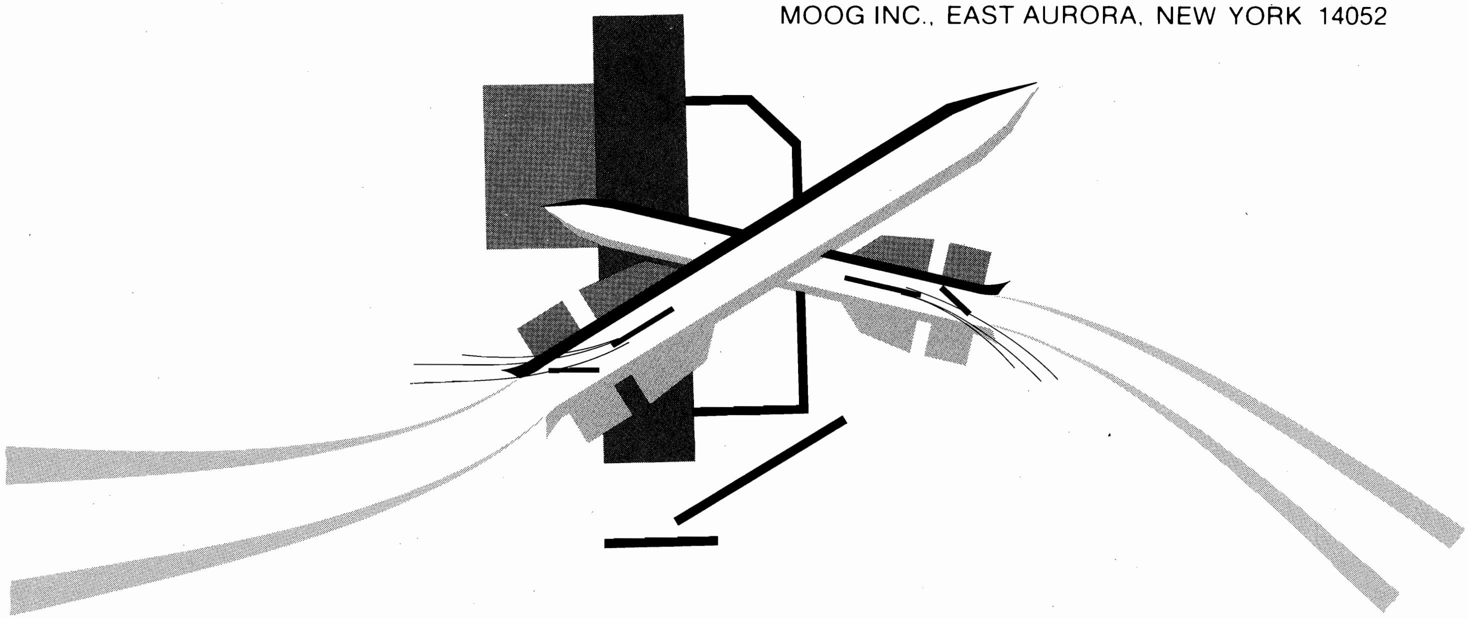
TECHNICAL BULLETIN 151

MOOG

MOOG INC., EAST AURORA, NEW YORK 14052



ELECTROPNEUMATIC SERVOACTUATION
AN ALTERNATIVE TO HYDRAULICS FOR SOME LOW POWER APPLICATIONS

William J. Thayer
August 1984
Revised February 1988

ABSTRACT

Use of compressed gas to power a high-performance servoactuation system may be an attractive, lower-cost alternative to a more conventional electrohydraulic servo system. This paper discusses the present state of the technology of high-performance electropneumatic servoactuation. Particular emphasis is placed on dynamic response, both analytical modeling to predict response and special hardware configurations to improve response.

Considerations of electropneumatic servoactuation are treated from the systems standpoint where the source of the pneumatic power, power consumption, power modulation (i.e., valve controls) and nature of the servoelectronics are important in determining the suitability and potential cost of an electropneumatic servo system.

INTRODUCTION

Electrohydraulic position servos can provide excellent performance for a wide range of actuation power requirements. However, a distinct drawback in the choice of electrohydraulic servos is cost. The higher cost associated with electrohydraulic servos is due, primarily, to: (a) the need for a high-pressure hydraulic supply having good contamination control, and (b) the use of high performance, two-stage servovalves.

Two alternative technologies for servoactuation of low power level loads are receiving considerable attention because they may offer acceptable performance at lower cost. These are: (1) electromagnetic servos using brushless, samarium cobalt, electric motors, and (2) electropneumatic servos. This paper describes recent developments at Moog Inc. with electropneumatic servos, and a companion paper does the same for electromagnetic servos.[1]

TECHNOLOGY COMPARISONS

The potential for alternate technologies should be assessed in light of the well-known capabilities of conventional electrohydraulic servos. Figure 1 shows typical power level and dynamic response requirements for a variety of aerospace servoactuator applications. The performance available with electrohydraulic servos encompasses every application shown. This is easily explained because electrohydraulic servoactuation systems have been (and will, undoubtedly, continue to be) designed and developed to accomplish essentially every task that has appeared.

[1] See References

Figure 1 indicates that applications in the lower ranges of power and dynamic response may also be satisfied with electromagnetic and electropneumatic servos. The best choice, then, is determined by other considerations such as those listed in Table I. Putting aside Customer Preference (i.e., "bias") as the often prevailing concern, the aspect of Cost is generally dominant. Experience indicates that, in many applications, the cost of either electromagnetic or electropneumatic servoactuation will be lower than electrohydraulic.

This cost differential rapidly dissipates for applications that require high power and/or high dynamic response. The present practical limits for electromagnetic and electropneumatic aerospace servoactuators are approximately those shown in Figure 1.

In comparing costs, one must be careful to consider the total cost of the entire servoactuation system. If a servoactuation system is defined by the components that relate an electrical command to the motion being controlled, together with the power source and power conversion equipment necessary to run the servo, then the block diagram of Figure 2 describes the system. The components commonly used for each type of servo are named in Table II. The need for accessories such as a pump, filter, reservoir, battery, etc. is self evident. Unfortunately, the higher cost of an electrohydraulic servo often results from the power conversion equipment needed to provide high-pressure, hydraulic fluid having low contamination.

The relative costs of alternate actuation systems designed for a specific application will depend, primarily, on the actuation power level. The term **actuation power,** as used in this paper, is the maximum power that can be delivered by the actuator to move the load. This is not the product of stall force (or torque) and no-load velocity, but, rather, is the maximum power the actuator is capable of delivering to the load. For electropneumatic and electromagnetic systems, the maximum continuous* actuation power is approximately ¼ of the product of stall load and no-load velocity, whereas for an electrohydraulic system, theoretical maximum actuation power is 0.38 (stall force x no-load velocity).

Generalized cost trends for the three types of actuation systems discussed here are illustrated in Figure 3. It should be emphasized that these cost comparisons relate to flightworthy aerospace hardware and the costs include all components of the actuation system (see Table II.) In the range below ½ hp, an electropneumatic actuation system is on the order of one-half the cost of an equivalent electrohydraulic system.

