Ac}{Vc}\right)/M}{S(\tau S+1)\left[S^2+\frac{C}{M}S+\frac{B}{M}\left(\frac{A_r^2}{V_r}+\frac{A_c^2}{V_c}\right)\right]} \tag{23}$$

Which may be put in the form;

$$\frac{X}{Y_{cmd}} = \frac{W_N^2 Kv\left(\frac{A_rV_c+A_cV_r}{A_r^2V_c+A_c^2V_r}\right)\sqrt{\frac{P_{sp}}{2}}}{S(\tau S+1)\left[S^2+2\xi W_N S+W_N^2\right]} \tag{24}$$

yeilding system natural frequency and damping.

$$W_N^2 = \frac{B}{M}\left(\frac{A_r^2}{V_r}+\frac{A_c^2}{V_c}\right) \tag{25}$$

$$\xi = C/(2W_N M) \tag{26}$$

For the case $A_r = A_c$ and $V_r = V_c$, such as a double acting linear actuator operating about midstroke the transfer function would reduce to:

$$\frac{X}{Y_{cmd}} = \frac{W_N^2 Kv\sqrt{\frac{P_{sp}}{2}}/A}{S(\tau S+1)\left[S^2+2\xi W_N S+W_N^2\right]} \tag{27}$$

$$W_N^2 = \frac{2BA^2}{VM} \tag{28}$$

$$\xi = C/(2W_N M) \tag{29}$$

A rotary motor can also be modeled if the mass of the load is replaced with inertia of the rotary motor.

3.0 Method of Solution

A computer model of the above equations (Appendix 7.0) was implemented using the ACSL code. ACSL was written to facilitate the solution of a system of nth order, nonlinear ordinary differential equations which describe a physical system. Special functions are available in addition to standard Fortran commands to provide flexibility in the problem description. The code sorts the commands into an executable sequence such that a variable is calculated before it is used, allowing the system of equations to be inputed in a logical sequence. Nine integration algorithms are available (default is fourth order Runge-Kutta) in addition to a user supplied routine. The code assigns lower order derivatives a variable name then simultaneously solves the system of first order equations and determines the system states.

A post run command invokes a linear analysis which determines the system poles and damping. The linearized forcing matrix is determined for perturbations about a particular operating point. This capability is particularly useful for designing or developing control strategies.

4.0 Results

A typical use for a servovalve controlled hydraulic actuator system would be that of model injection into the test chamber of a supersonic wind tunnel. Table I shows system parameters for such an application. Dynamic behavior for this model was simulated by solving the nonlinear equations described in section 2 using the ACSL code. Results are presented for three different servovalve command functions.

4.1 Case 1, Step Command to Servovalve

Figure 3a-e presents results for a step command to the servovalve. The servovalve was given a step command of 30%, held for 0.5 second than returned to the neutral position. Table I shows the ACSL equation for the step command. Simulated response to this command shows large initial acceleration as the servovalve opens rapidly subjecting the actuator piston to large differential pressure. While the servovalve spool position is held fixed, the actuator piston velocity remains fairly constant, decreasing slightly as accumulator pressure decreases. Upon return of the servovalve spool to the neutral position, the actuator again undergoes large acceleration/deceleration. Oscillation (second order response) of the piston and load occurs due to the compressibility of fluid (fluid bulk modulus). These oscillations are damped by viscous and coulomb friction. The maximum flow for this simulation occurs to the cap end of the actuator, about +15GPM, while the rod end experiences about a −7GPM flow rate (flow in is taken as positive, flow out is negative). The difference in these flow rates is due to the unequal piston area of the rod and cap ends.

4.2 Case 2, Steep Ramp Command to Servovalve

Figure 4a-e presents results for a steep ramp command to the servovalve. The servovalve was given a command of 150%/sec for 0.2 second, then held at 30% for 0.2 second and returned to the neutral position at a rate of −150%/sec. Table I shows the ACSL equation for this ramp command. From these results it is seen that the actuator piston and load is subjected to much lower peak acceleration than in Case 1. Maximum velocity and flows are similar due to the same command hold state. Piston oscillation is much less pronounced and of shorter duration than that of Case 1.

4.3 Case 3, Shallow Ramp Command to Servovalve

Figure 5a-e presents results for a shallow ramp command to the servovalve. The servovalve was given a command of 60%/sec for 0.5 second, then held at 30% for 0.1 second, and then returned to the neutral position at a rate of −60%/sec. Table I shows the ACSL equation for this ramp command. Similar to the results of Case 2, peak acceleration of the piston and load is again reduced with the less abrupt servovalve spool movement. Maximum velocity and flows are again about the same due to the same command hold state as the previous simulations. Piston oscillation is almost non-existent for this case.

These simulations indicate that the acceleration of the piston and load are determined by the rate of change of the servovalve command. The magnitude of the servovalve command determine piston and load slew rate.

5.0 Conclusion

A general purpose math model of a servovalve controlled hydraulic actuator system is derived for use with the Advanced Continuous Simulation Language (ACSL). Parameters typical of a hydraulic system were input and the model exercised. The math model is of a general hydraulic actuator system which can be easily changed. This allows for quick assessment of alternate system hardware or control strategy.

6.0 References/Acknowledgement

1. Burrows, C. R., "Fluid Power Servomechanisms," Van Nostrand Reinhold Co., London, 1972.
2. Franklin, G. F., Powell, J. D., Emami-Naeini, A., "Feedback Control of Dynamic Systems," Addison-Wesley Company, Menlo Park, California, 1986.
3. Mitchell and Gauthier Associates, "Advanced Continuous Simulation Language," 4th ed., Concord, Mass., 1987.
4. Crane Co., "Flow of Fluids", 1981.

Acknowledgement is extended to Mr. J. F. Watson Jr. of NASA Langley Research Center. Mr. Watson provided the author with an ACSL model and schematic of a hydraulic injection system which became the basis of this memorandum. Appendix 7.0 contains the code listing and revisions incorporated into the model.

Table I Example Simulation Parameters

Actuator		
Cylinder Length	17	Inch
Stroke	15	Inch
Mass of Piston & Load	2050	Lbm
Cylinder I.D.	2.0	Inch
Rod Diameter	1.375	Inch
Kq_{act}	0.0	$in^5/(sec\ Lbf)$
Fstk	75	Lbf
Fcol	50	Lbf
Servo Valve		
Bandwidth	25	Hz
Max Flow @ 1000 psid	40	GPM
Deadband	+/−2	%
Accumulator and Piping		
Volume	5	Gal
Initial Gas Charge	1800	psia
Supply Pressure	3000	psig
Return Pressure	25	psig
Kqsup	0.0125	$Lbf\ sec^2/in^8$
Kqret	0.0125	$Lbf\ sec^2/in^8$
Kqcap	0.0125	$Lbf\ sec^2/in^8$
Kqrod	0.0125	$Lbf\ sec^2/in^8$

Command to Servo Valve

Case 1. (Figure 3), Step Command

$$Y_{cmd} = 30(\text{step}(.2)\text{-step}(.7))$$

Case 2. (Figure 4), Steep Ramp Command

$$Y_{cmd} = 150(\text{ramp}(.2)\text{-ramp}(.4)\text{-ramp}(.6)+\text{ramp}(.8))$$

Case 3. (Figure 5), Shallow Ramp Command

$$Y_{cmd} = 60(\text{ramp}(.1)\text{-ramp}(.6)\text{-ramp}(.7)+\text{ramp}(1.2))$$

