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AFFDL-TR-66-131

489455

DESIGN STUDY FOR A FLIGHTWORTHY PNEUMO-MECHANICAL SERVOMECHANISM

R. G. Read, K. W. Verge
Research Laboratories Division
The Bendix Corporation

TECHNICAL REPORT AFFDL-TR-66-131

August 1966

THIS DOCUMENT IS SUBJECT TO SPECIAL EXPORT CONTROLS AND EACH TRANSMITTAL TO FOREIGN GOVERNMENTS OR FOREIGN NATIONALS MAY BE MADE ONLY WITH PRIOR APPROVAL OF THE AIR FORCE FLIGHT DYNAMICS LABORATORY (FDCL) WRIGHT-PATTERSON AIR FORCE BASE, DAYTON, OHIO 45433.

Air Force Flight Dynamics Laboratory
Research and Technology Division
Air Force Systems Command
Wright-Patterson Air Force Base, Ohio 45433

BPSN: 6(618226-62405334)

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DESIGN STUDY
FOR A
FLIGHTWORTHY PNEUMO-MECHANICAL
SERVOMECHANISM

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FOREWORD

This report, dated August 1, 1966 prepared by The Bendix Corporation, Research Laboratories Division, Southfield, Michigan is the final report defining the results of a design study for a flight-worthy low pressure pneumo-mechanical servomechanism for actuation of flight control surfaces. This work was accomplished during the period of November 1, 1965 to August 1966, under Air Force Contract AF 33(615)-3309 Task Number 822604 sponsored by the Air Force Flight Dynamics Laboratory of the Research and Technology Division, Wright-Patterson Air Force Base, Ohio. This program was administered under the direction of Mr. James Hall, of the Air Force Flight Dynamics Laboratory, FDCL. The work was conducted at the Bendix Research Laboratories Division in the Energy Conversion and Dynamic Controls Laboratory, managed by Mr. L. B. Taplin. The project was directed by Mr. K. W. Verge, Assistant Department Head, Flight Controls Department, with Mr. R. G. Read, Senior Engineer assigned as Project Supervisor.

Publication of this technical report does not constitute Air Force approval of the findings or conclusions stated in the report. It is published only for the exchange and stimulation of ideas.

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ABSTRACT

This is a final report defining the results of a design study for a flightworthy low pressure pneumo-mechanical servomechanism for actuation of flight control surfaces.

The design concept presented is the unique DYNAVECTOR* Actuator,** Model PH-370-B1, designed for installation into a F101B test vehicle in parallel with the existing hydraulic rudder control system. The DYNAVECTOR Actuator is an integrated motor-epicyclic transmission servomechanism capable of meeting all specified performance requirements when operating on 50 psig compressor bleed air.

The analyses conducted during this study have established the characteristics and/or requirements of the following:

- Duty cycle and power supply characteristics
- Servo system characteristics
- Reliability and failure mode characteristics
- Bleed air consumption requirements
- Qualification test requirements
- Assembly and components design requirements

The conclusions of this study confirm the feasibility of a low pressure DYNAVECTOR Rudder Actuator.

* Trademark of The Bendix Corporation

** The Bendix Corporation has a patent application pending on this device.

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TABLES

NOMEN

A = Spool end area, in^2

A_c = Control port throat area, in^2

A_d = Valve feedback orifice area, in^2

A_e = Vortex valve exit area, in^2

A_f = Valve feedback port area, in^2

A_g = Commutation area, in^2

A_l = Leakage area, in^2

A_m = Stress area, in^2

A_n = Nozzle ramp annular area, in^2

A_p = Cylinder piston effective area, in^2

A_s = Annular area between spool and bore, in^2

A_v = Valve supply area, in^2

a = Automatic valve feedback output, psid

b = Gear face width, in.

b_o = Output gear face width, in.

b_r = Reaction gear face width, in.

C = Thermodynamic coefficient, $deg^{1/2}/sec$

C_d = Discharge coefficient

c = Number of displacement chambers

D = Spring mean diameter, in.

D_m = Motor displacement, in^3/rev

DP = Gear diametral pitch

D_p = Gear pitch diameter, in.

d = Piston diameter, in.

d_f = Diameter on which clutch gear force acts, in.

d_m = Motor displacement, in^3/rad

d_w = Wire diameter, in.

e = Eccentricity, in.

F = Hydraulic cylinder output force, lb

F_f = Friction force, lb

F_k = Clutch kickoff spring force, lb

F_n = Normal force, lb

F_p = Clutch piston force, lb

F_s = Gear separating force, lb

F_t = Tangential force, lb

F_{ts} = Tension spring force, lb

f = Frequency, cps

G_s = Servovalve transfer function

g = Volumetric flow constant, $in^4/sec\text{-}lb^{1/2}$

K = SFC constant, HP-HR/lb-in

K_a = Automatic valve amplifier gain, psi/psi

K_d = Gas density constant, $lb\text{-}min/in^3\text{-}sec$

K_f = Automatic valve feedback gain, psi/rad

K_s = Servovalve gain, in/psi

K_v = Vortex gain factor

k = Gas specific heat ratio

l = Length of displacement chamber, in.

l_1 = Input linkage length between manual valve and pilot input point, in.

l_2 = Input linkage length between manual valve and power actuator centerline, in.

M = Spool mass, $lb\text{-}sec^2/in$

M_t = Mach number - tangential, outer wall

N = Number of teeth

N' = Transmission ratio

N_l = Load speed, rpm

N_m = Zero load speed, rpm

P = Automatic valve amplifier output, psid

P_c = Control pressure, psia

P_e = Ambient pressure, psia

P_f = Feedback pressure, psia

P_i = Pressure at inside of vortex exit, psia

P_o = Pressure at outer wall of vortex, psia

P_s = Supply pressure, psia

P_1 = Upstream motor pressure at valve port, psia

P_2 = Downstream motor pressure at valve port, psia

P_1' = Upstream pressure inside motor, psia

NOMENCLATURE

lb

lb

action

nt, in^4/sec-$lb^{1/2}$

lb-in

lier gain, psi/psi

b-min/in^3-sec

ack gain, psi/rad

l

t chamber, in.

tween manual valve

l.

tween manual valve

terline, in.

l

tial, outer wall

fier output, psid

a

ia

vortex exit, psia

l of vortex, psia

ure at valve port, psia

essure at valve port, psia

side motor, psia

P_2' = Downstream pressure inside motor, psia

p = Autopilot input pressure signal, psid

p_c = Gear tooth circular pitch

Q = Volumetric flow rate, in^3/sec

R = Gas constant, in-lb_f/lb_m °R

R_b = Gear base circle radius, in.

R_p = Gear pitch radius, in.

r = Rudder horn torque arm, in.

S_c = Compressive stress, psi

SFC = Specific fuel consumption, lb/HP-HR

S_r = Radial stress, psi

S_s = Shear stress, psi

S_t = Tangential stress, psi

s = Laplace operation or d/dt

T = Temperature, °R

T_a = Automatic actuator output torque, lb-in

T_1 = Rudder load torque, lb-in

T_o = Power actuator output torque, lb-in

T_s = Stall torque, lb-in

t = Time, sec

t_p = Gear tooth thickness at pitch line, in.

t_w = Wall thickness, in.

V = Compression volume at spool end, in^3

W = Displacement flow, lb/sec

W_a = Stall leakage flow, lb/sec

W_b = Zero load displacement flow, lb/sec

W_c = Control throat weight flow, lb/sec

W_d = Weight flow displaced by spool, lb/sec

W_e = Actuator exhaust flow, lb/sec

w_f = Control flow entering control port, A_f, lb/sec

w_x = Supply flow entering feedback line, lb/sec

W_o = Vortex valve exit weight flow, lb/sec

W_p = Pressurization weight flow, lb/sec

W_s = Supply flow, lb/sec

w = Wave washer width, in.

X = Pilot input linkage displacement,

X_o = Quiescent nozzle clearance, in.

x = Gear tooth geometry factor, in.

Y_a = Automatic valve position, in.

Y_m = Manual valve position, in.

y = Lewis form factor

y_s = Spool position, in.

Z = Manual valve body position, in.

$\dot{Z}$ = Hydraulic piston linear velocity,

θ = Angular displacement, deg.

$\dot{\theta}$ = Angular velocity, deg/sec

$\ddot{\theta}$ = Angular acceleration, deg/sec^2

θ_o = Amplitude, deg

ρ = Fluid mass density, lb-sec^2/in^4

ω = Frequency, rad/sec

ΔP = Pressure differential, psid

δ = Swirl factor

σ = Nozzle gain parameter, in^{-1}

ψ = Ramp angle, rad

η = Efficiency

ϕ = Gear tooth pressure angle, deg

μ = Coefficient of friction

Δ = Deflection, in.

θ' = Rudder position, rad

θ_o' = Initial rudder position, rad

$\dot{\theta}'$ = Rudder velocity, rad/sec

β = Automatic actuator position, rad

τ_1 = Compressibility time constant, s

τ_2 = Amplifier time constant, sec

ωt = Angular displacement of forcing

ω = Harmonic motion angular velocit

γ = Weight density, lb/in^3

ν = Poisson's ratio

2

SECTION I

INTRODUCTION AND SUMMARY

This report summarizes the results of a nine-month study by which the design criteria for the fabrication of a flightworthy low pressure pneumo-mechanical servomechanism were established. The servomechanism has been designed for controlling the rudder of an F101B aircraft utilizing compressor bleed air from the Pratt and Whitney JT3 engines as the power supply. The servomechanism technical approach is based on a new concept in the field of servomechanisms, the Bendix DYNAVECTOR* Actuator.**

This design study was sponsored by the Systems Engineering Group, Research and Technology Division, Air Force Systems Command, United States Air Force, Wright-Patterson Air Force Base, Ohio, under Contract Number AF 33(615)-3309, BPSN 6(618226-62405334), Project 8226, Task Number 822604.

The major activities accomplished during this program are shown on the program schedule of milestones, Figure 1. The analyses and design efforts conducted were divided into two primary categories: application analysis, and actuator analysis and design.

The purpose of this study has been accomplished with the design of a pneumatic DYNAVECTOR rudder actuator, Model PH-370-B1, capable of being installed in an F101B aircraft in parallel with the existing hydraulic rudder actuation system and capable of meeting all specified performance requirements.

1. DESIGN DESCRIPTION

The DYNAVECTOR rudder actuator, Model PH-370-B1, is designed to mount concentric to the F101B rudder axis in parallel with the hydraulic integrated power actuator as shown in Figures 2 through 5. The DYNAVECTOR actuator, clutch and linkage assembly envelope is designed so that it does not interfere with operating space requirements of the integrated power actuator and lower damper cylinder packages. The DYNAVECTOR actuator system is capable of being declutched from the

*Trademark of The Bendix Corporation

**The Bendix Corporation has a patent application pending on this device.

MAJOR MILESTONES	FORECASTED ACHIEVEMENT DATE	ACTUAL * ACHIEVEMENT DATE
APPLICATION ANALYSIS		
1.1 INITIATE ANALYSIS OF INSTALLATION PARAMETERS	12-1-65	12-1-65
1.2 DEFINE INSTALLATION REQUIREMENTS	2-1-66	2-1-66
1.3 INITIATE EVALUATION OF JET ENGINE BLEED AIR CHARACTERISTICS	1-1-66	1-1-66
1.4 DEFINE POWER SUPPLY REQUIREMENTS	3-1-66	4-1-66
1.5 ISSUE PRELIMINARY SYSTEM & ACTUATOR SPECIFICATIONS	3-1-66	2-28-66
1.6 INITIATE SERVO SYSTEM ANALYSIS	1-1-66	1-1-66
1.7 COMPLETE SERVO SYSTEM ANALYSIS	4-1-66	4-1-66
1.8 INITIATE OPTIMIZATION ANALYSIS OF ACTUATOR & CONTROL SYSTEM INTERFACES WITH AIRCRAFT	1-1-66	2-1-66
1.9 DEFINE INTERFACE REQUIREMENTS	3-1-66	4-1-66
1.10 ISSUE SPECIFICATIONS FOR FLIGHTWORTHY ACTUATOR	5-15-66	5-15-66
ACTUATOR ANALYSIS AND DESIGN		
2.1 INITIATE ACTUATOR & SERVO COMPONENTS DESIGN	2-1-66	2-1-66
2.2 INITIATE ANALYSIS OF MECHANICAL PARAMETERS	11-15-65	11-15-65
2.3 INITIATE ANALYSIS OF FLUID POWER PARAMETERS	1-1-66	1-1-66
2.4 COMPLETE PRELIMINARY DESIGN LAYOUT	4-1-66	4-1-66
2.5 COMPLETE ANALYSIS OF MECH & FLUID POWER PARAMETERS	4-1-66	4-1-66
2.6 INITIATE FAILURE MODE ANALYSIS	3-15-66	3-15-66
2.7 COMPLETE FAILURE MODE ANALYSIS	4-15-66	5-15-66
2.8 CONDUCT CRITICAL DESIGN REVIEW	4-1-66	4-1-66
2.9 COMPLETE DESIGN REVIEW	4-15-66	4-15-66
2.10 COMPLETE FINAL SERVO SYSTEM DESIGN LAYOUT	6-1-66	6-1-66
PROGRAM DIRECTION AND REPORTS		
3-1 MILESTONE FORECAST SUBMITTAL	12-1-65	11-30-65
3-2 FINAL REPORT DRAFT SUBMITTAL	6-1-66	6-1-66
3-3 APPROVED FINAL REPORT SUBMITTAL	8-1-66	8-1-66

1965: NOV DEC — 1966: JAN FEB MAR APR MAY JUN JUL AUG

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LEGEND: PLANNED MILESTONES ▽
* COMPLETED MILESTONES ▼

Figure 1 - Program Schedule of Major Milestones

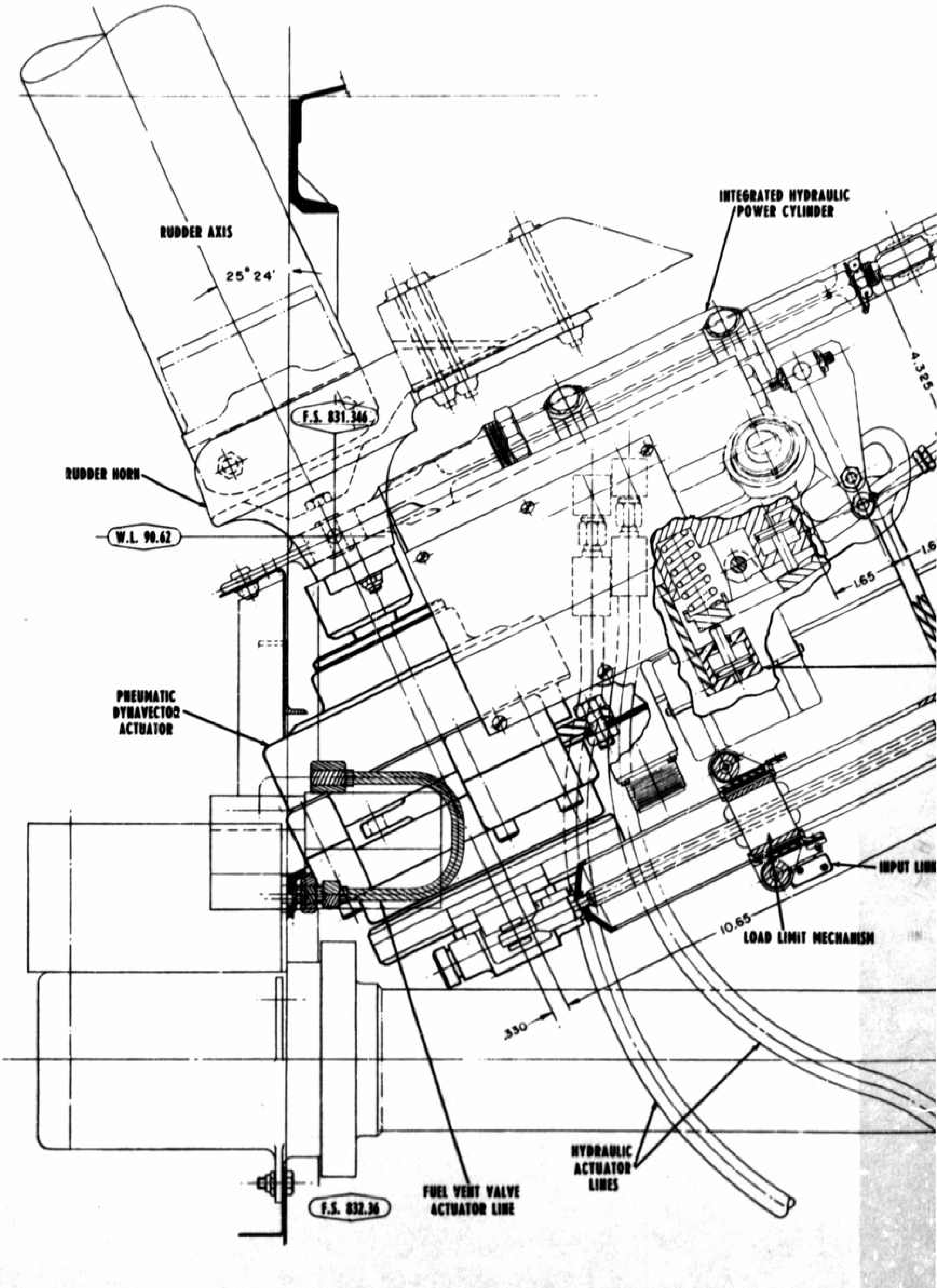
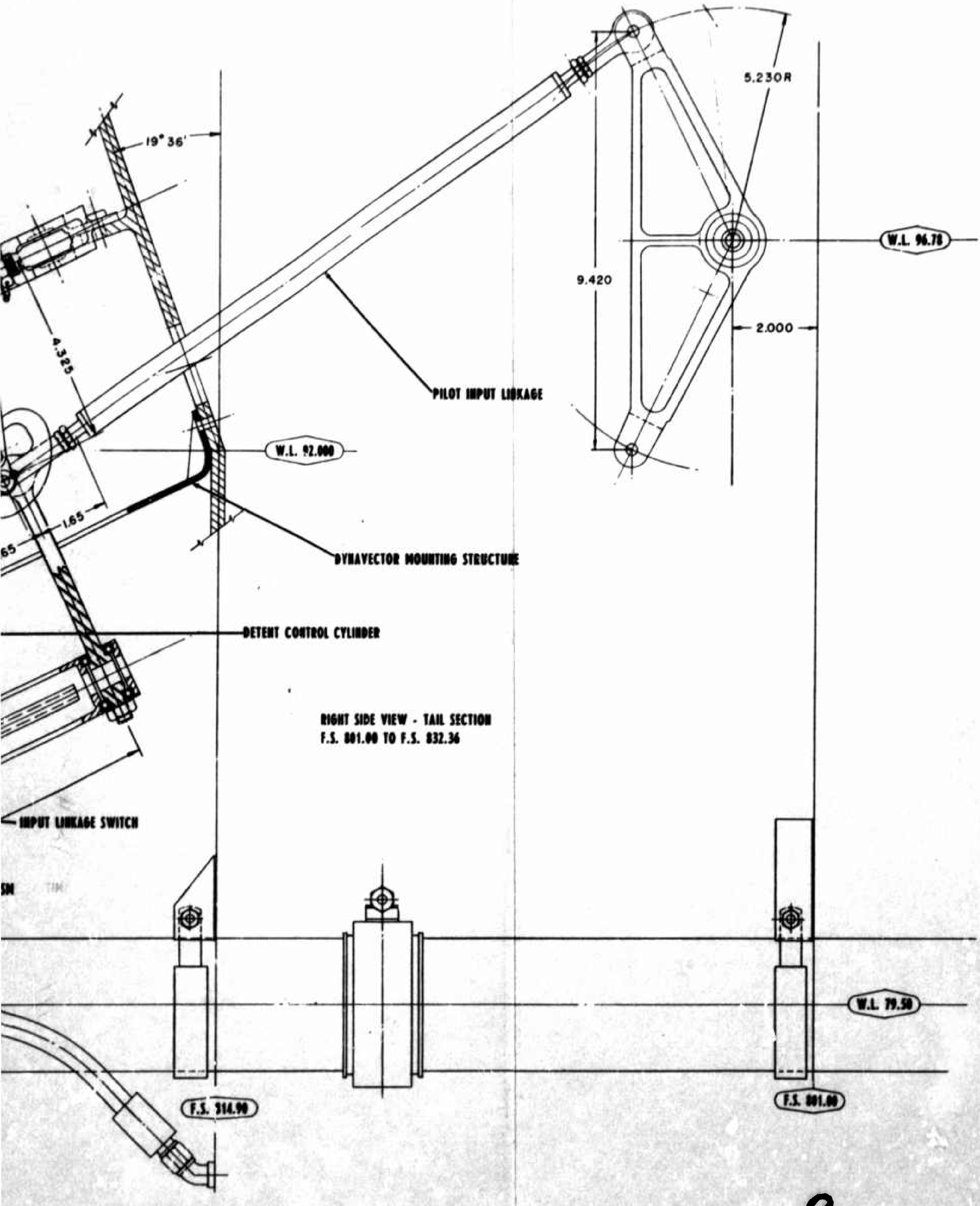


Figure 2 - DYNAVECTOR Installation



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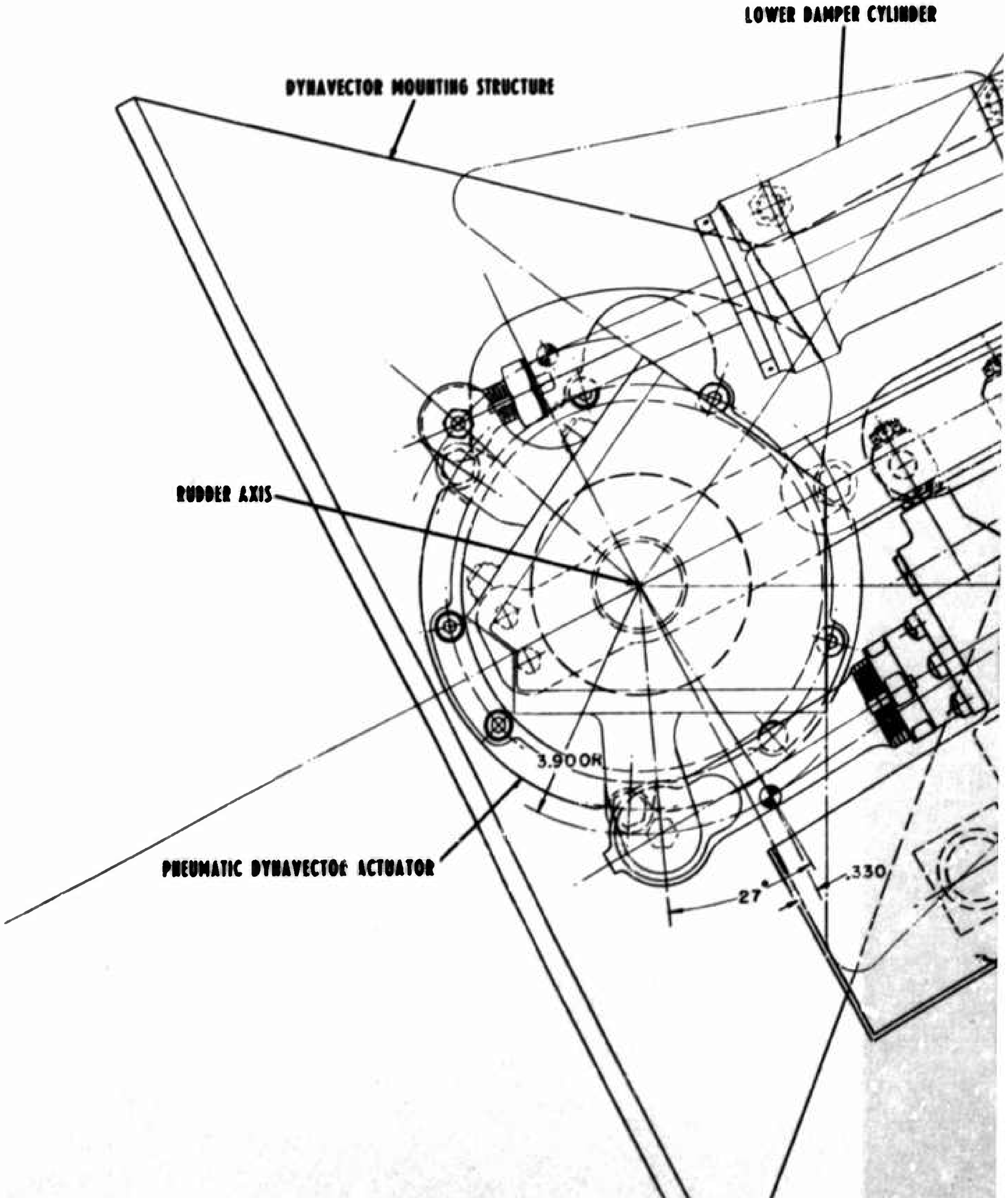
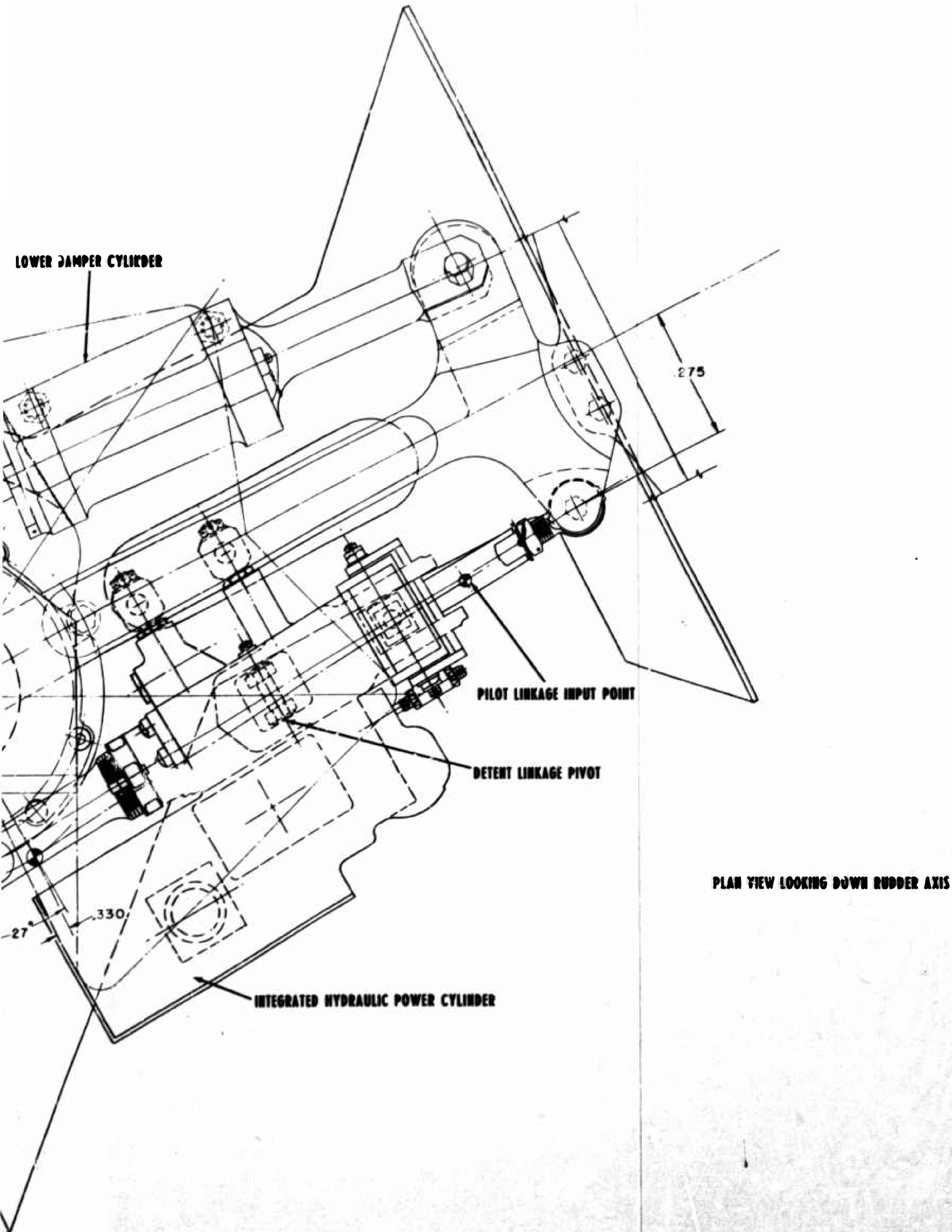