Table I

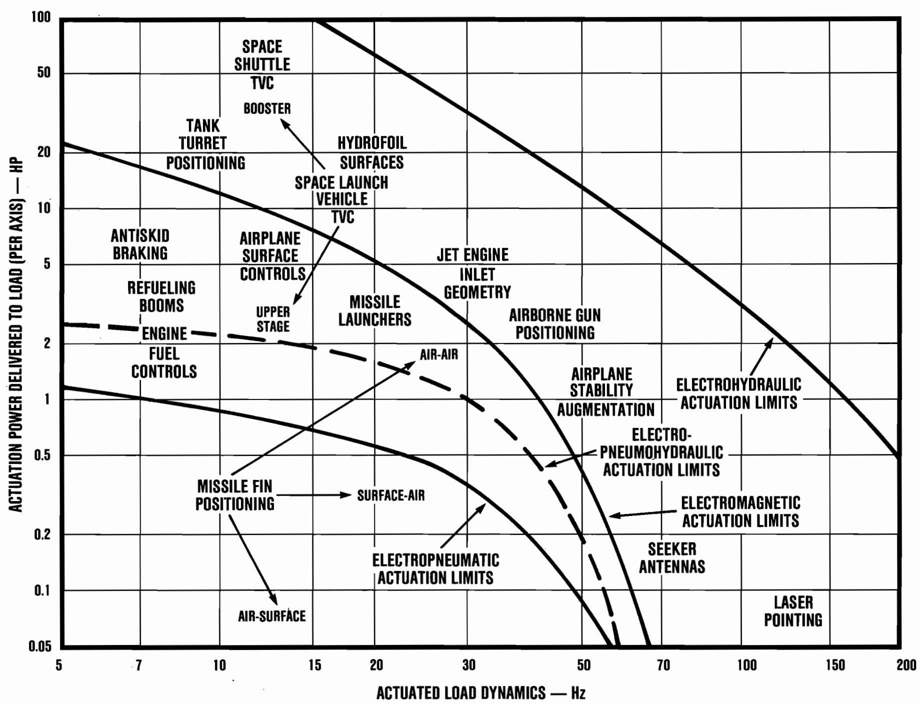


FIGURE 1. TYPICAL AEROSPACE APPLICATION REQUIREMENTS

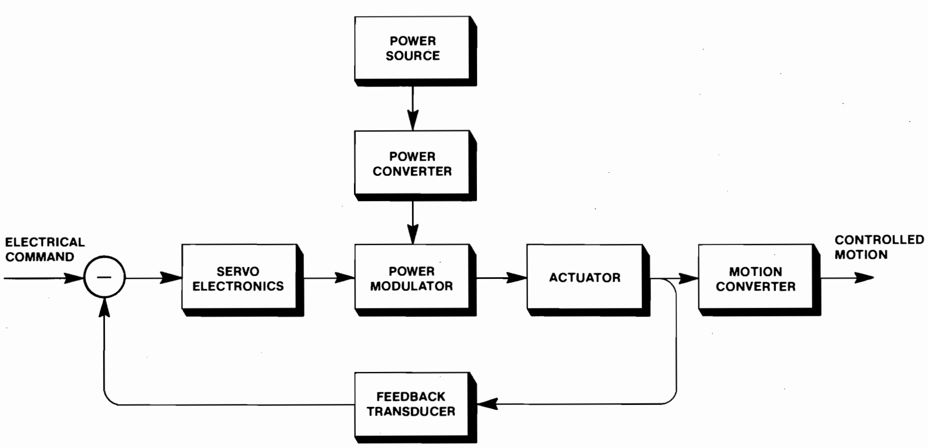


FIGURE 2. SERVOACTUATION SYSTEM COMPONENTS

**Electric motor actuators can usually be overdriven for a short period of time (hence, momentarily develop higher output power). The period of overdrive is limited by excessive internal heating of the motor or the drive electronics.*

TABLE I

OTHER SELECTION CRITERIA FOR SERVOACTUATION SYSTEMS

- Customer Preference
- Cost
- Size & Weight
- Duty Cycle
- Environment: vibration, shock & acceleration
  temperature
  nuclear hardening
  EMI
- Static Accuracy
- "ilities": storability
  verifiability
  reuseability
  maintainability
  transportability
  reliability

ELECTROPNEUMATIC SERVOACTUATORS

Consideration of an electropneumatic servo for a specific application generally focuses on dynamic performance. Obviously, the "fluid" used in a pneumatic servo is highly compressible and this compliance introduces severe performance degradation. The basic problem can be traced to the low-frequency time constant associated with pressure changes in the piston control chamber. This time constant not only reduces the dynamic response of an electropneumatic servo, but it also limits actuation loop gain which, in turn, degrades static accuracy (e.g., resolution and hysteresis from load friction, and positioning non-linearity due to external loading).

Another serious drawback with many electropneumatic servos is the dramatic change in actuation stiffness that occurs with variations in operating pressure and with movement of the actuator throughout its operating range. To assess the magnitude of these effects it is helpful to develop a simplified analytic model.

Consider the electropneumatic actuator illustrated by the partial drawing in Figure 4. This actuator, which might position fins for steering a tactical missile, is a simple push-push, three-way, double-piston configuration having a 2:1 area ratio. Servo operating pressure is supplied continuously to the half-area piston.

A pair of two-way, solenoid-operated poppet valves control gas flow into and out of the large piston. These solenoid valves are driven in an ON-OFF-ON fashion to create a closed-center condition so that continuous gas flow from supply pressure to exhaust is avoided.

A position transducer is attached to the output shaft of the actuator to provide a load position feedback signal. When an error exists between the feedback and command signals, a servoamplifier drives a proportional, pulse-width modulator (PWM) which supplies a series of electrical pulses to the appropriate solenoid for reducing the error. The modulation frequency of the PWM is typically 5 to 10 times higher than the highest servo operating frequency. The useful range of modulation extends from the shortest width pulse that will result in movement of a valve poppet, up to a pulse width that will give continuous poppet opening.

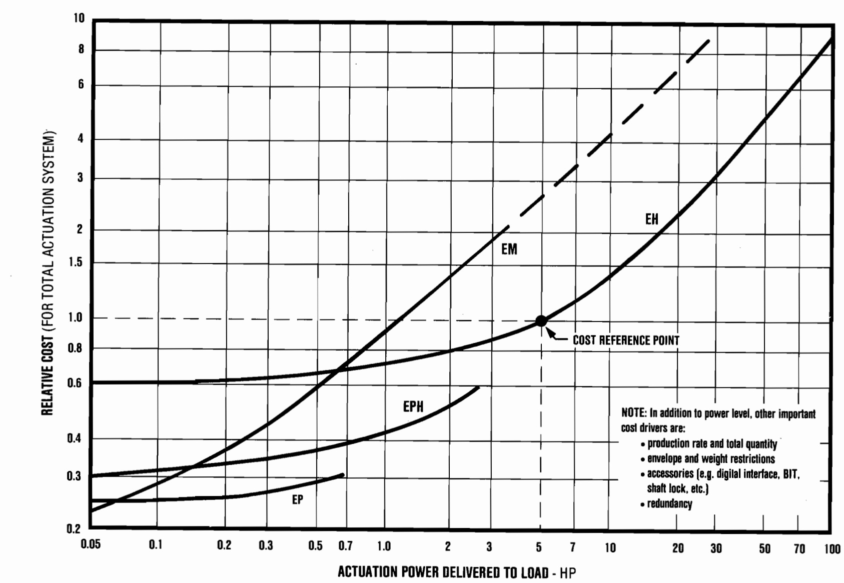


FIGURE 3. GENERAL COST TRENDS FOR ALTERNATE SERVOACTUATION SYSTEMS (USING MIL-SPEC COMPONENTS)