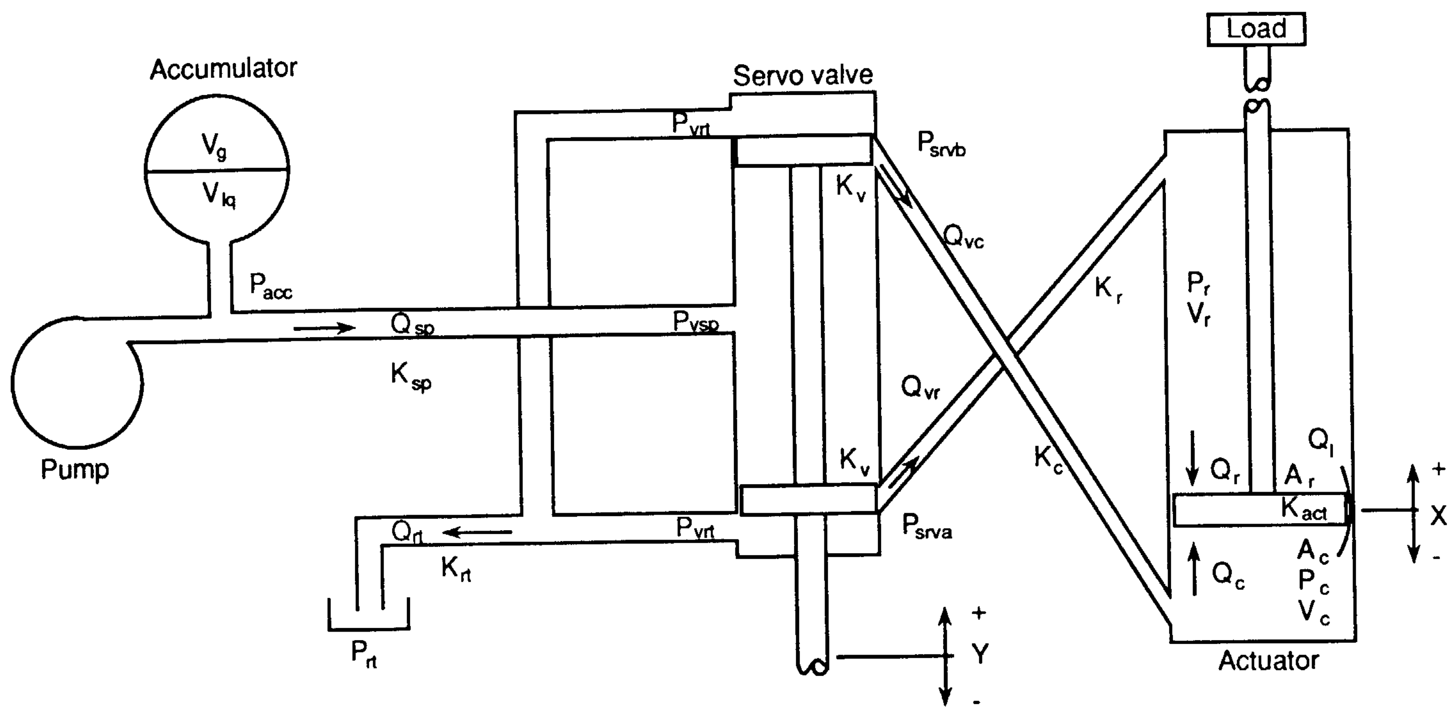
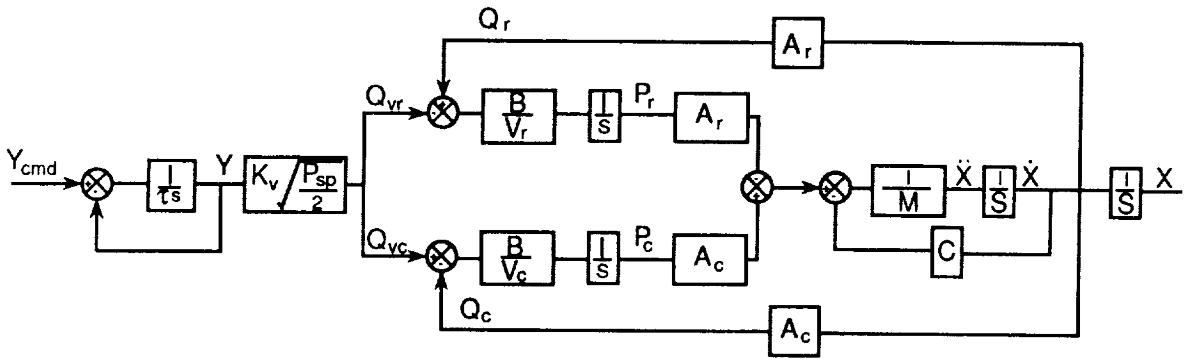


Figure 1. Schematic of Servovalve Controlled Hydraulic Actuator System.



Note: P_{sup} = Supply pressure: Return pressure assumed zero

Transfer function:

$$\frac{X}{Y_{CMD}} = \frac{BK_v\sqrt{\frac{P_{sp}}{2}}\left(\frac{A_r}{V_r}+\frac{A_c}{V_c}\right)/M}{S(\tau S+1)\left[S^2+\frac{C}{M}S+\frac{B}{M}\left(\frac{A_r^2}{V_r}+\frac{A_c^2}{V_c}\right)\right]}$$

For $A_r \cong A_c = A$
and $V_r \cong V_c = V$
Transfer function reduces to:

$$\frac{X}{Y_{CMD}} = \frac{2A\,BK_v\sqrt{2P_{sp}}\,/(VM)}{S(\tau S+1)\left(S^2+\frac{C}{M}S+2BA^2/(VM)\right)}$$

Figure 2. Block Diagram of Linearized Servovalve Controlled Hydraulic Actuator System for Small Perturbations.

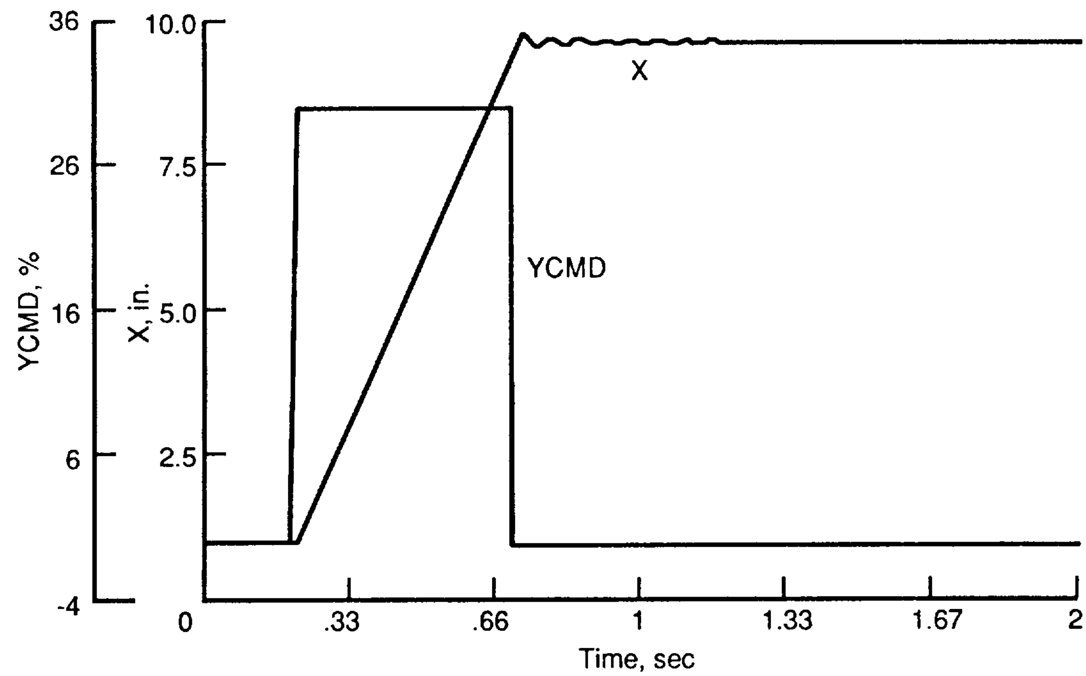


Figure 3a. Servovalve Command (YCMD) and Actuator Position (X) Versus Time for a Step Command to the Servovalve.