Figure 3 - DYNAVECTOR Installat



NAVECTOR Installation Plan View Looking Down Rudder Axis

2

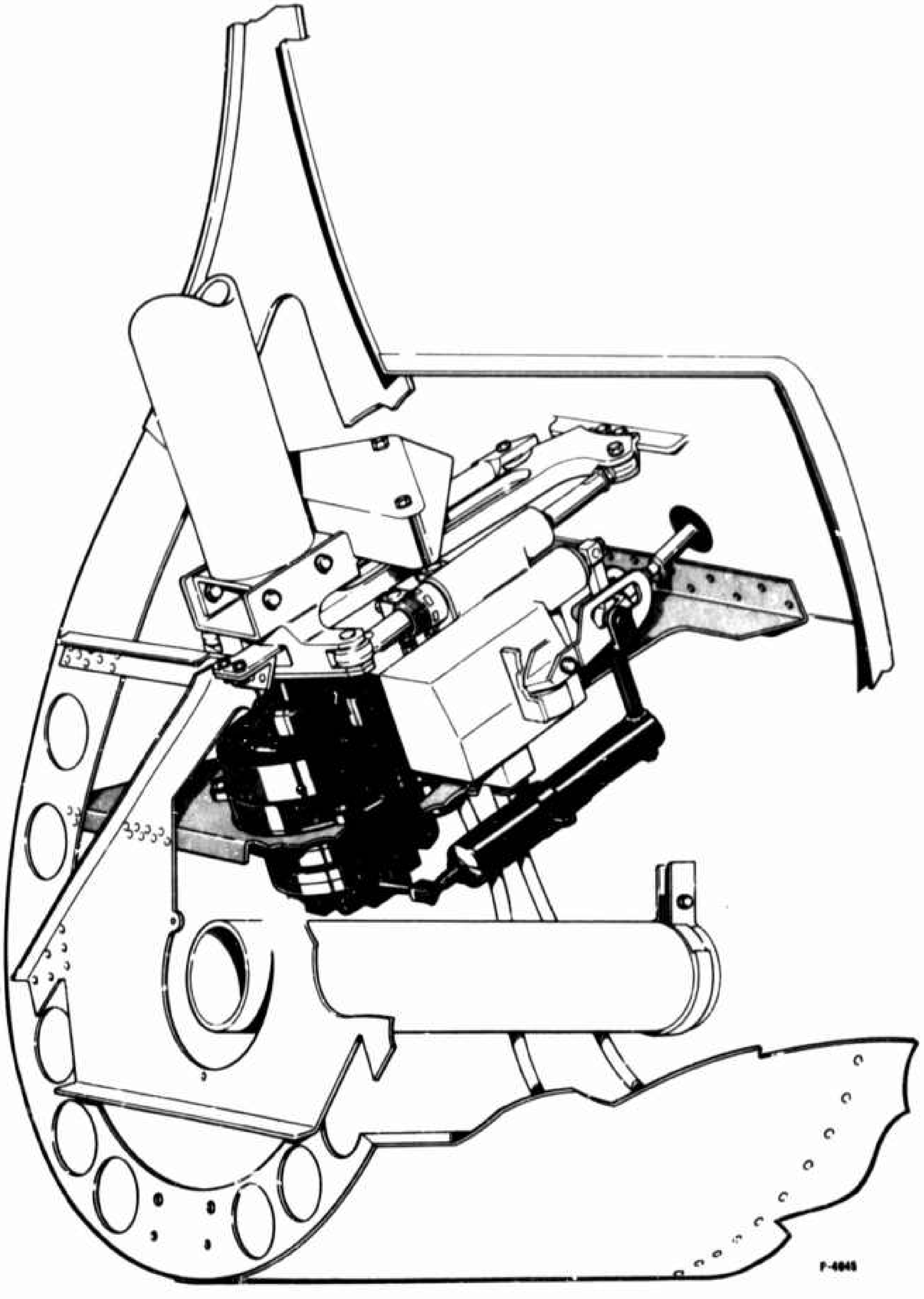


Figure 4 - Pneumatic DYNAVECTOR Rudder Actuator Installation

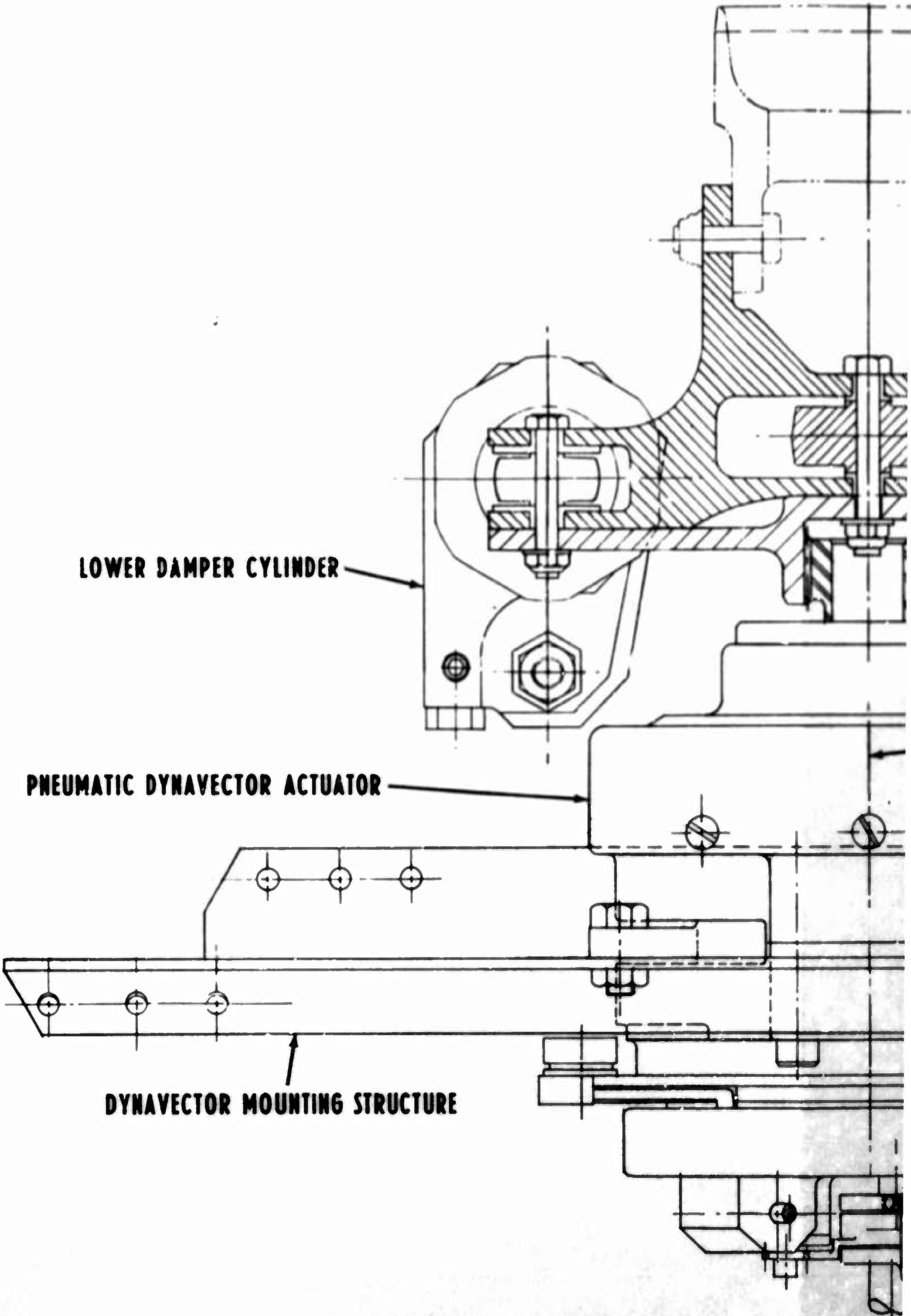
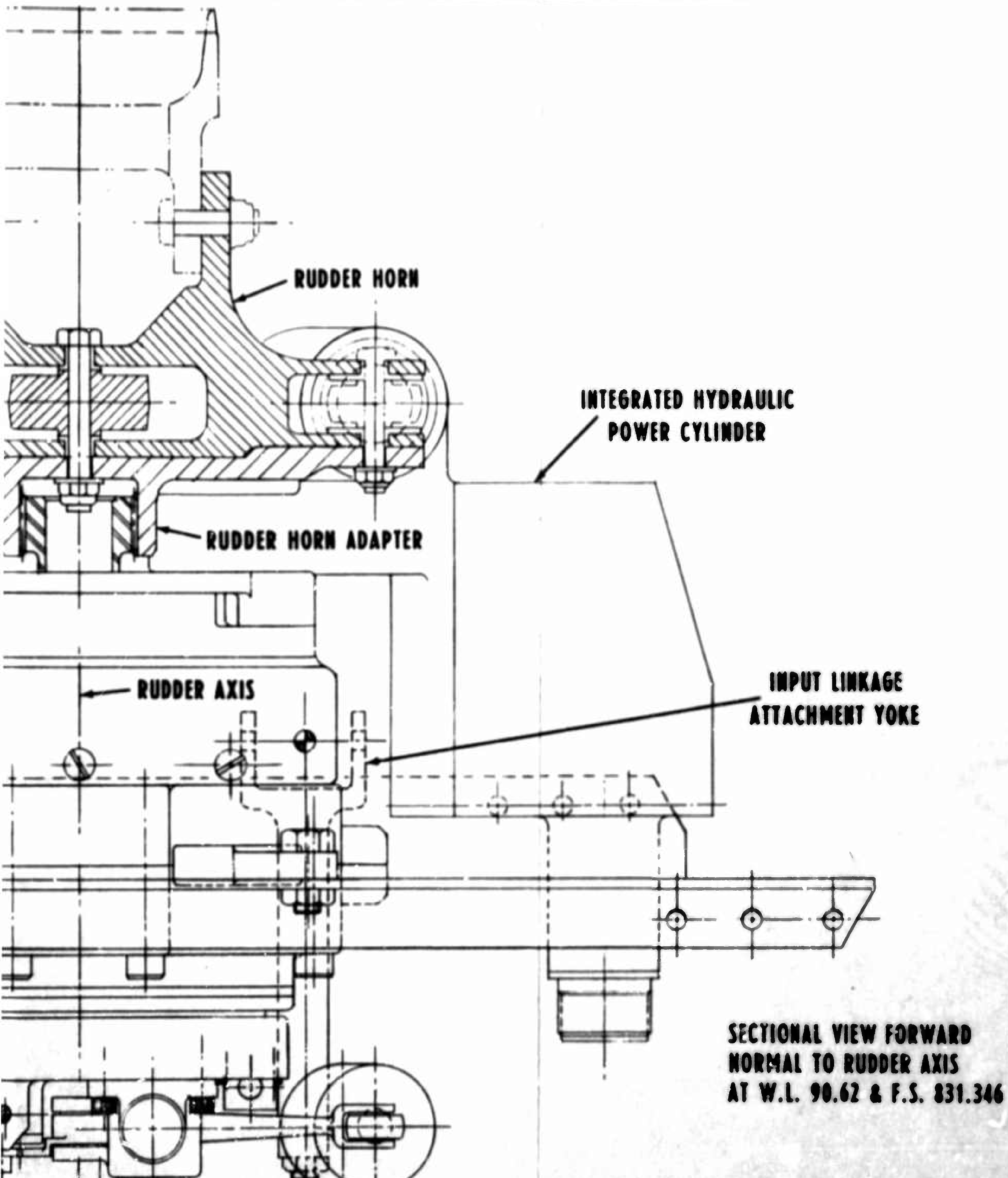


Figure 5 - DYNAVECTOR Instal
Rudder Axis at W.

/



TOR Installation Sectional View Forward Normal to
Axis at W.L. 90.62 and F.S. 831.346

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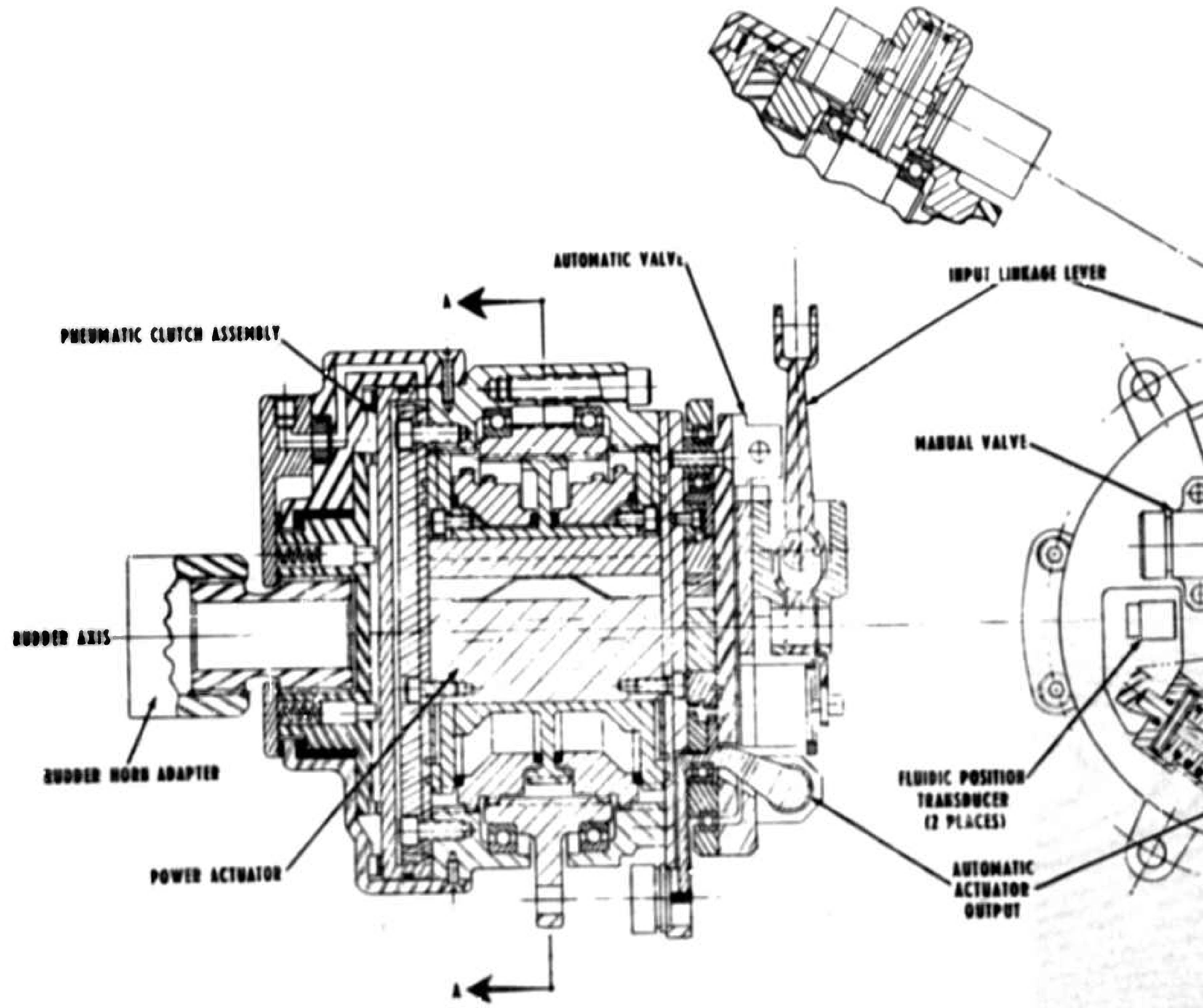

Figure 6 - DYNAVECTOR Pneumatic R

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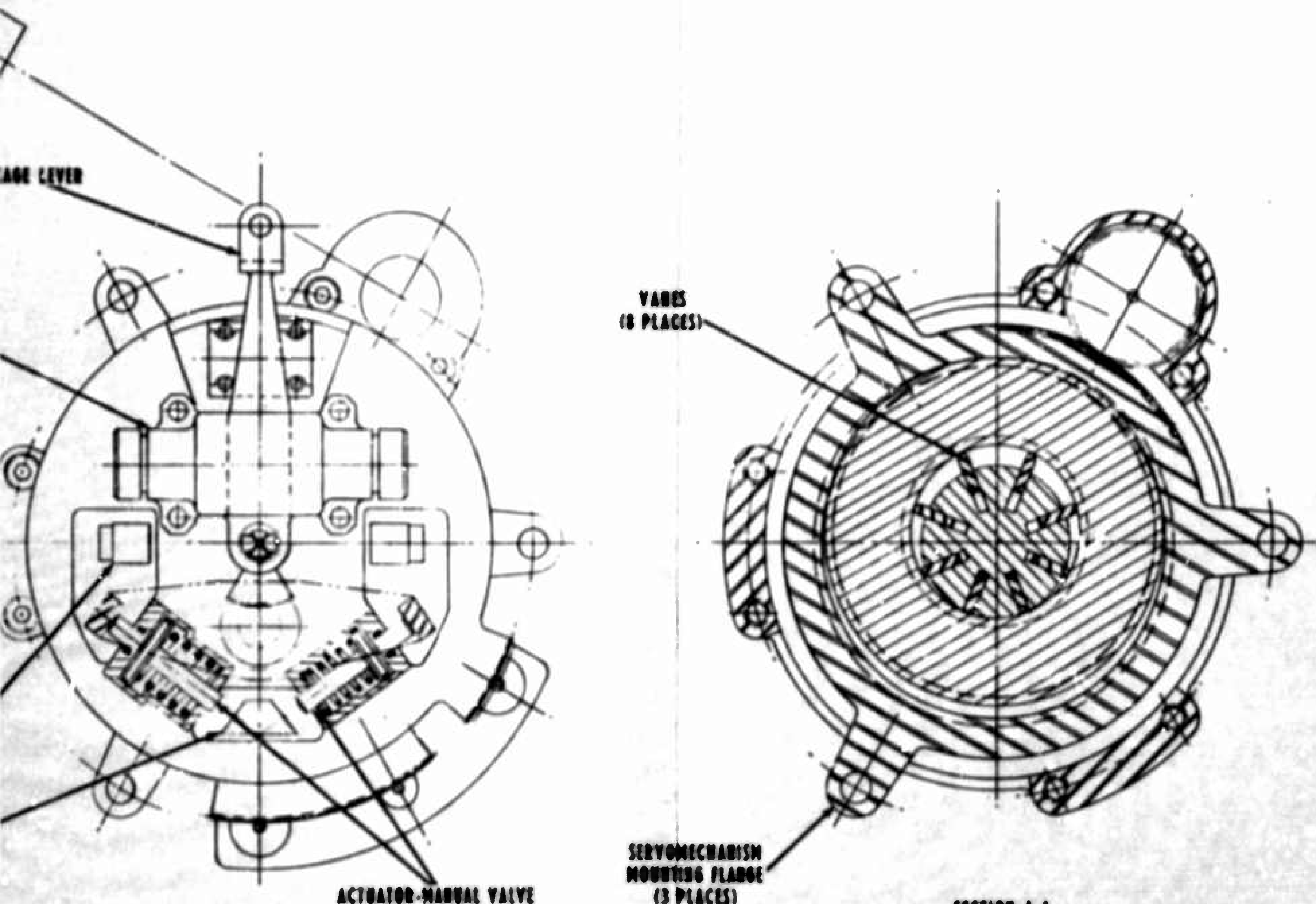


Pneumatic Rudder Actuator Assembly Drawing

2

rudder so that the rudder may be actuated in any of the following operative modes:

- Hydraulic system operative: pneumatic system shut down.
- Hydraulic system operative: pneumatic system operative, but declutched from rudder for monitor condition only.
- Hydraulic system inoperative: pneumatic system operative, and clutch engaged to rudder.

The layout of the DYNAVECTOR rudder actuation system is shown in Figure 6. The major components of the assembly consist of:

- Load limit mechanism
- Manual valve lever
- Manual valve
- Power actuator
- Single point engagement pneumatic clutch
- Rudder horn adapter
- Actuator-rudder interlock valve
- Actuator-manual valve latch
- Automatic valve
- Automatic valve amplifier
- Automatic actuator
- Fluidic position transducer
- Clutch, power supply, and latch switches
- Miscellaneous monitoring instrumentation

The functional relationships of these major components are shown schematically in Figures 7 through 10.

Figure 7 shows the manual power mode of operation.

Figure 8 shows the manual power mode of operation with stability augmentation operative.

Figure 9 shows the manual power mode of operation with stability augmentation in monitor condition only.