TABLE II TYPES OF COMPONENTS USED IN AEROSPACE FLUID POWER SERVOACTION SYSTEMS

COMPONENT / TECHNOLOGY	POWER SOURCE	POWER CONVERTER	POWER MODULATOR	SERVOELECTRONICS			ACTUATOR	MOTION CONVERTER	
				COMMAND	FEEDBACK TRANSDUCER	MODULATOR DRIVE		LINEAR OUTPUT	ROTARY OUTPUT
ELECTRO-HYDRAULIC (EH)	ENGINE OR BATTERY OR WARM GAS GENERATOR (SOLID OR LIQUID (THROTTLE-ABLE))	PUMP (FIXED DISPLACEMENT / VAR. DISPLACEMENT) MOTOR & PUMP TURBOPUMP (CENTRIFUGAL OR PISTON)	SERVOVALVE (1 STAGE OR 2 STAGE OR 3 STAGE)	ANALOG OR DIGITAL	ELECTRICAL OR MECHANICAL	PROPORTIONAL OR PWM*	SINGLE PISTON (3-WAY OR 4-WAY) OR PUSH-PUSH PISTONS (3-WAY OR 4-WAY) OR FLUID MOTOR (GEAR/VANE OR PISTON)	NOT NECESSARY N/A** BALLSCREW	CRANK ARM ROCKER ARM GEARS
ELECTRO-PNEUMO-HYDRAULIC (EPH)	COLD GAS BOTTLE OR GAS GENERATOR	LOW PRESSURE REGULATOR & BLOWDOWN FLUID TANK OR LOW PRESSURE REGULATOR & FREE PISTON PUMP HOT GAS RELIEF VALVE & BLOWDOWN FLUID TANK	DUAL SOLENOID VALVES OR SERVOVALVE	ANALOG OR DIGITAL	ELECTRICAL	PWM* OR PROPORTIONAL	SINGLE PISTON (3-WAY OR 4-WAY) OR PUSH-PUSH PISTONS (3-WAY OR 4-WAY)	NOT NECESSARY N/A**	CRANK ARM ROCKER ARM
ELECTRO-HYDROSTATIC (EHS)	ENGINE	ALTERNATOR & RECTIFIER OR DC GENERATOR	SOLID STATE ELECTRONIC POWER SWITCHES	ANALOG OR DIGITAL	ELECTRICAL	PWM* OR PWM PLUS COMMUTATION	ELECTRIC MOTOR DRIVEN REVERSIBLE PUMP WITH PISTON OR FLUID MOTOR	NOT NECESSARY N/A**	CRANK ARM GEARS
ELECTRO-MAGNETIC (EM)	ENGINE OR BATTERY OR WARM GAS GENERATOR (LIQUID OR SOLID)	ALTERNATOR & RECTIFIER NONE OR VOLTAGE STEP-UP TURBO ALTERNATOR & RECTIFIER	SOLID STATE ELECTRONIC POWER SWITCHES	ANALOG OR DIGITAL	ELECTRICAL	PWM* OR PWM PLUS COMMUTATION	BRUSH DC MOTOR: FIXED MAGNETS, ROTATING WINDINGS, COMMUTATOR 3 ø BRUSHLESS DC MOTOR: FIXED WINDINGS, ROTATING MAGNETS, ROTOR POSITION SENSOR	BALLSCREW	GEARS
ELECTRO-PNEUMATIC (EP)	ENGINE BLEED OR COLD GAS BOTTLE	LOW PRESSURE REGULATOR	DUAL SOLENOID VALVES OR TRISTABLE VALVE	ANALOG OR DIGITAL	ELECTRICAL	PWM*	3-WAY SINGLE PISTON OR 3-WAY PUSH-PUSH PISTONS	NOT NECESSARY N/A**	CRANK ARM ROCKER ARM

*PWM = PULSE WIDTH MODULATION

**N/A = NOT APPLICABLE

ANALYTICAL MODEL

The parameters that are used for a simple analytic model of this servo are illustrated in the sketch of Figure 5 and defined in Table III. The equations relating the variables of this system are:

SOLENOID COMMAND

$$e_S = K_A K_M (e_C - K_F \theta_L) \quad \text{(ASSUMING THAT NO ELECTRICAL COMPENSATION IS USED)} \qquad (1)$$

SOLENOID FLOW EQUATIONS[2] (SEE FIGURE 6)

$$\dot{W}_A = e_S K_S \left(\frac{P_S}{\sqrt{T_S}}\right) C_2 \qquad \left\{ \begin{array}{l} \text{FOR } e_S > 0 \\ \text{AND } \left(\frac{P_1}{P_S}\right) \le r_C \text{ (CHOKED)} \end{array} \right. \qquad (2)$$

$$\dot{W}_A = e_S K_S \left(\frac{P_S}{\sqrt{T_S}}\right) C_1 f_1 \qquad \left\{ \begin{array}{l} \text{FOR } e_S > 0 \\ \text{AND } \left(\frac{P_1}{P_S}\right) > r_C \end{array} \right. \qquad (3)$$

$$\dot{W}_B = e_S K_S \left(\frac{P_1}{\sqrt{T_1}}\right) C_2 \qquad \left\{ \begin{array}{l} \text{FOR } e_S < 0 \\ \text{AND } \left(\frac{P_E}{P_1}\right) \le r_C \text{ (CHOKED)} \end{array} \right. \qquad (4)$$

$$\dot{W}_B = e_S K_S \left(\frac{P_1}{\sqrt{T_1}}\right) C_1 f_2 \qquad \left\{ \begin{array}{l} \text{FOR } e_S < 0 \\ \text{AND } \left(\frac{P_E}{P_1}\right) > r_C \end{array} \right. \qquad (5)$$

where

$$K_S = C_O A_O \qquad \text{(EFFECTIVE ORIFICE SIZE)} \qquad (6)$$

$$r_C = \left(\frac{2}{k+1}\right)^{\left(\frac{k}{k-1}\right)} \qquad \text{(A CONSTANT)} \qquad (7)$$

$$C_1 = \sqrt{\frac{2gk}{R\,(k-1)}} \qquad \text{(ANOTHER CONSTANT)} \qquad (8)$$

$$C_2 = \sqrt{\frac{gk}{R}\left(\frac{2}{k+1}\right)^{\left(\frac{k+1}{k-1}\right)}} \qquad \text{(YET ANOTHER CONSTANT)} \qquad (9)$$

$$f_1 = \sqrt{\left(\frac{P_1}{P_S}\right)^{\left(\frac{2}{k}\right)} - \left(\frac{P_1}{P_S}\right)^{\left(\frac{1+k}{k}\right)}} \qquad (10)$$

$$f_2 = \sqrt{\left(\frac{P_E}{P_1}\right)^{\left(\frac{2}{k}\right)} - \left(\frac{P_E}{P_1}\right)^{\left(\frac{1+k}{k}\right)}} \qquad (11)$$

This imposing set of gas flow equations determine the actual valve gas flows for: (1) adiabatic conditions (no heat transferred between the gas and the valve or actuator parts), (2) the state of the gas flow (i.e., sonic for equations 2 and 4; subsonic for equations 3 and 5) which exists for valve pressure ratios above or below the critical pressure ratio, r_C, and (3) the gas characteristics, k and R.

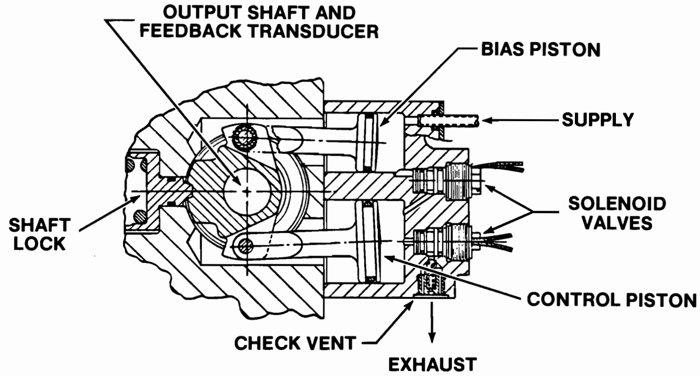


FIGURE 4. 3-WAY ROTARY ELECTROPNEUMATIC ACTUATOR

TABLE III

DEFINITIONS AND NOMENCLATURE

e_C =	piston command signal		volts
e_F =	position feedback signal		volts
e_S =	solenoid drive signal (% modulation)		%
$f_{1,2}$ =	various dependent functions		— — —
$C_{1,2}$ =	various constants		— — —
$\dot{W}_{A,B}$ =	weight of gas flow		lb/sec
$Q_{A,B}$ =	volumetric gas flow		in^3/sec
P_S =	supply pressure		psi
P_1 =	control piston pressure		psi
P_E =	exhaust pressure		psi
$T_{S,1}$ =	gas temperatures (°R = °F + 459.7)		°Rankine
V_1 =	control chamber volume		in^3
ρ_1 =	gas density in control piston chamber		lbs/in^3
R =	gas constant		in/°Rankine
k =	ratio of gas specific heats		— — —
r_C =	gas critical pressure ratio		— — —
g =	gravitational acceleration	386	in/sec^2
β_1 =	bulk modulus of gas in control chamber		psi
M_L =	actuator torque		in-lbs
θ_L =	actuator position		radians
K_A =	servoamplifier gain		volts/volts
K_M =	PWM gain		%/volt
K_S =	solenoid valve equivalent orifice size		in^2
K_F =	position feedback gain		volts/rad
$K_{1,2}$ =	other constants		— — —
C_O =	valve orifice flow coefficient	$\simeq 0.7$	— — —
A_O =	valve orifice area		in^2
A =	control piston area		in^2
r =	radius at which pistons act		in
I =	load inertia		$in\text{-}lb\text{-}sec^2$
B =	load damping		in-lb-sec
τ_L =	load time constant		sec
τ_G =	actuator pneumatic time constant		sec

Continuing,

VOLUMETRIC GAS FLOWS

$$Q_A = \frac{\dot{W}_A}{\rho_1} \qquad Q_B = \frac{\dot{W}_B}{\rho_1} \tag{12}$$

where GAS DENSITY $\rho_1 = \frac{P_1}{RT_1}$ (13)

CONTROL CHAMBER GAS TEMPERATURE

$$T_1 = T_S \left(\frac{P_1}{P_S}\right)^{\left(\frac{k-1}{k}\right)} = T_S\, f_3 \tag{14}$$

RATE OF CHANGE OF PRESSURE IN CONTROL CHAMBER

$$\dot{P}_1 = \frac{\beta_1}{V_1}\left(Q_A - Q_B - Ar\, \dot{\theta}_L\right) \tag{15}$$

where GAS BULK MODULUS $\beta_1 = k\, P_1$ (16)

ACTUATION TORQUE

$$M_L = Ar\left(P_1 - \frac{P_S}{2}\right) \tag{17}$$

AND LOAD VELOCITY

$$\dot{\theta}_L = \frac{M_L}{sI + B} \tag{18}$$

A block diagram representation of equations 1-18 appears in Figure 7.