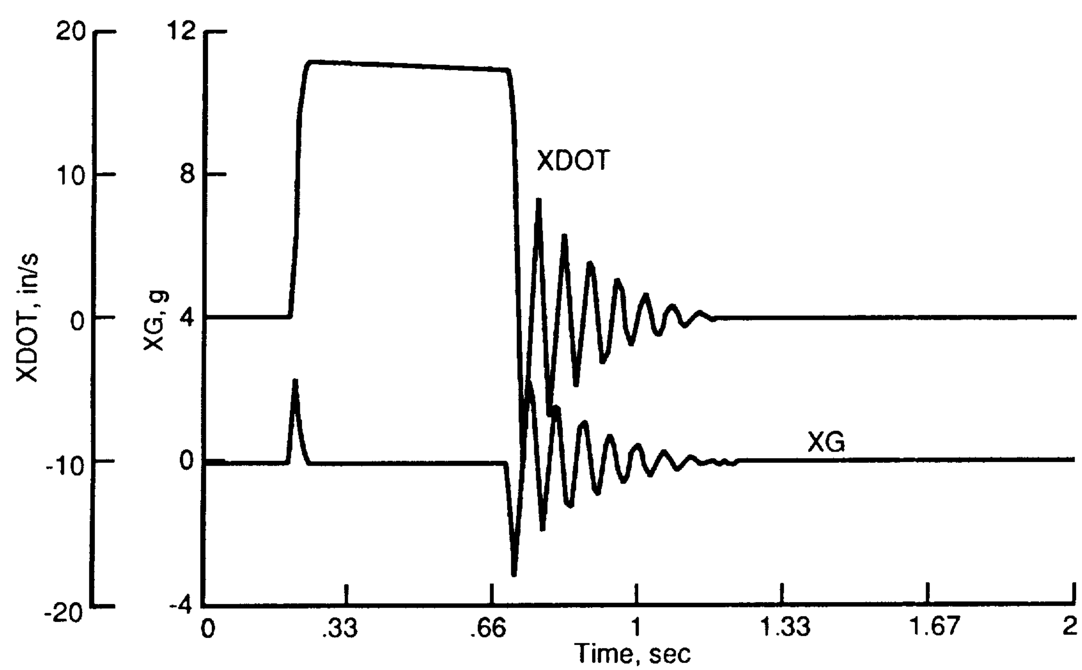


Figure 3b. Actuator Velocity (XDOT) and Acceleration (XG) Versus Time for a Step Command to the Servovalve.

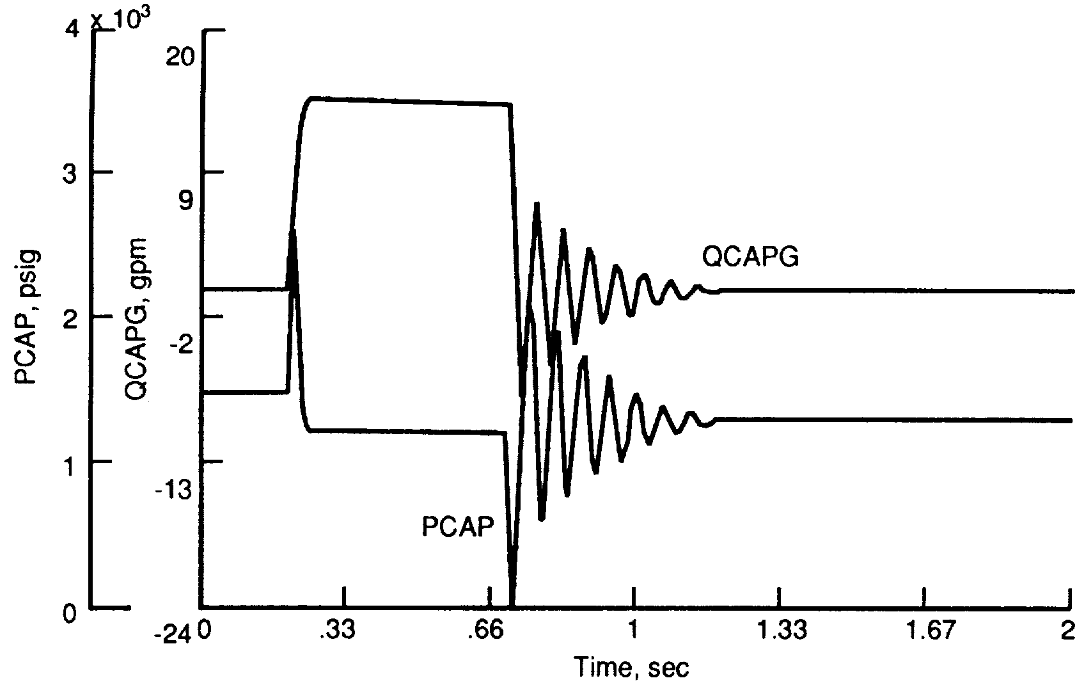


Figure 3c. Cap Side Flow (QCAPG) and Pressure (PCAP) Versus Time for a Step Command to the Servovalve.

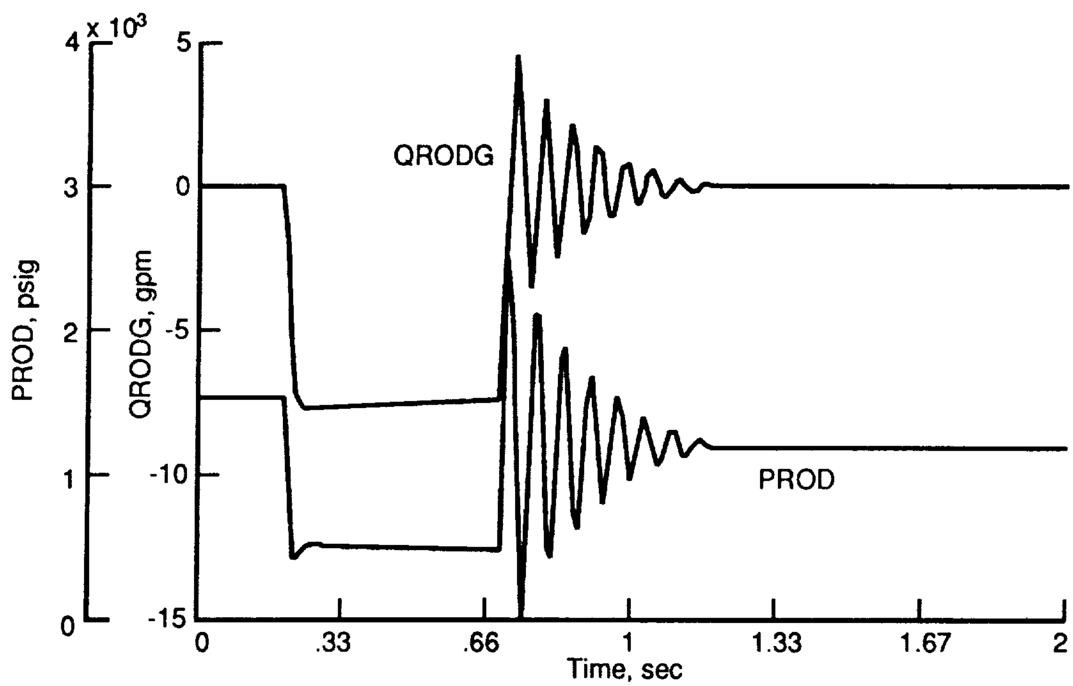


Figure 3d. Rod Side Flow (QRODG) and Pressure (PROD) Versus Time for a Step Command to the Servovalve.

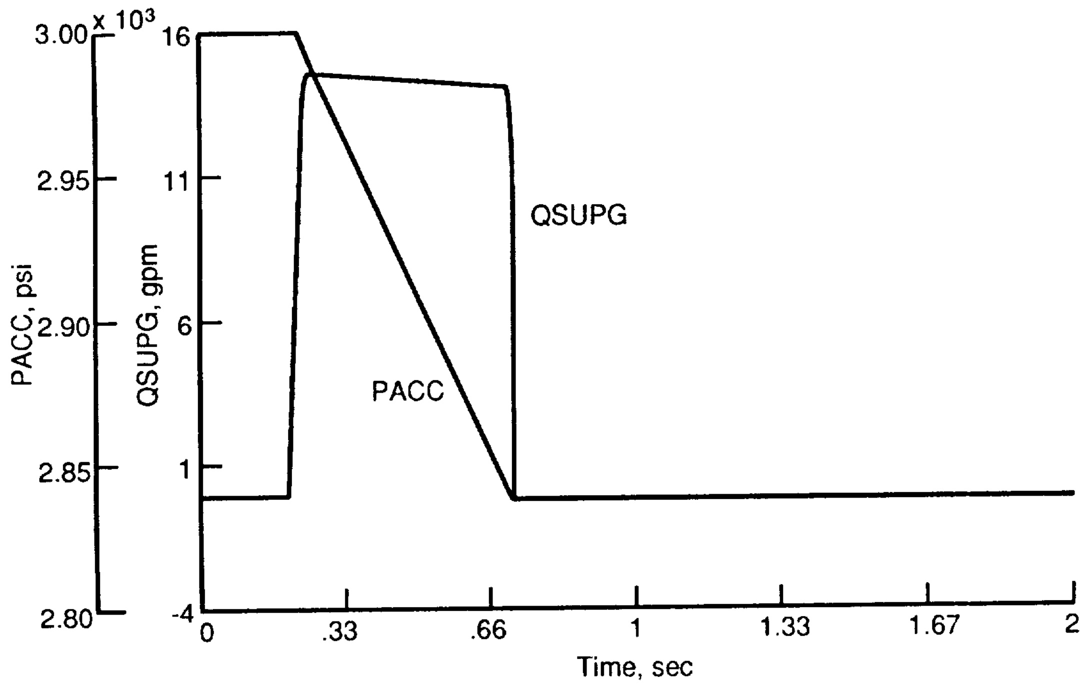


Figure 3e. Supply Pressure (PACC) and Supply Flow (QSUPG) Versus Time for a Step Command to the Servovalve.