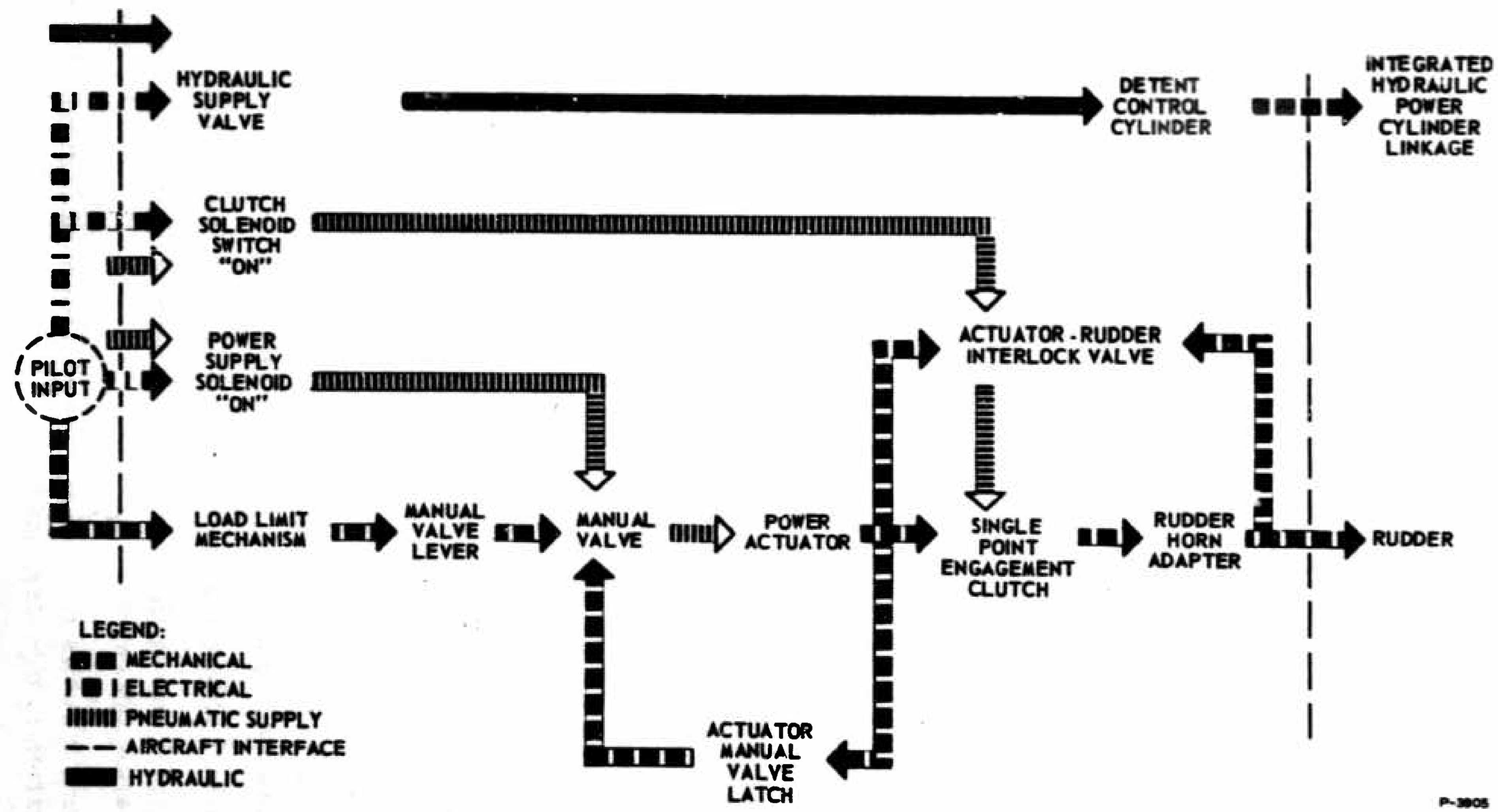


Figure 7 - Pneumatic Rudder Control System Schematic - Manual Power Mode

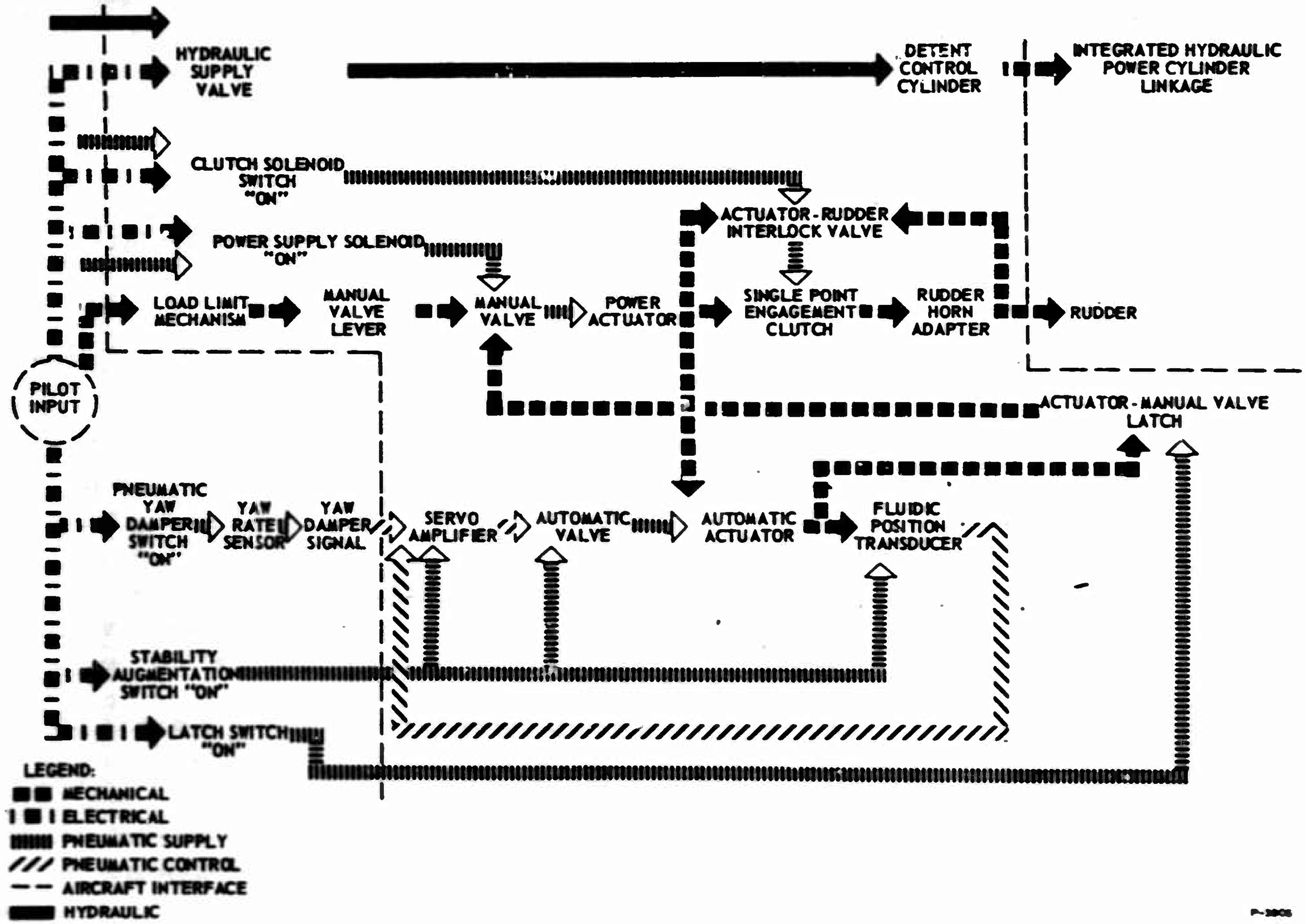


Figure 8 - Pneumatic Rudder Control System Schematic - Manual Power Mode with Stability Augmentation Operative

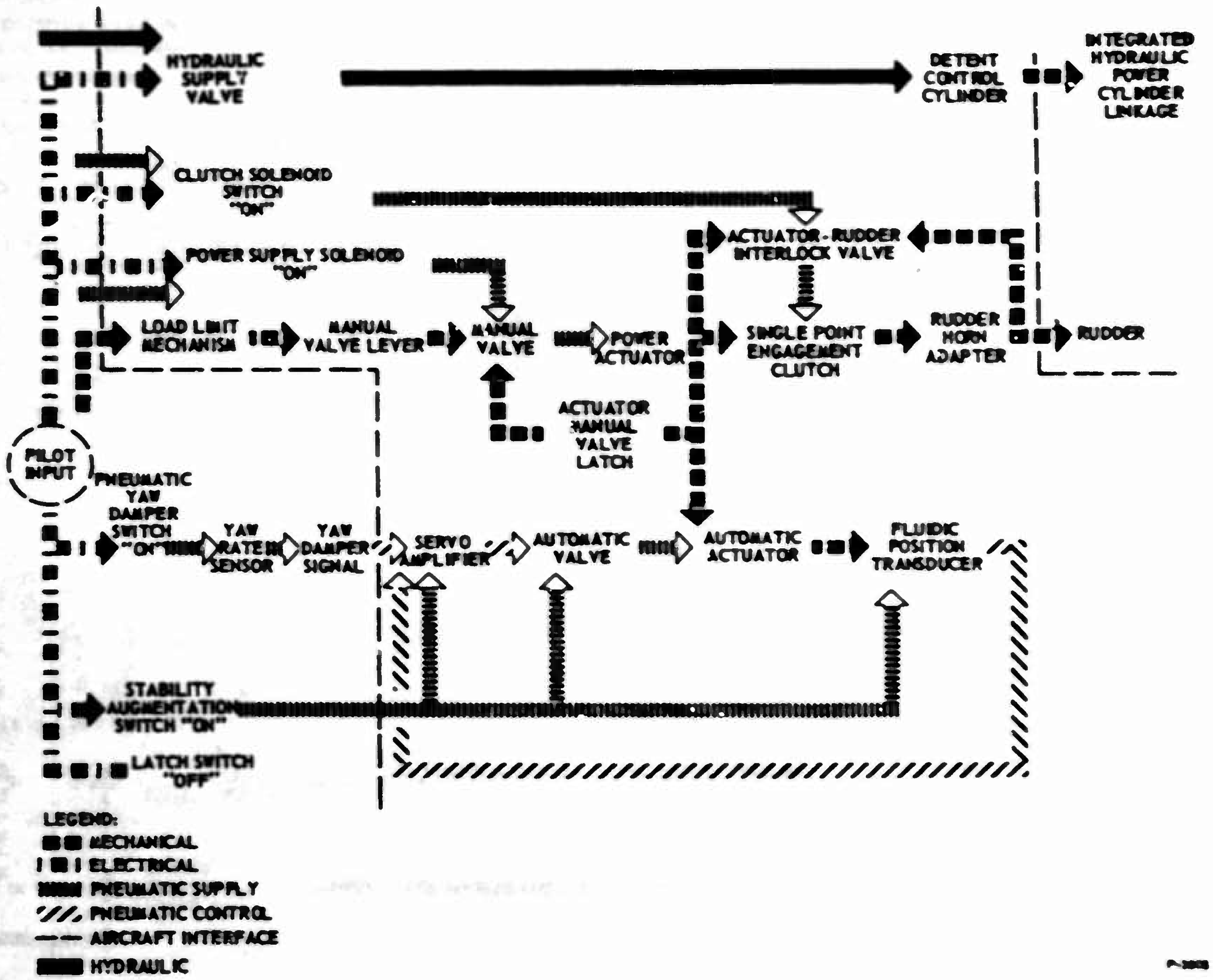


Figure 9 - Pneumatic Rudder Control System Schematic - Manual Power Mode with Stability Augmentation Monitored

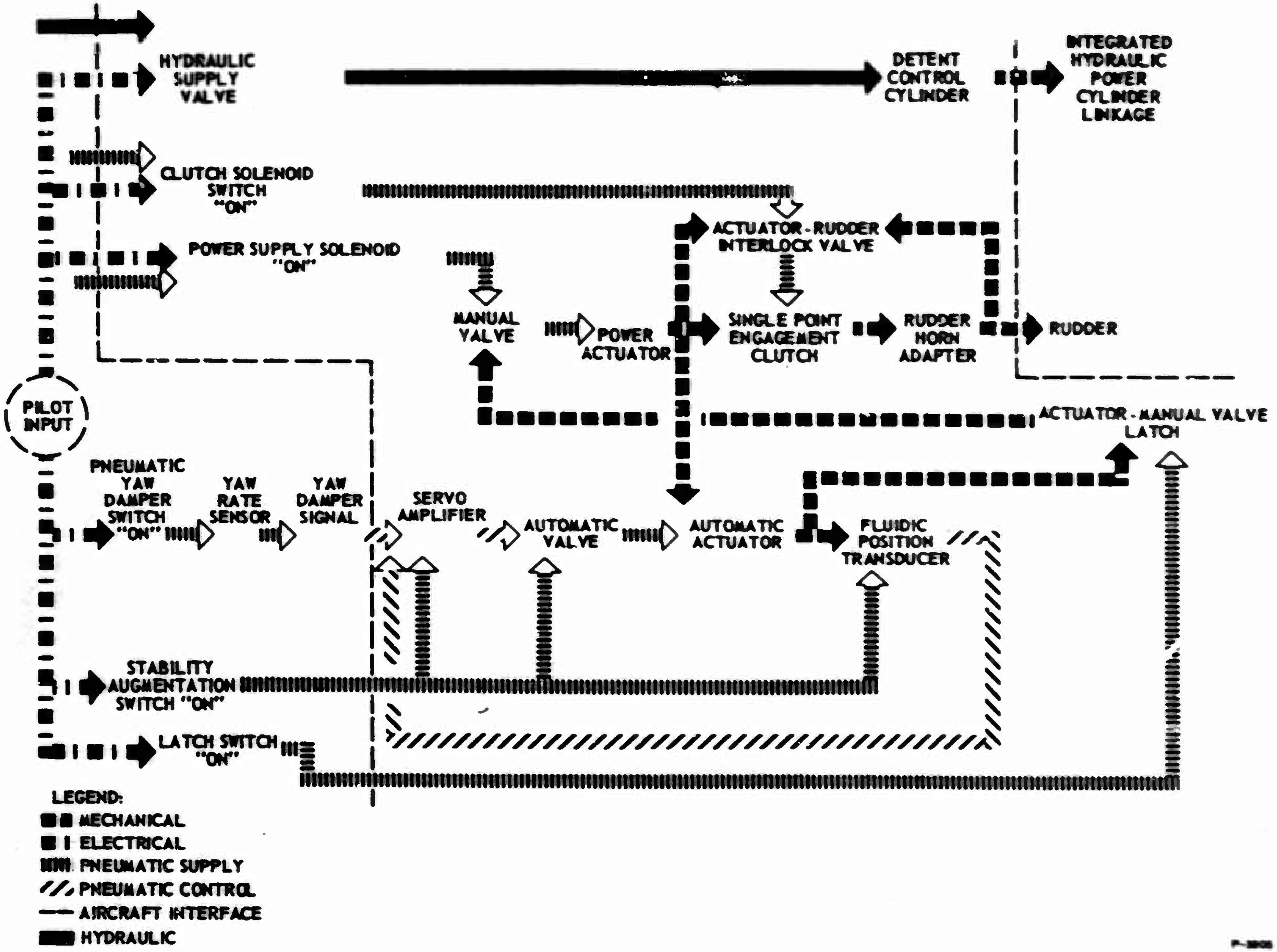


Figure 10 - Pneumatic Rudder Control System Schematic - Autopilot Operation

Figure 10 shows autopilot operation.

These operative modes are discussed in detail in Section III.

2. DESIGN REQUIREMENTS AND PERFORMANCE CHARACTERISTICS

The design requirements for the pneumatic rudder actuator servomechanism defined in the Statement of Work under Contract AF 33(615)-3309 at the start of this design study program are presented in paragraph A below. During the course of the program, refinements and modifications to the Statement of Work requirements were generated and these are summarized in paragraph B. The final flightworthy design requirements are presented in DS-747 (reference Appendix B). A summary of the performance characteristics of the flightworthy design are presented in paragraph C.

A. Statement of Work Pneumo-Mechanical Servomechanism Requirements

The following requirements are as stipulated in the subject Contract Statement of Work dated 5 May 1965.

(1) Scope:

(a) This exhibit defines the requirements for a design study leading to a pneumo-mechanical servomechanism capable of controlling an aircraft control surface.

(b) Pneumo-mechanical servomechanisms may be classified as linear or rotary depending upon the type of motion delivered to the control surface. It is desired that a linear output of the actuator be the design objective. Power to operate the unit shall be derived from bleed air of the compressor section of a turbojet engine.

(2) Objective:

(a) The objective of this study shall be the establishment of design criteria for fabricating a flightworthy pneumatic servomechanism.

(b) The study shall lead to a design which shall operate in concept with the existing aircraft hydraulic servo actuator subsystem.

(3) Applicable Documents:

(a) The following Government documents shall be used as a guide during the design study:

MIL-E-5272, Environmental Testing; Aeronautical and Associated Equipment

MIL-P-5514B, Packings; Installation and Gland Design of Aircraft Hydraulic and Pneumatic

MIL-A-8629 (AER), Airplane Strength and Rigidity

(b) Other documents which have not been covered by Item (3a) above and which have been generated by the contractor may be applied to this program.

(4) Requirements:

(a) General: The servomechanism design shall be of minimum size and weight consistent with the following requirements. Simplicity of operation and attaining the performance requirement of the specific function shall be the primary requirement.

(b) Weight: The unit shall not exceed twenty (20) pounds. This requirement may be relaxed provided the weight restriction penalizes the servomechanism's performance.

(c) Operating Conditions: The servo unit shall be designed to operate under the following conditions.

- Temperatures: Gas temperature 100°F to 600°F. Ambient temperature of -65°F to 270°F.
- Supply Pressure: Supply pressure shall vary from 50 psi to 200 psi. If it is necessary to operate at a fixed pressure level, consideration shall be given to implementing an accumulator and conventional pressure regulation subsystem.
- Altitude: Sea level to 50,000 feet.
- Flight Inertia Loads: The unit shall be structurally able to withstand without failure a 17.0g ultimate acceleration in any direction and shall operate satisfactorily without malfunction under a 12.0g acceleration in any direction.

(d) Dimensions: Figure 11 shall be used as a guide to establish the package design for the servomechanism assembly.

(e) Automatic Servovalve: The automatic servovalve shall operate with the following characteristics:

- A pneumatic input pressure differential signal of ± 5 psid full scale.
- Hysteresis limit of 1 percent.
- Natural frequency - 30 cps.
- Resolution - 1 percent full scale.
- A breakout force level of ±0.25 psid shall be a design requirement.

(f) The servomechanism in automatic mode shall provide an output which will deflect a control surface by ±5 degrees with ±0.25 degree positional accuracy.

(g) Manual Servovalve: The unit shall accept a command input by direct mechanical linkage to the pilot and a pneumatic signal from the automatic valve. It shall operate with the following characteristics:

- Static: The force applied by the pilot or automatic valve to the manual valve as seen by the manual to obtain 1.0 in/sec output velocity shall not exceed 0.5 of a pound.
- The manual valve shall be cascaded with the automatic valve so that the mechanical linkage to the pilot will track the operation of the automatic valve.
- The manual valve shall be the controlling valve of the servomotor/actuator assembly and shall be considered the primary valve.

(h) Output static force level of the servomotor shall be 3000 ± 50 pounds at the desired regulated input supply pressure.

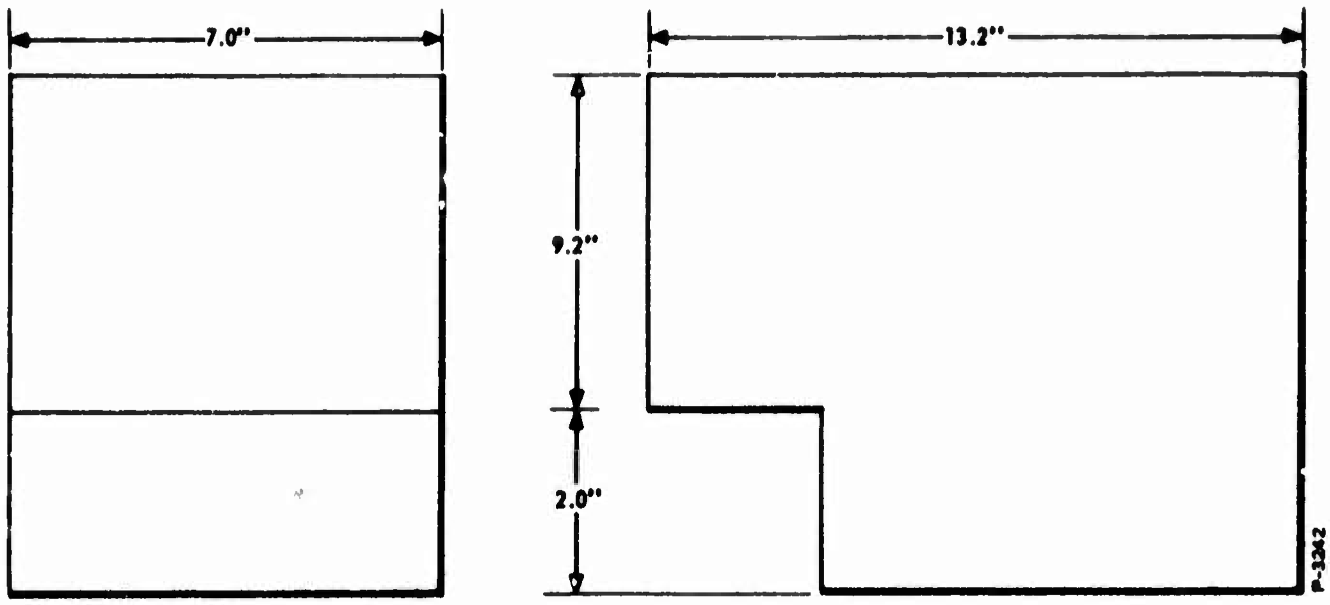


Figure 11 - Servomechanism Envelope

(i) The output motion of the servomotor/actuator assembly shall either be linear or rotary consistent with the following requirements:

- Linear manual operation output: Stroke ±1.65 inches with ±0.03 inch accuracy. Linear velocity 3.9 in/sec.
- Rotary manual operation output: ±20 degrees with ±1 degree accuracy. Angular velocity 60 deg/sec.
- Linear automatic operation output: ±0.42 inch with ±0.01 inch positional accuracy. Linear velocity 3.9 in/sec.
- Rotary automatic operation output: ±5 degrees with ±0.25 degree positional accuracy. Angular velocity 60 deg/sec.
- Maximum surface deflection velocity shall be 60 deg/sec for both manual and automatic operation.
- Maximum surface deflection acceleration shall be 150 deg/sec^2 for both manual and automatic operation.

(j) Dynamic Response: The unit under maximum loading conditions shall operate within the limits specified by Figure 12.

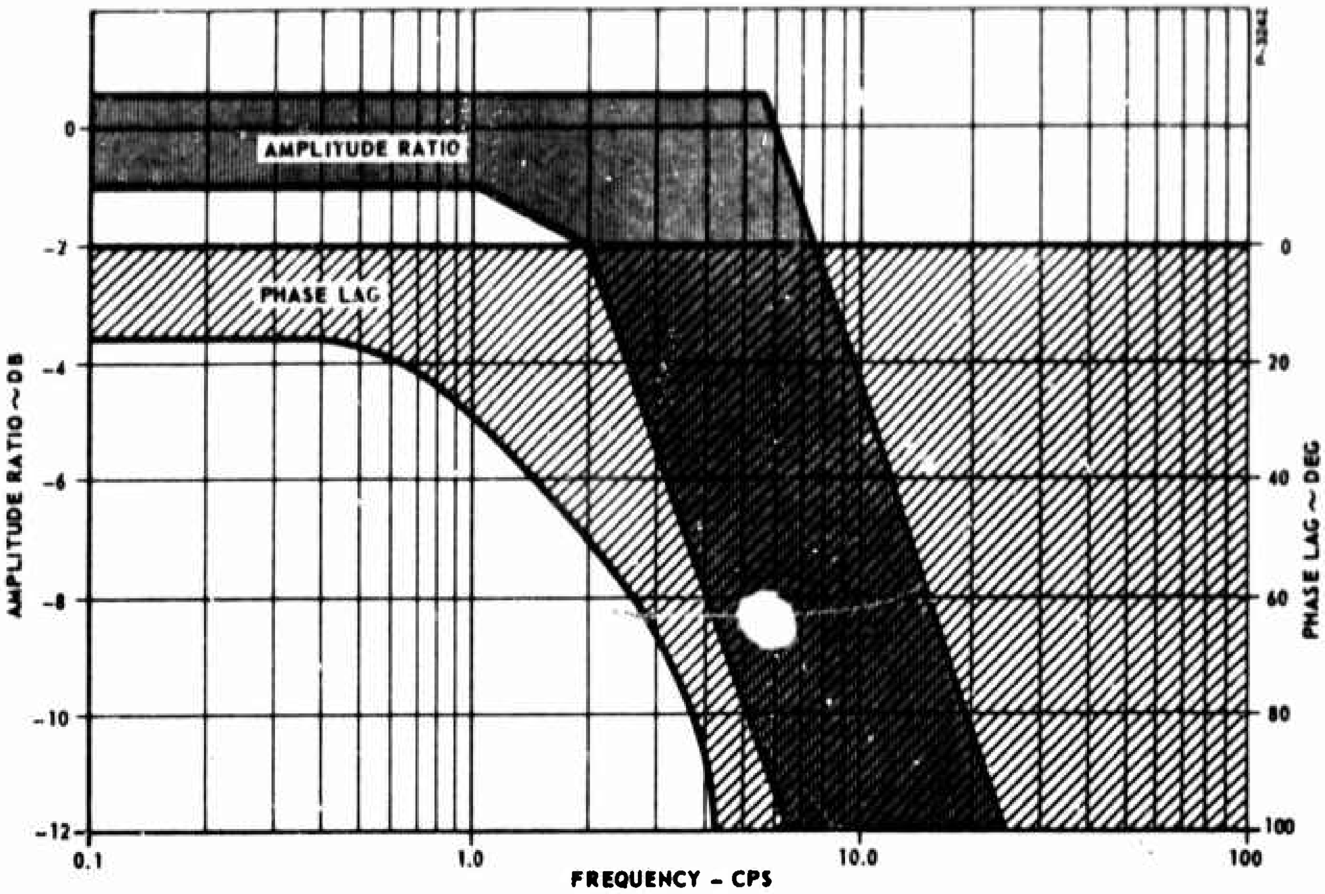


Figure 12 - Closed Loop Response of Servo Assembly

(k) Servo Actuator: The servo actuator shall deliver an output force of 2700 ± 150 pounds. It is desirable but not necessary that the output be in a linear form.

(l) Chatter and Instability: The unit shall operate smoothly without sustained chatter or instability under all operating conditions.

(m) Instrumentation: During the design of the servomechanism, provision shall be made to monitor the following parameters.

- Automatic servovalve input signal.
- Automatic servovalve position and velocity.
- Manual servovalve position and velocity.
- Manual servovalve input signal.

- Servomotor position and velocity.
- Servo actuator position and velocity.

(n) The pneumatic servomechanism shall be operated in parallel with the existing hydraulic servomechanism.

- Provisions for disengagement of the pneumatic servomechanism package shall be made when operation in the hydraulic mode is commanded.
- Additionally, a provision shall be made for disengagement when a hard-over signal in the automatic mode is experienced by the pneumatic servo package.
- A mechanical position follower shall be included in the design to prevent transients from occurring when switching modes of operation, e.g., pneumatic to hydraulic and hydraulic to pneumatic, and allow the passive servo package to follow the active servo package.

B. Modifications and Refinements of Requirements

During the course of this program, modifications and refinements to the original Statement of Work design requirements were required. These changes were either a result of a redefinition of the interface characteristics by Wright-Patterson Air Force Base or analyses conducted by Bendix and consist of the following:

(1) MIL-A-8629(AER), Airplane Strength and Rigidity, superseded by MIL-A-8860 through 8870.

(2) Gas temperature range of 100°F to 600°F revised to 100°F to 450°F.

(3) Dimensions: Figure 11 modified to allow concentric mounting of rotary actuator to rudder axis.

(4) Automatic servovalve input pressure differential signal of ±5 psid revised to ±2 psid.

(5) Pilot tracking of automatic valve. Automatic valve tracking will be provided by monitoring of electrical potentiometer signal rather than mechanical linkage to pilot rudder pedals.

(6) Maximum allowable fuel consumption requirement established as 0.018 lb/sec at 450°F gas temperature.

(7) Design life of 3000 hours established with duty cycle as defined per DS-742 (see Appendix A).

(8) Instrumentation requirements of manual and automatic valve to monitor valves' velocity and position modified to system providing direct read-out of position of valves. Velocity of valves will be obtained by differentiating valve displacement-time data from flight recorder.

C. Flightworthy Design Requirements

The flightworthy pneumatic DYNAVECTOR rudder actuator design requirements are defined in DS-747 (shown in Appendix B). A summary of these design requirements and the performance characteristics of this system are presented in Table I.

3. INTERFACE REQUIREMENTS

The interface requirements for the parallel installation of the pneumatic DYNAVECTOR rudder actuator may be categorized as electrical command, mechanical command, structural, hydraulic system modifications, pneumatic, and instrumentation.

A. Electrical Command Requirements

The electrical command interface requirements necessary to allow the pilot to activate the pneumatic rudder actuator system are:

- Clutch solenoid switch
- Manual valve power supply switch
- Actuator-manual valve latch switch
- Stability augmentation switch

The yaw damper switch and stability augmentation switch may be a single switch. However, in the event that the pilot chooses to monitor yaw rate sensor signals before monitoring the stability augmentation output of the automatic actuator, separate switching functions must be provided.