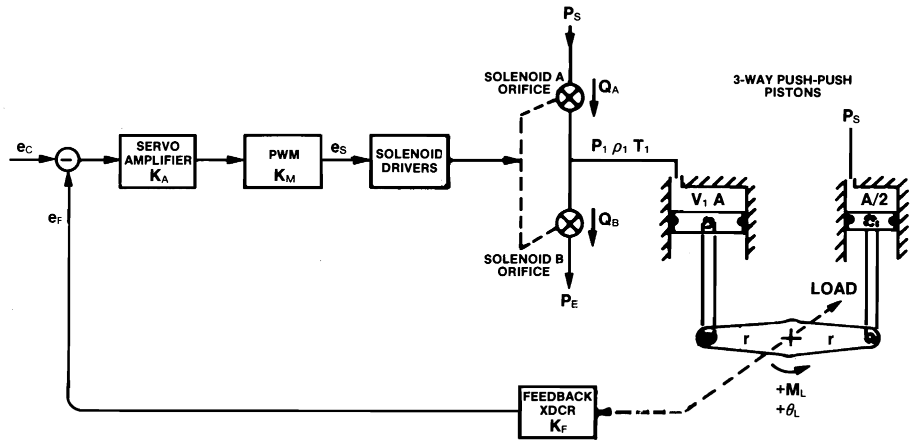


FIGURE 5. PARAMETERS FOR SERVOACTUATOR OF FIGURE 4

Clearly, a number of simplifying assumptions have been made to arrive at this analytic model. These include:

- no dynamic compensation is used in the electronics,
- the solenoid dynamics (both electrical and mechanical) are negligible,
- the speed of operation is such that adiabatic gas flow can be assumed,
- performance is modeled for small displacements so the control chamber volume, V_1, can be assumed constant,
- negligible shaft compliance exists so the actuator and load dynamics can be lumped,
- the load damping is viscous, and
- additional loads have been overlooked (e.g., spring rate effects on the load).

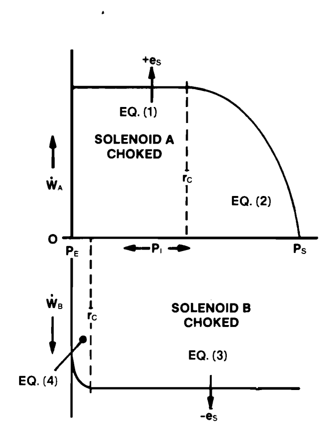


FIGURE 6. SOLENOID GAS FLOWS

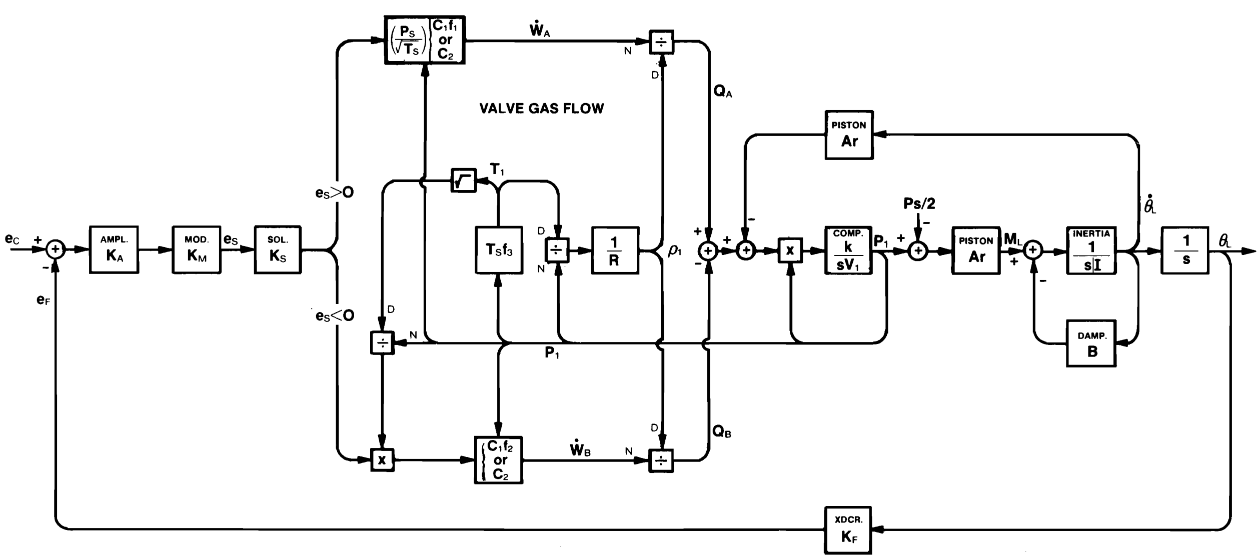


FIGURE 7. BLOCK DIAGRAM FOR ELECTROPNEUMATIC SERVOACTUATOR OF FIGURE 5

A better intuitive appreciation of this electropneumatic servo can be gained by linearizing the gas flow equations for small perturbations about a nominal set of operating conditions. In addition, assume that the gas flow through both solenoids is choked (i.e., sonic). Gas flow through solenoid B will be choked for all but extreme load torque conditions as $P_1 \gg P_E$. P_1 is nominally about ½ P_S, which is approximately the value of the gas critical pressure ratio, r_C. (See Table IV). This means that $P_1 \simeq r_C P_S$. Therefore, gas flow through solenoid A is close to choked. (Also, recognize that the amount of gas flow is not discontinuous as it changes from subsonic to sonic so the assumption of choked flow is reasonably accurate.)

In addition, it can be assumed that the gas temperature in the control chamber, T_1, is relatively constant at the operating point conditions. All of these assumptions lead to the following linearized equations where the various constants, K_1-K_7, are defined in Table V.

$$\dot{W}_A = K_1 e_S \qquad (19)$$

$$\dot{W}_B = K_2 e_S + K_3 P_1 \qquad (20)$$

$$\Delta Q = K_4 \Delta \dot{W} - K_5 P_1 \qquad (21)$$

where $\Delta \dot{W} = \dot{W}_A - \dot{W}_B$

$$\dot{P}_1 = K_6 P_1 + K_7 \Delta Q \qquad (22)$$

In practice, the solenoid orifice sizes are selected so that $K_1 \simeq K_2$. All of this leads to the simplified block diagram of Figure 8.