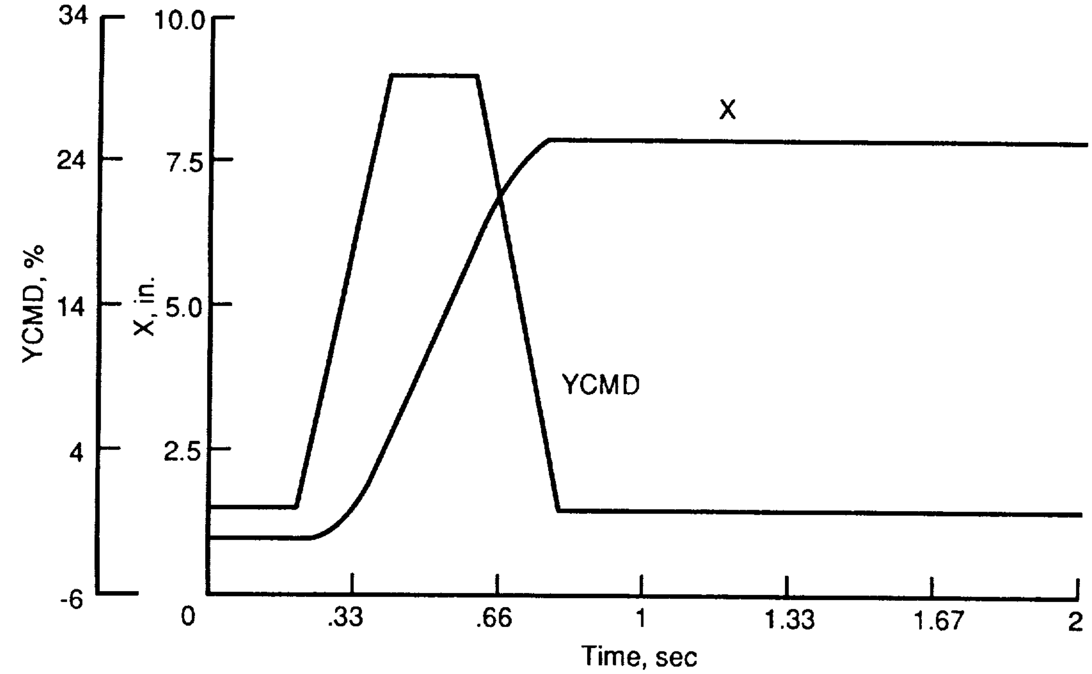


Figure 4a. Servovalve Command (YCMD) and Actuator Position (X) Versus Time for a Steep Ramp Command to the Servovalve.

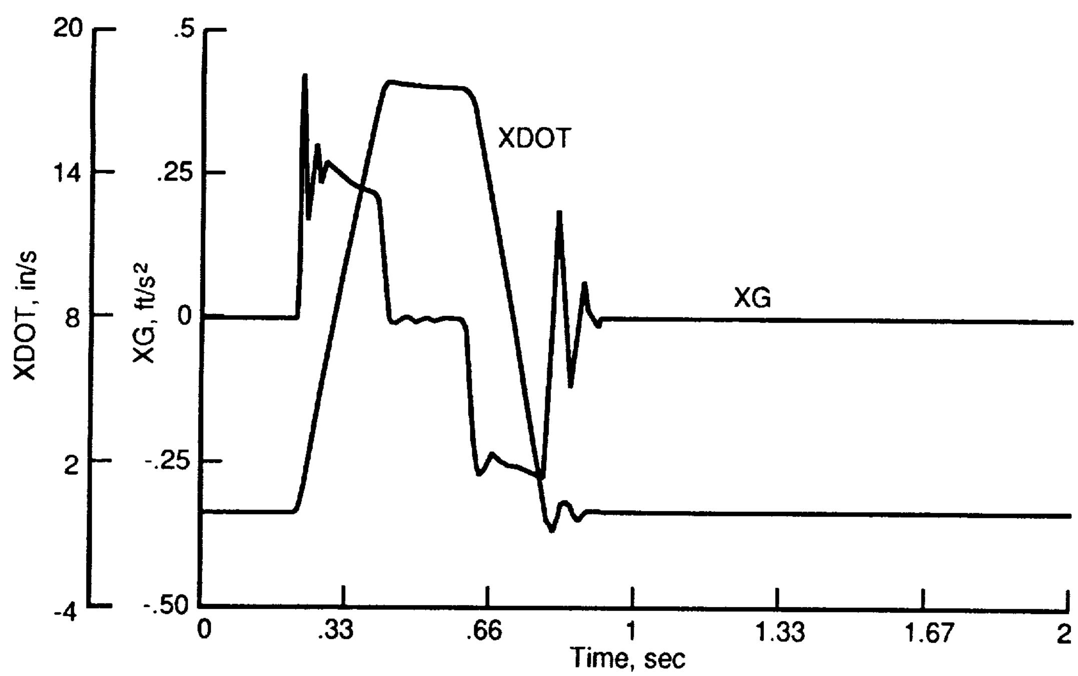


Figure 4b. Actuator Velocity (XDOT) and Acceleration (XG) Versus Time for a Steep Ramp Command to the Servovalve.

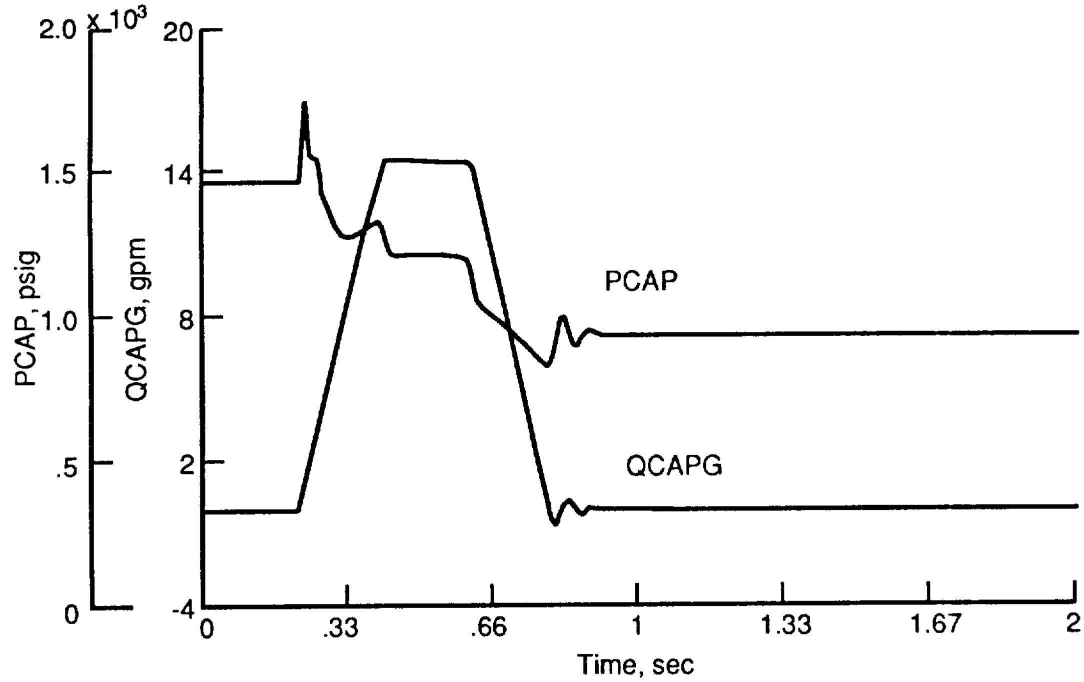


Figure 4c. Cap Side Flow (QCAPG) and Pressure (PCAP) Versus Time for a Steep Ramp Command to the Servovalve.

