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ADVANCED FLIGHT CONTROL ACTUATION SYSTEM (AFCAS - E/P)

Feasibility Investigation of an Electro/Pneumatic Dual Power Driven Concept

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1.0 SUMMARY

1.1 BACKGROUND INFORMATION

This final report presents the results of the BENDIX study and design effort on Contract No. N62269-77-C-0171, issued by NAVAL AIR DEVELOPMENT CENTER, Warminster, Pennsylvania. The Dual Mode Dynavector actuator that was produced by this program operates on 270 VDC electric power and/or 20 PSIG pneumatic power. This actuator consists of an electric stepper motor integrated with au anchored-vane pneumatic motor. The orbit rotor delivers the generated power to the output shaft through a step-down epicyclic gear transmission

The final configuration is equipped with a crank arm to convert the rotary output shaft motion to a linear drive that can control the rudder of a T-2C airplane (See Figure 1).

Computer studies have demonstrated that actuator systems based on the Dual Mode Dynavector have greater survivability than single mode actuators. Operation within specifications is achieved despite loss of either electric or pneumatic power. No interruption of control occurs during or after such power loss.

The Dual Mode Dynavector has a great size advantage over competing actuators. For example, the Dynavector produces stall torques that are twice those generated by a D.C. Torque Motor that is 4 times larger in volume.

Design parameters for the Dual Mode Dynavector are as follows:

Supply Voltage Supply Pressure Rated Torque, Single Mode Rated Torque, Dual Mode Rated Speed No-Load Speed Stall Torque, Single Mode Stall Torque, Dual Mode Resolution Diameter Length, Housing Air Flow, Rated Speed and Torque Electric Powr, Rated Speed and Torque Air Temperature, Pneumatic Supply Minimum Maximum Frequency Response, dual mode

270 VDC 20 PSIG (1.379x10⁵ Pa) 1500 In-Lbs. (169.5 Nm) 3000 In-Lbs. (338.95 Nm) 30°/Sec. (.52 rad/s) 50°/Sec. (.87 rad/s) 2200 In-Lbs. (248.6 Nm) 4400 In-Lbs. (497.2 Nm) + .048° (+ .838 millirad) 8.125 In. (206.38 mm) 5.2 In. (132.08 mm) .01946 lb./sec. (.0088 kg/s) 200 watts 0°F (-17.8°C) 350°F (177°C)

Flat Within + 3 DB Up to 20 Hz, (signal ampl. = .5° peak-to-peak)

1.2 EXTREME CONDITIONS

The Dual Mode Dynavector will function effectively under these extreme conditions:

1.2.1 Supply Voltage

The nominal supply voltage is 270 VDC. However, operation can be sustained continuously with the supply voltage in the range of 200 VDC to 280 VDC. Transient variations of .1 second duration or less can swing between 125 VDC and 475 VDC without damaging the Dual Mode Dynavector.

1.2.2 Supply Pressure

The nominal supply pressure is 20 PSIG (1.379 x 10^{5} Pa). However, operation can be sustained continuously with the supply pressure in the range 10 PSIG (6.895 x 10^{4} Pa) to 30 PSIG (2.07 x 10^{5} Pa).

1.2.3 Air Temperature, Pneumatic Supply

The nominal air temperature range of the pneumatic supply is $0^{\circ}F$ (-17.8°C) to 350°F (177°C). However, transient excursions of 30 seconds duration or less can vary up to $392^{\circ}F$ (200°C) without damaging the Dual Mode Dynavector.

The highest bleed-air temperature on the T2-C aircraft is $415^{\circ}F$ (212.8°C). This can be reduced to $350^{\circ}F$ (177°C) by flowing through a ten foot section of pnuematic tubing that is fitted with air-cooling fins.

1.3 SURVIVAL WITH THE DUAL MODE DYNAVECTOR

The Dual Mode Dynavector provides the pilot with a fast-response control system actuator with dual power options. Failure of primary electric power does not prevent the engine bleed air system from continuing the actuator function. Failure of the engine bleed air system does not prevent the primary electric power from continuing the actuator function.

Wide variations in delivered electric power or pneumatic power do not prevent the actuator from performing normally in the closed-loop control system. Abnormal power conditions are so easily absorbed by the Dual Mode Dynavector's redundancy that such problems might readily pass unnoticed by the pilot.

Wide variations in the temperature of the engine bleed air do not prevent the actuator from performing normally in a closedloop control application. Temperature swings from $0^{\circ}F$ (-17.8°C) to $350^{\circ}F$ (177°C) do not produce detectable changes in performance of the control system.

Aircraft survivability now has an ideal control system actuator the Dual Mode Dynavector.

1.4 ADVANCED AIRCRAFT APPLICATIONS

Advanced aircraft, such as STOL/VTOL and other high-performance mission fighters require dual-mode actuators which offer the following....

- compact packaging
- easily-changed form factors
- large angular control surface rotation
- high dynamic performance
- normal operation on poorly regulated power sources
- high reliability
- high survivability

The Dual Mode Dynavector meets these needs. For example, a design now under consideration produces high output torque at an outer member output ring. This leaves a 12 inch diameter hole through the actuator which permits thrust vector exhaust gases to exit through the actuator. Deflector vanes can be mounted to the outer member output ring for thrust control in a STOL or VTOL aircraft. The high-temperature environment in such an application can be safely handled by the Dual-Mode Dynavector without danger of fire or explosion in the actuator system. Environmental temperatures as high as $824^{\circ}F$ ($440^{\circ}C$) are feasible for the Dual Mode Dynavector and $1832^{\circ}F$ ($1000^{\circ}C$) temperatures are feasible for the Pneumatic Dynavector.

Another design now being reviewed provides high torques in a small diameter, lightweight unit which would be suitable as a hinge-line control surface actuator.

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1.5 OPERATION

The Dual Mode Dynavector consists of a pneumatic motor and an electric motor housed within a single rotary actuator assembly. The pneumatic motor is designed to operate on 20 PSIG pneumatic power and the electric motor is designed to operate on 270 VDC electric power. The pneumatic motor and the electric motor share the same rotor, and when electric power and pneumatic power are applied concurrently, the output torques produced by each add geometrically. Thus, each motor powered singly develops 1500 IN-LBS of torque at 5 RPM output shaft speed. When full electric and pneumatic power is supplied concurrently, the Dual Mode Dynavector develops 3000 IN-LBS of torque at 5 RPM.

Pneumatic Motor

Figs. 1, 2, & 3 show the essential features of the Dual Mode Dynavector. The electro-pneumatic valve (1) is controlled by a bipolar d.c. electrical current which is typically derived from the control system error signal. The d.c. current drives a lineardisplacement torque motor (2) which in turn strokes the singlestage spool valve (3). The spool valve provides modulated bidirectional flow of pneumatic power from inlet (4), through the pneumatic motor (5), then out the exhaust port (6). Reversal of flow direction by the spool valve through the motor ports (7), (8) reverses the direction of the pneumatic motor torque. Pneumatic force on the rotor (9) is converted by eccentric reaction into a rotary torque.

Commutation slots on the rotor ends act with the manifold plate (11) and the transfer plate (12) to transfer the air flow to adjacent chambers as the rotor orbits. This in turn rotates the pneumatic force vector so that the orbiting reaction of the rotor continues.

Electric Motor

Electric current flows through three adjacent coils (13), (14) (15). The magnetic flux lines crossing the air gap between the stator (16) and rotor (9) result in a radial force vector which attracts the rotor to the stator. This force is converted by eccentric reaction into a rotary torque.

The next electrical step produced by the electric controller deenergizes coil (15) and energizes coil (17) resulting in a force vector rotation, and the orbiting reaction of the rotor continues.

The pneumatic commutation slots of the rotor are designed to coordinate the force vectors of the pneumatic motor and the electric motor so that they act synergistically.

Gear Transmission

The integral epicyclic gear transmission consists of the ring gear (18), the ground gear (19), and the output gear and shaft (20). The input to the transmission from the rotor (9) is transferred through the roller bearings (21). Reduction ratio from rotating motor force vector speed to output shaft speed is 468.4 to 1.

Position Transducers

RVDT (22) and LVDT (23) are rotary and linear position transducers that are used as feedback elements in closed-loop instrumentation circuits.





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2.0 DISCUSSION

2.1 TRANSMISSION DESIGN

Ratio

Several factors must be considered in selecting the reduction ratio of the epicyclic gears. The ratio for a pneumatic Dynavector is usually selected to provide a maximum orbiting speed of about 3000 rpm. Electric Dynavectors generally use a higher ratio. The gear diameter should be as large as the space permits to reduce the gear tooth stresses. For a given gear diameter and diametral pitch only a few ratios can be obtained.

A computer study of gear tooth motion has indicated that 32 diametral pitch and 20° pressure should be used to minimize gear tooth interference. A tooth difference of 3 between the ring gear and pinion can be used to provide 3/64 inch eccentricity.

The eccentricity acts as a lever arm for converting the pole forces to torque. If the eccentricity is very small, very little torque will be developed. Also the epicyclic gearing will not work if the eccentricity is too small. On the other hand, increasing the eccentricity increases the air gap in the magnetic circuit. Electrical power loss becomes excessive when the gap is excessively large due to fringing and increased flux leakage. The number of ampere turns required for a given flux level increases as the gap increases.

Previous experience has indicated that an eccentricity of 3/64 inch is the best choice for electrical performance of a motor this size, and for a good gear design.

The transmission form is shown in Figure 4. The reduction ratio is given by

$$R = \frac{N_2 N_4}{N_1 N_3 - N_2 N_4}$$

where $N_1 = Number$ of teeth in ground gear

0

 N_2 = Number of teeth in ring gear-ground gear mesh

 N_3 = Number of teeth in ring gear-output gear mesh

 N_A = Number of teeth in output gear

The ratios obtainable are listed in Table 1.

A ratio of 468.4:1 was selected. This ratio should provide the specified torque and speed in both modes of operation. The orbiting speed at the rated speed is

 $\frac{deg}{sec} \times \frac{1 \text{ rpm}}{6 \text{ deg/sec}} \times 468 = 2340 \text{ rpm}$

TABLE 1 - GEAR RATIOS

Diametral Pitch = 32 3 Teeth Difference

NI	65	65	64	63	62	61	60	59
N2	63	68	67	66	65	64	63	. 62
N 3	65	64	64	63	62	61	60	59
14	62	-61	61	60	59	58	57	56
ATIO	463.4	345.7	454.1	440.0	426.0	412.4	399.0	385.8



FIGURE 4 TRANSMISSION CONFIGURATION



FIGURE 5 ROTOR FREE BODY DIAGRAM

2.2 STRESS ANALYSIS

1

The forces and moments acting on the rotor are shown in Figure 5. The equations for summing forces and moments are

$\Sigma F_{\mathbf{x}}$:	$F_{mx} - F_{gay} - F_{gay} - F_{px} = 0$	(1)
ΣF _v :	$F_{mv} - F_{gov} - F_{ggv} - F_{pv} = 0$	(2)
ΣM _C :	$AF_{DX} - BF_{GOX} - CF_{GGX} = 0$	(3)
ΣM _p :	$AF_{mx} - (A+B)F_{gox} - (A+C)F_{ggx} = 0$	(4)

The symbols used in this section are defined as follows.

$F_x =$		Net horizontal force acting on the rotor	lbs
$F_y =$		Net vertical force acting on the rotor	1bs
M _c =		Moment about point C, the center of in. the rotor	lbs
M _p =		Moment about point P, where the rotor contacts the pin, as shown in Fig in.	lbs
Fggx	=	Horizontal component of force acting on the rotor at the ground gear mesh point	1bs
F _{ggy}	=	Vertical component of force acting on the rotor at the ground gear mesh point	1bs
F _{gox}	=	Horizontal component of force acting on the rotor at the output gear mesh point	1bs
Fgoy	=	Vertical component of force acting on the rotor at the output gear mesh point	1bs
F _{mx}	=	Horizontal component of electro- magnetic or pressure force acting on the rotor	1bs
F _{my}	=	Vertical component of electro- magnetic or pressure force acting on the rotor	lhs
			100

F _{px}	= Horizontal component of pin force on the rotor	1bs
F _{py}	= Vertical component of pin force on the rotor	1bs
A	= Distance from center of rotor to contact point of pin with rotor	in.
В	= Distance from center of rotor to output gear mesh point	in.
С	= Distance from center of rotor to ground gear mesh point	in.
Т	= Torque in.	1bs
Fgo	= Output gear force	1bs
ⁿ t	= Torque efficiency	
R	= Gear ratio	
е	= Eccentricity	in.
θ _m	= Vector angle of force F _m	deg

 F_m is the force which causes the motor to have an output torque. F_{gg} and F_{go} are the reaction forces of the ground gear and the output gear on the rotor. F_p is the reaction force of the pin on the rotor. In this analysis all of the load is considered to be taken by one pin.