TABLE IV REPRESENTATIVE GAS PARAMETERS

Parameter	Nomenclature		Air	Helium	Nitrogen
gas constant	R	in/°R	640	4636	662
ratio of specific heats	k	— —	1.41	1.66	1.41
critical pressure ratio	r_C	— —	0.528	0.488	0.528
gas flow constant	C_1	$\sqrt{°R}/sec$	2.06	0.647	2.00
gas flow constant	C_2	$\sqrt{°R}/sec$	0.533	0.209	0.524

TABLE V LINEARIZATION COEFFICIENTS

$K_1 = \left(\frac{K_S P_S C_2}{\sqrt{T_S}}\right)$ $\frac{\text{lbs/sec}}{\%/100}$

$K_2 = \left(\frac{K_S P_{10} C_2}{\sqrt{T_{10}}}\right)$ $\frac{\text{lbs/sec}}{\%/100}$

$K_3 = \left(\frac{K_S e_{SO} C_2}{\sqrt{T_{10}}}\right)$ $\frac{\text{lbs/sec}}{\text{psi}}$

$K_4 = \frac{1}{\rho_{10}}$ in^3/lb

$K_5 = \left(\frac{\Delta \dot{W}_O}{\rho_{10} P_{10}}\right)$ $\frac{\text{in}^3/\text{sec}}{\text{psi}}$

$K_6 = \left(\frac{k \Delta Q_O}{V_{10}}\right)$ $\frac{1}{\text{sec}}$

$K_7 = \left(\frac{k P_{10}}{V_{10}}\right)$ $\frac{\text{psi}}{\text{in}^3}$

SECOND SUBSCRIPT 0 DENOTES NOMINAL OPERATING CONDITION

NOTE: K_6 and K_7 vary with actuator position,
K_5 and K_6 vary with nominal solenoid flow,
K_2 and K_7 vary with nominal actuator load,
K_3 varies with nominal servoloop gain,
and for static conditions, K_3, K_5 and K_6 approach zero

The forward leg of this block diagram includes two dynamic effects: (1) the first-order dynamics associated with gas compressibility and valve damping, and (2) the first-order dynamics associated with the load. The time constant associated with the load is generally small, or may cease to exist when the load damping is insignificant.

$$\tau_L = \frac{I}{B} \quad \text{sec} \qquad (23)$$

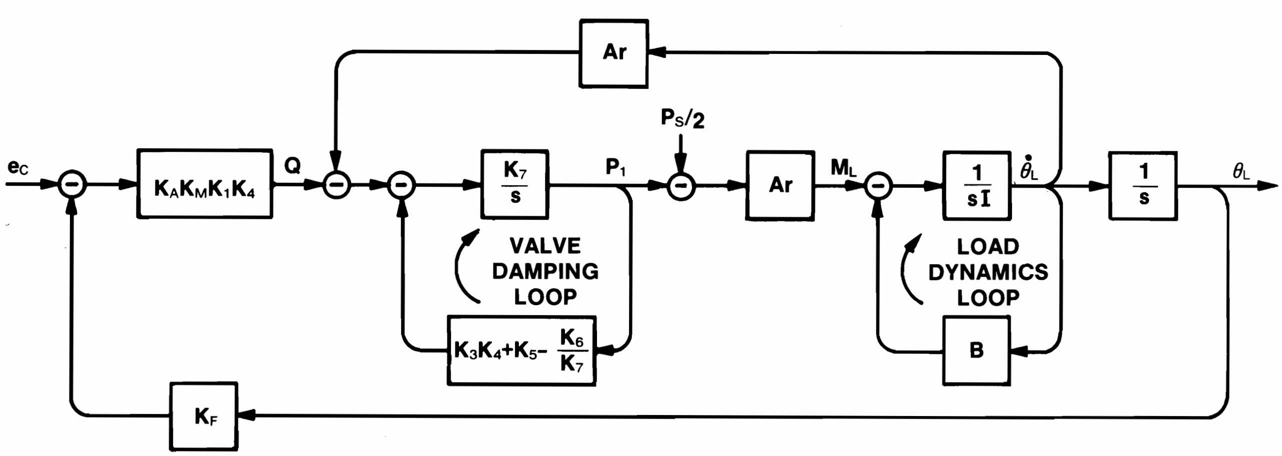


FIGURE 8. SIMPLIFIED DYNAMIC REPRESENTATION OF ELECTROPNEUMATIC SERVOACTUATOR FOR SMALL AMPLITUDE MOTION ABOUT A SPECIFIC OPERATING POINT

The time constant associated with the idiosyncrasies of the pneumatic medium is long (and so, quite troublesome) and varies widely with nominal operating conditions.

$$\tau_G = \frac{1}{K_7 (K_3K_4 + K_5 - K_6/K_7)} \qquad (24)$$

This gas time constant contains the following:

a) K_7 is the fluid bulk modulus which represents the increase in P_1 due to incremental increases of fluid into the control chamber volume,

b) K_3K_4 is the increase in flow through solenoid B due to increased P_1,

c) K_5 is the fluid density effect that reduces the relationship of volumetric flow to weight flow as P_1 increases, and

d) K_6/K_7 is the increase in bulk modulus with higher P_1 that, in effect, makes the fluid volume stiffer.

In a typical application, τ_L may be 0.003 sec (corresponding to a corner frequency of 50 Hz) whereas τ_G may vary from 0.08 to 0.008 sec (2 to 20 Hz) as the control piston moves from fully extended to fully retracted and as other operating conditions change. The variation of loop dynamics is illustrated by the frequency response plots in Figure 9.

The reader should be cautioned about use of the simplified block diagram shown in Figure 8. Several of the linearization coefficients vary in value as the actuator operating conditions change (specifically: position, velocity and torque). Also it should be noted that when the actuator approaches a static condition, the feedback in the Valve Damping Loop approaches zero. This leaves the basic integration of flow into the control chamber with near-zero position. This integration reduces static inaccuracies to reasonable values.

COMPENSATION TECHNIQUES FOR PUSH-PUSH PNEUMATIC SERVOACTUATOR

The most effective technique for improving the performance of a push-push electropneumatic servoactuator being practiced today is to attempt to linearize performance by compensating for the wide variations in the gas time constant. A practical scheme is to utilize the half-area biasing piston as another gas damper that also varies with piston stroke. This approach is illustrated by the schematic in Figure 10.

Moog has modeled this system for accurate design analysis by an extension of the basic system equations (1-18) to include two more gas control volumes, V_2 and V_3. Digitally derived solutions using CSMP[3] with IBM VM370 software on an IBM 3033 mainframe computer give good correlation with hardware test data.

Intuitively it can be seen that the affect of the damped biasing piston is to add two more minor loops that relate additional piston torques to piston velocity. These loops have gas time constants, τ_{G2} and τ_{G3}, that act in parallel to that of the control piston. When the gas time constant of the control piston is longest (i.e., when the control piston is extended), that of chamber V_2 (which is about the same size as the control chamber volume) is shortest. This offsetting affect reduces the variation of dynamic response throughout the range of stroke.