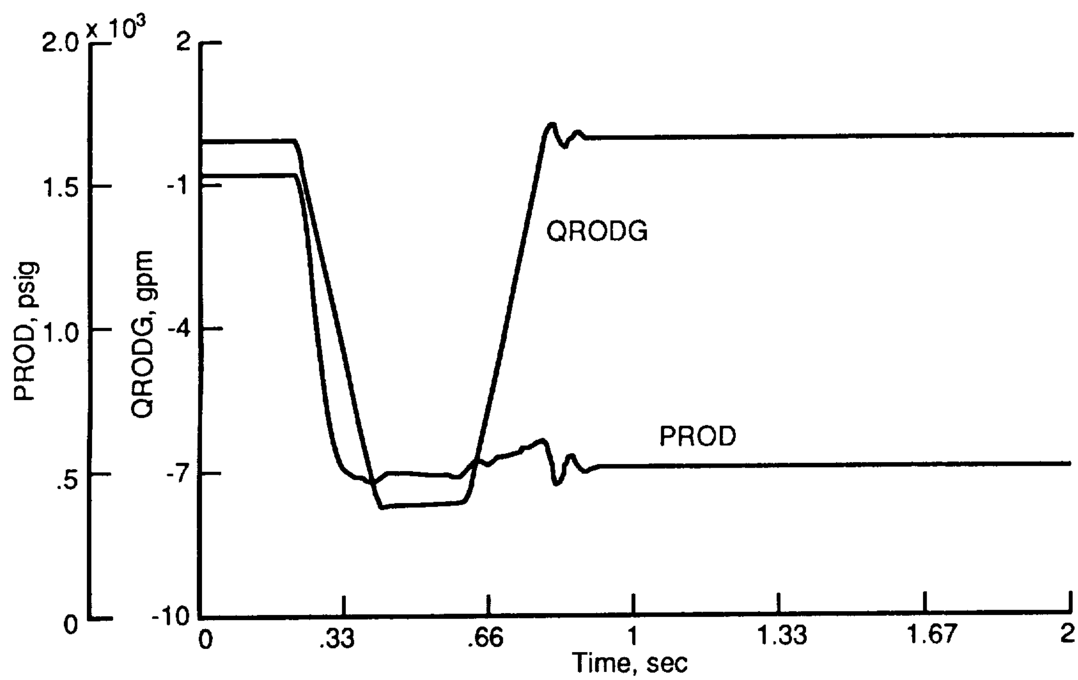


Figure 4d. Rod Side Flow (QROD) and Pressure (PROD) Versus Time for a Steep Ramp Command to the Servovalve.

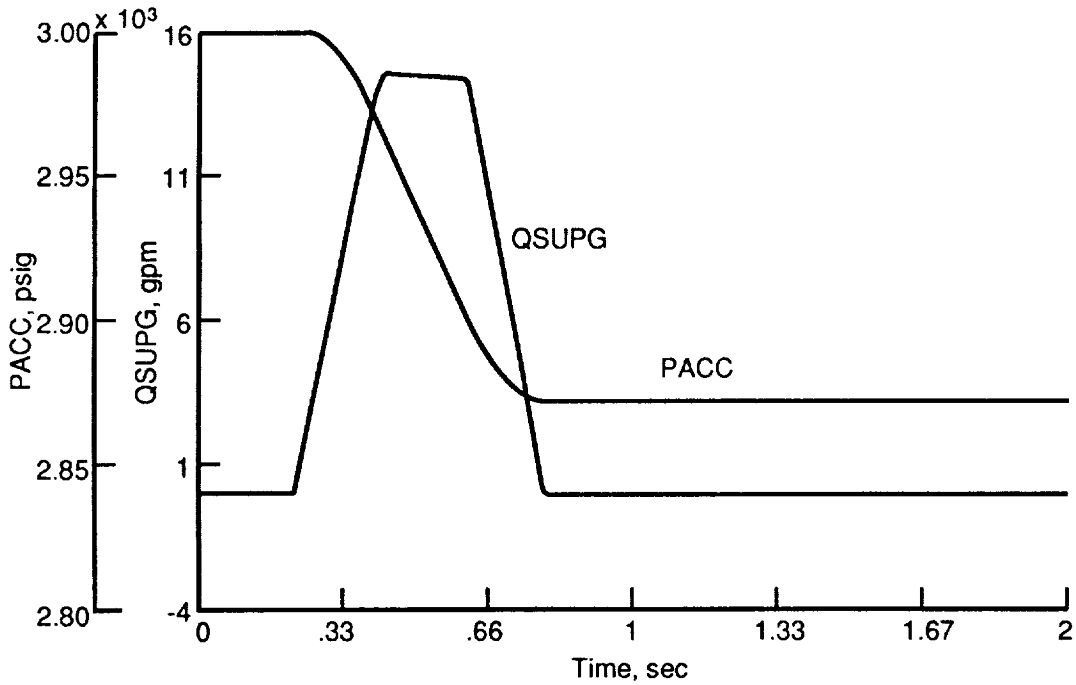


Figure 4e. Supply Pressure (PACC) and Supply Flow (QSUPG) Versus Time for a Steep Ramp Command to the Servovalve.

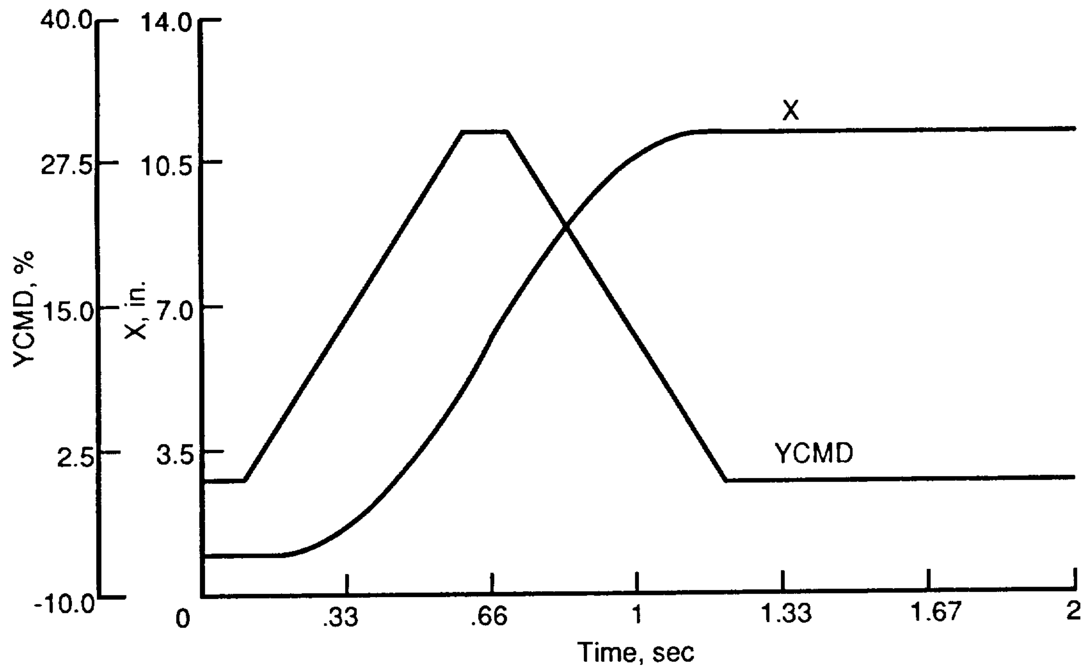


Figure 5a. Servovalve Command (YCMD) and Actuator Position (X) Versus Time for a Shallow Ramp Command to the Servovalve.

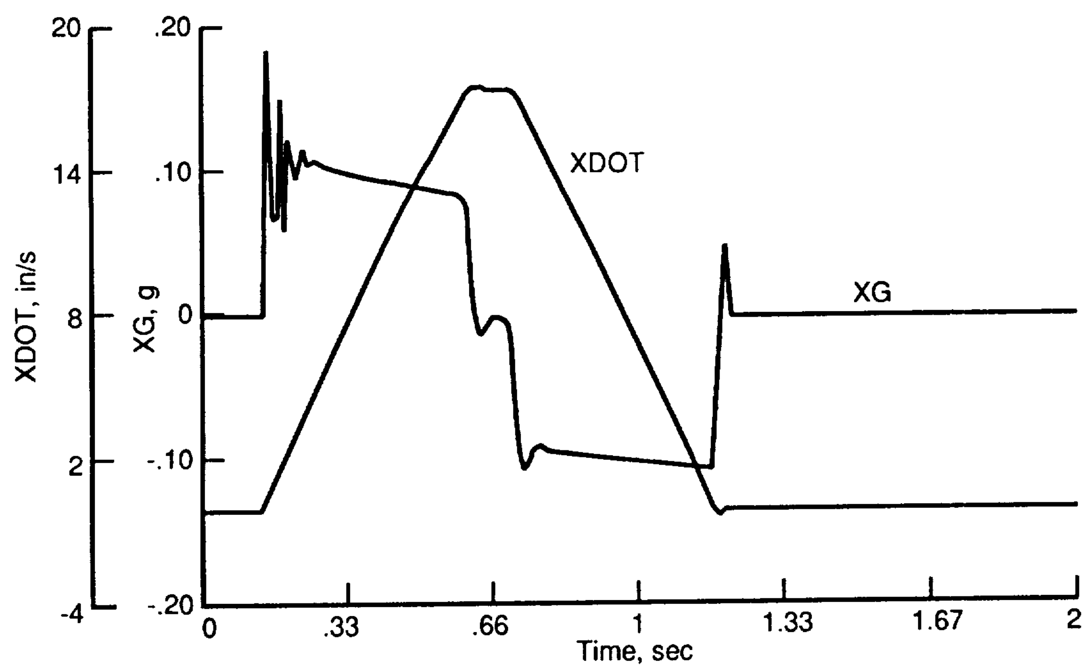


Figure 5b. Actuator Velocity (XDOT) and Acceleration (XG) Versus Time for a Shallow Ramp Command to the Servovalve.