The output gear force for the maximum stall torque of $T_0 = 4400$ in-lbs. is

 $F_{go} = \frac{T_o}{B_{go} \cos 20^{\circ}} = \frac{4400}{.970 \cos 20^{\circ}} = 4827 \text{ lbs.}$ $F_{gox} = F_{go} \cos 20^{\circ} = 4536 \text{ lbs.}$ $F_{goy} = F_{gox} \tan 20^{\circ} = 1651 \text{ lbs.}$

The torque is given by

1

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 $T_o = n_t \text{ Re } F_{mx}$

where n_t = efficiency R = gear ratio e = eccentricity

$$F_{mx} = \frac{T_0}{n_t Re}$$

 $F_{mx} = \frac{4400}{.48 \times 470 \times .047} = 415$ lbs.

 $\mathbf{F}_{my} = \mathbf{F}_{mx} \tan \Theta_{m}$

From equation (4),

 $F_{ggx} = \frac{A}{A+C} \quad F_{mx} - \frac{A+B}{A+C} \quad F_{gox}$ $F_{ggx} = \frac{1.812}{2.8745} \quad x \quad 415 - \frac{2.8276}{2.8745} \quad x \quad 4536$ $F_{ggx} = -4200 \quad 1bs.$ $F_{ggx} = -F_{ggx} \quad tan \quad 20^{\circ} = 1529 \quad 1bs.$

From equation (3),

$$F_{px} = \frac{B}{A} F_{gox} + \frac{C}{A} F_{ggx}$$

$$F_{px} = \frac{1.0156}{1.812} \times 4536 - \frac{1.0625}{1.812} \times 4200$$

 $F_{px} = 80$ lbs.

From equation (2),

$$F_{py} = F_{mx} \tan \Theta_{m} - F_{goy} - F_{ggy}$$

$$F_{py} = 415 \tan \Theta_{m} - 1651 - 1529$$

$$F_{py} = 415 \tan \Theta_{m} - 3180$$

 F_{py} is negative for Θ_m less than 83 degrees. When the sign is negative there is no^m vertical force component on the reaction pin. The vertical force acts instead on the gearing.

Reaction Pin Stress

Since θ_m should never exceed 83 degrees, the pin force is

$$F_{p} = F_{px} = -80$$
 lbs.

The reaction pin stress is given by

$$S_s = \frac{F_p}{A_p}$$
 where $A_p = pin$ cross-section area.

The stress in shear is

$$S_s = \frac{80}{0.0276} = 2,899 \text{ psi.}$$

Yield strength of the reaction pin steel in shear is 65,000 psi.

Steady-State Gear Mesh Stress

The maximum steady-state gear mesh bending stress S_b is found from the Lewis Formula,

$$S_b = \frac{4 T_0}{Ybcd^2}$$

where $T_0 = maximum stall torque, 4400 in-lbs.$

 $S_{b} = \frac{(4) (4400)}{(0.37)(1.13)(0.16)(1.94)^{2}}$ Y = Lewis form factor, 0.37 b = Face width, 1.13 in. c = Percent teeth driving the load, 16%d = Pitch diameter, 1.94 in.

Yield strength of the gear steel is 330,000 psi.

I_o, Inertia from Rotor to Output Gear

The moment of inertia of the actuator mechanism referenced to the output gear is expressed by:

 $I_{o} = \left[\frac{Wmp}{g}(e)^{2} + \frac{I_{BB}}{(R_{r})^{2}}\right] \cdot (RATIO)^{2}$

.

The first term in the bracketed expression represents the kinetic reaction due to the eccentric orbiting motion of the rotor The second term represents the kinetic reaction due to the rotational motion of the gear mechanism. In this equation,

Wmp = weight of rotor, lbs.

 $g = 386 \text{ inches/sec.}^{2}$ $R_{r} = \text{transmission ratio of ring rotation to ground, 22}$ e = eccentricity, 3/64 inch RATIO = overall actuator ratio, 468:1 $I_{BB} = \text{inertia of ring gear, in-1b. sec.}^{2}$ $I_{o} = \left[\frac{Wmp}{g}(e)^{2} + \frac{I_{BB}}{(R_{r})^{2}}\right] \cdot (RATIO)^{2}$

 $I_{o} = \left[\frac{7.7}{386} \left(\frac{3}{64}\right)^{2} + \frac{.0045}{(22)^{2}}\right] \cdot (468)^{2}$ $I_{o} = \left[.0000438 + .0000093\right] \cdot (468)^{2}$ $I_{o} = 11.63 \text{ in.-lb.-sec.}^{2}$

Dynamic Gear Mesh Stress

Maximum gear mesh bending stress occurs when the crank arm impacts a limit stop while slewing at maximum speed. The crank arm, gearing and resisting members absorb the kinetic energy of the Dynavector while stopping through .005 radian of arc of the output shaft. Thus $\theta_{om} = .005$ rad as shown below.

Assume a simple harmonic motion of the output shaft upon stopping.



 V_0 max. = 10 RPM

= 1 rad/sec.

For simple harmonic motion

$$\Theta_{o} = \Theta_{om} \sin \omega t$$

$$V_{o} = \frac{d \Theta_{o}}{dt} = \Theta_{om} \quad \omega \cos \omega t$$

Maximum velocity occurs when $\cos \omega t = 1$; this occurs at t =0 and,

 $V_{o max} = \Theta_{om} \omega$ (6)

22

(5)

Acceleration, $A_0 = \frac{d^2 \theta_0}{dt^2} = -\theta_{om} \omega^2 \sin \omega t$

Maximum acceleration occurs when sin $\omega t = 1$; this occurs at $\omega t = \pi/2$ and,

$$A_{o} max = - \theta_{om} \omega^{2}$$
 (7)

from (6), $V_{o \text{ max.}} = \Theta_{om} \omega$,

at 10 RPM, $V_{o \text{ max.}} = 1 \text{ rad/sec.}$, 1 rad/sec = .005 radian x ω $\omega = 200 \text{ sec.}^{-1}$

from (7)

0

0

$$A_{o} \max = - \left[.005 \text{ radian } (200 \text{ rad/sec.})^2 \right]$$
$$A_{o} \max = - \left[200 \text{ rad/sec.}^2 \right]$$

The worst-case loading on the gear mesh upon stopping is equal to the maximum steady-state torque plus the maximum dynamic torque at the output shaft.

> $T'_{o} = T_{o max} + I_{o} A_{o max}.$ where $T'_{o} = (stall + energy torque), max.$ $T'_{o} = 4400 + 12 (200)$ $T'_{o} = 6800 \text{ in-lbs}.$ $T_{o} = stall torque, in-lbs.$

- $A_{o} = \max.$ output deceleration, rad/sec²
- I_o = inertia of Dynavector ref. to output gear, in-lb-sec²

The worst-case dynamic gear mesh stress based on the Lewis Formula is

$$S'_{b} = \frac{4 T_{0}}{ybcd^{2}}$$

$$S'_{b} = \frac{(4)(6800)}{(0.37)(1.13)(0.16)(1.94)^{2}}$$

$$S'_{b} = 108,000 \text{ psi}$$

Yield strength of the gear maraging steel is 330,000 psi.

Dynamic Crank Arm Stress

The crank arm impacts the limit stop at a point 4 inches from the output shaft centerline. The bending force

 $P_{max} = \frac{6800 \text{ in-lbs.}}{4 \text{ inches}} = 1700 \text{ lbs.}$

The maximum deflection of the crank arm is

$$\Delta = \frac{1}{3} \times \frac{P_{max} L^{3}}{EI}$$
(8) where L = 4 inches

$$\Delta = \frac{1}{3} \times \frac{1700 (4)^{3}}{3 \times 10^{7} \times .06}$$
moment of inertia,

$$\Delta = .02 \text{ inch}$$
I = .0833 bh³
I = .0833(7.5)(1)³
I = .06 in⁴

The maximum stress in the crank arm is

$$S_{b} = \frac{Mc}{I}$$
$$S_{b} = \frac{6800 \times .5}{.06}$$

 $S_{b} = 57,000 \text{ psi}$

Yield strength of the crank-arm steel is 110,000 psi.

θ_{om} Calculation

1

From equation (8) above, the maximum deflection of the crank arm upon stopping is Δ = .02 inch.

$$\Theta_{\text{om}} = \frac{\Delta}{\text{circumf.}} \times 2\pi \text{ rad}$$

$$\Theta_{\text{om}} = \frac{.02 \text{ inch}}{2\pi (4 \text{ inches})} \times 2\pi \text{ rad} = .005 \text{ rad}$$

$$\Theta_{\text{om}} = .005 \text{ radian.}$$

Ring Gear Stress and Deflection

The ring gear shear stress and deflection are given by

$$J = \frac{\pi}{32} \left(d_{1}^{4} - d_{0}^{4} \right)$$

$$S_{s} = \frac{Td_{i}}{2J}$$

$$\phi = \frac{TL}{JG} \times \frac{180}{\pi}$$
where d_{i} = inside diam., in.
 d^{i} = outside diam., in.
 G^{0} = modulus of elasticity in shear, psi
 J = polar moment of inertia, in.⁴
 L = length, in.

$$J = \frac{\pi}{32} \left(2.5^{4} - 2.15^{4} \right) = 1.74 \text{ in.}^{4}$$

$$S_{s} = \frac{4400 \times 2.15}{2 \times 1.74} = 2718 \text{ psi}$$

$$\phi = \frac{4400 \times 2.625}{1.74 \times 10.5 \times 10^{6}} \times \frac{180}{\pi} = .036 \text{ deg.}$$

Bearings

The actuator has two roller bearings supporting the output shaft with a load of 2420 lbs. These bearings rotate at the low speed of the output shaft. Torrington bearing HJ-142216 was selected for this application. The dynamic load rating for this bearing is 4860 lbs.

Two roller bearings separate the ring gear and rotor. These bearings support a maximum load of 2860 lbs. each and rotate at the output shaft speed. The bearing selected for this application, Torrington WJ-40416, has a 9110 lb. dynamic load rating.

The bearings selected have a safety factor of 2 or 3 and should have a long life at the low speeds of this application.

2.3 ELECTRIC MODE

The computer program described in Appendix B was used to compute torque-speed curves for the electric mode. The correlation of this computer program with test data for the Bendix Model EH-441-Ul motor is shown in Figure 6.

Stator Design

A stator length of 2.6 inches was selected for the Dual Mode Actuator. It is theoretically possible to develop the required torque with a shorter stator. However, more electrical power would be required. Conversely, if the stator is too long, the inductance of the coils will be unacceptably high and the speed of the electric motor will be reduced. The 2.6 inch length appears to be the best compromise between inductance and power input.

Stator Voltage Waveform

The motor excitation voltage waveform used for this design is shown in Figure 7. A high-voltage pulse is used for fast current buildup. To develop maximum torque with the least possible current, the high-voltage pulse should be of duration to just cause magnetic saturation at low speed. A computer simulation indicates that this time is about 7 milliseconds. The voltage is then reduced to the value that just maintains saturation, then reduced linearly with time to a value of about





5 volts at 180 degrees (3.14 radians) vector rotation. This voltage waveform maintains a constant flux density for maximum efficiency.

Torque and current at 30 deg/sec vs gear ratio are shown in Figure 8 for 48% and 64% torque efficiency. An efficiency of 64% was obtained with previous Electric Dynavectors, and 48% is the expected efficiency for the pneumatic mode. Torque increases and current decreases as the ratio increases. The highest ratio considered was 470, because higher ratios could reduce performance in the pneumatic mode.

A reduction ratio of 470:1 gives an orbiting speed of 4008 rpm at an output rate of 55 deg/sec. Previous experience has shown that the pressure loading on the rotor will result in reduced bearing life if the maximum orbiting speed is much in excess of 4000 rpm. The loading is different in the electrical mode and higher speeds could be used for electrical operation only.

Figure 9 shows the effect of the number of turns per coil. Both the torque and current decrease as the number of turns is increased. However, the current decreases more than the torque. Therefore, increasing the number of turns should improve efficiency.

A ratio of 468:1 and 450 turns per coil were selected. The torque-speed curve with these parameters, and 48% efficiency, is shown in Figure 10.

The space available for the coil wires is 0.188 sq in. The wire size selected, AWG 25, has the following characteristics:

dia = 0.0179 inch

circular mils = 320

turns per square inch = 2475

The space needed is

450 turns/2475 = 0.182 sq. in.

Therefore, the area available is adequate.

At the specified maximum torque, 2200 in lbs., the current is 14 amps.

Amps per coil = 14/8 = 1.75 amps/coil

320/1.75 = 183 circular mils/amp

At the rated torque, 1500 in lbs., the current is 10 amps, or 1.25 amps per coil.