Table I - Summary of Flightworthy Pneumatic DYNAVECTOR Rudder Actuator Design and Performance Characteristics

DYNAVECTOR Model	PH-370-B1
Transmission Ratio	370:1
Stall Torque	10,200 lb-in
No-Load Speed	60 deg/sec
Maximum Horsepower	0.405 hp
Weight	14 lbs
Design Life	3000 hours
Maximum Fuel Consumption @450°F	0.018 lb/sec
Supply Pressure	50 psig
Gas Temperature	100°F to 450°F
Manual Input	1.65 in (±25 deg rudder motion)
Automatic Input	±2 psid @ 15 psig
Automatic Output (Relative to Manual Position)	±5 deg
Power Actuator Response to Manual Inputs (-3db point)	25 cps
Power Actuator Response to Stability Augmentation (-3db point)	6.2 cps
Automatic Valve Response (-3db point)	44 cps

B. Mechanical Command Requirements

The only form of mechanical command input required for the pilot to operate the pneumatic rudder actuator is rudder pedal movement via the existing feel cylinder and bellcrank-linkage system. The linkage input to the DYNAVECTOR actuator manual valve is provided by a yoke mounting of the DYNAVECTOR actuator linkage to the integrated power cylinder linkage input point.

C. Structural Interface Requirements

The structural interface requirements consist of the attachment of the rudder horn adapter to the rudder horn and the attachment of the DYNAVECTOR actuator mounting structure to the aircraft bulkheads at fuselage sections 832.36 and 814.90.

D. Hydraulic System Modifications

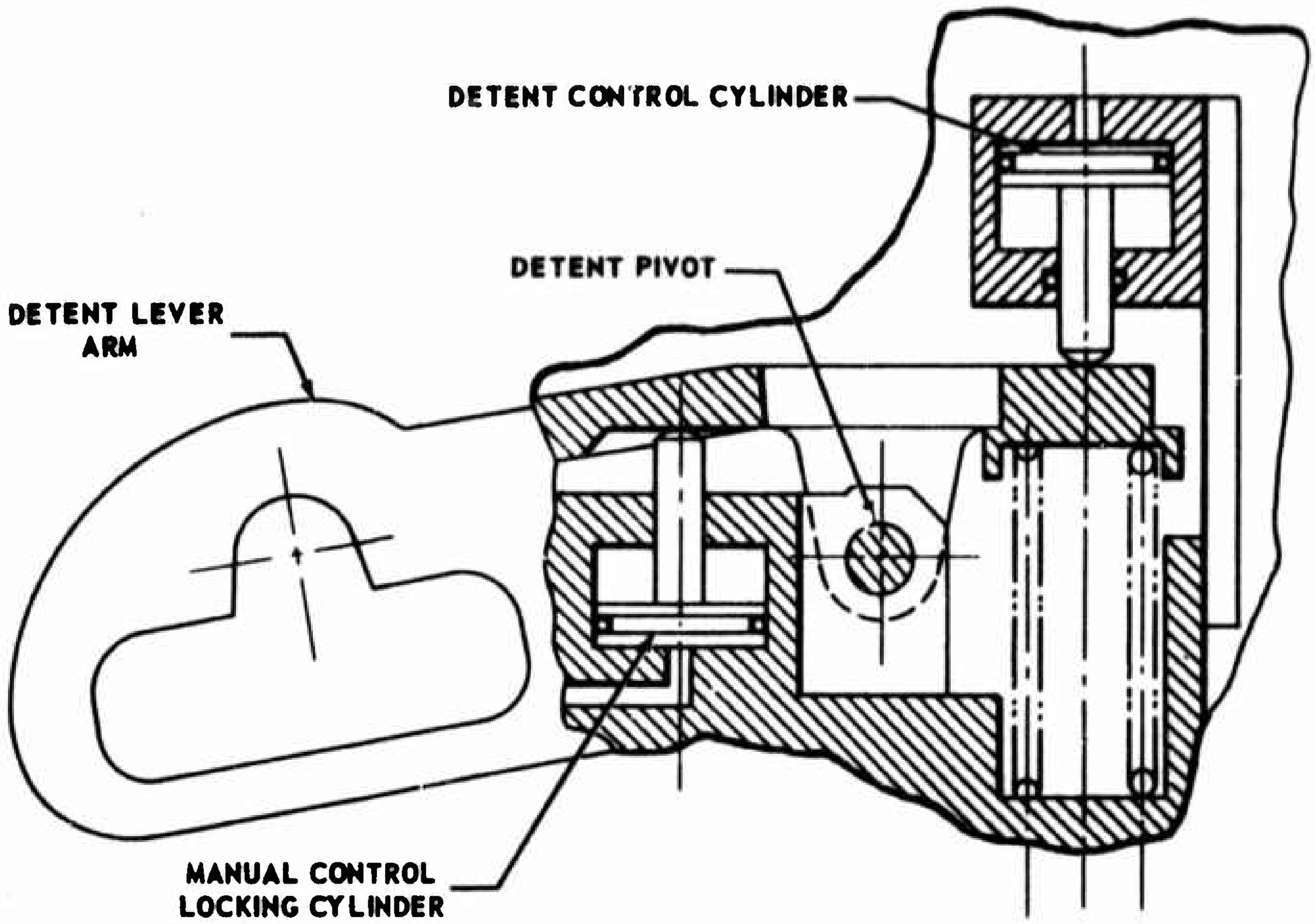
There are three modifications required on the existing hydraulic system, two of which are required because of physical interference of hydraulic lines, while the third was necessitated from operational considerations.

The two interference modifications are considered minor as they merely require a rerouting of hydraulic lines.

The hydraulic supply and return lines to the integrated power cylinder interfere with the DYNAVECTOR actuator as shown in Figure 2. Interference may be eliminated by routing these lines outside of the DYNAVECTOR actuator package diameter. The location of the attachment points for these lines need not change.

The hydraulic line from the fuel vent solenoid valve to the fuel vent valve actuator interferes with the DYNAVECTOR envelope at the location where the hydraulic line mounts to the vent valve actuator. This interference may be eliminated by providing a vent valve actuator manifold block with the hydraulic line intake located off of the longitudinal centerline of the aircraft.

The third modification of the hydraulic system is required to allow shut-off of hydraulic rudder control and subsequent control by the pneumatic actuation system. The hydraulic supply pressure must be shut-off from the integrated power cylinder to allow pneumatic system operation. Such a shutdown is equivalent to a utility hydraulic system failure, and the integrated power cylinder would normally respond so as to allow the pilot to manually drive the rudder. The cylinder bypass valve would open, thereby preventing the cylinder from becoming hydraulically locked, and the manual linkage input point would become spring-loaded into a detent position, thus allowing the pilot to move the cylinder and rudder as a solid link directly by foot pedal displacements. Detent engagement can be prevented upon intentional hydraulic pressure shutdown by pilot activation of a pneumatic cylinder package mounted to the integrated cylinder detent lever as shown in Figure 13. The switching



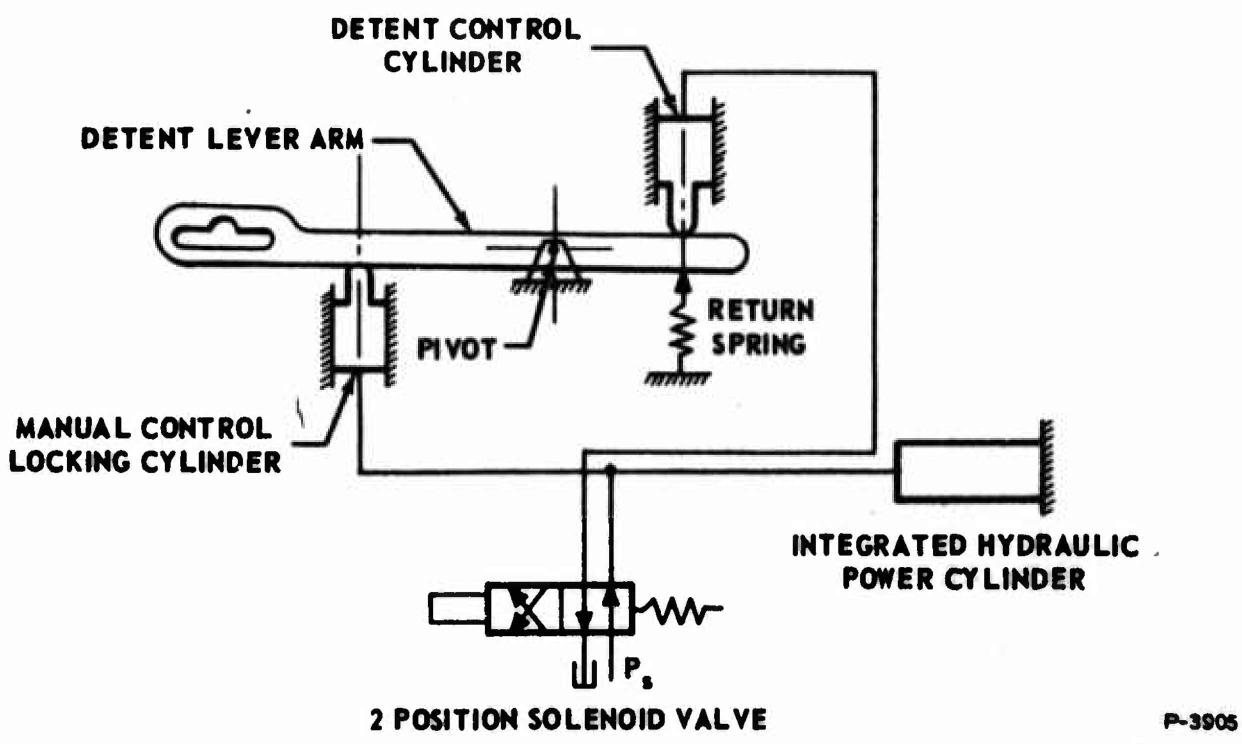


Figure 13 - Detent Control Cylinder Design

function would be integrated with the clutch solenoid switch so that hydraulic linkage detent would be prevented only when the DYNAVECTOR actuation system was clutched to the rudder. The detent control cylinder as shown in Figure 13 would be normally spring-loaded out of engagement, and only upon chamber pressurization would the piston be actuated to hold the hydraulic linkage out of detent.

E. Pneumatic Interface Requirements

An extensive analysis of the anticipated power supply requirements of the DYNAVECTOR actuator was conducted during this program and is detailed in Section III, paragraph 6, of this report. The results of this power consumption study, combined with a review of the characteristics of the available power supply afforded by the JT3 compressor bleed air, indicate that the flight-qualified pneumatic DYNAVECTOR actuator system would require a maximum consumption of 0.018 lb/sec at a 450°F supply temperature. This consumption requirement is less than 2 percent of the mass flow available when compressor bleed pressures are at the minimum required operating level of 50 psig.

The temperature of the pneumatic supply, as monitored during flight tests, was found to range between 100°F and 600°F. Precooling of this bleed air will be required so as to limit the gas temperature to 450°F maximum before porting to the rudder actuator interface.

F. Instrumentation Interface Requirements

The instrumentation requirements necessary for monitoring the operation of the pneumatic actuation system consist of both electric and pneumatic read-out signals. The installation of these monitoring devices is shown in Figure 6. The signals to be generated are:

(1) Power actuator position relative to airframe (electrical dual ganged potentiometer).

(2) Automatic actuator position relative to power actuator (pneumatic pressure differential signal).

(3) Automatic actuator position relative to power actuator (electrical)

(4) Manual valve body and input linkage pivot relative to power actuator (electrical) position.

(5) Input linkage position relative to power actuator (electrical dual ganged potentiometer).

(6) Automatic valve spool position relative to automatic valve body (electrical).

(7) The signals of items (1) and (5) may be summed to provide a signal of input linkage position relative to airframe.

(8) Since the input linkage and manual valve spool are an integral assembly, the signals of items (4) and (5) may be summed to provide a signal of manual valve spool position relative to valve body. This summed signal and item (6) may both be differentiated with respect to time to obtain valve spool velocities.

(9) Power actuator velocity relative to airframe.

(10) Input linkage limit switch to signal mechanical input to pneumatic system is in phase with hydraulic system.

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SECTION II

GUIDELINES AND ASSUMPTIONS

1. TORQUE-SPEED REQUIREMENTS

A. Summary

The torque-speed requirements for the pneumatic rotary DYNAVECTOR actuator must be developed to permit proper sizing of the actuator.

The Statement of Work requirements, as summarized in Section I, require a linear actuator output force of 2700 ± 150 lb, maximum velocity capability of 60 deg/sec, and maximum acceleration limit of 150 deg/sec^2. The performance characteristics of the hydraulic cylinder presently installed are used as a basis for establishing the DYNAVECTOR requirements and are developed from information presented in the McDonnell Aircraft Corporation Specification Control Drawing for Integrated Power Control Cylinder (Rudder) #20-91023. The stall torque requirement is found to be 10,200 lb-in at full rudder deflection using the maximum output force specified in the above document at a differential pressure of 3,000 psi. The force-speed characteristic of the hydraulic cylinder, developed from the flow information and the above output force information, approximates a straight line similar to the DYNAVECTOR actuator steady-state torque-speed characteristic with a torque-speed characteristic with a stall torque of 10,200 lb-in and a no-load speed of 60 deg/sec. The DYNAVECTOR actuator must be able to drive a spring type load in oscillatory motion for the following most critical conditions:

1 - Oscillations of ±20-degree amplitude about zero (or null).

2 - Oscillations of ±5-degree amplitude at a maximum acceleration of 150 deg/sec^2 about a bias displacement of 20 degrees.

The actuator is found capable of meeting the above requirements when driving a torsional spring load having a linear spring rate of 270 lb-in/deg. The torque-speed characteristics of the system and the actuator are summarized in Figure 21.

B. Nomenclature

A_p - Effective piston area of the hydraulic cylinder (in^2)

C_d - Orifice discharge coefficient

F - Hydraulic cylinder output force (lb)

P_s - Supply pressure (psia)

Q - Valve flow or flow through hydraulic cylinder (in^3/sec)

r - Torque arm (in)

t - Time (sec)

T_1 - Torque (lb-in)

$\dot{Z}$ - Piston velocity (in/sec)

ΔP - Pressure differential across hydraulic cylinder piston (psi)

θ - Angular displacement (degrees)

$\dot{\theta}$ - Angular velocity (deg/sec)

$\ddot{\theta}$ - Angular acceleration (deg/sec^2)

θ_o - Amplitude of displacement (degrees)

ρ - Fluid mass density ($lbs\text{-}sec^2/in^4$)

ω - Frequency of oscillation (rad/sec)

C. Analysis

The present hydraulic rudder control actuator is capable of producing a force of 2,700 ± 150 pounds at a differential pressure of 3,000 psi. This is assumed to be the stall torque requirement for the DYNAVECTOR actuator. The stall torque is assumed to occur at full rudder deflection where the aerodynamic loading and the resultant hinge moment are normally maximum. The torque output of the hydraulic cylinder versus rudder displacement with a differential pressure of 3,000 psi across the piston is plotted in Figure 14. The torque capability drops off with rudder position because of the reduction in effective torque arm as the hydraulic cylinder rotates with respect to the airplane axis. For sizing the pneumatic rudder control actuator, the torque at the maximum displacement is selected and a linear variation of load

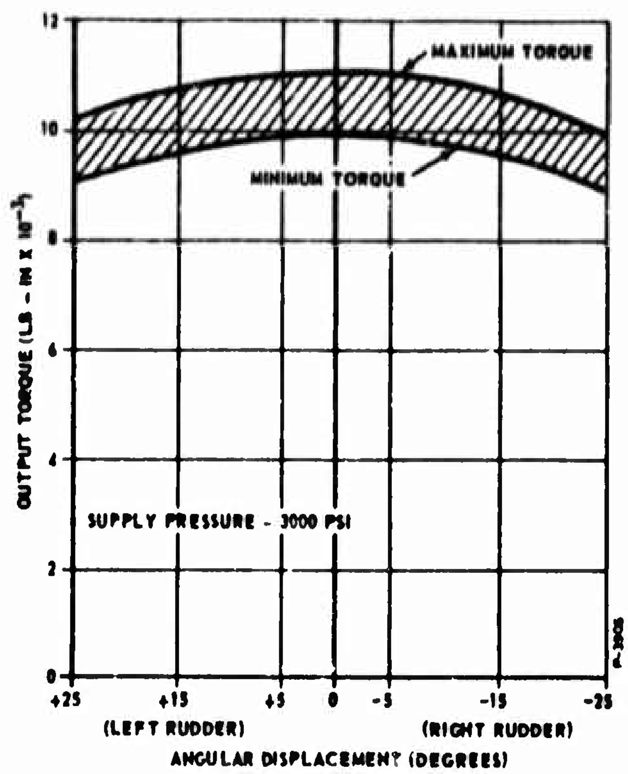

Figure 14 - F101B Hydraulic Rudder Control Output Torque Versus Rudder Position

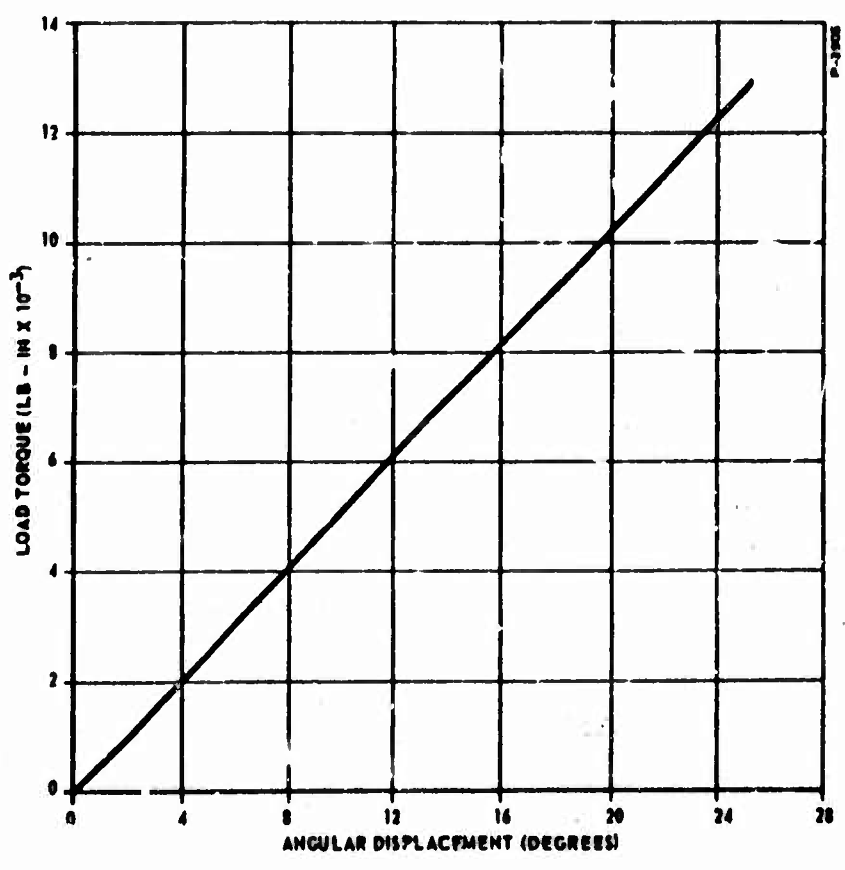

Figure 15 - Load Torque-Displacement F101B Rudder-Spring Rate 510 lb-in/degree

torque from zero at null to 10,200 lb-in at 20 degrees deflection is assumed. A plot of this load torque is presented in Figure 15.

The torque-speed characteristic of the hydraulic rudder control actuator is obtained as a guide for establishing the performance requirements for the pneumatic DYNAVECTOR actuator. For a hydraulic cylinder, the output force is expressed as

$$F = \Delta P A_p \tag{1}$$

and the linear velocity of the piston as

$$\dot{Z} = \frac{Q}{A_p} \tag{2}$$

To convert the output force and linear velocity to torque and angular velocity, respectively, the relationships below are used:

$$\dot{\theta} = \frac{\dot{Z}}{r} \tag{3}$$

$$T_l = Fr \tag{4}$$

A constant torque arm is assumed, although this is not the case in actual operation. However, the variation is deemed not large enough to affect the torque-speed curve. The flow through the cylinder is the same as the flow through the servovalve. The following assumptions are made in determining the flow:

- Constant pressure source
- Series circuit in the valve
- Zero or positive lap in the valve
- The valve ports are rectangular and identical so that the valve is symmetrical.

The flow is determined from the following equation:

$$\Delta P = P_s - \frac{2Q^2}{g^2} \tag{5}$$

where

$$g = A_p C_d \sqrt{\frac{2}{\rho}} .$$

At stall, the flow is zero and, therefore, the supply pressure is equal to the differential pressure in the cylinder, or 3,000 psig. The maximum flow for a differential pressure of 900 psi is 1.125 gpm or 4.33 in^3/sec. (Ref: McDonnell Specification Control Drawing #20-91023, Sheet 25). To obtain the value for g, substitute the above numbers in equation (5).

$$g^2 = \frac{2(4.33)^2}{3,000 - 900} = 0.0179$$

or

$$g = 0.1338 .$$

The effective piston area is obtained from equation (1) using the median value for output force.

$$A_p = \frac{F}{\Delta P} = \frac{2,700}{3,000} = 0.9 \text{ in}^2$$

The output torque is obtained by multiplying the output force by the torque arm of 3.9 inches.

$$T_1 = Fr = \Delta P A_p r = (0.9)(3.9)\Delta P = 3.51 \Delta P \qquad (6)$$

and the speed is obtained from

$$\dot{\theta} = \frac{360}{2\pi r} \dot{Z} = \frac{360}{2\pi r} \left(\frac{Q}{A_p}\right) = \left(\frac{Q}{(0.9)(3.9)}\right) \frac{360}{2\pi} = 16.3\ Q \qquad (7)$$

Substituting the numerical values 3000 and 0.1338 for P_s and g, respectively, in equation (5) yields

$$Q \doteq 0.0946 \sqrt{3,000 - \Delta P} \qquad (8)$$

and therefore, combining equations (7) and (8)

$$\dot{\theta} = 1.54 \sqrt{3,000 - \Delta P} . \qquad (9)$$

Various values of ΔP are substituted into equations (6) and (9), and the resultant torque-speed curve is plotted in Figure 16.

The operational mode of the DYNAVECTOR actuator must be considered in establishing performance characteristics. Actuator output is limited to oscillations of ±20 degrees for manual mode with an additional ±5 degrees on automatic mode. Therefore, a steady-state

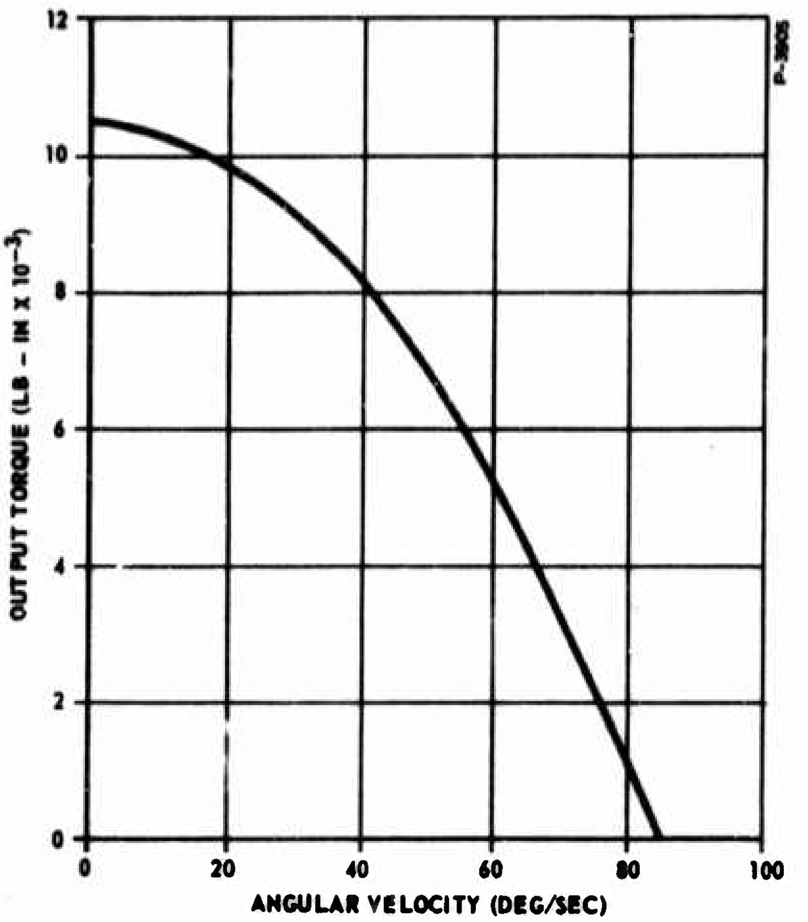

Figure 16 - Torque-Speed Characteristic - Hydraulic Actuator

analysis is not applicable and the system performance will be established for oscillatory motion only, assuming a harmonic input. For angular harmonic motion, the following equations are applicable:

$$\theta = \theta_o \cos \omega t \tag{10}$$

$$\dot{\theta} = -\omega \theta_o \sin \omega t \tag{11}$$

$$\theta = -\omega^2 \theta_o \cos \omega t . \tag{12}$$

The maximum control surface deflection velocity of 60 deg/sec is assumed to occur at the null position ($\theta = 0$) and the maximum acceleration of 150 deg/sec^2 is assumed to occur at the full deflection position of 20 degrees.

To obtain the load torque-speed characteristic for the DYNA-VECTOR actuator with angular harmonic motion, it is necessary to obtain the velocity-displacement profile and then combine this with the load torque-displacement profile. The system is assumed to be velocity-limited at 60 deg/sec and, by combining equations (10) and (11), the velocity-displacement profile for a number of frequencies is established and plotted in Figure 17. The load torque-speed characteristics plotted in Figure 18 are the result of combining the load torque-displacement and velocity-displacement profiles of Figures 15 and 17. The maximum accelerations are computed for various frequencies by combining equations (11) and (12) and all are found to be in excess of 150 deg/sec^2, and are plotted in Figure 19. Combining these curves with the load torque-displacement of Figure 15 produces the load torque-speed characteristics of Figure 20.

In addition to oscillation about the rudder zero position, the actuator in automatic mode must be capable of oscillations of ±5 degrees amplitude about any manual setting, up to and including 20 degrees. The maximum power is required for automatic operation of maximum amplitude at 20-degree rudder position, and curve F of Figure 20 represents the load torque-speed characteristic for this operational mode. Curve A of Figure 20 is shown only to indicate the frequency and amplitude that would be necessary to obtain a maximum velocity of 60 deg/sec; this amplitude is 24 degrees, which is greater than that stipulated for manual operation.