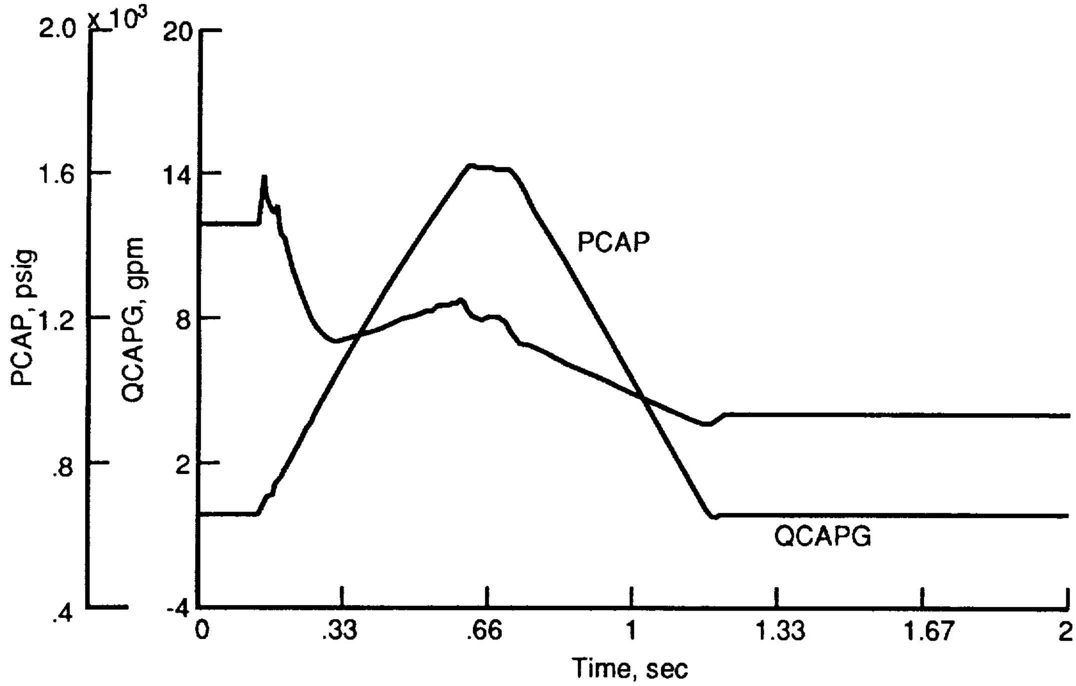


Figure 5c. Cap Side Flow (QCAPG) and Pressure (PCAP) Versus Time for a Shallow Ramp Command to the Servovalve.

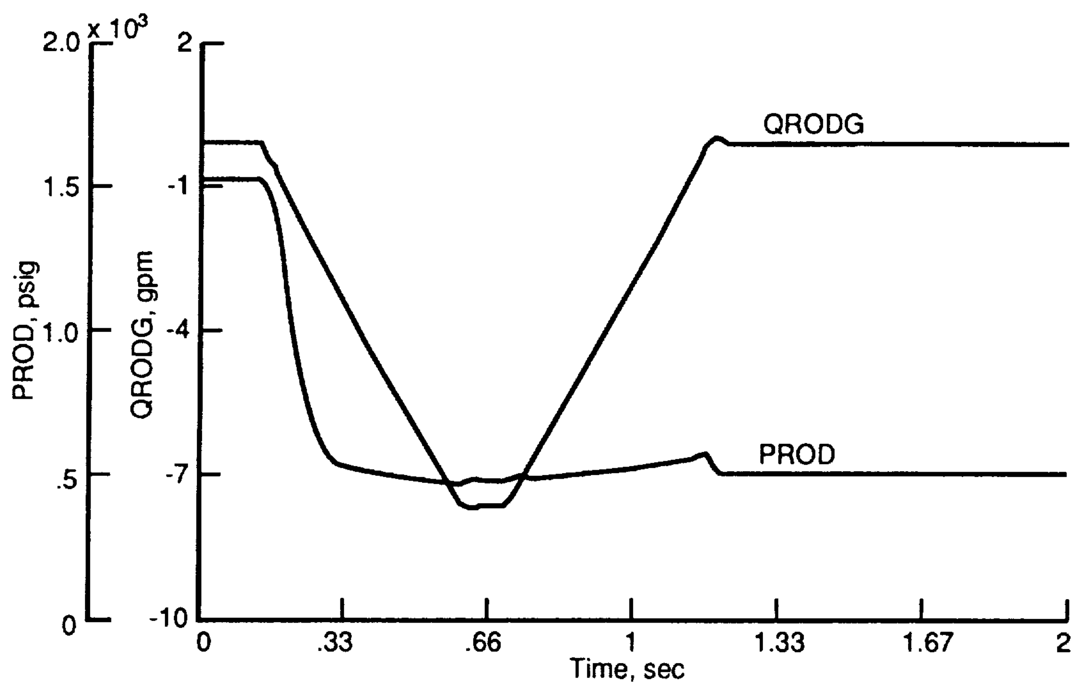


Figure 5d. Rod Side Flow (QRODG) and Pressure (PROD) Versus Time for a Shallow Ramp Command to the Servovalve.

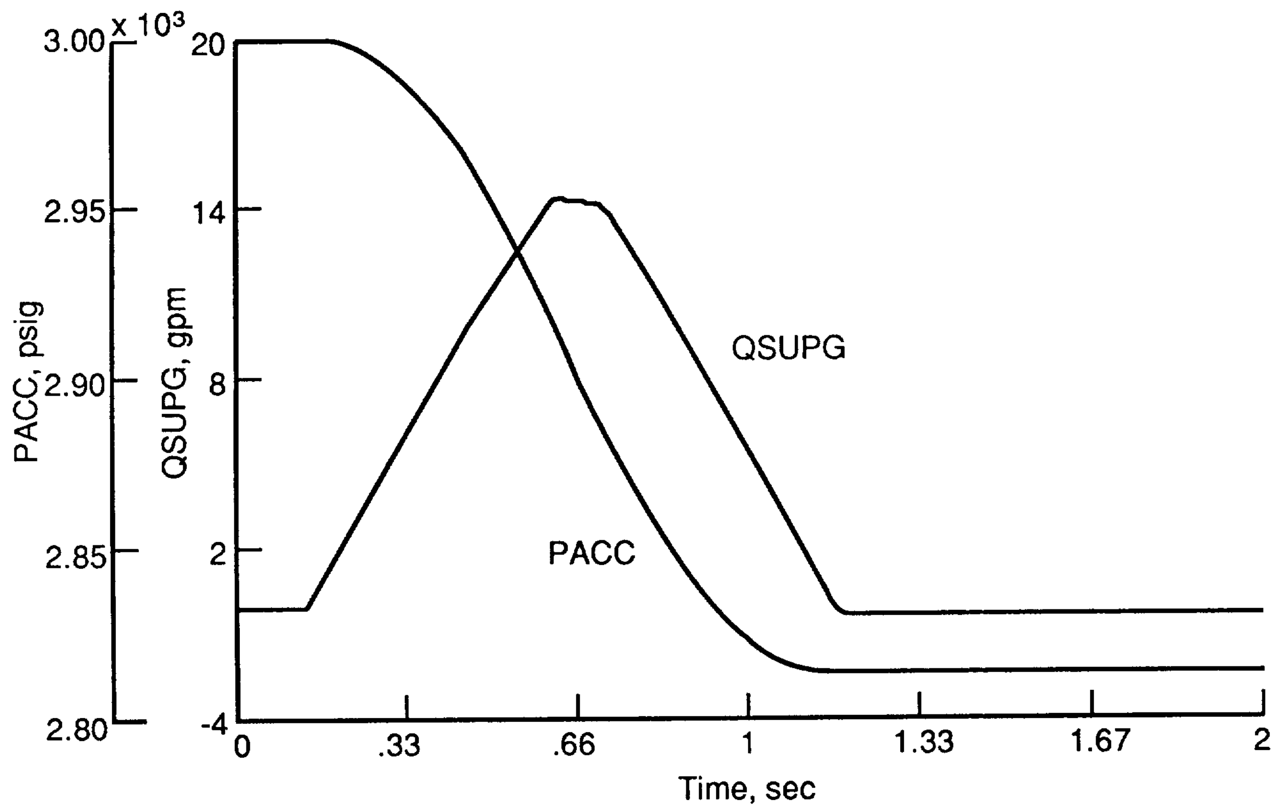


Figure 5e. Supply Pressure (PACC) and Supply Flow (QSUPG) Versus Time for a Shallow Ramp Command to the Servovalve.