320/1.25 = 256 circular mils/amp



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FIGURE 8 TORQUE AND CURRENT VS RATIO

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FIGURE 9 TORQUE AND CURRENT VS NUMBER OF TURNS


Previous Electric Dynavectors have operated continuously with 200 circular mils/amp. Therefore, heating of the Dual Mode Actuator should not be a problem.

A Study of the Electric Mode Dynavector with 4 poles energized in the holding phase, as in Figure 11, shows an unstable equilibrium point (slope is negative). The rotor will stabilize at either the pole on the left or the pole on the right of geometric center (depending on load disturbance torques). This results in a hysteresis of 45 degrees of orbiting motion or

 $\frac{45}{468.4}$ = .096 deg. of output shaft motion

Such hysteresis could provide the basis for a limit-cycle oscillation and is therefore unacceptable.

Figure 12 shows the torque vs. rotation with 3 poles energized in the holding phase. The geometric center is a stable equilibrium point. Therefore, the rotor will lock onto the geometric center and hysteresis will be eliminated.

Rotor Design

The rotor design utilizes a solid (instead of laminated) steel form that can be case-hardened to support the roller bearing and reaction pin stresses. The solid steel rotor design costs less to manufacture than the laminated steel rotor design since the inserts and liners of a laminated structure are not required.

There is some decrease in efficiency due to the increased eddy current flow in the solid steel. However, this increase in eddy current flow is minimized because

- a) high resistivity silicon steel resists current flow,
- b) eddy current flow occurs only during servo action,
- c) crowding of magnetic flux near the surface of the rotor results in low flux density at the center of the rotor.
- d) laminated structure of stator restricts eddy current flow.

Rotor Power Loss

The flux density cycles from 0 to 15,000 gauss with amplitude of 7,500 gauss. The core loss of M-19 silicon iron 0.025 inch thick at this amplitude and at 60 Hertz is 0.4 watts per pound. This loss is approximately 1/3 eddy current loss and 2/3 hysteresis loss. Eddy current power loss is proportional to frequency squared; hysteresis power loss is proportional to frequency. The rotor frequency at rated speed is 39.2 Hertz. The power loss for a laminated rotor is

$$Pr = D L w \left[\frac{1}{3} \left(\frac{n}{\bar{N}}\right)^2 + \frac{2}{3}\left(\frac{n}{\bar{N}}\right)\right]$$

where Pr = power loss, watts

D = servo action duty cycle L = power loss per pound at 60 Hertz, watts w = weight of rotor, lbs n = rotor frequency, Hertz N = 60 Hertz Pr = 0.15x0.4x8.29 $\left[\frac{1}{3}\left(\frac{39.2}{60}\right)^2 + \frac{2}{3}\left(\frac{39.2}{60}\right)\right]$

Pr = .29 watts

The eddy current loss for a solid rotor theoretically varies as the square of the thickness. However, the rotor is not an ideal flux generator, and the losses will be less than for an ideal generator. The rotors of previous Dynavectors have demonstrated an efficiency as an eddy current generator of 20%. Thus, the eddy current power loss for a solid rotor is

$$P_{e} = \left(\frac{\text{mean rotor thick}}{\text{lamination thick}}\right)^{2} E D L w \left[\frac{1}{3} \left(\frac{n}{N}\right)^{2}\right]$$

where E = eddy current efficiency, 20%

$$P_{e} = \left(\frac{.875}{.025}\right)^{2} \times 0.20 \times 0.15 \times 0.4 \times 8.29 \left[\frac{1}{3} \left(\frac{.39.2}{.60}\right)^{2}\right]$$

 $P_o = 17.3$ watts

The hysteresis power loss for the solid rotor will be the same as for a laminated rotor, about 0.225 watt.



FIGURE 11 TORQUE VS ROTATION - 4 POLES EXCITED



Electric Mode Stiffness

The force developed by a pole has been determined by test to be given by

 $F = 3.75 \times 10^{-7} A_p B_a^2$ lbs.

where

$$B_a = air gap/flux density, Gauss$$

 $A_p = pole face area, in.^2$

The back torque T_s is given by

 $T_{s} = \frac{eR}{n_{t}} \Sigma F \sin \theta$ where

 $n_t = mech.$ efficiency R = transmission ratio

The results of calculations for 3-pole energization and 4-pole energization are shown in Figure 13. By definition,

Stiffness,
$$K_s = \frac{\Delta T}{\Delta \Theta/R} \times \frac{180}{\pi}$$

where

 ΔT = back torque, in.-lbs. $\Delta \Theta$ = orbit displacement, deg. R = transmission ratio



For 4-pole energization,

Stiffness, $K_{s4} = \frac{4529}{15/468} \times \frac{180}{\pi}$ $K_{s4} = 8 \times 10^6 \text{ in.-lbs/radian}$

For 3-pole energization,

Stiffness,
$$K_{s3} = \frac{3600}{15/468} \times \frac{180}{\pi}$$

 $K_{s3} = 6.4 \times 10^6$ in.-lbs./radian

2.4 PNEUMATIC MODE

The torque-speed curves for the pneumatic mode were computed with the computer model described in Appendix A. Pressure loss in the motor passages is not included in the computer model. The passages were sized so that these losses would not be significant.

Computer Model Input Data

The pneumatic mode torque is given by

$$T = \eta_{+} RD_{m} \Delta P$$

where

 $n_t = torque efficiency, 0.48$ $D_m = displacement, in³/rad$

 ΔP = differential pressure, psi

R = ratio, 468

The displacement is

 $D_m = D_v eb$

where

b = rotor face width, in

 D_{v} = vane chamber diameter

e = eccentricity

Inserting numerical values

 $D_{\rm m} = 4.953 \times 0.0468 \times 3.1888 = 0.740 \text{ in}^3/\text{rad}$

The differential pressure at the rated torque is

$$\Delta P = \frac{T}{n_{+}RD_{m}} = \frac{1500}{0.48 \times 470 \times 0.74} = 9 \text{ psi}$$

The principal case drain leakage path is from the commutator slot ends to the rotor inside diameter. Thus, the case drain leakage area equals the commutator slot width times the end clearance times the sum of the number of commutator feed slots and one-half the number of commutator slots.

 $A_{cd} = 0.125 \times 0.0015 \times 24 = 0.0045 \text{ in.}^2$

There are two commutator slots for each chamber. The total commutator flow area for the motor inlet flow is equal to two times the maximum area for one chamber. Thus, the effective commutator area is

 $A_c = 4 x$ slot length x eccentricity

 $= 4 \times 0.438 \times 0.0468 = 0.082 \text{ in.}^2$

There are two cross port leakage paths. The first is from the high pressure commutator slots to the low pressure slots. The flow area equals the slot length times the end clearance times the number of slots at upstream pressure. Thus

 $A_{cpl} = 0.875 \times 0.0015 \times 16 = 0.021 \text{ in.}^2$

The second cross port leakage path is across the ends of the vanes. This leakage area is

 $A_{cp2} = 2 x$ vane height x end clearance = 2 x 0.391 x 0.0015 - 0.0012 in.²

The total cross port leakage area is then

 $A_{cp} = A_{cp1} + A_{cp2} = 0.021 + 0.0012 = 0.0222 \text{ in.}^2$

Computer Results

The pneumatic mode torque-speed curves for several value areas at 0 deg. F are shown in Figure 14. A value area of 0.072 sq. in. provides the specified torque and speed. The effect of gas temperature is shown in Figure 15. At 140° F there is about a 12% increase in speed.

The effect of reducing the end clearance to 0.001 inch is shown in Figure 16. With the reduced clearance, the valve area can be reduced to 0.060 sq. in.

The motor torque will be a little lower than shown in these curves due to pressure drop in the motor passages. This pressure drop is shown in the next section to be about 2 psi.

Figures 17 and 18 show the effect of reduced supply pressure. These curves can be used to estimate the effect of line pressure drop. A 2 psi line pressure drop is approximately equivalent to lowering the supply pressure 2 psi.

Figure 19 shows the effect of of varying the commutator flow area. It appears that reducing the commutator flow area 50%, to 0.082 sq. in., will result in more torque at 30 deg/sec., due to the decreased leakage area. This would also reduce the total flow, and thus would reduce the pressure drop in the passages.









FIGURE 16 EFFECT OF REDUCED END CLEARANCE













2.4.1 Pressure Drop in Passages

The computer model for the torque-speed curves does not include pressure drop in the motor passages. Due to the low supply pressure, the motor passages must be sufficiently large to cause little pressure drop.

The pressure drop in a passage is given by

$$AP = f \frac{L}{d} - \frac{1}{2g\rho} \left(\frac{W}{A}\right)^2$$

where

A = flow area, sq in d = diameter, in f = friction factor g = acceleration of gravity, 386 in/sec^2 L = passage length, in

The air density is given by

ρ

$$= \frac{P}{RT} \quad \frac{1b}{3}$$
in

where

P = pressure, psi R = gas constant, 640 in/ O R T = air temperature, O R

The procedure is to assume a value for f, calculate the flow and Reynolds number N_R , and then find the correct value of f from curves of f vs N_R .

From the computer results, at rated speed and torque:

Air flow = 0.01946 lb/sec PV1 = 32.67 psi, ρ = 0.0000963 lb/in³ PM1 = 31.03 psi, ρ = 0.0000915 lb/in³ PM2 = 19.69 psi, ρ = 0.0000580 lb/in³ PV2 = 17.75 psi, ρ = 0.0000523 lb/in³

The calculated pressure drops are

PASSAGE	DIAMETER in	LENGTH in	∆p psi	VOLUME cu-in
Inlet	0.312	1.000	0.07	0.0764
Discharge	0.312	1.000	0.13	0.0764
Bypass-inlet	0.250	3.812	0.21	0.2383
Bypass-discharge	0.250	3.812	0.40	0.2383
Commutator manifold	0.312	4.950	0.10	0.4822
inlet	0.312	11.000	0.19	1.0700
Sum			1.10	2.1816

There is an additional pressure drop due to direction change of the flow. This pressure drop is given by

$$\Delta P = \frac{0.4\rho}{2g} \frac{V^2}{2g} = \frac{0.2}{\pi^2 \rho g} \frac{W^2}{4} = \frac{0.2}{d}$$

n = number of 90 degree bends

There are approximately ten 90 degree bends. Therefore, the pressure loss is

 $\Delta P = \frac{0.2}{\pi^2 (0.52 \times 10^{-4}) (386)} \frac{(0.01946)^2}{(0.312)^4} \times 10 = 0.40 \text{ psi}$ Total pressure drop = 1.10 + 0.40 = 1.5 psi Percent of motor differential pressure $= \frac{1.5}{12.98} \times 100 = 11.6\%$

The volume under compression is one-half the motor clearance volume plus one-half the passage volume. Thus

One-half	passage volume	=	1.09	sq.	in.
One-half	motor clearance volume	=	1.17	sq.	in.

Volume under compression = 2.26 sq. in.

2.4.2 Servovalve

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A single state electro-pneumatic servovalve was developed for this program.

Figure 20 shows the assembly drawing for this valve. The frequency response is 25 Hz at 90 deg. phase shift. Rated torque motor power is 2 watts.

The torque motor provides a \pm 0.015 in. stroke. The spool diameter of 1.75 inches generates a maximum flow area,

 $A_{y} = \pi Dx = \pi (1.75)(.015) = 0.0825 \text{ in.}^{2}$



If the flow of 70° F air is sonic-limited by the valve,

 $W = .53 \text{ AcP}_1$

1

VT_R

where

$V = \frac{.53 \times .0825 \times .75 \times 35}{\sqrt{460 + 70}}$	W = flow, lb./sec. A = flow area, in.2
$V = .05 \text{lb./sec.}, 70^{\circ} \text{F air}$	C = discharge coeff. $P_1 = upstream pressure, psia$
V = .053 lb./sec., 0 ⁰ F air	T_R^{-} temperature of air, R
$I = .047 \text{lb./sec., } 140^{\circ} \text{F air}$	

Thus, the valve control range adequately covers the computerrated air flow requirement of W = .01946 lb/sec., 70° F air.

One noteworthy feature of this valve design is that the spool is connected to a drive rod which is guided by ball bushings. This suspension method effectively eliminates many of the problems that are associated with stiction and friction in pneumatic spool valves.

An LVDT transducer, Schaevity 025 MHR, provides position feedback information for the torque motor-spool position loop. This inner servo loop acts to linearize and smooth the control action of the servo valve assembly.

3.0 RESPONSE

3.1 PNEUMATIC MODE

The block diagram for pneumatic mode response is shown in Figure 21.