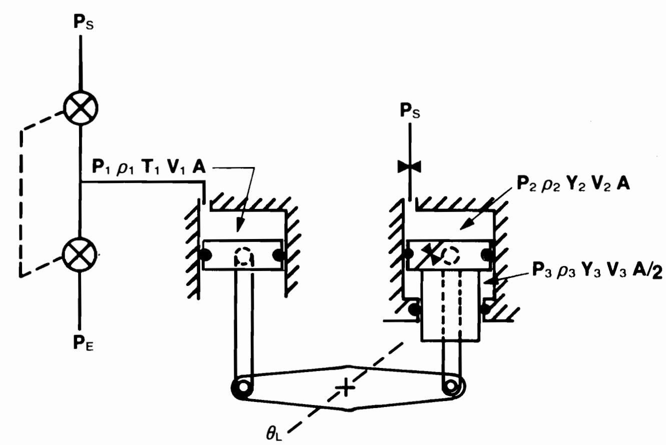


FIGURE 10. PUSH-PULL ROTARY PNEUMATIC ACTUATOR WITH DAMPING PISTON

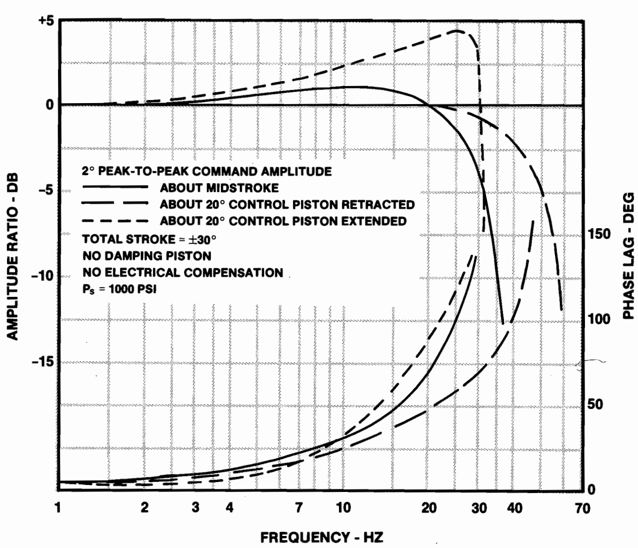


FIGURE 9. PNEUMATIC SERVO FREQUENCY RESPONSE

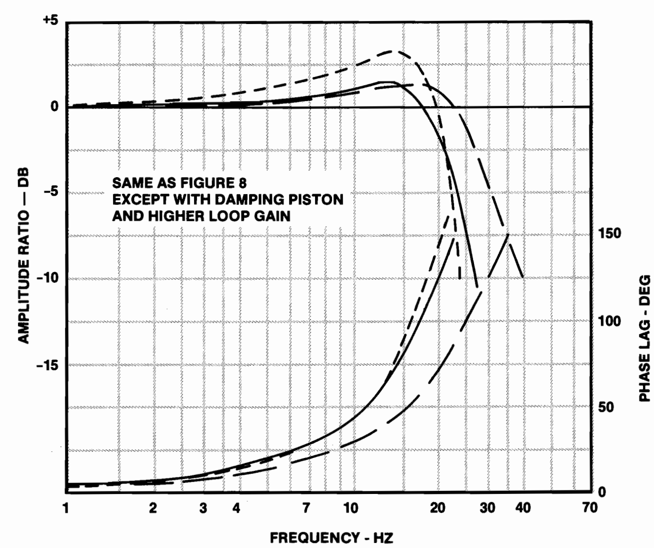


FIGURE 11. PNEUMATIC SERVO WITH DAMPING PISTON

Figure 11 illustrates the improvement in dynamic response achieved by adding a damping piston to the servo whose performance was given in Figure 9.

ALTERNATE MECHANIZATIONS OF AN ELECTROPNEUMATIC SERVO

Clearly, the rotary-motion electropneumatic actuator of Figure 10 can be adapted for rectilinear motion by an arrangement as indicated in Figure 12.

Poppet-type valves are generally used in pneumatic servos because of their inherent simplicity, lower cost and low leakage when closed. Spool-type valves, on the other hand, inherently have some laminar leakage when closed. Also, they have an annoying tendency to gall (followed by seizure) during rapid and repeated cycling while valving gas. Three and 4-way spool valves can have a closed-center null configuration which reduces gas consumption. A closed-center valve is almost essential for a blow-down, stored-gas electropneumatic servo as illustrated in Figure 13. In this case, a closed-center condition is obtained by having deadzone (created by spring preload) in the two, 2-way solenoid valves.

Another way to accomplish 3-way valving with deadzone (ON-OFF-ON) is to have a 3-way, closed-center poppet valve driven by a 3-position solenoid or force motor. Two valve configurations are illustrated by the schematics in Figure 14 and two types of drivers are shown in Figure 16. The primary difference between the two valve arrangements is the relative ease of having either the supply or exhaust orifice the larger.

A double solenoid for creating the ON-OFF-ON gas valve positions requires two coils and separate drive amplifiers (unless diodes are used with a polarity sensitive, single-ended drive). Typically, each solenoid will require about 25 watts electrical power and current