The curves shown in Figure 20, with the exception of curve A, must fall under the curve defining the steady-state torque-speed characteristic of the DYNAVECTOR actuator, which is assumed to be a straight line with a no-load speed of 60 deg/sec. If it is desirable that the system operate at maximum acceleration at all times, the minimum actuator steady-state torque-speed curve must be tangent to curve B; however, this results in an actuator with a stall torque capability of 25,200 in-lbs, far greater than the present hydraulic cylinder and the stipulated stall torque requirement. If the performance is reduced slightly at 20 degrees amplitude, the system performance in automatic mode and at a bias of 20 degrees is the limiting parameter. For this operational capability, the actuator torque-speed curve must be tangent to curve F in Figure 20, resulting in a stall-torque capability of 19,400 in-lbs, again almost double the output of the present actuator. The load torque requirement must therefore be lowered so that the actuator is not overdesigned.

An actuator steady-state torque-speed curve having a stall torque of 10,200 in-lbs and a no-load speed of 60 deg/sec (curve C of Figure 21), which approaches the torque-speed characteristic of the

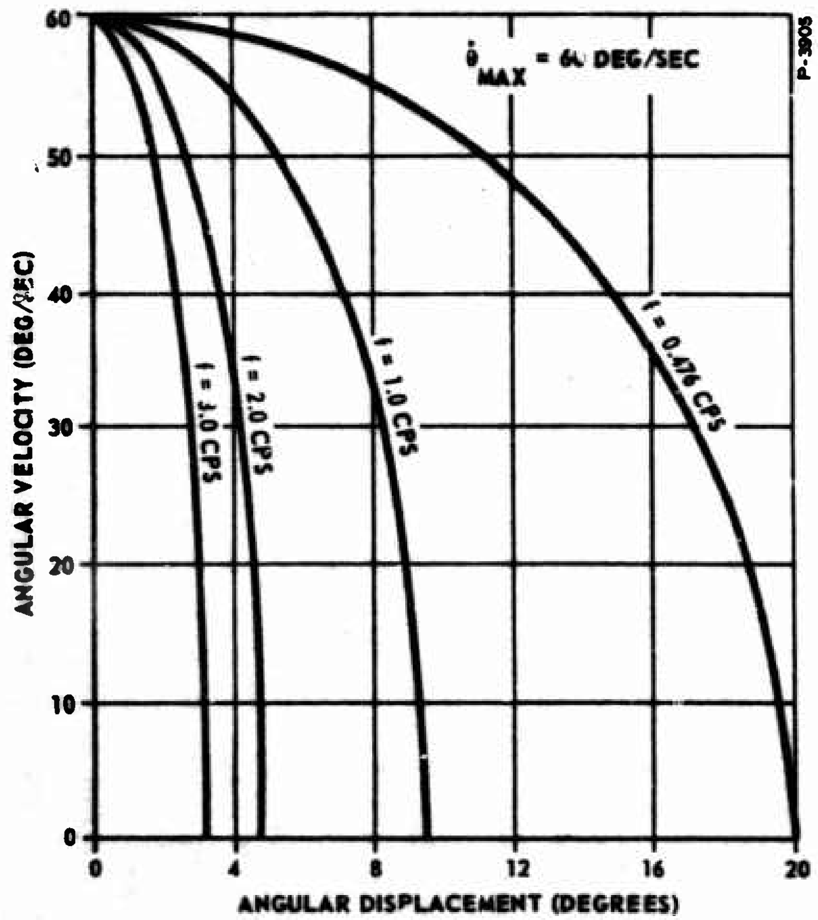

Figure 17 - Velocity - Displacement Curves
F101B Rudder - Velocity Limited

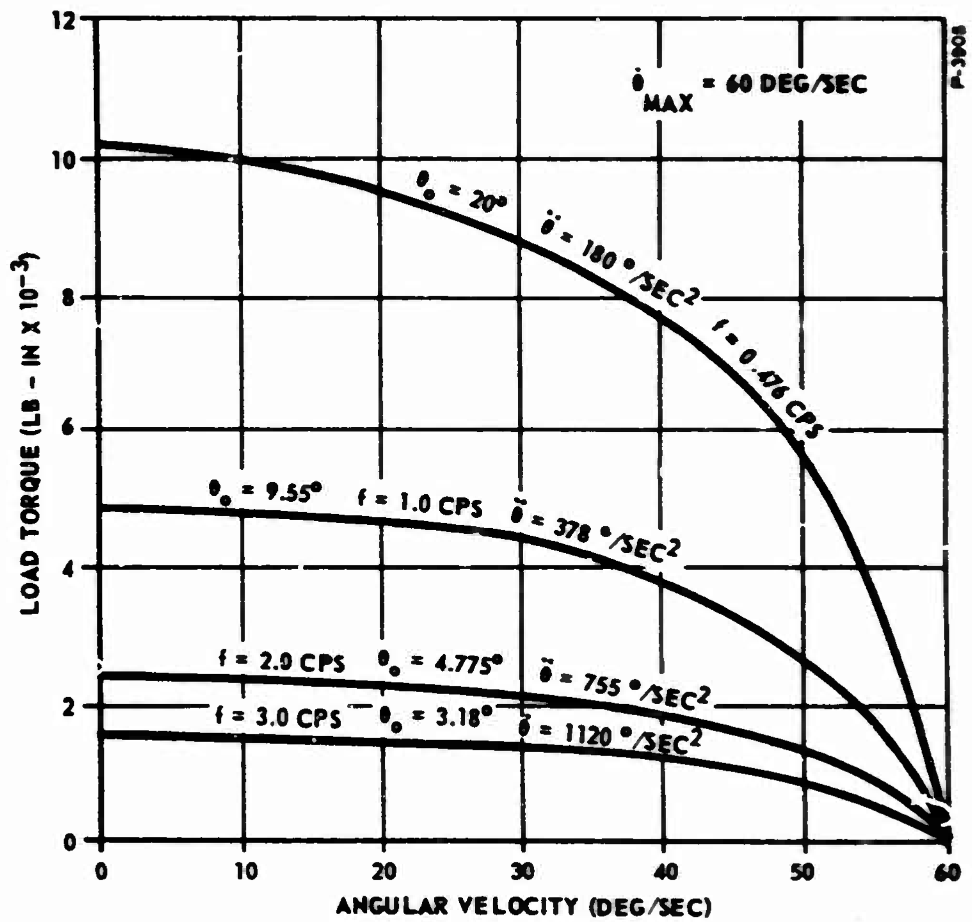

Figure 18 - Load Torque-Speed Curves
Harmonic Motion

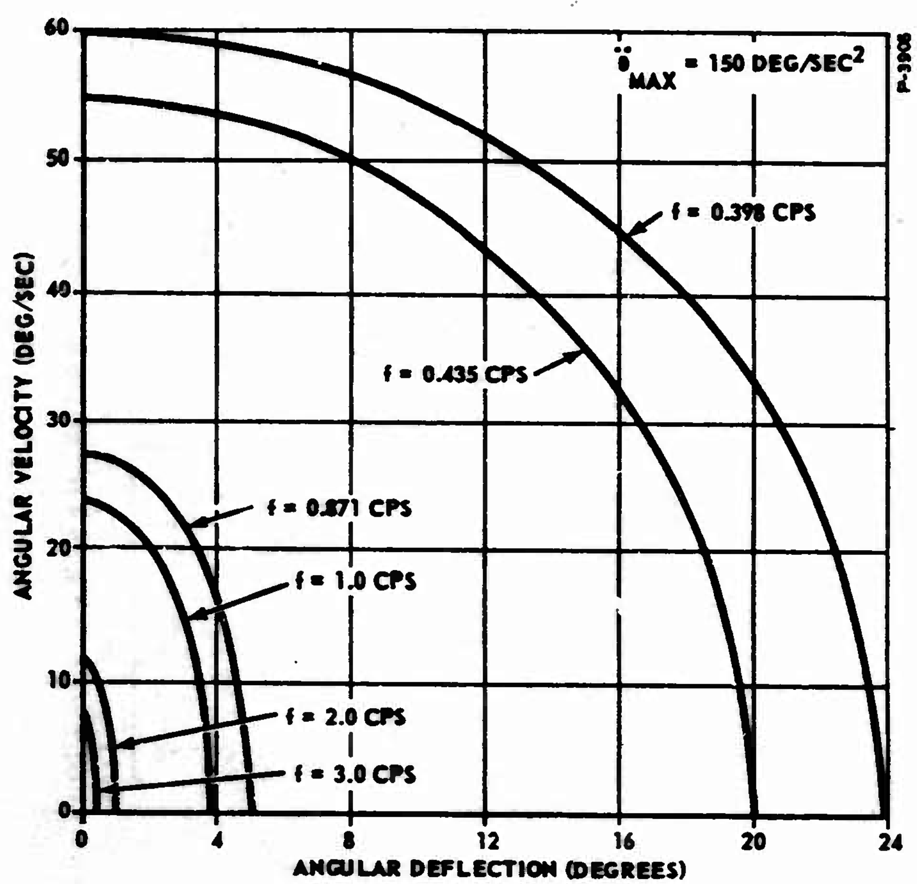

Figure 19 - Velocity-Displacement Curves F101B Rudder - Acceleration Limited

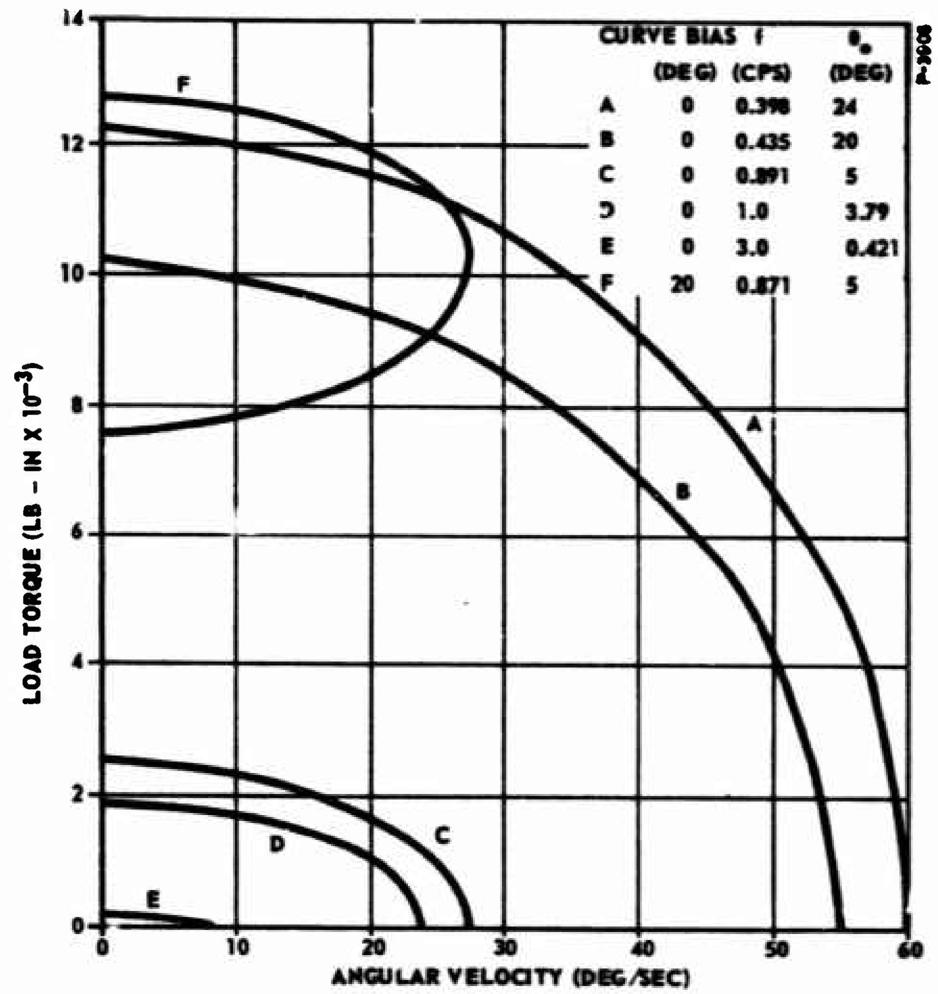

Figure 20 - Load Torque-Speed Curves Harmonic Motion - Acceleration Limited

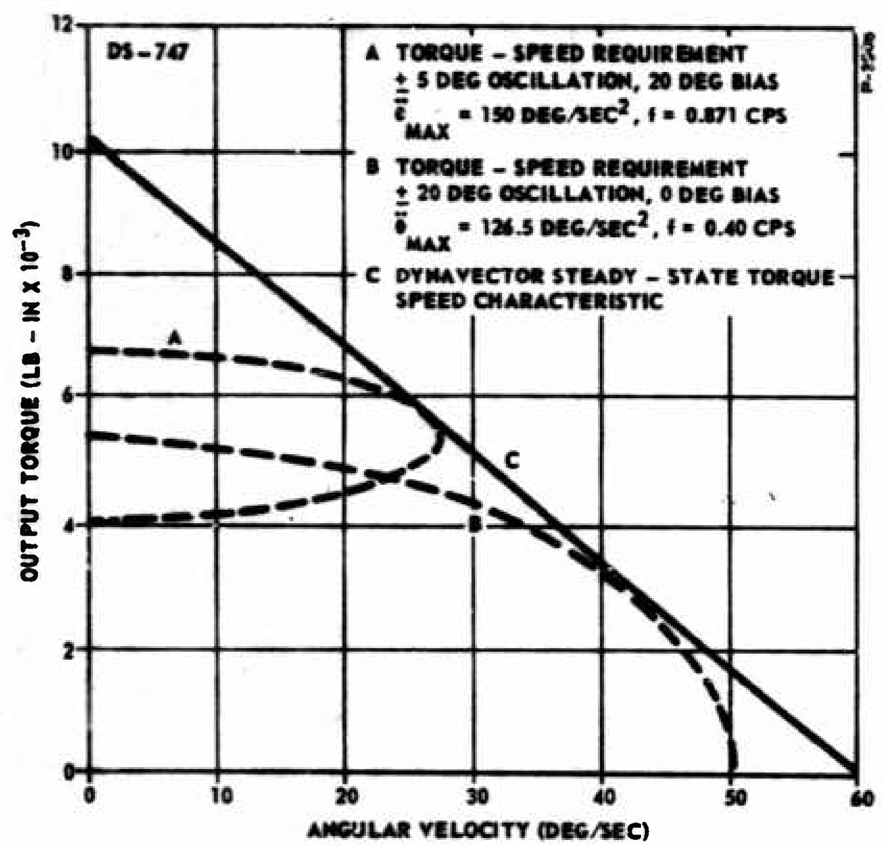

Figure 21 - Torque-Speed Capability Pneumatic DYNAVECTOR Actuator

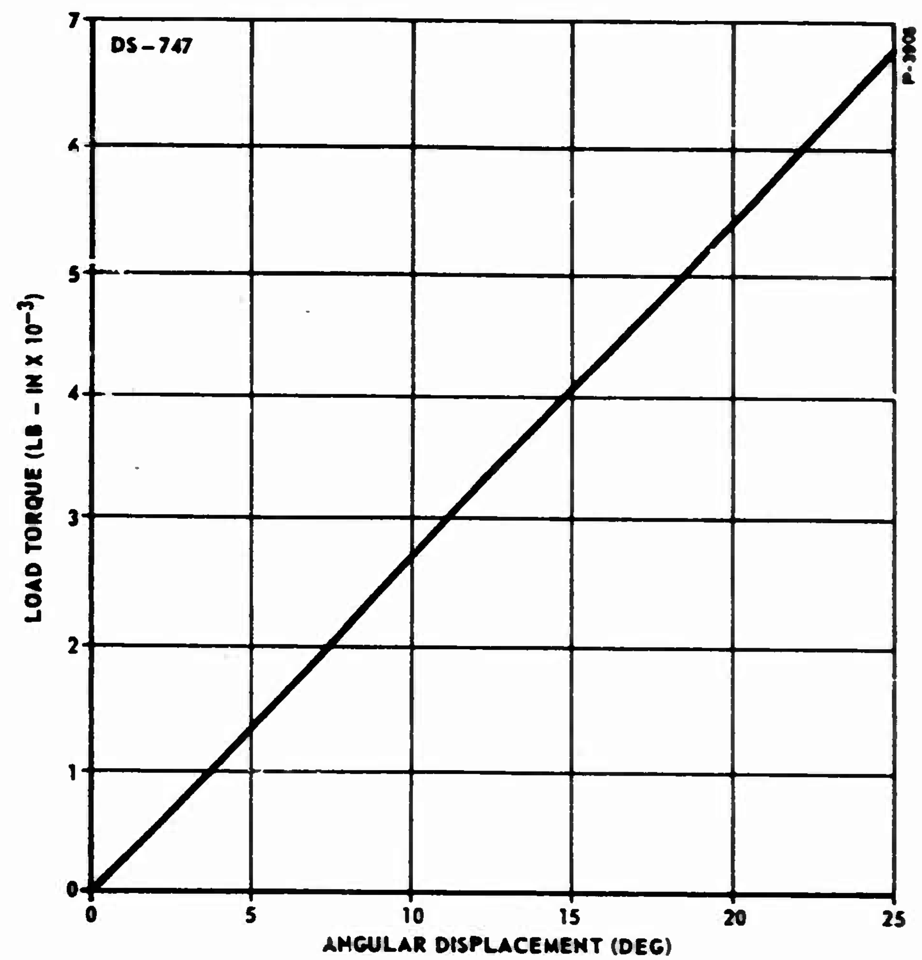

Figure 22 - Load Torque-Displacement F101B Rudder-Spring Rate 270 lb-in/degree

present hydraulic actuator shown in Figure 16 is chosen as the torque-speed design characteristic for the pneumatic actuator. To determine the load torsional spring rate and the maximum load torque to obtain the desired automatic mode performance of ±5 degrees at a bias of 20 degrees and a maximum acceleration of 150 deg/sec^2, a number of load torsional spring rates are selected and the resultant torque-speed characteristics plotted until a curve tangent to the steady-state torque-speed curve results. A torsional load spring rate of 270 lb-in/deg results in a torque-speed curve tangent to the steady-state performance, and this curve is plotted in Figure 21 as curve A. The load torque-displacement curve for the spring rate of 270 in-lbs/deg is plotted in Figure 22. The load torque at 20 degrees output deflection for this load is 5,400 lb-in.

To determine the performance criteria for harmonic motion with an amplitude of 20 degrees and a maximum load torque of 5,400 lb-in, performance curves at several frequencies are calculated until one is found that is completely within the steady-state torque-speed requirements of the motor. The frequency for which this occurs is 0.40 cps, which results in a maximum acceleration of 126.5 deg/sec^2. The torque-speed curve is shown as curve B of Figure 21.

2. DUTY CYCLE DEFINITION

An assumed duty cycle has been derived for the DYNAVECTOR rudder actuator application. This duty cycle has been established to provide actuator design information, to assist in the reliability and failure mode analyses, and to establish the basis by which the estimated power consumption requirements of the DYNAVECTOR may be compared to alternative pneumatic actuation systems.

A. Flight Mission Definition

A four-hour flight mission has been assumed and consists of the four operative conditions and times shown in Table II. The total design life of the actuator has been assumed as 3,000 hours, made up of such four-hour flight missions.

Table II - Flight Mission Definition

Operative Condition	Time
Take-off and acceleration at constant altitudes less than 20,000 feet	5 minutes
Climb, cruise and descent at altitudes greater than 20,000 feet	2 hours and 30 minutes
Miscellaneous cruise, loiter, etc., at altitudes greater than 20,000 feet	1 hour and 18 minutes
Landing approach to touchdown	7 minutes
Total	4.0 hours

The rudder actuator may be in one of three operative modes during the four-hour flight time.

1 - Stall output torque with the rudder held at a command position from zero to ±20 degrees.

2 - Automatic mode rudder oscillations and yaw damper command signals at up to ±5 degrees about any rudder position from zero to ±20 degrees.

3 - Manual mode input commands for rudder displacements up to ±20 degrees.

B. Rudder Stall Mode Conditions

It has been assumed that stall rudder conditions will occur for the entire take-off and acceleration, duration of 5.0 minutes, and for the landing approach to touchdown, duration of 7.0 minutes, and for 20 percent of the remaining flight time. Therefore, the duration of rudder stall conditions for a four-hour flight is 58 minutes. The remainder of the flight time, 3 hours and 2 minutes, will comprise automatic or manual command oscillatory inputs.

Based on a rudder spring rate load characteristic of 270 lb-in per degree and the percentage of time at manual mode amplitudes from ±5 to ±20 degrees defined in DS-742 (reference Appendix A),

Table III - Stall Torque During Four-Hour Flight

Stall Torque (in-lbs)	% Load	Amplitude (degrees)	Duration (minutes)
5,400	100	20	4
4,860	90	20	4
2,430	90	10	10
1,620	60	10	10
810	60	5	30
		Total Time	58 minutes

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which was derived from the hydraulic actuator specification, McDonnell Aircraft drawing 20-91023, "Cylinder-Integrated Power Control (Rudder)", the stall torque durations shown in Table III have been derived.

C. Rudder Oscillatory Conditions

During the four-hour flight time when the rudder is not in a stall mode condition (3 hours, 2 minutes) it is assumed that the rudder is subjected to either automatic or manual oscillatory commands. The duration of time for the qualification test defined in DS-742 (reference Appendix A) is 210 hours, of which 54 percent is in automatic mode. Assuming this percentage distribution applies to the flight time oscillatory conditions of 3 hours, 2 minutes, Table IV defines the duration of oscillatory modes for the mission duty cycle.

3. POWER SUPPLY CHARACTERISTICS

The characteristics of the pneumatic power supply available for the DYNAVECTOR rudder actuator are defined in DS-743 (reference Appendix A). The pneumatic power is derived from bleed air of the compressor section of the Pratt and Whitney JT3 turbojet engines. This bleed air is also the power supply for the cockpit pressurization and air conditioning systems. The power supply information defined in DS-743 and summarized below was derived from studies conducted by Honeywell, Inc., Aeronautical Division, under Contract AF 33(615)-2533, "Fluid State Yaw Damper System", for the period of March to June, 1965. The information presented herein is unclassified.

Table IV - Oscillatory Conditions During Four-Hour Flight

Mode	Amplitude (±degrees)	Frequency (cps)	Torque Variation (lb-in)	Time (minutes)
Automatic	5	0.871	0 to 810	35
Automatic	0.8	2.18	0	63
Total Time in Automatic Mode				98 minutes
Manual	20	0.435	0 to 4,860	84 minutes

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A. Supply Pressure

The availability of supply pressures between 50 and 200 psig is limited to normal engine operation at aircraft altitudes less than 35,000 feet. With the engine in an idle condition, the aircraft may operate at altitudes up to 20,000 feet and still provide 50 psig compressor bleed pressures, providing the aircraft flight speed is maintained. Figure 23 shows the normal and idle operational regimes for the pneumatic DYNAVECTOR actuator. The region under the normal and idle lines represent flight altitude and Mach number conditions under which the compressor bleed air is at least 50 psig. If the aircraft is required to operate in the region above the limit lines, actuator supply pressures will be less than 50 psig and degraded actuator performance must be anticipated.

B. Supply Temperature

Compressor bleed air temperatures, as monitored during flight tests and summarized in DS-743, vary from 100°F to 600°F. Wright-Patterson Air Force Base has stipulated that, for the F101B flight test vehicle, the maximum temperature for pneumatic flight surface control shall be 450°F. Consequently, precooling will be required before the bleed air is conducted aft to the DYNAVECTOR actuator interface.

C. Supply Flow

The maximum available supply flow as monitored during flight tests of an F101B by Honeywell is summarized in DS-743. The flow was found to vary from a minimum of 0.5 lb/sec in the descent mode at an altitude of 30,000 feet to 3.5 lb/sec for sea level take-off.

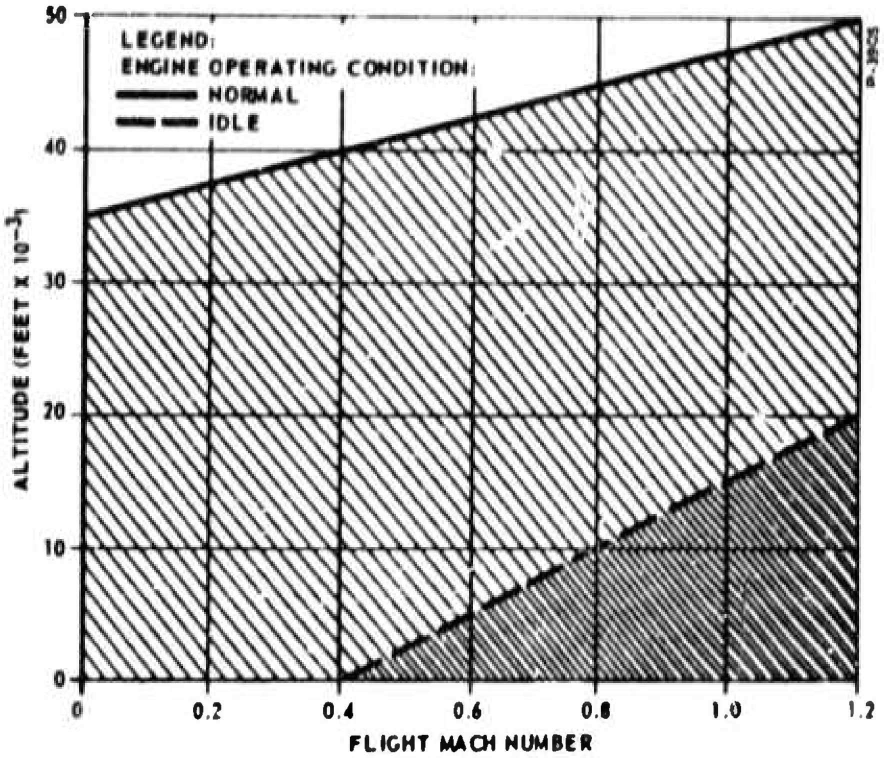


Figure 23 - DYNAVECTOR Operational Regimes Altitude Versus Flight Mach Numbers

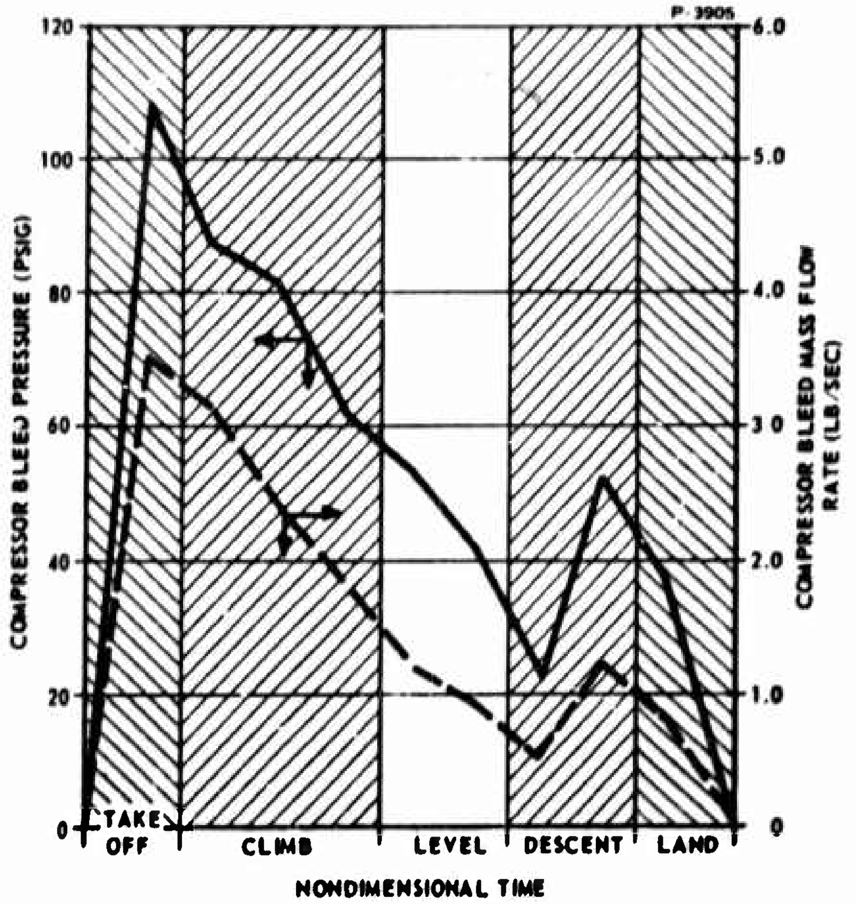


Figure 24 - Compressor Bleed Pressure and Mass Flow Versus Aircraft Operational Modes

Figure 24 summarizes the bleed pressure and concurrent supply flow values for five operative modes: take-off, climb, level, descent, and landing approach. The altitude and Mach number conditions for the flow and pressure values plotted in Figure 24 are defined in DS-743.