The open loop actuator transfer function, assuming the load spring rate to be negligible, is

$$\frac{\theta_{\mathbf{r}}}{T_{\mathbf{c}}} = \frac{1}{\partial T/\partial n} \qquad \frac{1}{S(1 + \frac{2\zeta}{\omega_{n}}S + \frac{1}{\omega_{n}^{2}}S^{2})}$$
(9)

where

$$\omega_{n}^{2} = \frac{1}{\tau J} \quad \frac{\partial T}{\partial n}$$
$$\zeta = \frac{1}{2} \quad \omega_{n} \frac{J}{\partial T / \partial n}$$

1

J = Combined inertia of motor and load, in 1b sec² $\frac{\partial T}{\partial n}$ = Slope of torque-speed curve

The compressiblility time constant is given by

$$r = \frac{v_{\partial n}^{\partial T}}{kP_{O}(RD_{in})^{2}}$$

$$D_{m} = Actuator displacement, in ^{3}/rad$$

$$k = Specific heat ratio, 1.4 for air$$

$$P_{O} = Nominal pressure, psia$$

$$R = Transmission ratio$$

$$V = Volume under compression, in^{3}$$



The slope of the torque-speed curve, from Figure 22, is

$$\frac{\partial T}{\partial n} = 3255$$

Substituting numerical values gives

$$\tau = \frac{(2.262)(3255)}{(1.4)(24.7)(0.74 \times 470)^2} = 0.00176 \text{ sec.}$$
$$\omega_n = \sqrt{\frac{3255}{0.00176 \times 20.47}} = 300 \text{ rad/sec.} = 47.75 \text{ Hz.}$$
$$\zeta = \frac{(0.5)(300)(20.47)}{3255} = 0.94$$

The servovalve has a second order transfer function with a natural frequency of 25 Hz. and a damping ratio of 0.7.

The open-loop and closed-loop frequency response of the actuator with valve are shown in Figure 23.

The open-loop gain is given by

$$K_0 = K_a \frac{\partial T/\partial Av}{\partial T/\partial n}$$

From the computer study, $\partial T/\partial Av = 71,600 \text{ in.-lb./in}^2$. K equals the open loop amplitude at 1.0 rad/sec. frequency, which is 46.77. Solving for the amplifier gain,

$$K_{a} = 46.77 \frac{\partial T/\partial n}{\partial T/\partial Av} = 46.77 \frac{3255}{71,600}$$
$$K_{a} = 2.13 \frac{in^{2}}{rad}$$

The error signal for full valve opening is

$$\mathcal{E} = \frac{Av}{K_{a}} = \frac{0.072}{2.13} \times \frac{180}{\pi} = 1.94 \text{ deg.}$$



FIGURE 22 PNEUMATIC MODE TORQUE-SPEED CURVE





Pneumatic Mode Stiffness

The pneumatic mode stiffness varies with amplitude. The small signal stiffness is

$$K_{s} = K_{a} \frac{\partial T}{\partial A} = 2.13 \times 71,600$$

 $K_{a} = 1.525 \times 10^{5} \text{ in.-lb./rad}$

The stiffness at maximum amplitude equals the peak torque divided by the error to saturate.

$$K'_{s} = \frac{2200}{1.94} = 6.5 \times 10^{4} \text{ in.-lb./rad}$$

Electric Mode Stiffness and Synchronization

The electric mode stiffness with 3 poles energized and magnetically saturated is

$$K_{s3} = 6.4 \times 10^6 \text{ in-lb/rad}$$

as shown in Section 2.3. Thus, the electric mode steppingmotor stiffness is 40 to 100 times greater than the pneumatic mode motor stiffness. Consequently the electric motor forces will dominate the rotor movement and the pneumatic commutation plates that are part of the rotor will then automatically synchronize the pneumatic motor force vector to the electric motor force vector.

Pneumatic Mode Torque Saturation

The amplitude at which pneumatic torque saturation will occur is determined from the open loop transfer function (9). Thus

$$\theta = \frac{2Ts}{\partial T/\partial n} \qquad \frac{1}{Re + Im} \frac{1}{1/2} \qquad \cdot \frac{180}{\pi} \deg$$

where

Re =
$$\frac{-2\zeta}{\omega_n} \omega$$
 = $\frac{-2 \times .94}{300} \omega^2$ = $-0.00626\omega^2$

$$Im = \frac{\omega - \omega^3}{\omega_n^2} = \omega - \frac{\omega^3}{90000}$$

Figure 24 shows the amplitude at which torque saturation occurs vs frequency. For frequency response testing to 20 hertz, the input amplitude should not exceed 0.5 degree peak to peak.

3.2 ELECTRIC MODE RESPONSE

The linear block diagram for response in the electric mode is shown in Figure 25. Figure 26 shows a nonlinear block diagram. The commanded step rate ω_c , which is actually the pulse frequency, is proportional to the error signal ε . Each pulse provides full excitation to the motor coils. The torque developed is proportional to the torque angle θ_t , which is the phase angle between the commanded angle θ_c and the rudder angle θ_r . $\partial T/\partial n$, the slope of the torque-speed curve, accounts for the decrease in torque with increasing speed due to the back emf. The controller makes the effect of inductance negligible.

The actuator open loop transfer function, with the load spring rate assumed negligible, is

$$\frac{\omega \mathbf{r}}{\omega_{\rm c}} = \frac{1}{1 + \frac{25}{\omega_{\rm n}} \,\mathrm{s} + \frac{1}{\omega_{\rm n}^2} \,\mathrm{s}^2}$$

where

$$\omega_{n}^{2} = \frac{R}{J} \frac{\partial T}{\partial \Theta_{t}}$$

$$\zeta = \frac{1}{2} \omega_{n} \frac{1}{R} \frac{\partial T/\partial n}{\partial T/\partial \Theta_{t}}$$

J = Combined inertia of motor and load, in lb sec²

R = Gear ratio

The actuator stall torque equals the maximum stall torque times the sine of the torque angle.

$$T = T_s \sin \theta_t$$





FIGURE 25 LINEAR BLOCK DIAGRAM – ELECTRIC MODE



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FIGURE 26 NON-LINEAR ELECTRIC MODE BLOCK DIAGRAM

Thus $\frac{\partial T}{\partial \theta_t} = T_s \cos \theta_t$ = $T_s = 2200$ in lbs $\theta_t = 0$

The slope of the torque-speed curve approximately equals the stall torque divided by the no load speed.

$$\frac{\partial T}{\partial n} = \frac{2200 \text{ in lbs}}{55 \text{ deg/sec}} \times \frac{180}{\pi} = 2291 \text{ in lb sec}$$

The natural frequency and damping are then

$$\omega_{n} = \sqrt{\frac{470 \times 2200}{20.47}} = 225 \frac{\text{rad}}{\text{sec}} = 35.8 \text{ Hz}.$$

$$\zeta = 0.5 \times \frac{225}{470} \times \frac{2291}{2200} = 0.249$$

The low damping indicates it may be desirable to use compensation. Figure 27 shows the open loop frequency response, the open loop frequency response with compensation added, and the closed loop frequency response. The compensation used was

$$\frac{1 + 0.0025 \text{ s}}{1 + 0.0054 \text{ s}}$$

Dynavector actuators generally have higher damping than indicated by analysis, due to friction and eddy currents. Therefore, the compensation may not be required.

NADC 77001-60



4.0 MATERIALS

4.1 MARAGING STEEL TYPE 18 NI 350

The <u>ring gear</u> and <u>output gear</u> will be fabricated from a high strength, gall-and wear-resistant maraging steel. A process specification for this steel is shown below. Yield strength for this steel in tension/compression is 330,000 psi.

Process Specification, Maraging Steel

General

This process specification is intended for Maraging Steel Type 18 Ni 350 grade (or similar grade) billet, bar or sheet in the as received condition and includes the machining sequences necessary to provide finished parts within drawing tolerances.

Procedure

- If the billet, bar or sheet is not in the solution annealed condition, solution anneal at 1650 degree Fahrenheit for one (1) hour and air cool to room temperature. Then solution anneal for one (1) hour at 1450 degrees Fahrenheit and cool. Hardness should be Rockwell 'C' 30-35.
- 2) Machine the billet, bar or sheet forging to finish dimensions, leaving only sufficient material to lap or dust grind the part. Allowance must be made for shrinkage of the part during the nitriding-aging process. Allow 0.001 in/in for shrinkage during aging.
- 3) Ultrasonically degrease in a trichlorethylene solution, followed by a deionized water rinse. Prepare surface with a 200 grain aluminum oxide blast, using air as the medium to a light matte finish. Surface must not be contaminated during this operation.
- 4) Nitride at 875-880 degrees Fahrenheit for forty-eight (48) hours in 25-30 percent dissociated ammonia.

5) Harness to be:

Case Rockwell "C' 66-67 Superficial Rockwell '15N' 92-93 Core Rockwell 'C' 55 MIN

6) Dust grind or lap to finish dimensions.

7) A test specimen of $\frac{1}{2}$ inch diameter bar one and one-half $(l\frac{1}{2})$ inches long centerless ground and copper plated for half the length to a thickness of 0.001/0.0015 inches shall accompany the parts to determine hardness and case characteristics.

8) Similar grade of steel: Teledyne VASCOMAX 350.

4.2 SILICON STEEL, $2\frac{1}{2}$ SILICON COMPOSITION

The <u>rotor</u> will be fabricated from $2\frac{1}{2}$ % silicon steel for optimum magnetic properties, then case-hardened and nickel-plated for wear and corrosion resistance.

Process Specification, Silicon Steel

General

This process specification is intended for an Electrical Steel with a nominal composition of 2.5% Silicon. The material may be received as a billet, bar, or sheet. This specification describes the heat treat and machining sequences necessary to provide finished parts within drawing tolerances.

Procedure

- 1) Machine the billet, bar or sheet forging to finish dimensions, leaving only sufficient material to lap or dust grind the part. Allowance must be made for .0003 inch nickel plate.
- 2) Ultrasonically degrease in a trichlorethylene solution, followed by a deionized water rinse. Prepare surface with a grain aluminum oxide blast, using air as the medium to a light matte finish. Surface must not be contaminated during this operation.

- 3) Carburize part in accordance with 1919225(PS) .018 to .020 inch thick.
- 4) Hardness, Case: Rockwell C60-64
- 5) Dust grind or lap to finish dimension allowing for .0003 inch nickel plate and final dust grind or lap.
- Electroless Nickel Plate all surfaces .0003 to .0005 inch per 1919253(PS).
- 7) Final dust grind or lap to finish dimensions.
- 8) A test specimen shall accompany parts to determine hardness and plating characteristics.
- 9) Suitable grades of steel for this application are:

Allegheny Ludlum "Relay No. 5" Carpenter "Silicon Core Iron B".

4.3 AISI M-19 TRANSFORMER C STEEL SHEET LAMINATE

The 8-pole stator will be fabricated from stamped 0.014 inch thick M-19 steel sheet stock. This steel is pre-coated with an electrical insulation.

Process Specification, Stator

General

This process specification describes the fabrication and assembly of M-19 Transformer C silicon steel laminates into an 8-pole stator. The material is received as stamped, insulation coated 0.014 inch thick laminates. This specification describes the assembly and machining sequences necessary to provide a finished part within drawing tolerances.

Procedure

 Eight-pole stator laminations are stamped from C-15 coated M-19 Transformer C silicon steel stock, 0.014 inch thick, then annealed at 1550 to 1650^oF for one hour.

- 2) Stack laminations in assembly jig to drawing dimensions and clamp tightly.
- 3) Vacuum bond lamination stack using Chrysler Chemical Division Cycleweld 55-9.
- 4) Grind inside diameter of assembled stator to pre-finish dimensions.
- 5) Assemble bobbin-wound coils onto stator poles and run hookup wires in accordance with assembly drawings.
- 6) Preheat stator assembly for one hour at 110°C.

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- Impregnate coils and pole structure with Dow Corning Sylgard 182, making sure all voids are filled flush. Cure at 110°C for 2 hours.
- 8) After air cooling to room temperature, finish grind inside diameter to drawing dimension.
- 9) Suitable grade of steel for this application is:

Allegheny Ludlum "A-L M-19 Transformer C".
5.0 RELIABILITY

A reliability prediction was performed on the Dual Mode Dynavector Actuator in accordance with MIL-STD-756A to provide a computed reliability estimate of the equipment Mean-Time-Between Failures (MTBF). The analysis was performed, utilizing an airborne uninhabited environment and two conditions of altitude:

CONDITION I - Component Ambient Temperature of $71^{\circ}C$ (160°F)

(45,000 ft. altitude operation)

CONDITION II - Component Ambient Temperature of 121°C (250°F)

(30,000 ft. altitude operation)

All piece parts were considered to be part of a series configuration. The total equipment failure rate and resultant MTBF were, therefore, computed by adding the individual piece-part failure rates.