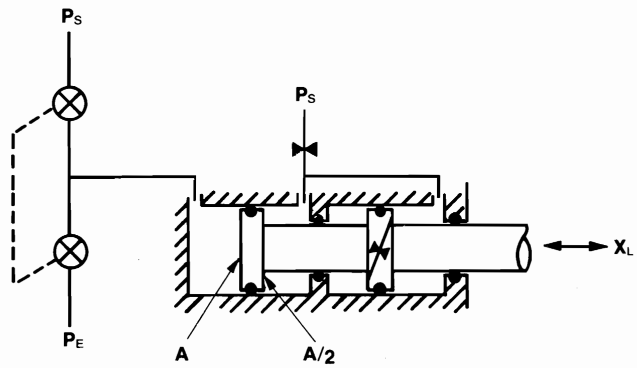


FIGURE 12. RECTILINEAR MOTION PNEUMATIC ACTUATOR WITH DAMPING PISTON

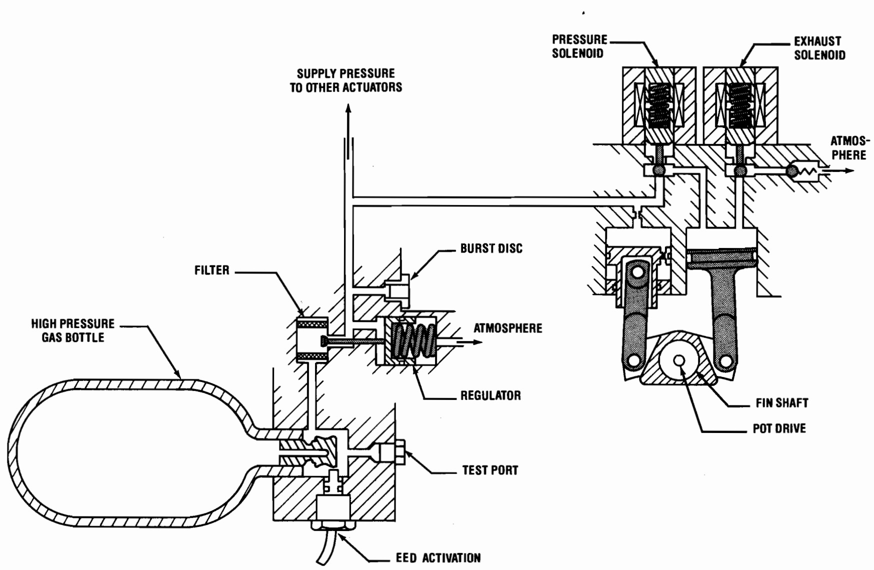


FIGURE 13. ELECTROPNEUMATIC FIN ACTUATION SCHEMATIC

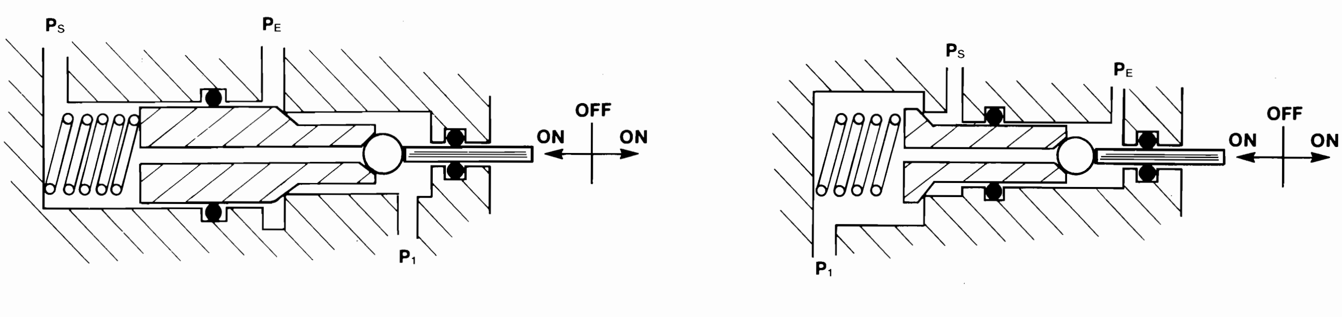


FIGURE 14. ALTERNATE CLOSED-CENTER, 3-WAY PNEUMATIC VALVES

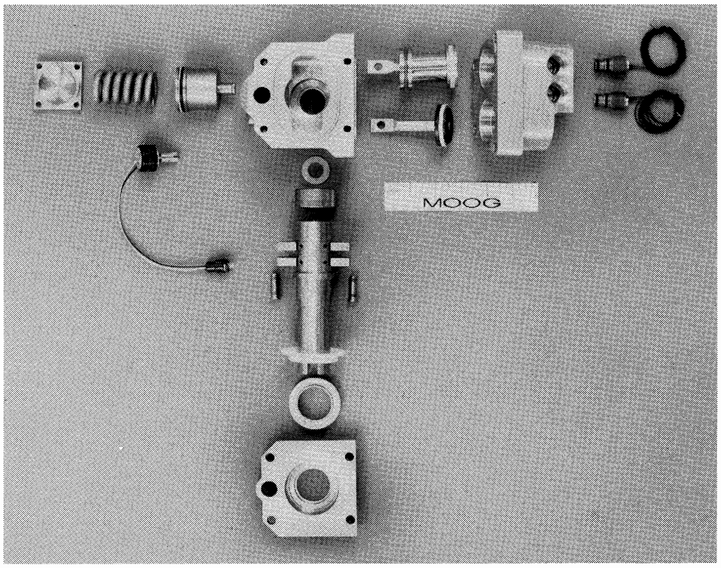


FIGURE 15. PUSH-PUSH ACTUATOR WITH DAMPING PISTON

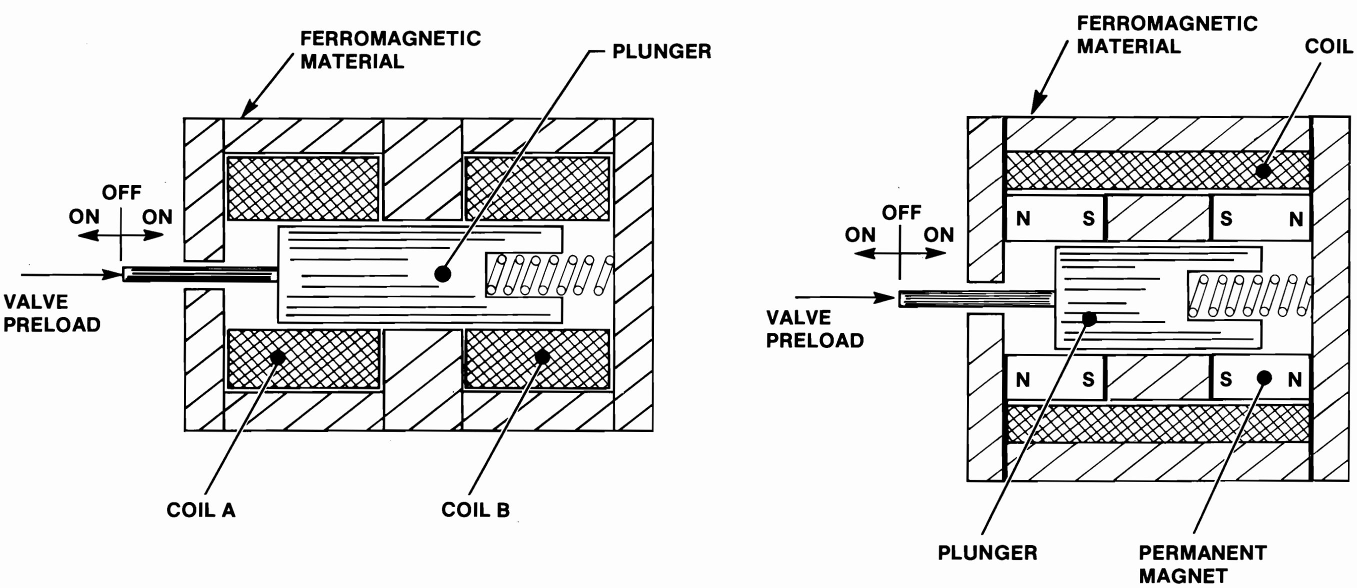
limiting is necessary to prevent excess electrical power at low temperatures (where the coil resistance is low). A force-motor type valve positioner, on the other hand, requires much less electrical power (2 to 5 watts) due to the use of permanent magnets. The magnets polarize the active air gaps so a single coil can be used if energized by a bidirectional drive amplifier.

APPLICATIONS

Moog applications of electropneumatic actuation have focused on fin control for medium-performance tactical missiles. To be more quantitative, "medium performance" can be defined as having peak actuation power (per fin) in the range from 0.07 to 0.4 hp with fin dynamic response in the range from 10 to 40 Hz. These performance parameters generally relate to air-to-surface and surface-to-air tactical missiles as illustrated in Figure 1 (Page 2).

Other reasons for focusing on this marketplace include: (a) the inherent fin duty cycle requirements for these missiles is relatively moderate, (b) operational life is characterized by one-shot usage following long-term storage and, most significantly, (c) the primary design emphasis for these missiles is low cost. If an electropneumatic system can meet the performance requirements, it is usually the lowest cost choice.

Most of the older, medium-performance tactical missiles use electrohydraulic fin actuation systems (e.g., Hawk, Maverick, and Patriot). These systems are relatively expensive due to the cost of hydraulic components such as servovalves, accumulators, filters and pumps. Today the competing technology for actuation in such missiles is either electromagnetic or electropneumatic. The cost of electromagnetic fin actuation is generally higher than electropneumatic due to the need for (a) high power, solid state switching devices, (b) precision motion reduction mechanisms, (c) a motor rotor position transducer, (d) separate fin lock/unlock devices, and (3) more complex control electronics (with the attendant cost of electrical components suitable for a wide thermal environment, adequate EMI suppression, and



(a) 3-POSITION DOUBLE SOLENOID **(b) 3-POSITION FORCE MOTOR**

FIGURE 16. VALVE DRIVERS

necessary effort to comply with DOD-2000 and MIL-STD-965.

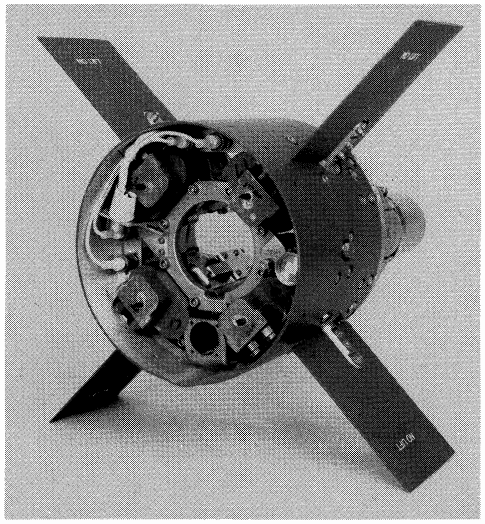
Electropneumatic fin actuation systems, on the other hand, can use simple on-off solenoid valves that operate on relatively low electrical power. Lowering the electrical power significantly reduces the cost of the electronics. The gas bottle power for an electropneumatic fin actuation is simple and inherently suitable for long-term storage with one-shot usage. Also, fin locks are a straightforward accessory with an EP actuator and do not require separate energization.

The relative advantages and disadvantages of electropneumatic fin actuation systems are summarized in Table VI. Examples of these systems are pictured in Figures 17, 18, and 19.

SUMMARY

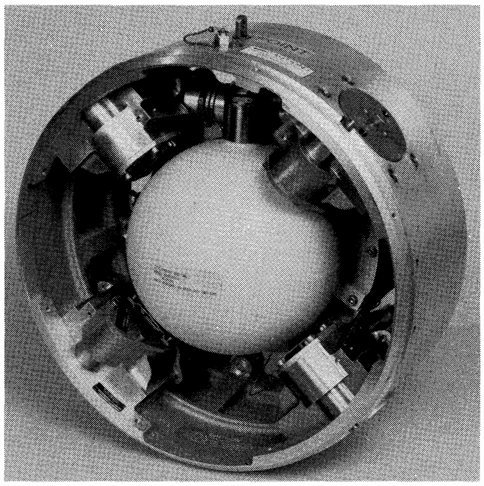
Electropneumatic actuation can claim a distinct segment of the overall market for electrically commanded servoactuators. The biggest potential advantage is lower cost than competing technologies. Other advantages are summarized in Table VI. The major disadvantages are the limited actuation power and performance obtainable.

The development efforts presently being expended by Moog and others to extend the performance capabilities of electropneumatic servoactuators are directed primarily at applications in the Defense Industry. However, the lessons learned and the advances made will, most likely, be adaptable to other markets. Robotics is a prime example of an industry that would welcome lower cost servocontrol if performance, operating life and safety were acceptable.