7.0 Code Listing

```
PROGRAM CYLINDER
"----------------------------------------------------------------------"
"     SERVOVALVE CONTROLLED HYDRAULIC ACTUATOR SYSTEM ANALYSIS        "
"           ORIGINAL VERSION BY J. F. WATSON, FENGD, FACS             "
"                         REVISED BY                                   "
"               D. A. BABCOCK   MAY 1989, FENGD, PSS                  "
"----------------------------------------------------------------------"
"                                                                      "
"  THIS UPDATED VERSION OF THE ORIGINAL CODE INCLUDES THE EFFECTS OF  "
"   ACTUATOR STICKING AT ZERO VELOCITIY; ACTUATOR COULOMB FRICTION;   "
"      ACTUATOR LEAKAGE FROM THE HIGH TO LOW PRESSURE PORTS; AND      "
"   SERVOVALVE DEADBAND AS A PERCENTAGE OF OVERLAP: THESE NONLINEAR   "
"          TERMS INCREASE THE SIMULATED SYSTEM DAMPING                "
"                                                                      "
"************************* VARIABLE TABLE ****************************"
"                                                                      "
"       X        LOAD DISPLACEMENT                        INCH        "
"       XDOT     LOAD VELOCITY                            IN/SEC      "
"       XDDOT    LOAD ACCELERATION                        IN/SEC**2   "
"       IX       INITIAL POSITION                         IN          "
"                                                                      "
"       CYSTK    ACTUATOR STROKE LENGTH                   IN          "
"       CYDIA    ACTUATOR PISTON DIAMETER                 IN          "
"       RODDIA   ACTUATOR ROD DIAMETER                    IN          "
"       WACTR    ACTUATOR PISTON AND ROD WEIGHT           LB          "
"       WLOAD    LOAD WEIGHT                              LB          "
"                                                                      "
"       FCYL     CYLINDER GENERATED FORCE                 LB          "
"       KFV      VISCOUS DAMPING                          LB-SEC/IN   "
"       FVISC    VISCOUS FRICTION FORCE                   LB          "
"       FSTAT    GRAVITY FORCE OF LOAD AND ACTUATOR       LB          "
"       FSTK     ACTUATOR STICKING FORCE                  LB          "
"       FCOL     ACTUATOR COULOMB FRICTION FORCE          LB          "
"       MTOTL    TOTAL MASS OF MOVING ITEMS               LB-SEC**2/IN"
"                                                                      "
"       PROD     ROD SIDE CYLDR PRESSURE                  PSIG        "
"       PCAP     CAP SIDE CYLDR PRESSURE                  PSIG        "
"       DPPROD   LINE PRESSURE LOSS ROD SIDE              PSID        "
"       DPPCAP   LINE PRESSURE LOSS CAP SIDE              PSID        "
"                                                                      "
"       QROD     ROD SIDE CYLDR OIL FLOW                  IN**3/SEC   "
"       QCAP     CAP SIDE CYLDR OIL FLOW                  IN**3/SEC   "
"       VROD     ROD SIDE CYLDR OIL VOLUME                IN**3       "
"       VCAP     CAP SIDE CYLDR OIL VOLUME                IN**3       "
"       AROD     ROD SIDE AREA                            IN**2       "
"       ACAP     CAP SIDE AREA                            IN**2       "
"                                                                      "
```

```
"       QL      ACTUATOR LEAKAGE                              IN**3/SEC        "
"       QSUP    OIL SUPPLY FLOW                               IN**3/SEC        "
"       QRET    OIL RETURN FLOW                               IN**3/SEC        "
"       PACC    ACCUM SUPPLY PRESSURE                           PSIG           "
"       POILCG  INITIAL SYSTEM PRESSURE                         PSIG           "
"       PRET    RETURN PRESSURE                                 PSIG           "
"       VOLACC  ACCUMULATOR VOLUME                              GAL            "
"       PN2CHG  INITIAL NITROGEN CHARGE                         PSIA           "
"                                                                              "
"       Y       SERVOVALVE SPOOL DISP                            %             "
"       YCMD    COMMAND TO SERVO VALVE                           %             "
"       YDOT    SERVOVALVE SPOOL VELOCITY                       %/SEC          "
"       TAU     SERVOVALVE TIME CONSTANT                         SEC           "
"       DBAND   SERVOVALVE DEAD BAND                             %             "
"       SERVFL  SERVOVALVE FLOW AT RATED PRESSURE               GPM            "
"       SERVPR  SERVOVALVE PRESSURE AT RATED FLOW               PSID           "
"                                                                              "
"       PSRVP   SERVOVALVE SUPPLY PRESSURE                      PSIG           "
"       PSRVT   SERVOVALVE RETURN PRESSURE                      PSIG           "
"       PSRVA   SERVOVALVE CONTROL PORT PRESSURE                PSIG           "
"       PSRVB   SERVOVALVE CONTROL PORT PRESSURE                PSIG           "
"       DPPSUP  LINE PRESSURE LOSS SUPPLY SIDE                  PSID           "
"       DPPRET  LINE PRESSURE LOSS RETURN SIDE                  PSID           "
"                                                                              "
"       QVR     SERVOVALVE ROD SIDE FLOW                      IN**3/SEC        "
"       QVC     SERVOVALVE CAP SIDE FLOW                      IN**3/SEC        "
"       KQSER   SERVOVALVE FLOW COEF                      IN**4/(SEC-LB**.5)   "
"       KQSUP   SUPPLY PIPING LOSS COEF                    LB-SEC**2/IN**8     "
"       KQRET   RETURN PIPING LOSS COEF                    LB-SEC**2/IN**8     "
"       KQCAP   CAP PIPING LOSS COEF                       LB-SEC**2/IN**8     "
"       KQROD   ROD PIPING LOSS COEF                       LB-SEC**2/IN**8     "
"       KQACT   ACTUATOR LEAKAGE COEF                       IN**5/(SEC-LB)     "
"                                                                              "
"       B       FLUID BULK MODULUS                              PSI            "
"       GAMM    IDEAL GAS SPECIFIC HEAT RATIO                    NA            "
"       ATM     LOCAL ATMOSP PRESSURE                           PSIA           "
"                                                                              "
"***************************  CONSTANTS  *****************************"
"                                                                              "
"--------------------------- SERVOVALVE ------------------------------"
"                                                                              "
CONSTANT SERVFL = 40.0, SERVPR = 1000.0
CONSTANT DBAND  = 0.02, TAU    = 0.006
"                                                                              "
"---------------------- ACCUMULATOR AND PIPING -----------------------"
"                                                                              "
CONSTANT PN2CHG =  1800.0,  VOLACC =    5.0,   GAMM = 1.4
CONSTANT KQSUP  = 0.01250,  KQRET  = 0.01250
CONSTANT KQCAP  = 0.01250,  KQROD  = 0.01250
"                                                                              "
"------------------------  FLUID CONSTANTS  --------------------------"
"                                                                              "
CONSTANT POILCG = 3000.0, PRET = 25.0, B = 100000.0
"                                                                              "
"---------------------------- ACTUATOR -------------------------------"
```

```
"
CONSTANT CYSTK  =  17.0,  CYDIA  =    2.0, RODDIA =   1.375