Attached as Tables 2 and 3 are tabulations of the individual piece-parts, quantity used, calculated failure rate and the data sources. Failure rates generated for the individual items are based on standard data sources and Bendix experience, and represent a fifty percent (50%) confidence level. A summary of the reliability estimate and design goal is indicated below:

CONDITION	UNIT	PART AMBIENT TEMP.	TOTAL FAILURE RATE	PREDICTED MEAN-TIME BETWEEN FAILURES (MTBF)	DESIGN GOAL MTBF
I (45,000 ft.) Alt.	Dual Mode Electro- Pneumatic Actuator	71 ⁰ C (160 ⁰ F)	40.6899 ppmh	24,576 Hours	20,000 Hours
II (30,000 ft.) Alt.		121 ⁰ C (250 ⁰ F)	85.0959 ppmh	11,751 Hours	2,500 Hours

TABLE 2.

RELIABILITY FAILURE RATE PREDICTION FOR THE DUAL MODE DYNAVECTOR ACTUATOR - CONDITION I

 $\begin{array}{r} \text{TEMPERATURE} \quad - \begin{array}{c} 71 \\ 0 \\ (160 \\ F \end{array}) \end{array}$

ENVIRONMENT-AIRBORNE UNINHABITED

Dart Tune	0+1	Failure Rate	Total Failure Rate	Failure Rate
Fait Type	gey.	x(ppini)	<u>(//. g(y)</u>	Data bource
Roller Bearings	4	. 284	1.136	1A
Torquer Motor, Brushless, DC	1	10.4575	10.4575	1B
Motor, Stepper, Brushless (pneumatics included)	1	8.68	8.68	2A
Gear Passes, Spur, Epicyclic	3	. 0202	.0606	5
LVDT (Solenoid Type)	1	14.22	14.22	2B
RVDT (Synchro Brushless)	1	2.358	2.358	2C
Valve, Spool	1	2.89	2.89	3
Needle Bearings	2	.284	. 568	1A
Connector (12 & 16 Pin)	2	.1587	.3174	4
Terminal Board	1	·0012	·0012	2D
Connections, Solder	10	.00012	.0012	2 E

Total Failure Rate - 40.6899 ppmh Mean-Time-Between-Failures (MTBF) - 24,576 Hours

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TABLE 3

RELIABILITY FAILURE RATE PREDICTION FOR THE DUAL MODE DYNAVECTOR ACTUATOR - CONDITION II

TEMPERATURE	- 121 ⁰ C (250 ⁰ F)	ENV	IRONMENT-AIRI	SORNE UNINHABITED
Dent Tuno	0.1.1	Failure Rate	Failure Rate	*Failure Rate
Part Type	QLY.	x(ppmn)	(x, Qty)	Data Source
Roller Bearings	4	.284	1.136	1A
Torquer Motor, Brushless, DC	1	10.4575	10.4575	18
Motor, Stepper, Brushless (pneumatics included)	1	8.68	8.68	2A
Gear Passes, Spur, Epicyclic	3	.0202	.0606	5
LVDT (Solenoid Type)	1	55.80	55.80	2B
RVDT (Synchro Brushless)	1	5.184	5.184	2C
Valve, Spool	1	2.89	2.89	3
Needle Bearings	2	.284	.568	1A
Connector (12 & 16 Pin)	2	.1587	.3174	4
Terminal Board	1	.0012	.0012	2D
Connections, Solder	10	.00012	.0012	2E

Total Failure Rate - 85.0959 ppmh

Mean-Time-Between-Failures (MTBF) - 11,751 Hours

* Connectors remain at 71°C.

*FAILURE RATE SOURCES

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CODE	FAILURE RATE SOURCE
1	RADC Non-Electrical Reliability Notebook (RADC-TR-75-22), dated Jan. 1975
	A) Pg. 2-40 B) Pg. 2-171
2	MIL-HDBK-217B, dated Sept. 7, 1976
	<pre>A) Pg. 2.8.1-10 B) Pg. 2.9-1 C) Pg. 2.8.3-1 D) Pg. 2.12 E) Pg. 2.13-2 F) Pg. 2.11-1</pre>
3	AVCO Reliability Handbook, AVCO Lycoming Corp.
4	MIL-HDBK-2178, dated April 1970
	Pg. 3-73 through 3-81
5	Bendix-Operational Reliability Failure Rate Report, RE-6886

6.0 MAINTAINABILITY

A maintainability analysis has been performed on the Dual Mode Dynavector in order to determine a design estimate in terms of Mean-Maintenance Man hours per Flight Hour (MMH/FH) and Mean-Time-To-Repair (MTTR). A description of the analysis performed and a breakdown of the task times to perform organizational level maintenance and item replacement at the Intermediate level are presented in Tables 4 and 5. A summary of the analysis results and design goals is indicated below.

Desig	n Goal		Predi	cted
Maintenance Level	MTTR	MMH/FH	MTTR	MMH/FH
Condition I:				
Organizational	-	-	.62 Hrs.	.000075
Intermediate	-		1.35 Hrs.	.000165
Total	2.0 Hrs.	.03	1.97 Hrs.	.000240
Condition II:				
Organizational	-		.62 Hrs.	.000157
Intermediate	-		1.23 Hrs.	.000315
Total	2.0 Hrs.	.03	1.85 Hrs.	.000472

All maintenance on the equipment will be unscheduled. Maintenance can be performed by one man so MTTR equals MMTR. Calculations for MMH/FH are based on a maintenance rate of 2.0, the failure rates determined by the Reliability predictions (Condition I and Condition II), and an operating hour per flight hour ratio (OH/FH) of 1.5.

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QUANTITATIVE DATA SHEET DESCRIPTION

The following Quantitative Data Sheets are used to present the times that would be required to perform each task required for maintenance. These task times then combined and used in further calculations of the mean-time-to-repair.

An explanation of the columnar headings follows:

- 1) Item/Part Number Used to identify the replaceable unit.
- 2) Qty. This shows the number of items of this type in the system, so located, physically and functionally, as to have the same task times.
- 3) Task Time, Hrs. This shows the amount of time in hours estimated to perform a task. These tasks are:

TP Test Preparation TS Trouble Shooting Disassembly DA RR Remove and Replace RA Reassemble AC Adjust/Calibrate FT Functional Test SS Secure Setup

- 4) Total Time (T) This is the summation of the task times in hours for each item.
- 5) Failure Rate This appears in two columns. The first is the reliability failure rate of maintenance rate in parts per million hours for one item. The second column indicatest the quantity of items associated with that failure rate.
- 6) T x λ x Q This column is the mean elapsed time in hours per million operating hours to perform the task. This column is summarized and used in the calculation of MTTR and MMH/OH.

TABLE 4 MAINTAINABILITY 1-44501CTTON DUAL MODE ELECTRO-PNEUMATIC ACTUATOR - CONDITION I

MAINTENANCE LEVEL				TASK	- SUIT	IICURS					FAILUR	E RATE	
ITEM/PART NO.	3	TP	S.I.	AU	RR	RA	AC	F7	SS	Ŧ	۲	2×9	Tr xà::Q
ORGANIZATIONAI.	I						1		1				
	1												
DME-PA	-		.250		.181			.184		.615	40.6899	40.6899	25.0243
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MAINTAINABILITY PREDICTION DUAL MODE ELECTRO-PNEUMATIC ACTUATOR- CONDITION I

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MAINTENANCE LEVEL	L			TASK	TIME -	HOURS					FAILUR	E LATE	
ITEN/PART NO.	3	τp	'TS	ĐΑ	RR	RA	AC	F.J.	SS	т	X	2×Q	Tr × A=Q
INTERMEDIATE													
Servo Valve Assy.		.088	. 333	.	.061		.	. 500	.083	1.065	27.5675	27.5675	29.3594
Angular Sensor		. 088	.250		.259		.167	.500	.083	1.347	2.3580	2.3580	3.1762
Connector, 16 Pin		.088	.167		.257	1.		. 500	.083	1.095	.1587	.1587	.1738
Connector, 12 Pin	-	.088	.250		.192	11		. 500	.083	1.113	.1587	.1587	.1766
Qutput End Fixed		.088	.500	610	.184	.008	.167	.500	. 083	1.549	.0151	.0151	. 0234
Sensor End Fixed Gear	1	.088	.500	. 022	.184	. 028	.167	.500	.083	1.572	.0151	.0151	.0237
Roller Bearings	1 21	.088	.500	.088	.008	.135	.167	. 500	.083	1.569	.2840	.5680	.8912
Roller Bearings	101	.088	. 500	.212	.008	.340	.167	. 500	.083	1.898	.2840	. 5680	1.0781
Thrust Bearings	0	.088	. 500	.181	.008	.274	.167	. 500	.083	1.801	.2840	. 5680	1.0230
Output Gear	-	.088	. 500	.176	.010	.269	.167	.500	.083	1.793	.0151	.0151	.0271
Ring Gear	-	.088	.500	.203	.018	.330	.167	.500	. 083	1.889	.0151	.0151	. 0285
Stator Assy.		. 088	.250	.402	.301	.628	.167	.500	.083	2.419	8.6812	8.6812	20.9998

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E DATE	720	(85.0959)	
FAILUR	X	(85.0959)	
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TASK 1	μi,		
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	TP		
-	3		
EXTENANCZ LEVEL	TEMPART NO.	ORGAN I ZAT TONAL DME - PA 3854020-1	

		CONDITIC
TABLE 5	MAINTAINABILITY PREDICTION	L MODE ELECTRO-PNEUMATIC ACTUATOR -
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				TOLT	du La	SUM							
r NO.	8	TP	TS	DA	un .	UNI VI	AC	1.1	SS	T	γ γ	AK Q	T ₇ x2xQ
DIATE													
Jalve Assy.	1	.088	. 333	1	.061		.	. 500	.083	1.065	69.1475	69.1475	73.6421
Sensor	1-1	.088	.250		.259		.167	.500	.083	1.347	5.1840	5.1840	6.9828
tor, 16 Pin		.088	.167		.257			. 500	.083	1.095	.1587	.1587	.1738
or, 12 Pin	-	.088	.250		.192	.		. 500	.083	1.113	.1587	.1587	.1762
End Fixed		.088	.500	.019	.184	.008	.167	.500	. 083	1.549	.0151	.0151	.0234
Eng Fixed	1-1	.088	.500	. 022	.184	.028	.167	.500	. 083	1.572	.0151	.0151	. 225.
Bearings	1 01	. 088	.500	.088	.008	.135	.167	.500	.083	1.569	.2840	.5680	. 8912
Bearings	10	.088	.500	.212	.008	.340	.167	.500	.083	1.898	.2840.	.5580	1.0751
Bearings	0	.088	.500	.181	.008	.274	.167	.500	.083	1.801	.2840	.5680	1.0230
Gear	1-1	.088	.500	.176	.010	.269	.167	.500	.083	1.793	.0151	.0151	.0271
ear		.088	. 500	.203	.018	.330	.167	.500	.083	1.889	.0151	.0151	.025
Assy.	_ ·	.088	.250	.402	.301	.628	.167	. 500	. 083	2.419	8.6812	8.6812	20.9998

Sample calculations and applicable formulas used in the analysis are as follows for Condition I:

Organizational Level

Maintenance Data read directly from table

$$\Sigma T x \lambda x Q = 25.0243 x 10^{-6}$$

= .0000250 MMH/OH
MTTR = .615

Intermediate Level

Summations from Data Summary Table

System Repair

 $\Sigma \left(\lambda \times \mathbf{Q} \right) = 40.6885 \times 10^{-6}$ $\Sigma \left(\mathbf{T} \times \lambda \times \mathbf{Q} \right) = 54.9808 \times 10^{-6}$

Mean Time to Repair (MTTR)

$$\frac{\Sigma}{\lambda} \frac{(T \times \lambda \times Q)}{\lambda \times Q} = \frac{54.9808 \times 10^{-6}}{40.6885 \times 10^{-6}}$$

= 1.35 Hrs.

Mean Maintenance Man Hours per Operating Hour

Σ (T x λ + Q) x 10⁻⁶ = .000055 MMH/OH

80

Convert to Flight Hours and at a frequency based on Mean Time Between Maintenance actions.

OH/FH = 1.5 MR = 2.0 MMH/FH = $\Sigma(T \ge \lambda \ge Q) \ge OH/FH \ge MR$ Organizational Level = .000075 Intermediate Level = .000165

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The values for Condition II MTTR, and MMH/FH were calculated as above but using the figures from Table 5.

APPENDIX A

COMPUTER MODEL OF DYNAVECTOR ACTUATOR

A flow model was constructed for the servovalve-Dynavector motor combination and was programmed on a digital computer. By means of this routine, the pressure and flow characteristics of the valve and motor were determined for any specified operating condition. The approach was sufficiently general to allow a broad range of servovalve-motor designs to be investigated.