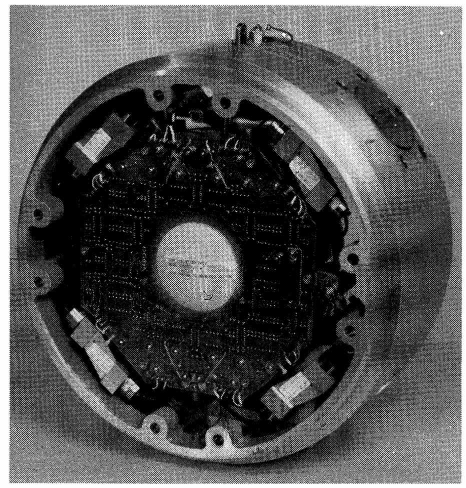


FIN INERTIA	0.009 IN-LB-SEC²
FREQUENCY RESPONSE (90° PHASE)	14 Hz
RESOLUTION	0.15 DEG
HELIUM STORAGE PRESSURE	7450 PSI
OPERATING PRESSURE	550 PSI
OPERATING TIME	105 SEC
DIAMETER	12 INCHES
FIN TRAVEL	±25 DEG
PISTON AREA-RADIUS	1.13 IN³/RAD
STALL TORQUE	620 IN-LBS
NO-LOAD VELOCITY	300 DEG/SEC
ACTUATION POWER (PER AXIS)	0.12 HP

FIGURE 17. MAVERICK EP ACTUATION PACKAGE



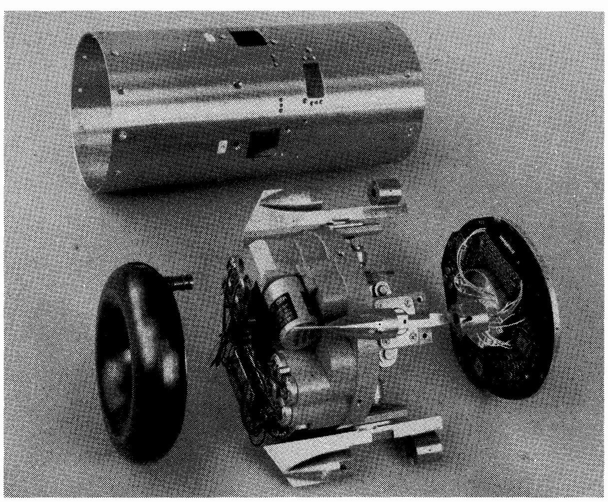
BACK SIDE



FRONT SIDE

DIAMETER	16 INCHES	FIN INERTIA	0.052 IN-LB-SEC²
FIN TRAVEL	±22 DEG	FREQUENCY RESPONSE (90° PHASE)	9 Hz
PISTON AREA-RADIUS	1.13 IN³/RAD	RESOLUTION	0.1 DEG
STALL TORQUE	960 IN-LBS	HELIUM STORAGE PRESSURE	8000 PSI
NO-LOAD VELOCITY	150 DEG/SEC	OPERATING PRESSURE	850 PSI
ACTUATION POWER (PER AXIS)	0.08 HP	OPERATING TIME	600 SEC

FIGURE 18. GBU-15 RP FIN ACTUATION PACKAGE



DIAMETER	6.5 INCHES	FIN INERTIA	0.008 IN-LB-SEC²
FIN TRAVEL	±20 DEG	FREQUENCY RESPONSE (90° PHASE)	20 Hz
PISTON AREA-RADIUS	0.684 IN³/RAD	RESOLUTION	0.05 DEG
STALL TORQUE	820 IN-LBS	HELIUM STORAGE PRESSURE	10,000 PSI
NO-LOAD VELOCITY	450 DEG/SEC	OPERATING PRESSURE	1200 PSI
ACTUATION POWER (PER AXIS)	0.24 HP	OPERATING TIME	12 SEC

FIGURE 19. VT-1 SURFACE-TO-AIR MISSILE EP FIN ACTUATION PACKAGE

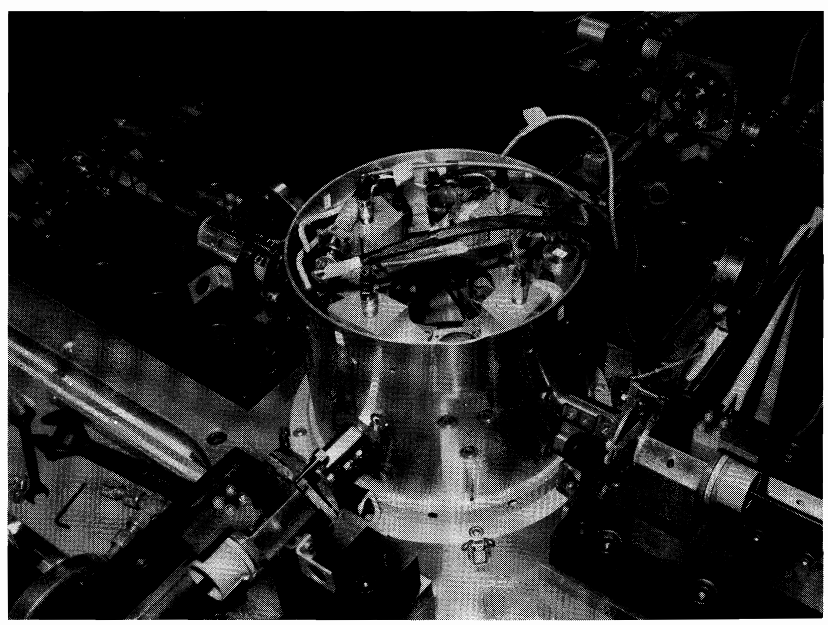


FIGURE 20. MAVERICK FOUR AXIS TEST RIG

REFERENCES

1 Moog Technical Bulletin #150 "High Performance Electromagnetic Servoactuation Using Brushless DC Motors," M. A. Davis 1984

2 "Fluid Power Control" Blackburn, Reethof & Shearer 1960
The Technology Press of M.I.T.; Library of Congress Catalog No. 59-6759

3 Continuous System Modeling Program (CSMP copyright of IBM)

ACKNOWLEDGEMENTS

The author wishes to acknowledge the following co-workers who have spearheaded the development activity that is described in this paper: Aurelius (Marc) Chaves, George Chenault, Charles Greaves, Arthur Phillips and L. Jack Williams.

TABLE VI

ADVANTAGES AND DISADVANTAGES OF ELECTROPNEUMATIC ACTUATION SYSTEMS

ADVANTAGES	DISADVANTAGES
• generally lowest cost	• limited to low power applications
• simple energy storage and Power Conversion	• poorer accuracy
• wide temperature capability	• low dynamic response
• high vibration and acceleration capability	• low backdrive stiffness
• long-term storability	• often requires damping and/or electronic servo compensation
• nuclear hardenable	• more difficult dynamic modeling
• low EMI emissions	• more difficult to check-out
• simple servoelectronics	• bottle transportability\approval
• medium power, pulsed solenoids	

OTHER TECHNICAL BULLETINS AVAILABLE FROM MOOG

TB 101 Controlled Damping through Dynamic Pressure Feedback

TB 103 Transfer Functions for Moog Servovalves

TB 104 Design Considerations for Mechanical Feedback Servoactuators

TB 108 Pulse Operated Bipropellant Reaction Control Valves

TB 114 Control Contamination in Hydraulic Systems

TB 115 Fluid Contamination Effects on Servovalve Performance

TB 116 Majority Voting Servoactuator for Space Launch Vehicles

TB 117 Specifications Standards for Servovalves

TB 119 Supply Pressure Considerations for Servoactuators

TB 121 The Deflector Jet Servovalve

TB 123 Time-Optional Electrohydraulic Servo Positioning System

TB 124 Remote Control of Hydraulic Equipment

TB 125 Lift/Lower Control Valve Package for Lift Trucks

TB 126 Performance Estimation for Electrohydraulic Control Systems

TB 127 Redundant Electrohydraulic Servoactuators

TB 128 Computer Controlled Testing of Servovalves

TB 129 Stepover Needlebar Positioner

TB 141 Brief History of Electrohydraulic Servomechanisms

TB 142 Microprocessors in Closed Loop Electrohydraulic Control Systems

TB 143 Vertical Stabilization of a Ship-Mounted 200 Ton Derrick

TB 144 Electrohydraulic High-Rate Impact Tester

TB 145 Controls for Injection Molding of Thermoplastics

TB 146 High Performance Electrohydraulic Control System for Large Grinding Machines

TB 147 MX Stage 1 Thrust Vector Actuation System

TB 148 Electrohydraulic Control Applied to Hydrostatic Transmissions

TB 150 High Performance Electromagnetic Servoactuation Using Brushless DC Motors

MOOG INC., EAST AURORA, NEW YORK 14052 • 716/652-2000 • FAX 716/687-4467 • TELEX 443-2023