CONSTANT WACTR  = 500.0,  WLOAD  = 2000.0, KFV    =  50.0
CONSTANT FSTK   =  75.0,  FCOL   =   50.0, KQACT  =   0.01
CONATANT ALL    =   0.5   $" ACTUATOR LOWER INTEG LIMIT "
CONATANT AUL    =  16.5   $" ACTUATOR UPPER INTEG LIMIT "
"
"------------------------------ OTHER --------------------------------"
"
LOGICAL STUCK               $ " TRUE IF VELOCITY IS ZERO OTHERWISE FALSE "
CONSTANT TSTRT   =     0.0
CONSTANT TSTP    =     1.0
CONSTANT ATM     =     14.7
CONSTANT PI      =     3.1415927
CONSTANT INPGAL  =     231.0     $" IN**3 PER GALLON "
CONSTANT CNVRT   =     3.85      $" IN**3/SEC TO GPM "
CONSTANT ACCG    =     386.4     $" GRAVITATIONAL CNST IN/SEC**2 "
CONSTANT LL      =     5.0       $" LOWER PRES INTEGRATION LIMIT "
CONSTANT UL      =     5000.0    $" UPPER PRES INTEGRATION LIMIT "
"
"*********************************************************************"
"
INITIAL    $" SET INITIAL CONDITIONS "
"
"---------------------------- ACCUMULATOR ----------------------------"
"
VLN20    =    VOLACC * INPGAL * PN2CHG / ( POILCG + ATM )
KACCM    =    ( POILCG + ATM ) * ( VLN20 ** GAMM )
"
"------------------------------ ACTUATOR -----------------------------"
"
RESET("NOEVAL")
STUCK = XDOT .EQ. 0.0  $" ACTUATOR STICKS WHEN VELOCITY IS ZERO "
FVISC    =    0.0
"
FSTAT    =    WACTR + WLOAD
MTOTL    =    FSTAT / ACCG
".
ACAP     =    (PI / 4.0) * (CYDIA ** 2)
AROD     =    ACAP - (PI / 4.0) * (RODDIA ** 2)
"
IPROD    =    (POILCG * ACAP - FSTAT) / (AROD + ACAP)
IPCAP    =    POILCG - IPROD
IX       =    0.0
"
"------------------------- SERVOVALVE --------------------------------"
"
KQSER    =    CNVRT * SERVFL / (SQRT( SERVPR ))
"
END $"OF INITIAL"
"*********************************************************************"
"
DYNAMIC
DERIVATIVE
"
```

```
" YCMD = 0.3*(STEP(.2) - STEP(.7))"                        $"STEP COMMAND"
  YCMD = 1.5*(RAMP(.2)-RAMP(.4)-RAMP(.6)+RAMP(.8))         $"RAMP COMMAND"
" YCMD = 0.6*(RAMP(.1)-RAMP(.6)-RAMP(.7)+RAMP(1.2))"       $"RAMP COMMAND"
"                                                                        "
"********************** CONTROLLER SECTION ************************"
"                                                                        "
"               SERVOVALVE CONTROLLER MAY BE ADDED HERE                  "
"                                                                        "
"                                                                        "
"************************* SERVOVALVE *****************************"
"                                                                        "
"                         DYNAMIC SECTION                                "
"                                                                        "
YDOT   =  (YCMD - YV) / TAU
YV     =  LIMINT ( YDOT, 0.0, -1.0, +1.0 )
Y      =  DEAD ( -DBAND, DBAND, YV )
"                                                                        "
"                         FLOW SECTION                                   "
"                                                                        "
FLOW0  =  0.0
FLOW1  =  Y*KQSER*(SQRT(ABS(PSRVP-PSRVB)))*SIGN(1.,PSRVP-PSRVB)
FLOW2  =  Y*KQSER*(SQRT(ABS(PSRVB-PSRVT)))*SIGN(1.,PSRVB-PSRVT)
FLOW3  = -Y*KQSER*(SQRT(ABS(PSRVA-PSRVT)))*SIGN(1.,PSRVA-PSRVT)
FLOW4  = -Y*KQSER*(SQRT(ABS(PSRVP-PSRVA)))*SIGN(1.,PSRVP-PSRVA)
QVR    =  FCNSW(Y,FLOW4,FLOW0,FLOW3)
QVC    =  FCNSW(Y,FLOW2,FLOW0,FLOW1)
"                                                                        "
"***************** ACTUATOR FORCES AND POSITION ******************"
"                                                                        "
X       =  LIMINT ( XDOT, IX, ALL, AUL )
XDOT    =  INTEG ( XDDOT, 0.0)
XDDOT   =  ( FTOTL + FCF ) / MTOTL
"                                                                        "
FTOTL   =  FCYL - FVISC - FSTAT
FVISC   =  KFV * XDOT
FCYL    =  PCAP * ACAP - PROD * AROD
FCF     =  RSW ( STUCK, -FTOTL, FASF)
FIA     =  RSW ( STUCK, ABS ( FTOTL ) - FSTK, XDOT )
SCHEDULE STICK .XZ. FIA
"                                                                        "
"***************** ROD SIDE FLOW AND PRESSURE ********************"
"                                                                        "
PROD    =  LIMINT ( PDROD, IPROD, LL, UL )
QROD    =  AROD * XDOT
PDROD   =  ( QVR + QROD + QL ) * ( B / VROD )
VROD    =  AROD * ( CYSTK - X )
DPPROD  =  ( QROD ** 2 ) * KQROD * SIGN ( 1.0, QROD )
PSRVA   =  PROD - DPPROD
"                                                                        "
QL      =  KQACT * ( PCAP - PROD ) $" ACTUATOR LEAKAGE "
"                                                                        "
"***************** CAP SIDE FLOW AND PRESSURE *********************"
"                                                                        "
PCAP    =  LIMINT ( PDCAP, IPCAP, LL, UL )
QCAP    =  ACAP * XDOT
```

```
PDCAP   =  ( QVC - QCAP - QL ) * ( B / VCAP )
VCAP    =  ACAP * X
DPPCAP  =  ( QCAP ** 2 ) * KQCAP * SIGN ( 1.0, QCAP )
PSRVB   =  PCAP + DPPCAP
"                                                                       "
"****************  ACCUMULATOR / SUPPLY AND RETURN  ****************"
"                                                                       "
DELVOL  =  INTEG ( QSUP, 0.0 )
VOLN2   =  VLN20 + DELVOL
PACC    =  KACCM / ( VOLN2 ** GAMM ) - ATM
"                                                                       "
DPPSUP  =  ( QSUP ** 2 ) * KQSUP * SIGN ( 1.0, QSUP )
PSRVP   =  PACC - DPPSUP
QSUP    =  FCNSW ( Y, -QROD, 0.0, QCAP )
"                                                                       "
DPPRET  =  ( QRET ** 2 ) * KQRET * SIGN ( 1.0, QRET )
PSRVT   =  PRET + DPPRET
QRET    =  FCNSW ( Y, -QCAP, 0.0, QROD )
"                                                                       "
END     $   "OF DERIVATIVE"
"                                                                       "
DISCRETE STICK
PROCEDURAL
STUCK   =  .NOT. STUCK .AND. ABS( FTOTL ) .LT. FSTK
FASF    =  RSW (STUCK, 0.0, SIGN( FCOL, -FTOTL))
XDOT    =  0.0 $" SET VEL 0 IF STICKING "
END     $  "OF PROCEDURAL"
END     $  "OF DISCRETE"
"                                                                       "
"********************** ENGINEERING VARIABLES ***********************"
"                                                                       "
" CONVERT IN**3 TO GALLON PER MINUTE                                    "
QCAPG   =   QCAP  /  CNVRT
QRODG   =   QROD  /  CNVRT
QSUPG   =   QSUP  /  CNVRT
QRETG   =   QRET  /  CNVRT
" CONVERT IN/SEC**2 TO G                                                "
XG      =   XDDOT /  ACCG
"                                                                       "
TERMT (T .GE. TSTP)
"                                                                       "
END     $   "OF DYNAMIC"
END     $   "OF PROGRAM"
```

Report Documentation Page

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15. Supplementary Notes

16. Abstract

A general purpose math model of a servovalve controlled hydraulic actuator system is derived. The system consists of a linear actuator with unequal piston areas, a single stage servovalve, a gas charged hydraulic accumulator, and the interconnecting piping. The state equations are integrated using the Advanced Continuous Simulation Language (ACSL) for determining the system's dynamic response characteristics. Using this generalized hydraulic actuator system model, response characteristics were determined for various servovalve commands.

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