The pneumatic Dynavector actuator flow model determines the pressure drops, motor and leakage flows, and output torque for a given supply pressure and motor displacement and an assumed speed and valve opening. The model is depicted in Figure A-1. Leakage from the motor commutation ports is represented by orifices A_{ll} and A_{cpm} in the figure. The leakage area from the high-pressure ports to ambient is essentially one-half the motor circumference multiplied by the total axial clearance between the ring gear and the manifold plates. The area of the leakage path from the low pressure side of the motor to ambient is determined likewise. The cross-port leakage area of the Dynavector actuator is the product of the total periphery of the exhaust slots and the total axial clearance between the ring gear and manifolds. The motor commutation area, Ac, is determined directly by the commutation slot dimensions and the number of chambers. The motor is represented in terms of its displacement, D_m , and the working flow is denoted W_m . Cross-port leakage in a well-designed servovalve is very small and consequently is ignored in the flow model. Also tending to justify this assumption is the fact that the servovalve lands are generally underlapped to achieve linearity and high gain about the on-center spool position.

The equations for describing the operation of the servovalve-Dynavector motor combination are written directly, using the pneumatic orifice flow equations and node equations for the flows. The symbols are defined in Table A-1 and the equations are:

$$W_{r} = \frac{C_{d} C_{2} P_{v2} A_{v} f_{1} \left(\frac{P_{a}}{P_{v2}}\right)}{\sqrt{T}}$$
(A-1)

 $W_{c2} = W_r$ (A-2)

$$P_{m2} = \frac{W_{c2} \sqrt{T}}{c_{d} c_{2} A_{c} f_{1} \left(\frac{P_{v2}}{P_{m2}}\right)}$$
(A-3)





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Table A-1 - Servovalve-Dynavector Motor Torque-Speed Curve Parameters

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Parameter	Definition
A _c	Dynavector motor commutation area - in ²
A _v	Servovalve metering area - in ²
Acpm	Dynavector motor cross-port leakage area - in ²
A _{ll}	High pressure motor leakage to ambient area - in ²
A ₂₂	Low pressure motor leakage to ambient area - in ²
c _d	Orifice discharge coefficient
c2	Thermodynamic constant for the working fluid - $R^{1/2}/sec$
. D _m	Dynavector motor displacement relative to output shaft - in^3/rev
f f	Compressibility function for flow of working fluid through an orifice dependent on ratio of pressures across orifice
n	Dynavector motor torque efficiency for pressures measured directly across chambers
ė	Output shaft speed - rad/sec
P _{m1} (P _{m2})	High (low) motor pressure - lb/in ²
P _{v1} (P _{v2})	High (low) servovalve pressure - lb-in ²
Pa	Ambient pressure - 1b/in ²
Ps	Supply pressure - lb/in ²
R	Gas constant for the working fluid - in-lb/lb - *R
т	Temperature of the working fluid - "R
To	Dynavector motor output torque - in-1b

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$$W_{L2} = \frac{D_{d} C_{2} P_{m2} A_{L2} f_{1} \left(\frac{P_{a}}{P_{m2}}\right)}{\sqrt{T}} \qquad (A-4)$$

$$W_{\rm m} + W_{\rm cpm} = W_{\rm L2} + W_{\rm c2}$$
 (A-5)

$$W_{\rm m} = 2\pi D_{\rm m} \dot{\theta}_{\rm o} \frac{P_{\rm m1}}{RT} \qquad (A-6)$$

$$W_{cpm} = \frac{D_d C_2 P_{ml} A_{cpm} f_1 \left(\frac{P_{m2}}{P_{m1}}\right)}{\sqrt{T}}$$
(A-7)

$$W_{l1} = \frac{C_d C_2 P_{m1} A_{l1} f_1 \left(\frac{P_a}{P_{m1}}\right)}{\sqrt{T}} \qquad (A-8)$$

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(A-9)

$$W_{c1} = W_{m} + W_{cpm} + W_{\ell 1}$$

 $P_{v1} = \frac{W_{c1} \sqrt{T}}{C_{d} C_{2} A_{c} f_{1} \left(\frac{P_{m1}}{P_{v1}}\right)}$ $W_{s} = \frac{C_{d} C_{2} P_{s} A_{v} f_{1} \left(\frac{P_{v1}}{P_{s}}\right)}{\sqrt{T}}$ (A-10)

These equations are solved iteratively until the flows and pressures balance and then the output torque is computed from

 $T_o = 2\pi D_m n (P_{m1} - P_{m2})$ (A-12)

The algorithm for solution of the flow model equations is shown in the flow diagram of Figure A-2.

The algorithm was programmed in a very general manner to maximize its usefulness as a design tool. In addition to the input parameters given in Figure A-2 the temperature and other thermodynamic properties of the working fluid may readily be changed. A four-way servovalve was assumed and provisions were made for individually selecting supply and return land areas and their degree of underlap. For convenience, normalizing values for speed and torque are computed from the input supply pressure, motor displacement, and valve area. The normalizing values are theoretical maximum values; for computing normalizing speed, θ_n , it is assumed that there is no leakage and that the entire pressure drop from supply to ambient occurs across the motor, or:

$$\dot{\theta}_{n} = \frac{R \sqrt{T} C_{d} C_{2} A_{v} f_{1} \left(\frac{P_{v}}{P_{v} + P_{s}}\right)}{2\pi D_{m}}$$
(A-13)

1

The normalizing torque value is based upon 100 percent torque efficiency and the assumption that the full pressure drop occurs across the motor, or:

$$T_{m} = 2\pi D_{m} (P_{s} - P_{v})$$
 (A-14)

The program requires, in addition to the input parameters already mentioned, percentage values of normalizing speed and valve opening. A sample printout with legends added is shown below; the data are for valve areas of 0.1 in², 0 percent land underlap, 155 psia supply pressure, 14.7 psia ambient pressure, and a motor displacement of 180.0 in³/rev. The commutation, leakage to ambient, and cross-port leakage areas are: 0.100, 0.00275, 0.00267 in², respectively.

NORMALIZING VALU	ES ARE		
SPEED RAD/SEC=,	.2571E+02	TORQUE LB-INCH	= .4019E+04 =TN
PERCENT MAX AREA	= .1000E+0	3	
Al A	2 4	A3 A	4
.1000E+00	.0000E+00	· . 1000E+00	.0300E+00
PERCENT SPEED=	.2500E+02		
P 1513E+03	P 1474E+03	PM2 . 5380E+02	Pv2 .4341E+02
WS - 1088E+00	W4 . 1089E+00	WCPM . 8755E-02	W11 . 9018E-02
WR .9655E-01	Wer 9655E-01	Wm-9109E-01	W 3290E-02
To .2682E+04	%TN . 6674E+82		



Figure A-2- Algorithm for Solution of Flow-Model Equations

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High accuracy is achieved in the routine by using a half-interval search in the iterations.

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BENDIX RESEARCH LABORATOPIES

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(REV K/77FEB15) SEL EXTENDED FORTRAN 3 V

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PNEUMATIC MOTOR WITH 4-WAY VALVE AND SERIES COMMUTATION ORIFICES PS=PSIA SUPPLY, PV=PSIA VENT R=GAS CONSTANT IN-LB/LB-DEG C3=CONSTANT DEPENDING ON SPECIFIC HEAT RATIO OF GAS (INCH)(1/2)/SEC C *** PNEUNATIC NOTOR WITH 4-MAY VALVE AND SERIES COMMUTATION C C *** PS=PSIA SUPPLY, PV=PSIA VENT C *** R=GAS CONSTANT IN-LB/LB-DEG C *** CONSTANT IN-LB/LB-DEG C *** CONSTANT DEPENDING ON SPECIFIC HEAT RATIO OF GAS C *** NEG H=UNDERLPP(OPEN CENTER, PERCENT OF ZERO LAP MAX AREA) C *** NEG H=UNDERLPP(OPEN CENTER, PERCENT OF ZERO LAP MAX AREA) C *** D=DISPIACEMENT/REV (INCH**3/REV) C *** D=DISPIACEMENT/REV (INCH**3/REV) C *** D=DISPIACEMENT/REV (INCH**3/REV) C *** D=DISPIACEMENT/REV (INCH**3/REV) C *** D=DISPIACEMENT/REV (INCH**3/REV) C *** D=DISPIACEMENT/REV (INCH**3/REV)

DATA PI. G.' 3. 141592. THE SEAV DATA PI. G.' 3. 141592. State 264/ C *** NORMALIZING VALUES BASED ON ZERO TOROUE. ZERO LEAK, ZERO LAP C *** RV1 THRU AV4-VALVE MAX PORT AREAS. ZERO LAP. NAMELIST/CONS/R, 1, CD. XX NAMELIST/CONS/R, 1, CD. XX NAMELIST/CST1/AP1, AV2, AV3, AV4, H1, H2, H3, H4, PS, PV, D, AC, AL, ACPM. 4XJ1M, XDJ1, M1, XJ2M, XDJ2, M2 READ(5, CONS)

C3=C33(XK, G)

RRT=SQRT(R+T)

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444 THIS IS RETURN POINT FOR EXTRA LST1 SETS 102 READ(5, LST1, END=99) WRITE(6, 150)

150 FORMATCIAL. * PNEUMATIC MOTOR WITH 4-LAY VALVE. INPUT DATA')
WRITE(6, 151)R, T, CD, XK
151 FORMATCIAH , 'GAS CONSTANTS', 2X, 'R=', F10, 3, 2X, 'T=', F10, 3, 2X,
151 FORMATCIAH , 'GAS CONSTANTS', 2X, 'R=', F10, 3, 2X, 'T=', F10, 3, 2X,
152 FORMATCIAH , 'YALVE CONSTANTS', 1A, AV1=', E15, 4, 2X, 'AV2=', E15, 4, 2X,
2'H3 =', F15, 4, 2X, 'H4 =', F15, 4)
WRITE(6, 1533)PS, PV

153 FORMATCIH , EXTERNAL PRESSURES, PSIR, /, PS=', E15. 4, 2X, 'PV=',

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1E15. 4

INFITE(6, 154)0, AC, AL, ACPM
154 FORMAT(1H , 'MOTOR PARAMETERS, D INCH++3/REV, AREAS IN++2', /' 1X,
1'D=', E15. 4, 2X, 'AC=', E15. 4, 2X, 'AL=', E15. 4, 2X, 'ACPM=', E15. 4)
INFITE(6, 155)XJ1N, XDJ1, M1

155 FORMAT (1H , 'VALUE AREA PANGE IN PERCENT', /' INITIAL PERCENT AREA', 1511. 3. 2X, 'INCREMENT IN PERCEN1', F10. 3. 2X, 'NUMBER OF VALUES', 14)

LATITE(6, 156)XJZN, XDJZ, M2 156 FORMAT(1H , SPEED RANGE IN PERCENT, /, / INITIAL PERCENT SPEED', 1510. 3, 2X, / INCREMENT IN PERCENT', F10. 3, 2X, / NUMBER OF VALUES', 14) DM=D/(2. *PI)

CON=1. 10

NORMALIZIND VALUES DELP=1. E-5*PS WPSM=W(PV, PS, XK, CDN, AV1, C3, RR1) DELT=1. E-5*WPSM PM1H=PS

PM1L=0.

PH1=, 54(P5+PV) DO 18 J=1, 38 MP5=W(PM1, P5, XK, CDN, AV1, C3, RRT)

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APPENDIX B

COMPUTER MODEL FOR ELECTRIC MODE TORQUE-SPEED CURVES

The voltage applied to a motor coil equals the sum of the voltage drop in the coil resistance and the back emf.

$$\mathbf{V} = \mathbf{R} \, \mathbf{i} + \mathbf{E}_{\mathbf{h}} \tag{1}$$

The back emf is proportional to the product of the number of turns and the rate of change of magnetic flux.

$$E_{\rm b} = 10^{-8} \, \mathrm{N} \, \frac{\mathrm{d}\phi}{\mathrm{d}t} \tag{2}$$

The flux crossing a pole face equals the pole face area times the flux density. Thus

$$\phi = 6.452 \text{ A}_{\text{p}} \text{B}$$
 (3)

$$E_{\rm b} = 10^{-8} \, {\rm N} \, {\rm x} \, 6.452 \, {\rm A}_{\rm p} \, \frac{{\rm d}B}{{\rm d}t}$$
 (4)

$$V = R 1 + 6.452 \times 10^{-8} NA_p \frac{dB}{dt}$$
 (5)

The flux density is proportional to the number of ampere turns divided by the air gap length.

$$\mathbf{B} = \frac{0.4\pi \,\mathrm{N}\,\mathbf{i}}{2.54\mathrm{g}} = 0.496\,\frac{\mathrm{N}\,\mathbf{i}}{\mathrm{g}} \tag{6}$$

Differentiating equation 6 gives

$$\frac{dB}{dt} = 0.496 \frac{N}{g} \frac{d1}{dt} - 0.496 \frac{N1}{g^2} \frac{dg}{dt}$$
(7)

The air gap length is given by

$$g = C + (1 + \cos \theta)e \tag{8}$$

Differentiating gives

$$\frac{dg}{dt} = e \ \omega \sin \theta \tag{9}$$

Combining equations 5, 7, and 9 and rearranging terms gives

$$\frac{di}{dt} = 3.125 \times 10^7 \frac{g}{A_N N^2} (V-Ri) - \frac{e\omega I}{g} \sin \theta$$
(10)

The instantaneous current is

The torque is computed by determining the torque for an incremental pole face area dA, and integrating this torque over the complete area. The incremental pole face area is

$$dA = radius x length x d\theta$$

The torque developed by area dA is

 $dT = 3.75 \times 10^{-7} eB dA sint x ratio x eff$

The efficiency term eff accounts for mechanical losses. The constant $3.75 \ge 10^{-7}$ includes an <u>empirical</u> factor to account for the difference between the theoretical force developed by a pole and the actual force.