Figure 24 indicates that, for the flight operational conditions during the monitoring tests, the available mass flow exceeds 1.0 pound per second for supply pressures above 50 psig, the specified design supply pressure for the pneumatic DYNAVECTOR actuator.

SECTION III

DESIGN AND PERFORMANCE ANALYSIS

1. SERVOMECHANISM ASSEMBLY DESIGN

A. DYNAVECTOR Operation

The DYNAVECTOR actuator is an integral high speed motor and transmission without high velocity mechanical elements. The major components of the DYNAVECTOR actuator assembly consist of a series of displacement chambers, a unique integral epicyclic transmission, and commutation porting. The transmission and motor use elements common to both, resulting in a much simpler and more reliable design.

In a low ratio DYNAVECTOR actuator the power element is a positive displacement, very low inertia, non-rotating vane motor. Its output is a radial force vector that rotates at high speed and in either direction of rotation. The displacement chambers formed by the vanes and the housing expand and collapse at the same speed as the force vector, but do not rotate. The motor is self-commutating but does not contain a rotating porting plate or spindle. The absence of high velocity members in the motor significantly reduces the inertia, resulting in high acceleration capability.

The unique epicyclic transmission converts the rotating force vector directly into low speed, high torque rotary motion without the use of high speed mechanical input stages. The transmission also has zero backlash without using preloaded members.

The integration of the power element and epicyclic transmission into an integral actuator design results in an ideal servo actuator with a high torque-to-inertia ratio and high constant efficiencies for both small and rated loads.

The operation of a low ratio DYNAVECTOR actuator is illustrated by Figure 25. The basic components are the ring gear, the ground gear and housing, the center output gear, and the vanes. The displacement chambers are formed between the ground gear and the ring gear mesh by the vanes. This gear mesh provides displacement motion without rotation because both gears have exactly the same number of teeth. It may be considered as a loose spline but is a true involute gear mesh. The internal portion of the ring gear forms the transmission between the motor and the output shaft and represents the epicyclic transmission.

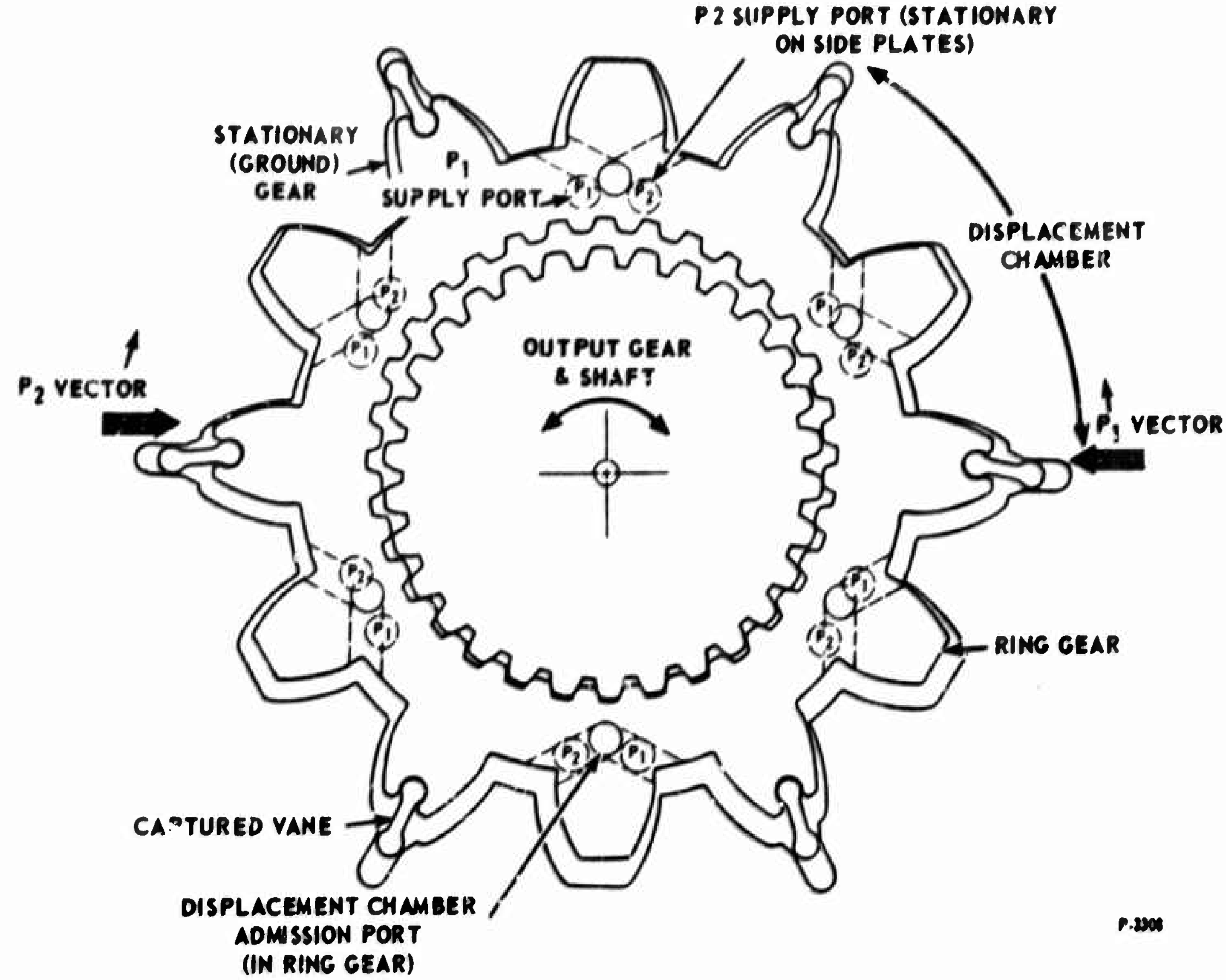


Figure 25 - Basic Operation and Design of Low Ratio DYNAVECTOR Actuator

A force vector is generated by pressurizing three adjacent displacement chambers and venting the remaining three. The vector is made to rotate by pressurizing a vented chamber adjacent to the original three pressurized chambers while simultaneously venting the diametrically opposite one. If the force vector on the ring gear is located at approximately 90 degrees to the ring and output gear contact point, the ring gear will move, causing the output gear to turn and the contact point to move. If the force vector is also rotated and remains 90 degrees to the contact point, the motion will be continuous and the output shaft will turn continuously but at a much lower speed than the force vector. The ratio will be determined by the difference in number of teeth between the ring gear and the output gear. The gears in Figure 25 have 30 and 32 teeth; thus, the reduction ratio is 15:1.

The available differential pressure in the form of two motor port pressures P_1 and P_2 must be commutated to the proper displacement chambers to produce a rotating force vector in phase with the ring gear motion. To ensure that this phase relationship always hold true, the motion or position of the ring gear is used to provide this commutation through a series of ports. Each displacement chamber has a pair of supply ports or a P_1 and P_2 port associated with it. The P_1 ports are all interconnected in the housing and brough out to a single inlet port, as are all the P_2 ports. These ports are in the housing and, therefore, stationary with respect to the displacement chambers. They are also located under the ring gear face, as shown in Figure 25 and a port connecting the displacement chamber to the ring gear face is located opposite them.

By locating these P_1 and P_2 ports as shown in Figure 25, the ring gear ports will open P_1 ports to half the displacement chambers and P_2 ports to the remaining half. The resulting pressure force on the ring gear from the displacement chambers connected to P_1 is 180 degrees opposite P_2 and 90 degrees from the output gear contact point. Therefore, pressurizing P_1 and venting P_2 produces rotation in one direction, while interchanging pressure and return reverses the motor. This also satisfies the desired relationship between force vector and ring gear position. Because this commutation is created by the displacement member or ring gear itself, it will always rotate in phase with the motor, producing maximum efficiency.

One of the primary advantages of the DYNAVECTOR actuator is the potential efficiency, especially outstanding at high ratios. The unique ring gear transmits the load reaction forces at close to one-to-one correspondence to ground and, therefore, is actually an output or high torque member. On the other hand, it is also the dynamic member of the motor, which is the low torque component of the system.

Two other factors present in conventional rotary motor plus transmission systems are significantly reduced by the DYNAVECTOR actuator design and operation. The relative velocities between dynamic and static members are very small, because of the small amplitude epicyclic motion. In a DYNAVECTOR actuator, the relative velocity between the ring gear and the housing is only a function of the eccentricity, which is usually less than one-tenth of an inch, times the angular velocity. Whereas, in a conventional motor, there are usually components with a radius more than an inch rotating at the same angular velocity. This also holds true for the transmission which does not have the conventional input gear running at high pitch line velocities. The relative velocities between the meshing teeth correspond to those found in the last stage of a conventional transmission.

The absence of high relative velocities results in:

- Friction losses at high motor input speeds are significantly reduced.
- Because of low friction losses, high mechanical efficiencies can be obtained.
- Wear is greatly reduced, resulting in longer life of the actuator.

The other factor significantly reduced is the actuator or motor inertia. In conventional high speed motors, the motor inertia resulting from significant mass rotating at high angular velocity has always limited the motor acceleration or response capabilities. The small volumes under compression have generally made up for lack of response due to inertia and have placed rotary servos on equal terms with piston-cylinder servos having very little inertia. However, the spring rate of the transmission has in some cases presented unwanted decoupling between the load inertia and the motor inertia, resulting in load resonance. The problem is usually solved by stiffening the transmission at the expense of added weight, as it is usually the load-carrying output members that are too weak.

The DYNAVECTOR actuator has no mass rotating at input or force vector speed and only a small reflected inertia, due to the small eccentric rotation of the ring gear, and the low speed output shaft. Therefore, it has an inertia equal to a similar capacity piston-cylinder actuator and a volume under compression equivalent to a conventional similar capacity rotary servo. This combination results in a servo with a response potential many times that obtained by present day systems.

DYNAVECTOR actuators can be designed utilizing any high ratio in the epicyclic transmission by substituting a ratio other than one to one in the reaction mesh. In compound epicyclic transmissions of this type the ratio is computed from the following expression:

$$N' = \frac{\dot{\theta}_i}{\dot{\theta}_o} = \frac{1}{1 - \dfrac{N_1 N_4}{N_2 N_3}} \qquad (13)$$

The nomenclature is given in Figure 26, Unbalanced High Ratio DYNAVECTOR actuator. In this device the basic description and operation is the same as in the low ratio actuators. However, the ring gear will have an angular rotation which is a function of the gear pitch diameters and is equal to $\left(\frac{D_3 - D_1}{D_3}\right)\theta_i$. Slightly increased motor friction will be noted from vane tip sliding, although this will be offset through more efficient conversion of the force vector to output torque.

Porting commutation is provided from the motion of the ring gear as in the low ratio design. Due to the rotation of the ring gear, the P_1 and P_2 ports in the end plates are not holes or slots but concentric rings. These rings will commutate P_1 and P_2 to opposite sides of the ring gear by uncovering porting slots in the ring gear at its extreme inward and outward radial positions.