The portion of the circumference occupied by the pole face is

$$T_{m} = 8 \times \frac{\text{pole face area}}{\text{stator length}} \times \frac{1}{\text{circumference}}$$
$$T_{m} = \frac{A_{p}}{\text{length}} \times \frac{8}{2\pi \times \text{radius}}$$

The motor torque is

$$T = T_m \int_0^{\pi} dt$$

Eddy current power loss is given by

$$P_e = Kt^2 B^2 f^2 W_1$$

where

- f = frequency
- t = iron thickness
- W, = weight of iron

The power loss for magnetic sheet steel is given by a curve of watts per pound at 60 hertz versus flux density.

For analysis, the eddy current power loss is generally accounted for by an equivalent resistor R_{\perp} . Thus

$$I^{2}R_{e} - P_{e} - Kt^{2}B^{2}f^{2}$$
$$R_{e} - K\left(\frac{tBf}{I}\right)^{2}$$

R is added to the coil resistance.

Due to flux leakage, all of the flux generated does not cross the working air gap. A flux leakage factor of 0.65 was established for Dynavectors. Using this leakage factor, the flux density is given by

$$B = \frac{NI}{3.113 g}$$

Fringing reduces the effective flux density in the air gap. The effective flux density with fringing is given by

$$B_{g} = \frac{L \times W}{(L + 2G)(W + 2G)} B$$

where

B = Flux density without fringing

B = Flux density with fringing

G = Air gap length

L = Length of pole face

W = Width of pole face

Iron reluctance reduces the effective number of ampere turns. The effective ampere turns are given by

$$NI_e = NI - 0.029 \left(\frac{B}{1000}\right)^2 L_1$$

where L, is the length of the flux path through iron.

CORRELATION OF COMPUTER PROGRAM WITH TEST DATA FOR SOLID ROTOR

The model EH-441-Ul Electric Dynavector, which was correlated with the computer model, has a solid rotor. According to theory the power loss will increase as the square of the thickness. Thus the theoretical increase in power loss due to the solid rotor, compared with 0.025 sheet stock, is

$$C_{\bullet} = \left(\frac{0.375}{0.025}\right)^2 \times \frac{\text{rotor weight}}{\text{total iron weight}} = 60$$

B-3

The actual power loss of the solid rotor will be less than indicated by theory, because the magnetic flux travels near the surface of the rotor, and the flux density at the center of the rotor iron is low. C_e equal to 1/3 the calculated value (20) was therefore used for the solid rotor model. C_e equals 1.0 for the laminated rotor. The equivalent resistor R_e in the computer program was multiplied by this factor.