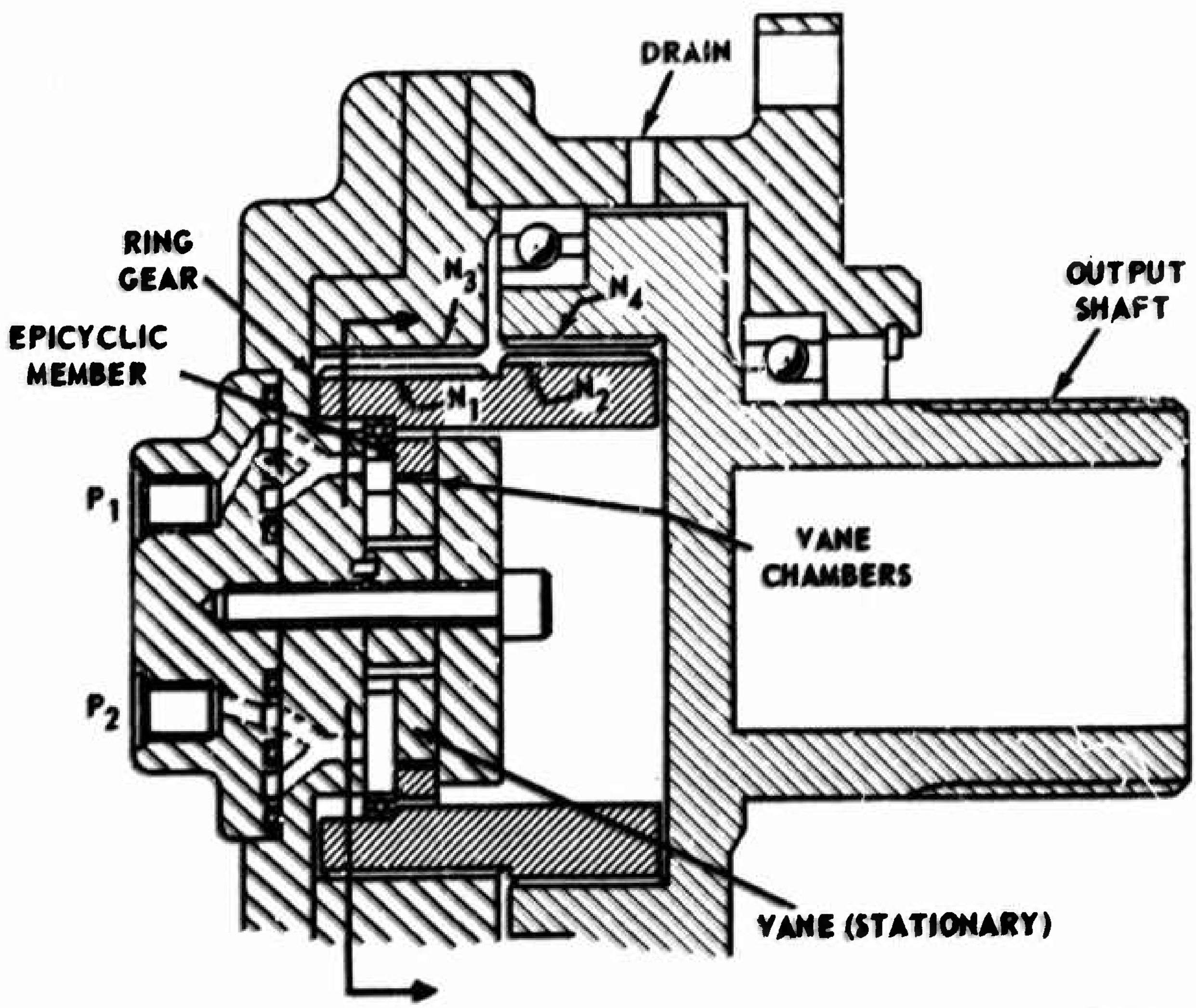


Figure 26 - Unbalanced High Ratio DYNAVECTOR

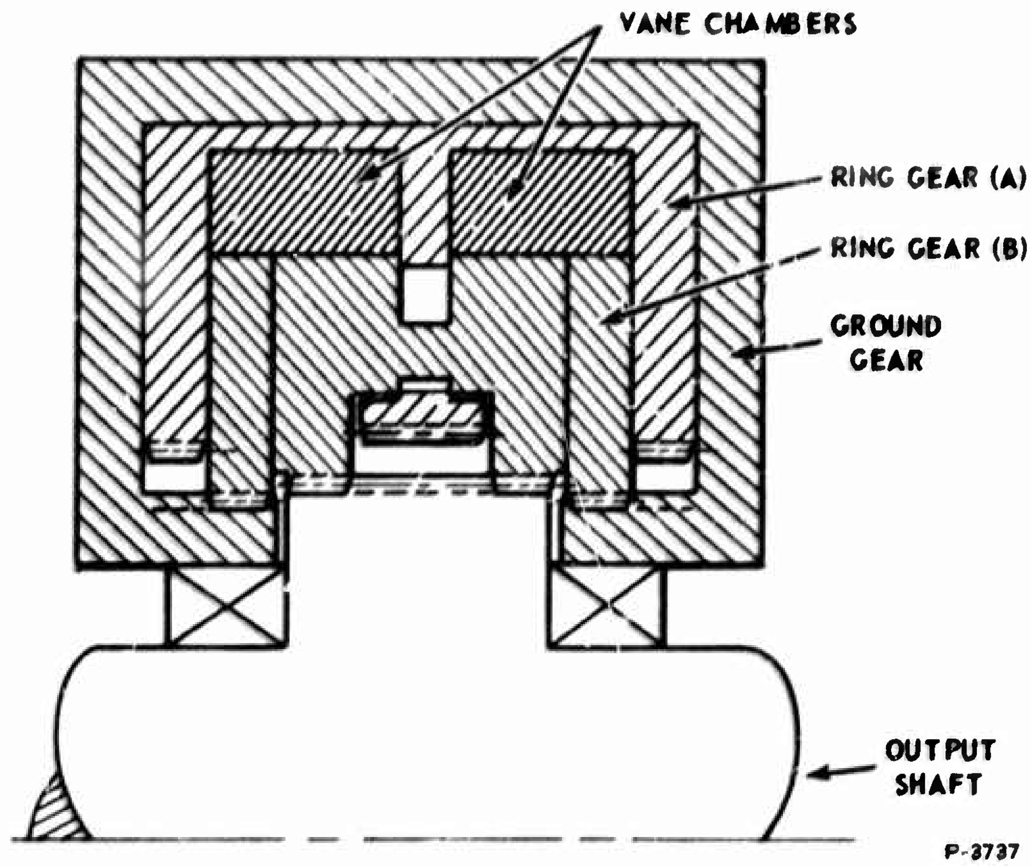


Figure 27 - Internal Mesh Balance Configuration

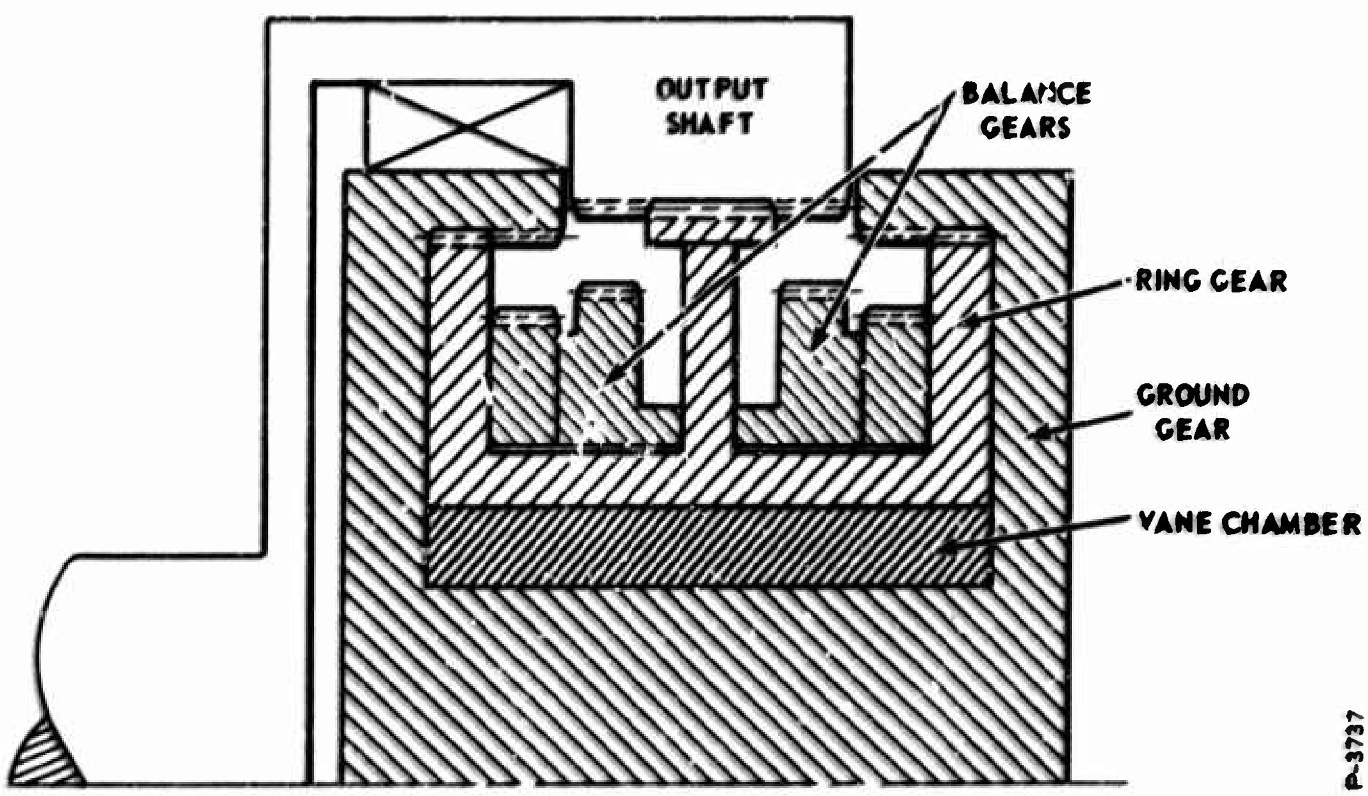


Figure 28 - External Mesh Balance Configuration

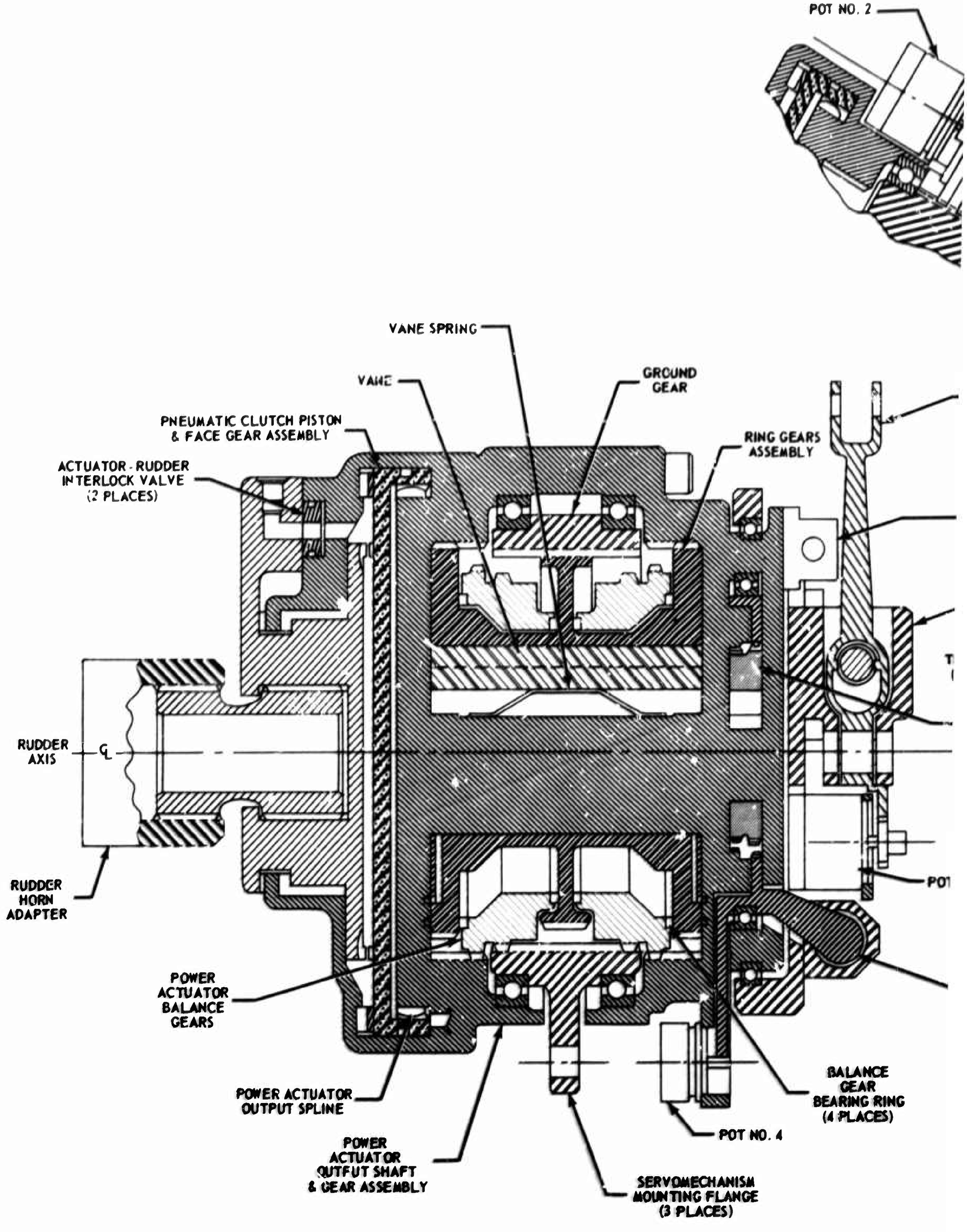
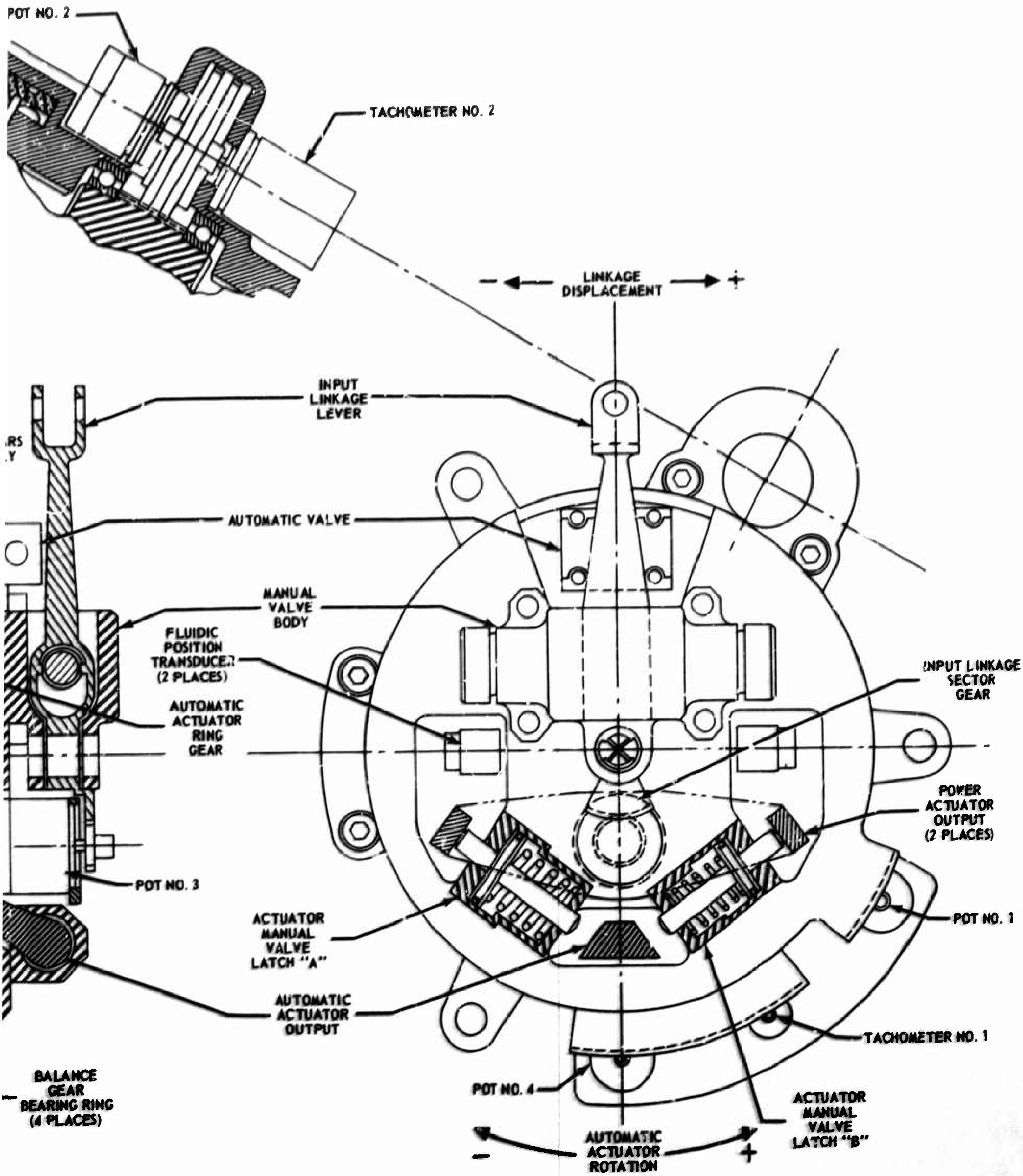


Figure 29 - DYNAVECTOR Pneumatic



OR Pneumatic Rudder Actuator Design Drawing

2

The number of vanes or displacement chambers only determines the very low speed torque ripple present. The number of chambers need not be odd or even since the starting torque is only a function of the force vector angle which varies through an angle equal to the angle included by one displacement chamber.

In many applications involving high output speeds and power, the unbalanced ring gear inertia force will be significant, and cause vibration. This rotating inertia force vector can be eliminated by designing the actuator with an equal unbalanced force vector located 180° out of phase with the ring gear. This can be accomplished by splitting the ring gear and taking half of the torsion on diametrically opposite sides of the motor as in Figure 27. This design will provide complete force balance as well as mass balance.

Another design which provides mass balance but not complete force balance is shown in Figure 28. In this design the inertia effect of the ring gear are canceled by counterweights having mass and epicyclic motion equivalent to the ring gear but vectorially opposed. These counterweights are gear driven from the output and reaction members of the actuator.

B. Servomechanism Assembly Design and Operation

The pneumatic rudder actuator servomechanism, DYNAVECTOR actuator model PH-370-B1 is shown in Figure 29. The major components of this assembly are summarized in Section 1 and the functional interrelationships are shown schematically in Figure 7 through 10. The functioning of these components for the four modes of operation; manual power, manual power with stability augmentation in monitor and operating, and autopilot, will be described in detail below.

(1) Load Limit Mechanism

The load limit mechanism shown in Figure 2, Section 1, couples the pilot rudder pedal hydraulic control linkage to the pneumatic actuator. The load limit mechanism is designed to allow hydraulic control linkage movement when the pneumatic system is not operative. The mechanism consists of two negator springs and acts as a rigid link when transmitting loads up to 0.5 pounds. When the pneumatic system is not operative, and the pilot displaces the hydraulic control linkage, the pneumatic input linkage lever is displaced until the manual valve spool is bottomed out in the valve body. The linkage microswitch assures proper phasing of the pneumatic servomechanism with the hydraulic system.

Since the pneumatic manual valve body cannot track the linkage, further hydraulic linkage motion could not occur. The load limit mechanism allows further hydraulic linkage travel by deflection of the negator springs until the desired rudder position is attained.

(2) Power Actuator Design

Upon energization of the pneumatic power supply solenoid by the pilot, pneumatic supply pressure is ported to the manual valve supply ports. Displacement of the input linkage lever shifts the valve spool in the valve body thereby producing a pressure differential in the DYNAVECTOR power actuator eight vane chambers. The vanes are spring loaded radially outward, and the pneumatic force vector created by the vane chambers pressure differential drives the ring gears assembly. The epicyclic motion produced by the ring gear about an eccentric axis offset from the rudder axis drives the power actuator output shaft through an internal gear mesh concentric to the rudder axis. Mounted to the power actuator output shaft is the output spline which engages the pneumatic clutch piston and face gear assembly. Both the clutch piston and piston cylinder rotate integrally with the power actuator output shaft.

Upon energization of the clutch solenoid switch, pneumatic pressure would engaged the piston face gear with the rudder horn adapter face gear provided proper interlock valve position.

(3) Actuator - Rudder Interlock Valve

Prior to actual engagement of the pneumatic rudder actuator to the F101B rudder during flight tests, monitoring tests of the pneumatic actuation system will be conducted to assure proper pneumatic system functioning. Engagement of the pneumatic system to the rudder and shutdown of the hydraulic rudder actuation system is a controlled fail-safe procedure such that if a failure of the pneumatic supply occurs or the pneumatic actuator is not positionally phased properly with the hydraulic system command position, reversion to hydraulic mode will occur immediately. Proper positional phasing is accomplished by a redundant fail-safe design of the pneumatic clutch used for engagement of the pneumatic power actuator to the rudder horn. The clutch teeth are of a single point engagement design such that they cannot become engaged unless the ruder and power actuator output shaft positions are identical. The actuator-rudder interlock valve also assures proper actuator positioning before engagement can occur. Clutch piston pressurization for engagement by 50 psig compressor bleed air cannot occur until the interlock valve ports are properly aligned.

(4) Actuator-Manual Valve Latch

Upon displacement of the input linkage lever and manual valve spool and subsequent power actuator output shaft rotation, the manual valve body tracks the power actuator rotation through the actuator-manual valve latch. The latch consists of two spring loaded piston and cylinder assemblies denoted in Figure 29 as latch "A" and latch "B". In manual power mode where the automatic actuator is not operative, the latches are de-energized. The spring loading of these latches locks up the manual valve body to the power actuator output. Thus, as the power actuator output attains the commanded rudder angular displacement position, the manual valve body is driven with the power actuator until the manual valve body displacement nulls out the commanded displacement of the linkage and valve spool.

(5) Automatic Actuator Design

The automatic actuator is an unbalanced DYNAVECTOR actuator capable of introducing stability augmentation or autopilot commands to the power actuator.

The reaction gear for the automatic actuator is integral with the power actuator output shaft. Thus, the null position for the automatic actuator is always maintained at the rudder position existing at any given time. The output of the automatic actuator is used only for biasing the manual valve body relative to the spool. Therefore, when the pilot wishes to stabilize rudder displacements commanded by input linkage lever motion or maintain a heading under autopilot mode, the actuator-manual valve latch switch must be energized thereby porting pneumatic supply to latches "A" and "B". Pressurization of the latches mechanically links the output of the automatic actuator to the manual valve body. Automatic actuator output rotation is physically limited to ± 5 degrees about the existing rudder position as the stroke of the latch pistons is equivalent to 5 degrees rotation of the rudder. A plus (+) linkage displacement as shown in Figure 29 would produce a clockwise rudder rotation when viewing up the rudder axis. The automatic actuator direction of rotation required to produce such a clockwise rudder rotation would be opposite, that is counterclockwise viewing up the rudder axis. Such a rotation of the automatic actuator would displace the manual valve body relative to the spool in an identical manner as if the linkage lever were displaced in a plus (+) direction. Thus, the power actuator output shaft would always rotate in such a direction as to limit automatic actuator rotation; in this case to 5 degrees rotation.

(6) Instrumentation Design

The electrical potentiometer and tachometer installation is as shown in Figure 29. The relative displacement and velocity functions generated by these components is summarized in Table V.

2. SERVOMECHANISM COMPONENTS DESIGN

A. Power Actuator

(1) Description

The power actuator as proposed is designed as a balanced DYNAVECTOR actuator with counterweights driven in an epicyclic motion. The output power is transmitted through a single ring gear. In this design, force couples will not be produced in the loaded members and there will not be a tendency for the ring gear to skew. The epicyclic counterweights will individually produce a skewing couple resisted by the ring gear through thrust bearing ring surfaces made of Delrin AF material. The couples produced by the two counterweights are small in magnitude and opposite in direction and will not have a significant net effect upon the ring gear.

The heavily loaded members are located near the outer diameter of the package, providing maximum torque capacity. The virtual

Table V - Potentiometer and Tachometer Functions
(Reference Figure 29)

Potentiometer No.	Function
1	Manual Valve Body to Power Actuator Output
2	Power Actuator to Airframe
3	Manual Valve Spool to Manual Valve Body
4	Power Actuator to Automatic Actuator
Tachometer No.	
1	Power Actuator to Automatic Actuator
2	Power Actuator to Airframe

motor is located in the center of the actuator and consists of eight displacement chambers formed by sliding carbon vanes. The motor is ported from both ends thereby balancing the pressure forces acting on the ring gear.

(2) Design Analyses

(a) Gear Pitch Diameter Sizing

The power actuator is sized on the basis of transmission torque capacity and motor displacement. Minimum size is of prime importance due to the available aircraft space and specified weight limitations. The actuator is required to have a stall torque capability of 10,200 in-lbs and produce a maximum angular output velocity of 60°/sec.

In sizing the transmission first on the basis of torque capacity, the approximate gear sizes may be established from the following relationships:

$$F_t = \frac{\pi S_s y b}{DP} \quad \text{(Lewis Equation for gear tooth stress)} \qquad (14)$$

where:

F_t = tangential load applied to the tooth

b = face width

S_s = tooth stress

DP = diametral pitch

y = tooth form factor

D_p = gear pitch diameter

and,

$$F_t = \frac{4 T_o}{N D_p} \qquad (15)$$

where:

T_o = maximum torque

N = number of teeth sharing load

$$\frac{\pi S_s y b}{DP} = \frac{4 T_o}{N D_p} \tag{16}$$

$$N = \frac{DP (D_p)}{10} \quad \text{(assumption to be verified with computer run)} \tag{17}$$

$$\frac{\pi S_s y b}{DP} = \frac{40 T_o}{DP (D_p)^2}$$

$$D_p = \sqrt{\frac{40 T_o}{S_s y b}} \tag{18}$$

Use of a high strength alloy steel such as AISI 4340 will permit a gear design stress as high as 75,000 without exceeding the endurance limit. The tooth form factor "Y" for a stubbed tooth gear will be assumed to be 0.5 based upon previous experience.

The relationship between the pitch diameter and gear face width for a torque capacity of 10,200 in-lbs becomes:

$$D_p = \sqrt{\frac{40 \times 10{,}000}{75{,}000 \times 0.5 \times b}}$$

$$D_p = 3.3 \sqrt{\frac{1}{b}} \tag{19}$$

The tooth face width and pitch diameters can be selected from the following chart:

b	D_p
1.50	2.71
1.25	2.96
1.00	3.31
0.75	3.82
0.50	4.70

(b) Transmission Ratio

The power actuator can be designed to operate with a force vector angular velocity of 420 radians/sec or 4,000 RPM. This velocity is established from the actuator commutation porting area and the valve orifice area. Although this maximum force velocity might be considered arbitrary, velocities in this range provide the most desirable balance between motor displacement, porting area, and bearing P-V values.

The actuator maximum output velocity must be 60°/sec or 10 RPM. The transmission must therefore have an approximate reduction ratio of:

$$N' = \frac{\dot{\theta}_i}{\dot{\theta}_o} = \frac{4,000}{10} \quad \text{or} \quad 400/1$$

The motor displacement (D_m) required for the power actuator to provide an output torque of 10,200 in-lbs, assuming an overall actuator mechanical efficiency (η_t) of 80 percent is:

$$D_m = \frac{2 \pi T}{N' (\Delta P) \eta_t} \tag{20}$$

$$D_m = \frac{10,200 \times 2 \pi}{400 \times \Delta P \times 0.80} = \frac{200}{\Delta P}$$

Assuming a recovery ΔP of 45 psig

$$D = 4.45 \text{ in}^3$$

At this point, due to the restrictions on the power actuator size for this application, preliminary layouts must be made to establish a combination of gear sizes, transmission ratio, and motor displacement providing specified motor performance within the smallest package size.

The transmission ratio for this actuator configuration is determined from equation (13)

$$N' = \frac{\dot{\theta}_i}{\dot{\theta}_o} = \frac{D_2 D_3}{D_2 D_3 - D_1 D_4}$$

By adjusting the various design parameters and observing their effect upon the design trend, the following gear sizes were selected:

Gear	D_p	DP	N
D_1	4.17	24	100
D_2	4.37	24	105
D_3	4.42	24	106
D_4	4.62	24	111

The exact transmission ratio is

$$N' = \frac{105 \times 106}{105 \times 106 - 100 \times 111} = \frac{11{,}130}{11{,}130 - 11{,}100}$$

$$N' = 371/1$$

(c) <u>Gear Face Width</u>

Using the gear addendum sizing computer program to eliminate tooth tip interferences through the arcs of approach and recession in the reaction gear mesh, the gear pitch radii are $R_o = 2.1166$, $R_i = 2.1786$, with a pressure angle of 20° and contact ratio of 1.496. Figure 30 describes the relation between teeth and the nomenclature.

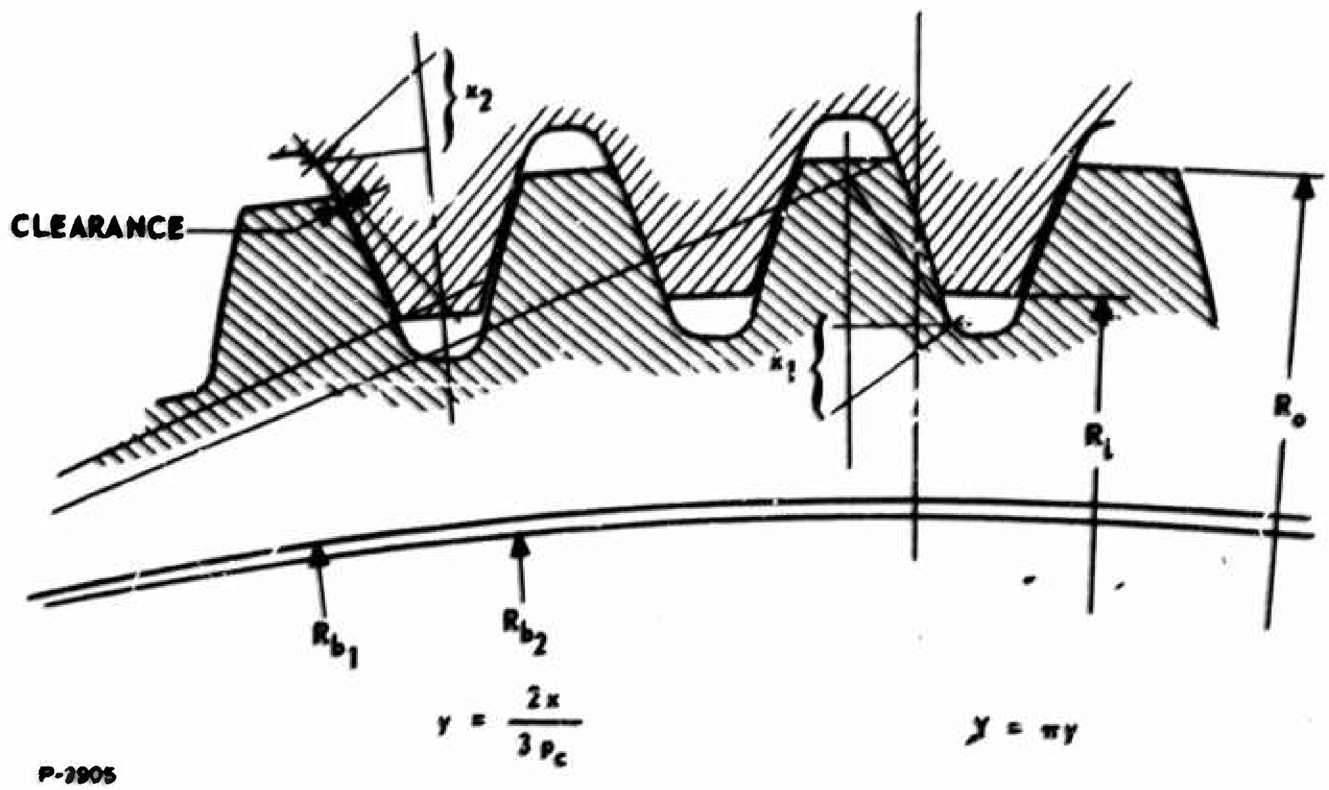


Figure 30 - Tooth Form Factors

Based upon gear designs involving these gear addendums, trochoids of the gear tooth motion can be computed giving the clearance between teeth as they engage and disengage. By computing the total deflection between two teeth transmitting a force and comparing with the clearance between other teeth, the total number of teeth capable of sharing the load can be more accurately predicted.

According to Buckingham[1], the empirical equation for the combined bending and compressive deformation of a mating pair of gear teeth when the contact is at the middle of the gear tooth heights is:

$$\Delta = \left(\frac{F_t}{b}\right)\left[\frac{E_1 Z_1 + E_2 Z_2}{E_1 Z_1 E_2 Z_2}\right] \tag{21}$$

[1] Buckingham, Earle, Analytical Mechanics of Gears, First Edition, McGraw-Hill Book Company, Inc., 1949, p. 342.

where:

Δ = combined deformation

F_t = tangential tooth load

b = tooth face width

E = Young's Modulus

y = tooth form factor

$$Z = \frac{y}{0.242 + 7.25\, y}$$

From the geometry of the gear teeth, y_1 and y_2 are calculated to be 0.168 and 0.194 for teeth having equal tooth tip thicknesses. Solving equation (21)

$$Z_1 = 0.115$$

$$Z_2 = 0.118$$

$$\Delta = \left(\frac{F_t}{b}\right) \left[0.574 \times 10^{-6}\right]$$

The maximum allowable tooth tip load is determined from equation (14)

$$\frac{F_t}{b} = \frac{S_s\, Y}{DP} = \frac{75{,}000 \times 0.526}{24} = 1{,}645 \text{ lbs/in}$$

where:

$Y = \pi y$

and the combined deformation is;

$$\Delta = 1{,}645 \text{ lbs/in} \left[0.574 \times 10^{-6}\right]$$

$$\Delta = 0.000945 \text{ in.}$$

The arc of teeth which will share the torsion load can be determined from a comparison of the tooth trochoid clearance computer results and the combined deformation. In this case 13 degrees of arc in the approach and 11 degrees on the recession arc will share in transmitting torque. Correcting out initial estimate of the number of teeth sharing the load from 10 percent to 6.6 percent, and the "Y" form factor from 0.5 to 0.527

$$D_p = \sqrt{\frac{60.5\ T_o}{S_s\ Y\ b}} = \sqrt{\frac{60.5 \times 10{,}200}{75{,}000 \times 0.527\ b}}$$

$$D_p = 3.96\sqrt{\frac{1}{b}}$$

In the reaction mesh, the pitch diameter (D_p) in the higher stressed external gear is 4.17 inches, resu[illegible]g in a gear face width (b_r) :

$$b_r = \left(\frac{3.96}{4.17}\right)^2 = 0.902$$

In the output mesh, the percent of teeth sharing the load is nearly the same as in the reaction mesh and therefore, the required face width of the gears is:

$$b_o = \left(\frac{3.96}{4.37}\right)^2 = 0.82 \text{ in.}$$

(d) Porting Area

The commutation porting area must be sized such that it will not limit the free running motor speed. Assuming that the force vector has a maximum angular velocity of 418 rad/sec, flow (Q) through each displacement chamber is:

$$Q = (\text{in}^3/\text{sec}) = \frac{\pi D_m\ e\ \dot{\theta}'\ (\ell)}{c} \qquad (22)$$

where:

D_m = mean diameter of the displacement chamber (in)

ℓ = length of the displacement chamber (in)

c = number of displacement chambers

e = eccentricity of ring gear (in)

$\dot{\theta}'$ = angular velocity of motor (rad/sec)

$$Q = \frac{\pi \times 2.66 \times 3.12 \times 0.125 \times 418}{8}$$

$$Q = 170 \text{ in}^3/\text{sec}$$

The commutation porting area is calculated from the following expression for gas flow out of an upstream region based upon upstream gas density:

$$Q = C_d A_g \sqrt{T_u} \, R \, C_2 \, f_1 \left(\frac{P_d}{P_u}\right) \qquad (23)$$

The $f_1 (P_d/P_u)$ is established by the acceptable pressure drop through the commutation porting. If this is limited to 1 psi at maximum motor speed, (P_d/P_u) will be (39/40) and f_1 (0.975) is 0.31.

Solving for the commutation orifice area (A_g) at the worst case condition with the supply air at -65°F, and assuming an orifice coefficient of discharge (C_d) equal to 0.7 we have

$$A_g = \frac{170}{0.7 \times \sqrt{395} \times 340 \times 0.31}$$

$$A_g = 0.116 \text{ in}^2$$

(e) <u>Bearing Analysis</u>

The proposed actuator has two radial ball bearings maintaining alignment of the reaction and output gears. These bearings operate at the low speed of the output shaft. The operating temperatures for the bearings is expected to be no higher than 300°F allowing grease lubrication and common bearing materials.

The bearings will be subjected to individual radial loads which will not exceed 1,150 lbs. The quiet running radial load capacity for the bearing design, (Kaydon KC-50-CP bearing), is 3,450 lbs. static, thus, providing a factor of safety of 3.

B. Automatic Actuator

The purpose of the automatic actuator is to convert the amplified output differential pressure signals from the autopilot into proportional rotary motion. This rotary output creates an error signal in the manual control valve by shifting the position of the valve body with respect to the valve spool.

The specification for the automatic actuator response is 5 cps at an amplitude of 0.001 radians. The torque required to provide this response is insignificant, and the actuator can be sized to provide torque for positional stability of the manual control valve body and overcome friction losses in the drive mechanism.

By selecting a transmission ratio equal to the power actuator, 371/1, and designing the automatic actuator to provide 250 in-lbs torque, the required motor displacement can be calculated from equation (20).

$$D_m = \frac{T_o \times 2\pi}{N' \times \Delta P \times \eta_t} = \frac{250 \times 2\pi}{371 \times 45 \times 0.72}$$

$$D_m = 0.13 \text{ in}^3/\text{rev}$$

Selecting a transmission ratio equal to the power actuator will produce equal force vector velocities in the two actuators and porting areas and gear designs can be proportional. The motor displacement of 0.13 in^3 results in a package having a mean diameter of 0.88 inches and a length of 0.375 inches.

The unbalance due to the ring gear inertia is negligible and the automatic actuator will be designed as an unbalanced high ratio device.