B--4

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DELMO4-ELECTRIC DYNAVECTOR TOROUE+SPEED CURVES
10 +
          DIMENSIONS IN INCHES, POUNDS, GAUSS (VOLTAGE WAVE FORM)
20 ·
          PROGRAM GIVES TORQUE FOR EACH SPEED OMEGA(K)
30 +
50
          DIMENSION GMEGA(6), DES(6)
          REAL LENGTH. I.N.
60
          DATA ACAP/1.000/,BS/15000./,C/.005/,CAP/.0000/,EFF/.64/
62
          DATA PI/3.14159/,AMAX/10./,RATID/360./
64
          DATA DES/10..20..30..40..50..60./
70
          DATA DIAMO/8./.DIA/4.875/.ECC/.048/.VD/265./
71
          04TA HC01L/.750/.WC01L/.500/.WIDTH/.8/,TC0L/144./
72
73
          DATA V1/90.0/
      600 WRITE(9,98):98 FORMAT('LENGTH,RATIO,N,V1')
130
          READ (9,99) LENGTH . RATIO, N. VI
140
          DO 19 K=1.6
141
          OME64(K) = DES(K) + P1+ RATIO/180.
142
      19 CONTINUE
:43
      99
          FORMAT (7F10.3)
150
          ACOIL=MCOIL+HCOIL
154
          TSI=N/ACGIL
155
          CDEFR=8.046+.0136+(TSI-688.)
155
          WIRLEN=N+2. + (LENGTH+W1DTH+2.+WCDIL)
157
158
          R=COEFR+WIRLEN/12000.
159
          R=10.
          HC=LENGTH+WIDTH
160
          D1=DIA-WIDTH
161
162
          WI=.29+.785+(D1AMO++2.-DI++2.)+LENGTH
163
          WIC=.00155+W1
          XLI=2.8+DIAMO-1.5+DIA
164
          AP=ACZACAP
170
130
          PERI=2. . (LENGTH+WIDTH)
190
          RADIUS=.5+DIA
0.05
          DA=RADIUS+LENGTH+.01
          CALCULATE IM TO COPRECT FOR SPACE BETWEEN POLES
210 +
          TM=.9+(AP/LENGTH)+8./(2.+PI+RADIUS)
220
230
          WRITE(9.101)
240 101 FORMAT(3X.7HDEG/SEC,4X.6HIN-LBS,6X.4HAMHS.5X.5HWATTS.7X.3HEFF)
          DU 720 K=1.6
259
260
          V=VQ:THETA=0.0;T=0.0;SDT=0.0;SIDT=0.0;B=0.0;I=0.0
           XHI=0.0; J=0; POW=0.0; POUT=0.0
261
270
           DT=.01/CMEGR(K)
           ST1=.010+(1.-.00122+DMEGA(K))
271
           STMAX=PI/DMEGA(K)
272
274
           ST2=.33+STMAX
           1F(ST2.GT.ST1)60 TD 666
275
075
           ST2=ST1
277
       666 CONTINUE
       615 G=C+(1.+COS(THETA)) +ECC
085
281
           IF (SDT.LT.ST1) GD TO 634
283
           IF (SDT.LT.ST2) 60 TO 635
284
           V=V1+(1.-(SDT-ST2)/(STMAX-ST2))
235
           60 TO 634
      625 V=V1
885
685
     634 CONTINUE
290
           AG=AP+6+PEP1+1.57+6++2.
291
           Y=AP/AG
395
           B6=8+Y
```

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294		RE=RED+DMEGA(K)
295		IF(V-5.)638,639.639
536	638	V=5.
297	639	CONTINUE
333		B5=B/100000.
599		RE=WIC+B5++2+(DMEGA(K)/(I+.1))++2
300		D11=3.125E07+6+(V-I+(R+RE))/(AP+N+N)
31.0		D12=I+ECC+OMEGA(K)+SIN(THETA)/G
0×0		0101=011-012
330		I=I+0I0T+0T
335		IF(I-AMAX)616,616,617
333	616	60 TC 618
334	617	I=AMAX
335	618	CONTINUE
340		$B = (N \bullet I - X H I \bullet X L I) / (3.113 \bullet 6)$
350		BI=B/ACAP
360		IF(BI-BS)619,620,620
370	619	60 TO 681
330	620	B=BC+HCHP
395		XHI=.029♦(B/1000.)♦♦2.
333	621 0	ONTINUE
3.3.0		[0P(]W=A+[
460		₽©⊌=₽₽₩+₽₽₽₩•₽1
420		60 TO 626
440	623	CONTINUE
443		BG=B♦AP/AG
450	626	DELT=0.375E-06+DA+ECC+RATIO+BG+B6+S1N(THETA)+EFF
460		T=T+DELT
479		THETA=THETA+DMEGA(K) + DT
430		SIDT=SIDT+I+DT
490		SDT=SDT+DT
491		J= J+1
492		IF(J-10)511,512,512
493	511	60 TO 513
4-4	512	CONTINUE
495		J=()
49-	513	CONTINUE
447	514	FCRMAT(F10.2,F10.4,3F10.2)
500		IF(THETA-P1)615,700,700
510	700	AVGI=SIDT/SDT
511		WATTS=4.*PGW/SDT
520		T1=TM+T-TCCL
533		POUT=T1+OMEGA(K) +.113/RATIO
524		XMEFF=POUT/WATTS
530		AMPS=4. ◆AVGI
540		SPS=(OMEGA(K) +4.)/PI
550		WRITE(9,710) DES(K), T1, AMPS, WATTS, XMEFF, SPS
560	057	CONTINUE
570	710	FORMAT(4F10.2,F10.4,F10.2)
572		MRITE(9,222);222 FORMAT(/KY/)
574		READ(9,823)KY:823 FORMAT(112)
530		60 10 (600,800x,KY
290	800	STOP
500		END

man

APPENDIX C

DUAL MODE DYNAVECTOR WITH COMMUTATED PNEUMATIC/COMMUTATED ELECTRIC DRIVE

C.1 FOREWORD

This report has been written to describe preliminary studies investigating the feasibility of operating a Bendix Combination Electric-Pneumatic Dynavector Rudder Actuator in both electric and pneumatic modes simultaneously, each acting as an implicit standby for the other to improve the probability of aircraft rudder control survival when this device is installed.

It will be noted that the basic dynavector design can be implemented as either"commutated-proportional" or "pure-stepper" in both the pneumatic and electrical modes. However, the arrangement choice is subtly restricted by the fact that electric designs are more cost-effective as steppers whereas pneumatic versions work better with the commutated proportional approach. And as a further complication, the successful marriage of dissimilar types would appear to impose complex dynamic problems on the common control system which, in addition to its normal duties, must also deal with load sharing (a problem which resolves itself naturally when similar devices are used).

Because of its apparent simpler load-sharing control problem, the combination of commutated-proportional pneumatic with commutated-proportional electrical has been chosen as the first target of this investigation. Stepper modes and dissimilar combinations will be presented later.

C.2 OBJECTIVES

It is the objective of this phase of the Bendix Dynavector Development Program to create a CSMP dynamic model of paired proportional electric and pneumatic actuators integrated into the T-2C airplane rudder channel. The model is exercised in a manner that will demonstrate performance capability, dynamic response, load-sharing characteristics and the impact of simulated battle-damage on the pilot's ability to safely control the aircraft rudder channel.

C.3 CONCLUSIONS

Operation of the Bendix Dynavector Dual-Mode T-2C Rudder Channel Model demonstrated satisfactory compliance with the aircraft dynamic requirements while exhibiting acceptable load-sharing throughout. Simple failures such as the loss of an electrical (or pneumatic) power source or an electrical controller (or pneumatic amplifier) were so easily absorbed by the system's redundancy that such problems might readily pass unnoticed by the pilot. On the other hand, off-center malfunctions such as a sticking displaced pneumatic valve or an amplifier hard-over failure yield more severe problems to the extent that the pilot must exert considerable effort just to maintain a center position and can only achieve full rudder displacement by applying maximum physical effort.

To prevent any hardovers caused by a malfunction in electrical signal paths it is necessary to make the signal paths redundant and employ in-line monitoring and voting schemes.

C-2
To take care of situations where the pneumatic valve is stuck open, a provision must be made to cut off the pneumatic air supply.

C.4 RECOMMENDATIONS

-

C.4.1 Flight Safety

Based on these findings, it is recommended that the system arrangement of the Dual-Mode Dynavector Actuator ultimately selected for flight test be configured in such a manner that either (or both) channel(s) will be automatically decoupled in case of emergency.

C.4.2 Further Investigation

This study should be continued by creating a CSMP Electrical Dynavector Stepper Model operating in dual-mode with a Commutated Pneumatic partner. The program would resolve load-sharing problems and simulate critical battle damage to determine the related impact on flight-safety.

C.4.3 Flight Test Configuration

It will be necessary to compare relative merits of paired Commutated-Pneumatic/Commutated -Electric with paired Commutated-Pneumatic/Stepper-Electric Dynavector Actuators to identify the best combination for eventual flight test on the T-2C airplane.

C-3



C-4

FIGURE C-1

C.5.1 <u>Dual-Mode Commutated-Pneumatic/Commutated-Electric</u> Dynavector Model Configuration

Please refer to Figure C-1 describing the T-2C Rudder Channel CSMP Model used for this study. The overall approach is a classic application of angular dynamic equilibrium theory. The instantaneous rudder hinge-pin torques applied by the pilot via pedals and cable, cross-wind from the rudder, centering spring, pneumatic actuator and electric actuator are all added arithmetically and the unbalanced sum applied to impart angular acceleration to the combined rudder and actuator moments of inertia. The resultant angular acceleration is iteratively integrated first to obtain shaft angular speed and again to obtain output shaft position. Pilot effort is measured by a torque sensor, compared with a displacement schedule of pedal 'feel', and residual errors directed to simultaneously drive the electrical controller and the amplifier-positioned pneumatic transfer valve. Because it is technically difficult to measure rotor orbit-speed on a dynavector-type mechanism, the natural damping provided by pumping losses, back EMF and friction is used to the fullest extent and open-loop gains are adjusted accordingly. The short-term lags associated with the transfer-valve, compressibility, and winding inductance are rigorously simulated and, in fact, were the cause of considerable load-sharing difficulties before compensating networks were devised to correct the problem.

C.5.2 Test Methods

To obtain sufficient test data on the above Dynavector-Assisted Rudder Channel Configuration to properly evaluate its true performance capability, the simulated human pilot was programmed to impose a series of critical forcing-functions into the cockpit pedal mechanism, including small stepchanges, sinusoidal motion at 1, 2, 4 and 8 Hz, and finally a rated-speed full-left, full-right rudder position traverse to demonstrate slew-speed capability. Baseline visibility into the system functions was provided by means of sets of time-based parametric data plots describing exactly how each parameter responds during the programmed disturbances.

Next, destructive logic was implemented to simulate the occurrence of certain failures during the early stages of each forcing function. By this method, it was possible not only to determine how much the channel performance would be degraded but also the absolute capability of the pilot to cope with the problem and maintain acceptable standards of flight safety. In addition to a baseline-run, the following failures were simulated during the testing.

- Pneumatic pressure loss
- Transfer valve hang-up at point of peak test-mode excursion.
- Pneumatic amplifier gain loss of 75%.
- Electrical power loss.
- Electrical controller hard-over type failure.

C.5.3 Test Results

C.5.3.1 Commentary

In general, the Dual Mode Dynavector System was found to be highly tolerant of catastrophic failures such as the loss of basic electrical or pneumatic power, the occurrence of a completely "dead" controller or the gross deterioration of pneumatic amplifier gain. On the other hand, unusual failures such as a stuck transfer valve or the loss of one side of a push/pull electrical driver creates the condition of the actuators opposing each other, backing up not only the full rudder load but also the force-sensor thrust on the overburdened pilot. An automatic disconnect system has been proposed to enable the pilot to free such a failure should one be encountered during flight test.

C.5.3.2 Specific Findings

Base-line system performance and the impact of the various failure modes discussed above are presented numerically in Figure C-2.

C-7

requires an automatic disconnect for com- plete safety.	+103.3 (340% Incr)	+48. +40. (1746%Incr)	34.6 (7% Incr)	1.0	lectrical antroller ard-Over ailure
This type failure is an acceptable risk	+40.0 (3.8% Decr)	+ 3.1 (21% Incr)	31.1 (3% Decr)	0.5	omplete oss of lectrical ower Supply
This type failure requires an automatic disconnect for com- plete safety.	+179.1 (330% Incr)	+ 7.0 - 1.9 (150% Incr)	54.4 (69% Incr)	ve 1.0	ung Up neumatic ransfer Val t Maximum xcursion
This type failure is an acceptuble risk	68.4 (64% Incr)	+ 5.3 (103% Incr)	57.2 (77% Incr)	1.0	omplete oss of neumatic upply ressure
This type failure is an acceptable risk	+56.6 (36% Incr)	+ 4.3 (50% Incr)	47.1 (45% Incr)	1.0	5% Loss f Pneumatic mplifier ain
Baseline	+41.6	+ 2.6	32.2	0.7	o-Failure aseline
FLIGHT SAFETY IMPACT	PEDAL FORCE TO ACHIEVE HARD OVER SLEW(POUNDS)	PEDAL FORCE AT 1 HZ (POUNDS)	PHASE LAG AT 4 Hz (DEGREES)	DAMPING RATIO	A I LUKE

Figure C-2

C-8

APPENDIX D

DUAL MODE DYNAVECTOR WITH COMMUTATED PNEUMATIC/STEPPER ELECTRIC DRIVE

D.1 FOREWORD

This report has been written to describe follow-on studies investigating the feasibility of operating a Bendix Combination Electric-Pneumatic Dynavector Rudder Actuator simultaneously in both electric and pneumatic modes, each acting as an implicit standby for the other to improve the probability of aircraft rudder control survival when this device is installed.

It will be noted that the basic dynavector design can be implemented as either 'commutated-proportional' or 'purestepper' in both the Pneumatic and Electric Modes. However, the arrangement choice is subtly restricted by the fact that electric designs are more cost-effective as steppers whereas pneumatic versions work better with the commutated proportional approach. And as a further complication, the successful marriage of dissimilar types would appear to impose complex dynamic problems on the common control system which, in addition to its normal duties must also deal with load-sharing (a problem which resolves itself naturally when similar devices are used).

Because of its apparently simplistic load sharing problems, the combination of Commutated Proportional Pneumatic with Commutated Proportional Electrical was studied first and the results presented in Appendix C. This document addresses the dynamic problems of the more cost-effective combination of stepper-electric with proportional pneumatic, and compares its technical risk with the all proportional alternate.

D.2 OBJECTIVES

It is the objective of this phase of the Bendix Dynavector Development Program to create a CSMP Dynamic Model of paired stepper-electric and proportional-pneumatic actuators integrated into the T-2C Airplane Rudder Channel. The model is exercised in a manner that will demonstrate performance capability, dynamic response, load sharing characteristics and the impact of simulated battle-damage on the pilot's ability to safely control the aircraft rudder channel.

D.3 CONCLUSIONS

Operation of the Bendix Dynavector Dual-Mode T-2C Rudder Channel Model demonstrated satisfactory compliance with the aircraft dynamic requirements. Simple failures such as the loss of an electrical (or pneumatic) power source or an electrical controller (or pneumatic amplifier) were so easily absorbed by the system's redundancy that such problems might readily pass unnoticed by the pilot. Even the occurrence of a pneumatic valve sticking at maximum displacement caused no trouble to the pilot who could readily overide the unwanted pneumatic torque making use of the abundant electrical stepper torque to continue operations. Only one failure-type produced a true flight-safety problem when it was shown that the maximum effort of the pilot reinforced by full pneumatic torque could not restrain the hard-over slew of the powerful stepper, inadvertently responding to the drive of a malfunctioning controller (which was assumed to have failed outputing an undesired train of pulses at maximum frequency).

It is probably necessary to make the electrical signal paths redundant with in-line monitoring and voting circuits to prevent any hardovers caused by a malfunctioning electrical controller.

D.4 RECOMMENDATIONS

D.4.1 Flight Safety

In view of these findings, it is recommended that the system arrangement of the Dual-Mode Dynavector Actuator ultimately selected for flight test be configured in such a manner that either (or both) channel(s) will be automatically decoupled in case of emergency.

D.4.2 Best Configuration Selection

Because paired Stepper-Electric and Proportional-Pnaumatic Dynavectors not only easily meet T-2C Rudder Channel Performance requirements but are beyond doubt the most costeffective candidates proposed, it is recommended that this combination be chosen as the best possible configuration for the detail design phase scheduled to begin during the upcoming period.

D.4.3

D.5 RESULTS

D.5.1 <u>Dual Mode Commutated-Pneumatic/Stepper-Electric</u> Dynavector Model Configuration

Please refer to Figure D-1 describing the T-2C Rudder Channel CSMP Simulation used for this study. The overall approach is a classic application of angular dynamic equilibrium theory. The instantaneous rudder hinge-pin torques applied by the pilot via pedals and cable, cross-wind from the rudder, centering spring, pneumatic actuator and electric stepper actuator are all added arithmetically and the unbalanced sum applied to impart angular acceleration to the combined rudder and actuator moments of inertia. The resultant angular acceleration is iteratively integrated first to obtain shaft angular speed and again to obtain angular shaft position. Pilot effort is measured by a torque sensor, compared with a displacement schedule of pedal 'feel' and residual errors directed to simultaneously drive the electrical controller and the amplifier-positioned pneumatic transfer valve. As might be expected, the basic stepper design requirement of having the abundent momentary torque peaks required to start(and stop) the entire inertial load at frequencies up to 500 pulses/ second tended to produce a motor with a stiffness factor capable of easily overpowering its relatively soft pneumatic partner. Under these circumstances, the original concept of using natural load-sharing tendency as a system figure-of-merit had to be completely abandoned.

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D-5

FIGURE D-1

ACTUATOR: DUAL MODE DYNAVECTOR CONTROLLERS: STEPPER ELECTRIC AND

PROPORTIONAL PNEUMATIC

1

D.5.2 Test Methods

To obtain sufficient test data on the above dynavectorassisted rudder channel configuration to properly evaluate its true performance capability, the simulated human pilot was programmed to impose a series of critical forcing-functions into the cockpit pedal mechanism including a very slow ramp to expose a single step, sinusoidal motion at 1, 2 and 4 Hz, and finally a rated speed full-left, full-right rudder position traverse to demonstrate slew-speed capability. Baseline visibility into the system function was provided by means of sets of time based parametric data plots describing exactly how each parameter responded during the programmed disturbances.

Next, destructive logic was implemented to simulate the occurence of certain failures during the early stages of each forcing function. By this method, it was possible not only to determine how much the channel performance would be degraded but also the absolute capability of the pilot to cope with the problem and maintain acceptable standards of flight safety. In addition to a baseline run, the following failures were simulated during the testing.

- Pneumatic Pressure Loss
- Transfer Valve Hang-up at Point of Peak Test Mode Excursion
- Pneumatic Amplifier Gain Loss of 75%
- Complete Electrical Power Loss
- Electrical Controller Failing With a Fixed Max Frequency Output Pulse Train

D.5.3 Test Results

D.5.3.1 Commentary

In general, the Electric Stepper varient of the Dual Mode Dynavector System was found to be even more tolerant of catastrophic failures than the proportional electric varient tested previously. The pilot would normally be completely unaware of the loss of basic electrical or pneumatic power, the occurrence of a completly 'dead' controller or gross deterioration of pneumatic amplifier gain. Even the conflicting output of a hung-up pneumatic transfer valve could be easily overpowered by the exceptionally stiff stepper. As might be expected, the one unacceptable malfunction proved to be the electrical controller failing hard-over with an unwanted pulsetrain output. Under these conditions, the pilot would be physically incapable of restraining the actuator from migrating against one of its maximum displacement stops unless the system was provided with automatic disconnect hardware.

D.5.3.2 Specific Findings

Baseline System performance and the impact of the various failure modes discussed above are presented numerically in Figure D-2

FIGURE D-2

x

		ilure is isk	ilure is DV	ilure is isk	ilure is isk	ilure omatic complete
FLIGHT SAFETY IMPACT	Baseline	This type of fa an acceptable r	This type of fa an acceptable r	This type of fa an acceptable r	This type of fa an acceptable r	This type of farequires an autodisconnect for safety.
PEDAL FORCE TO ACHIEVE HARD OVER SLEW(POUNDS)	51.6	51.6	51.0	51.6	34.5	Stepper Failure Induces a 708 Pound Cable Force
PEDAL FORCE AT 1 HZ (POUNDS)	8.9	6.9		9	8. E	Stepper Failure Induces A 795 Pound Cable Force
PHASE LAG AT 4 Hz (DEGREES)	ip 43.2	P 43.2	p 43.2	P 43.2	43.2	Non Responsive to Forcing Function
DAMPING RATIO	1.0(Approx) Hidden By Ste Coarseness	1.0(Approx) Hidden By Ste Coarseness	1.0(Approx) Hidden By Ste Coarseness	1.0(Approx) Hidden by Ste Coarseness	1.0	Non Responsive to Function
FATLURE	No-Failure Baseline	75% Loss of Pneumatic Amplifier Gain	Complete Loss of Pneumatic Supply Pressure	Hung Up Pneumatic Transfer valve at max excursion	Complete Loss of Electrical Power Supply	Electrical Controller Fails with Unwanted Puls Train Output

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DDC CAMERON STATION VIRGINIA 22314

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