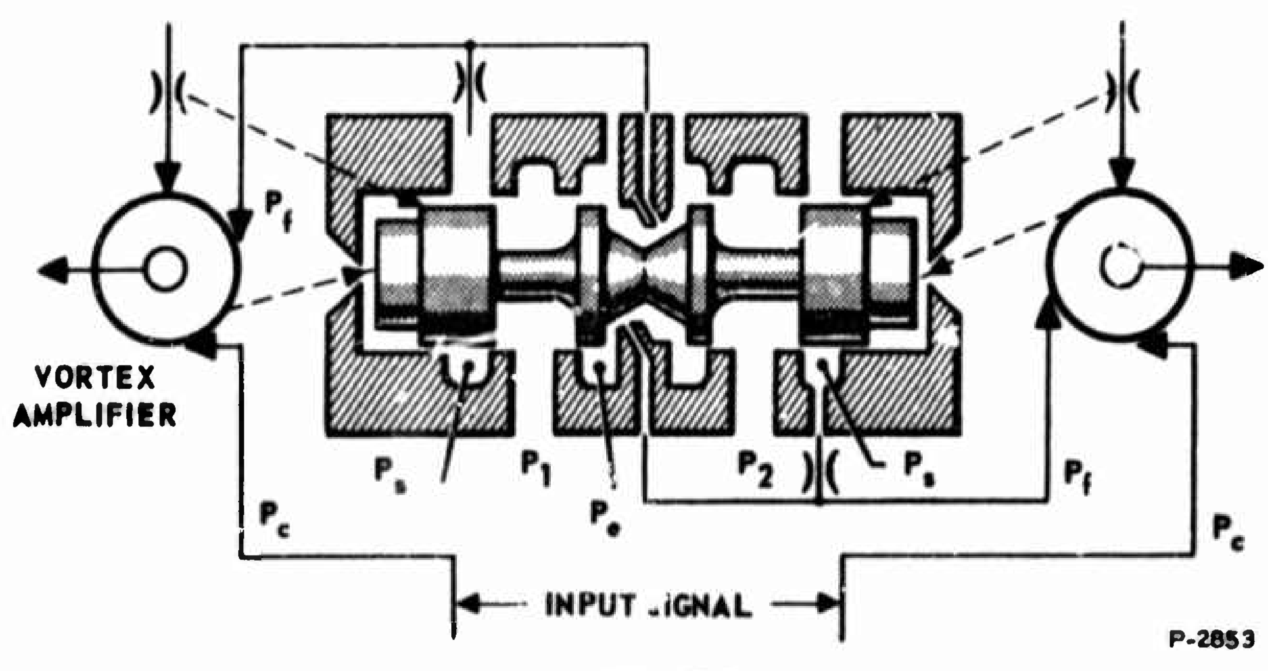
C. Automatic Servovalve

(1) Description

The automatic servovalve will be a pneumatic four-way spool valve actuated by a ± 6 psid pressure signal applied to a fluid state input. A ± 5 psid signal displaces the 0.250 diameter spool ± 0.010 inches in a direction dependent on the differential pressure in the system. The maximum spool supply area of 0.001 square inches; the maximum exhaust area is 0.0015 square inches. The valve is designed to operate within a temperature range of 70°F to 500°F. The valve spool and body are made of 440C stainless steel.

The schematic diagram, Figure 31, shows the internal mechanism of the servovalve. The basic concept for stroking the spool using vortex flow in the ram chambers was chosen for simplicity. The end lands of the spool become the buttons of vortex valves. The annular clearance between the spool and body provides the supply flow from the supply pressure land. Control flows are injected tangentially into this clearance area while flow exits from the center of the end caps.

Because the valve is a flow control valve, spool position must be a function of the input signal. This requirement was met by a



SCHEMATIC

Figure 31 - Automatic Vortex Servovalve Schematic

spool position feedback signal which is summed with the input signal at the vortex valves. A tapered ramp on the spool varies a nozzle area which provides a pressure signal that is a function of spool position for the feedback signal.

The vortex flow also provides the necessary damping of spool motion, eliminating the need for conventional damping tanks.

Lubrication is provided by a black oxide film applied to the spool and bore by preheating the parts in an oxidizing atmosphere. The end caps and manifolds will also be fabricated from 440 C stainless steel.

The valve seals used will be commercial metallic static seals fabricated from Inconel "X". They are silver plated, special care will be taken to lap the sealing surfaces to a fine finish.

(2) Design Analysis

A dynamic model of the automatic servovalve is shown in Figure 32. The nomenclature used is listed below:

A = Area of end of spool - in^2

A_c = Control port throat area - in^2

A_e = Exit hole area - in^2

A_f = Feedback port throat area - in^2

A_n = Nozzle ramp annular area - in^2

A_s = Annular area between spool and bore - in^2

C_d = Discharge coefficient

k = Ratio of specific heat of gas

K_v = Vortex gain factor

M = Spool mass - $lb\text{-}sec^2/in$

M_t = Tangential velocity of fluid at outer wall of chamber - Mach No.

P_c = Control pressure - psia

P_e = Ambient pressure - psia

P_f = Feedback pressure - psia

P_i = Pressure at inside periphery of exit hole - psia

P_o = Pressure at outer wall of vortex chamber - psia

P_s = Supply pressure - psia

R = Gas constant - in-lbf/lbm °R

s = Laplace operation of d/dt

$\mathcal{S}$ = Swirl factor = $\frac{W_c M_c}{W_N}$ - non-dimensional

T = Gas temperature - °R

V = Volume under compression at spool end - in^3

W_c = Control flow entering control port, A_c, - lb/sec

W_D = Weight flow displaced by spool - lb/sec

W_f = Control flow entering control port, A_f, lb/sec

W_n = Weight flow leaving nozzle-ramp area A_n - lb/sec

W_o = Exit flow leaving exit hole - lb/sec

W_P = Pressurization weight flow - lb/sec

W_s = Supply flow entering annular chamber A_s - lb/sec

X_o = Quiescent normal clearance between nozzle and ramp surface - in.

y_s = Spool position - in.

δ = Nozzle gain parameter = $\frac{\theta}{X_o}$ - in^{-1}

ψ = Ramp angle - radians

"o" The presence of a subscript zero or additional subscript zero implies the quiescent value of the variable. When subscript zeros appear in equations the variables without subscript zeros are changes about the quiescent values.

$f_1\left[\frac{P_o}{P_c}\right]$ = Ratio subsonic to sonic gas flow for pressure ratio P_o/P_c

$f_2\left[\frac{P_e}{P_i}\right]$ = Ratio subsonic to sonic gas flow for pressure ratio P_e/P_i multiplied by pressure ratio P_i/P_e

$f_3\left[\frac{P_o}{P_c}\right]$ = Ratio of subsonic control momentum to sonic control momentum at pressure ratio P_o/P_c

$K_c = \dfrac{-f_1'\left[\frac{P_{oo}}{P_{co}}\right]}{f_1\left[\frac{P_{oo}}{P_{co}}\right]}$ Control port back pressure sensitivity coefficient

$K_s = \dfrac{-f_1'\left[\frac{P_{oo}}{P_s}\right]}{f_1\left[\frac{P_{oo}}{P_s}\right]}$ Supply annulus back pressure sensitivity coefficient

$K_m = \dfrac{-f_3'\left[\frac{P_{oo}}{P_{co}}\right]}{f_3\left[\frac{P_{oo}}{P_{co}}\right]}$ Control momentum back pressure sensitivity coefficient

$$K_i = -\frac{P_{eo}}{P_{io}} \frac{f_2'\left[\frac{P_{eo}}{P_{io}}\right]}{f_2\left[\frac{P_{eo}}{P_{io}}\right]} \quad \text{Exit hole gain parameter}$$

K_i = 1 if exit hole is flowed sonically.

Referring to Figure 32, the vortex chamber is considered to be a volume, V, at a chamber pressure, P_o. The exit hole is considered to be a fixed orifice with an upstream pressure, P_i, related to the chamber pressure, P_o, and the vortex flow's tangential Mach number. For normalization, the following defined quantity will be used.

$$W_N = \frac{C A_e P_e}{\sqrt{T}}$$

The spool position, y_s, response to small input signals, P_c, is given by,

$$C_1 \frac{P_c}{P_{co}} = \left[\frac{VM}{2 W_{oo} K_i k RT A} s^3 + \left(\frac{C_3}{P_{oo}} - \frac{C_2 W_{fo} K_f}{C_4 P_{fo}}\right) \cdot \right.$$

$$\left. \ldots \frac{M}{2A} s^2 + \frac{P_{oo} A}{W_{oo} K_i RT} s + \frac{C_2 W_{no} \delta}{C_4}\right] y_s . \qquad (24)$$

where:

$$C_1 = \frac{W_{co}}{W_{oo}} \left(1 + K_c \frac{P_{oo}}{P_{co}}\right)\left(\frac{1}{K_i} - 2 K M_{too}^2\right) + \frac{2 K M_{too}^2}{1 + f} \left(1 + K_{m_c} \frac{P_{oo}}{P_{co}}\right) \qquad (25)$$

$$C_2 = \frac{W_{fo}}{W_{oo}} \left(1 + K_f \frac{P_{oo}}{P_{co}}\right)\left(\frac{1}{K_i} - 2 K M_{too}^2\right) + \frac{2 K M_{too}^2}{1 + f} \left(1 + K_{m_f} \frac{P_{oo}}{P_{fo}}\right) \qquad (26)$$

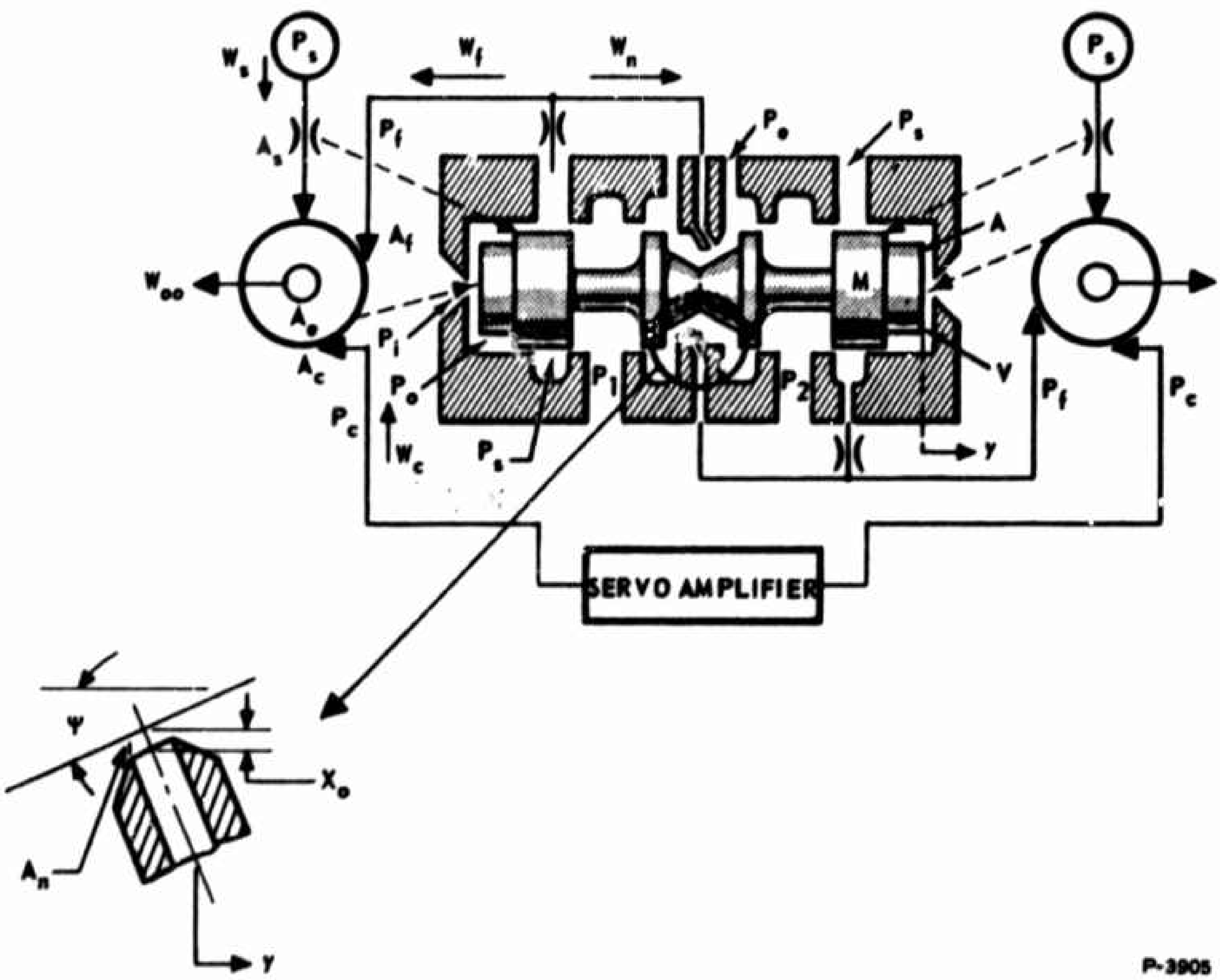


Figure 32 - Automatic Servovalve Schematic

$$C_3 = 1 + \left[\frac{W_{so} K_s P_{oo}}{W_{oo} P_{so}} + \frac{W_{co} K_c P_{oo}}{W_{oo} P_{co}} + \frac{W_{fo} K_f P_{oo}}{W_{oo} P_{fo}}\right]\left[\frac{1}{K_i} - 2\,KM_{too}^{2}\right] \cdots$$

$$\cdots + 2\,KM_{too}^{2}\left[\frac{1}{1+f}\,K_{m_c}\,\frac{P_{co}}{P_{co}} + \frac{f}{1+f}\,K_{m_f}\,\frac{P_{oo}}{P_{fo}}\right] \qquad (27)$$

$$C_4 = W_{fo}\left(1 + K_f\,\frac{P_{oo}}{P_{fo}}\right) + W_{no} + W_{xo}\,K_x\,\frac{P_{fo}}{P_{xo}} . \qquad (28)$$

Normalization of equation (24) yields,

$$\frac{y}{P_c} = \frac{K}{\frac{1}{\beta\omega_{ns}^{3}}\,s^3 + \frac{a}{\beta\omega_{ns}^{2}}\,s^2 + \frac{1}{\beta\omega_{ns}}\,s + 1} \qquad (29)$$

The third order factors a, β and ω_{ns} are given by

$$\omega_{ns}^{2} = \frac{2\,A^2\,k\,P_{oo}}{M\,V} \qquad (30)$$

$$a\,\omega_{ns} = \left[\frac{C_3}{P_{oo}} - \frac{C_2\,W_{fo}\,K_f}{C_4\,P_{fo}}\right]\frac{W_{oo}\,K_i\,k\,RT}{V} \qquad (31)$$

$$\beta\,\omega_{ns} = \frac{C_2\,W_{no}\,W_{oo}\,K_i\,RT\,\delta}{C_4\,P_{oo}\,A} \qquad (32)$$

$$K_v = \frac{C_1\,C_4}{C_2\,P_{co}\,W_{no}\,\delta} \qquad (33)$$

(3) Dynamic Design Factors

A 1/4 inch spool valve with a stroke of ± 0.010 inches was chosen to meet the required area. The critical parameters are,

Spool end area A	-	0.0491 sq. in.
Vortex exit hole area A_e	-	0.0005 sq. in.
Vortex chamber volume V	-	0.0025 cu. in.
Control and feedback port area	-	equal
Spool Mass M	-	0.00011 lb-sec^2/in.
Supply area A_s	-	0.0003 sq. in.

Setting the ram chamber quiescent pressure at 35 psia results in a natural frequency of the servovalve of,

$$\omega_{ns}^2 = \frac{2\,(0.0491)^2\,(1.4)(35)}{(0.00011)(0.0025)} = 860{,}000 \tag{34}$$

$$\omega_{ns} = 926 \text{ rad/sec} = 147 \text{ cps} \tag{35}$$

Maximum flow gain of the vortex pilot stage is desired and set by adjusting the constant $2\,K\,K_i\,M_{too}^{\;2}$ equal to one (1). Since the exit hole is sonic and K_i is equal to 1, $M_{too}^{\;2}$ must be set at 0.1179. Assuming the supply orifices to be choked, the constants in equations (25), (26), (27), and (28) become,

$$C_1 = \frac{1}{1+f}\left(1 + K_{m_c}\frac{P_{oo}}{P_{co}}\right) \tag{36}$$

$$C_2 = \frac{f}{1+f}\left(1 + K_{m_f}\frac{P_{oo}}{P_{fo}}\right) \tag{37}$$

$$C_3 = 1 + \frac{1}{1+f} K_{m_c} \frac{P_{oo}}{P_{co}} + \frac{f}{1+f} K_{m_f} \frac{P_{oo}}{P_{fo}} \tag{38}$$

$$C_4 = W_{fo} (1 + K_f \frac{P_{oo}}{P_{fo}}) + W_{no} \tag{39}$$

Substitution of the known quantities into these and the remaining equations of the analysis section enables one to obtain α and β. For example,

$$P_i = \frac{35}{e^{.5}} = 27 \text{ psia} \tag{40}$$

$$\frac{P_{io}}{P_{eo}} = \frac{27}{14.7} = 1.84 = \frac{W_{oo}}{W_n} \tag{41}$$

$$\mathscr{A}_o = M_{too} \frac{W_{oo}}{W_n} = (0.1179)(1.84) = 0.217 \tag{42}$$

$$A_c = A_f$$

$$\mathscr{A}_o = \frac{A_c}{A_e P_e} \left\{ P_{co} (f_3 \frac{P_{oo}}{P_{co}} + P_{fo} (f_3 \frac{P_{oo}}{P_{fo}}) \right\} \tag{43}$$

Choosing a control area of 0.00005 sq in. and setting $P_{co} = P_{fo}$ results in

$$P_{co} = P_{fo} = 45 \text{ psia} \tag{44}$$

also,

$$\frac{W_{no}}{W_{fo}} = \frac{A_{no}}{A_f f_1 \left[\frac{P_{oo}}{P_{fo}}\right]} \tag{45}$$

and for a 0.010 in. nozzle, a gain δ of 10 and a ramp angle of 5°

$$A_{no} = \frac{\pi (0.010)(0.0872)}{30} = 0.000091 \text{ sq. in.}$$

$$\frac{W_{fo}}{C_4} = \frac{1}{4.85}$$

$$C_1 = 2.09$$

$$C_2 = 2.09$$

$$C_3 = 5.18$$

$$K_f = 1.50$$

From equation (31)

$$a \omega_{ns} = \left[\frac{5.18}{35} - \frac{2.09}{40} \frac{1.50}{4.85}\right] \frac{(0.00020)(1.4)(640)(530)}{0.0025} = 6{,}200$$

$$a = 5.5$$

Similarly,

$$\beta = 1.0$$

Figure 33 shows the valve response characteristics.

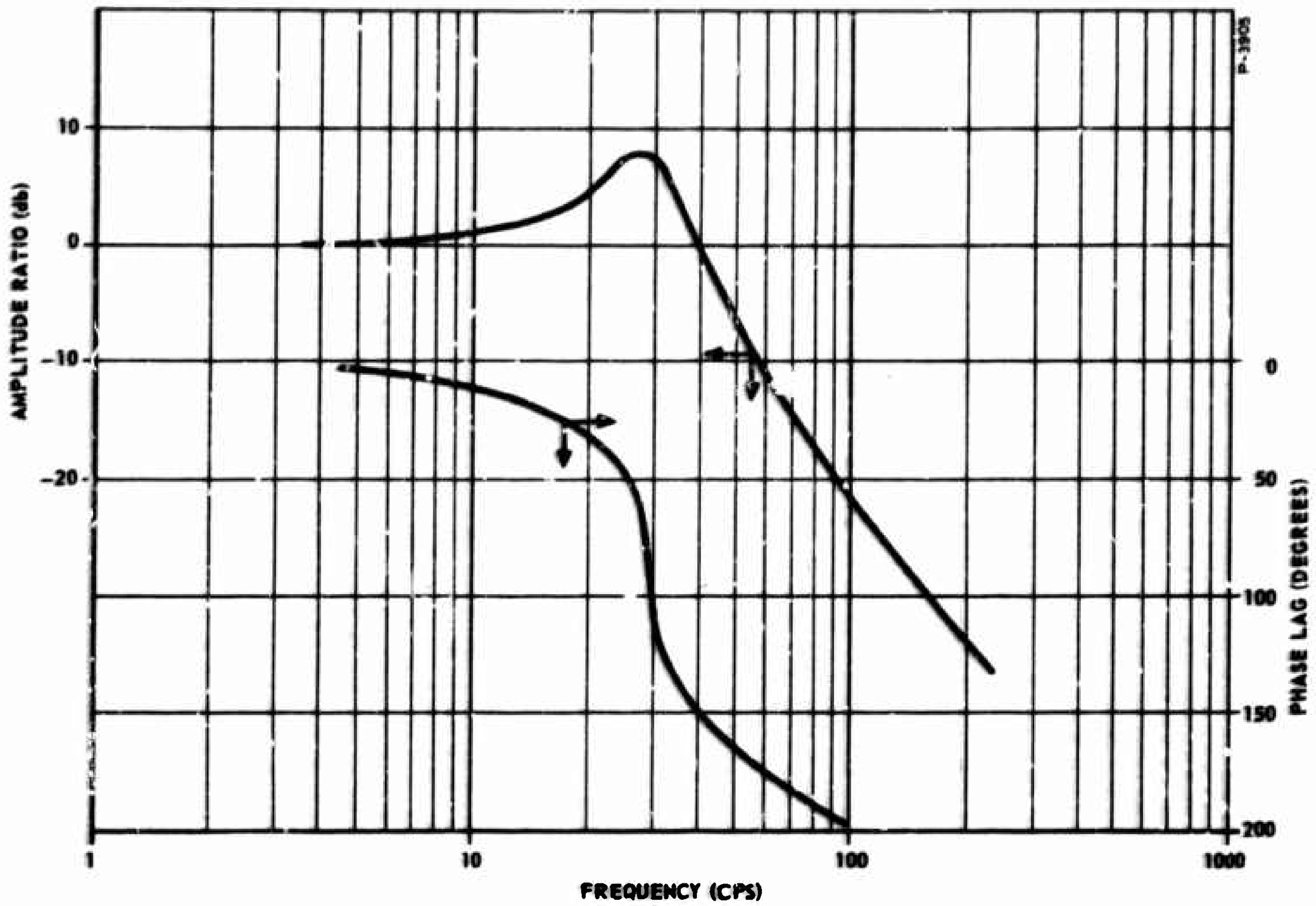


Figure 33 - Automatic Servovalve Frequency Response

<u>Static Design Factors</u>

From equation (33)

$$\left.\frac{y_s}{P_c}\right|_{ss} = \frac{\frac{C_1}{P_{co}}}{\frac{C_2 W_{no} \delta}{C_4}} = \frac{\frac{2.09}{45}}{\frac{2.09 \; 30}{4.85}} = \frac{1}{280} \qquad (46)$$

and for,

$$y_s = 0.010 \text{ in.}$$

$$P_c = 0.010 \, (280) = 3.1$$

Thus, the required differential pressure across the valve will be

$$2\,(3.1) = 6.2 \text{ psid}$$

for a full stroke error.

D. Manual Valve

The manual servovalve provides the power actuator supply flow and is driven by the pilot input linkage and by the atuomatic actuator during automatic operation. Figure 34 is a sectional view of the valve showing the design philosophy. A closed center, four-way spool forms the heart of the valve. The spool is directly connected to the pilot input linkage with the bell crank. As the pilot moves the linkage the spool is displaced with respect to the body, proportional to the linkage motion. The power actuator shaft which is directly connected to the valve body in manual operation must rotate the valve body to keep the spool at null. Since the input linkage pivot point is on the center line of the output shaft, the shaft must move through the same angle the input linkage does.

The maximum valve area is 0.040 in^2 on each land. The servo loop gain requirement is such that the valve must open proportionally to the bell crank displacement for the first 0.08 inches of travel at which time it must be fully open. However, the bell crank must be free to displace an additional 0.82 inches. In order to provide these requirements a special land design is required. Figure 35 is a sketch of the design. Slots equal in width to the required active stroke are milled into the spool sleeve. The spool will uncover these slots proportional to its stroke until the full area is open. Thereafter, the spool is free to continue without additional area being opened.

This design results in almost zero force required to actuate the valve thus the pilot will feel no additional effort when operating the pneumatic servo in parallel with the hydraulic servo.

The spool, spool sleeve, and body will be fabricated from 440C stainless steel and oxide coated for dry film lubrication between the spool and spool sleeve. The bell crank will be an integral part of the assembly thus, backlash will be eliminated as well as undesirable forces on the spool assembly. A flex pivot will be used to attach the bell crank to the valve body for a zero friction pivot point.

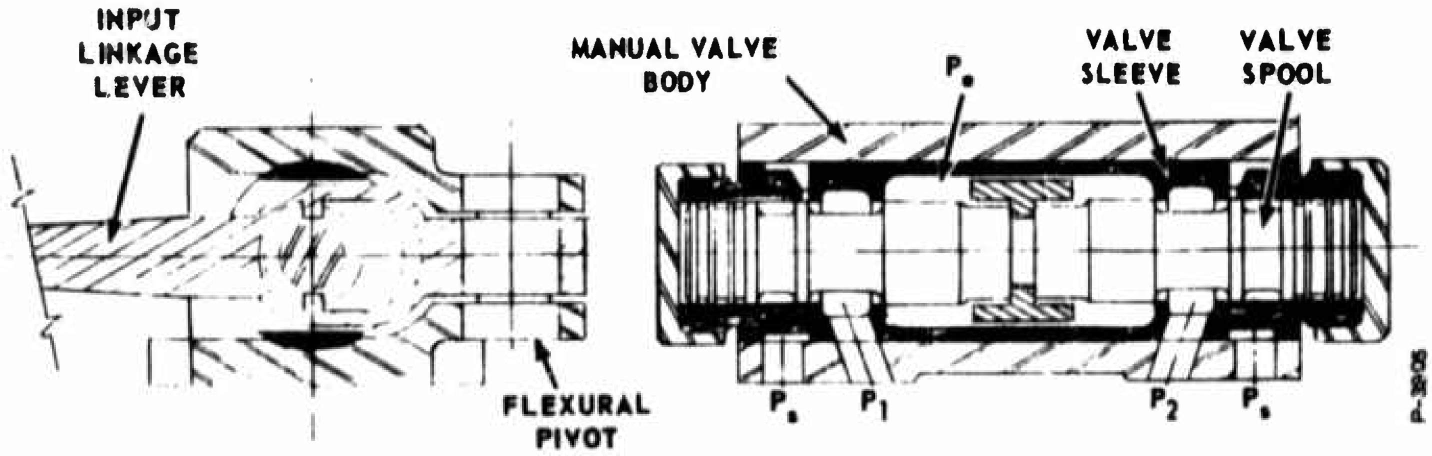


Figure 34 - Manual Servovalve Assembly

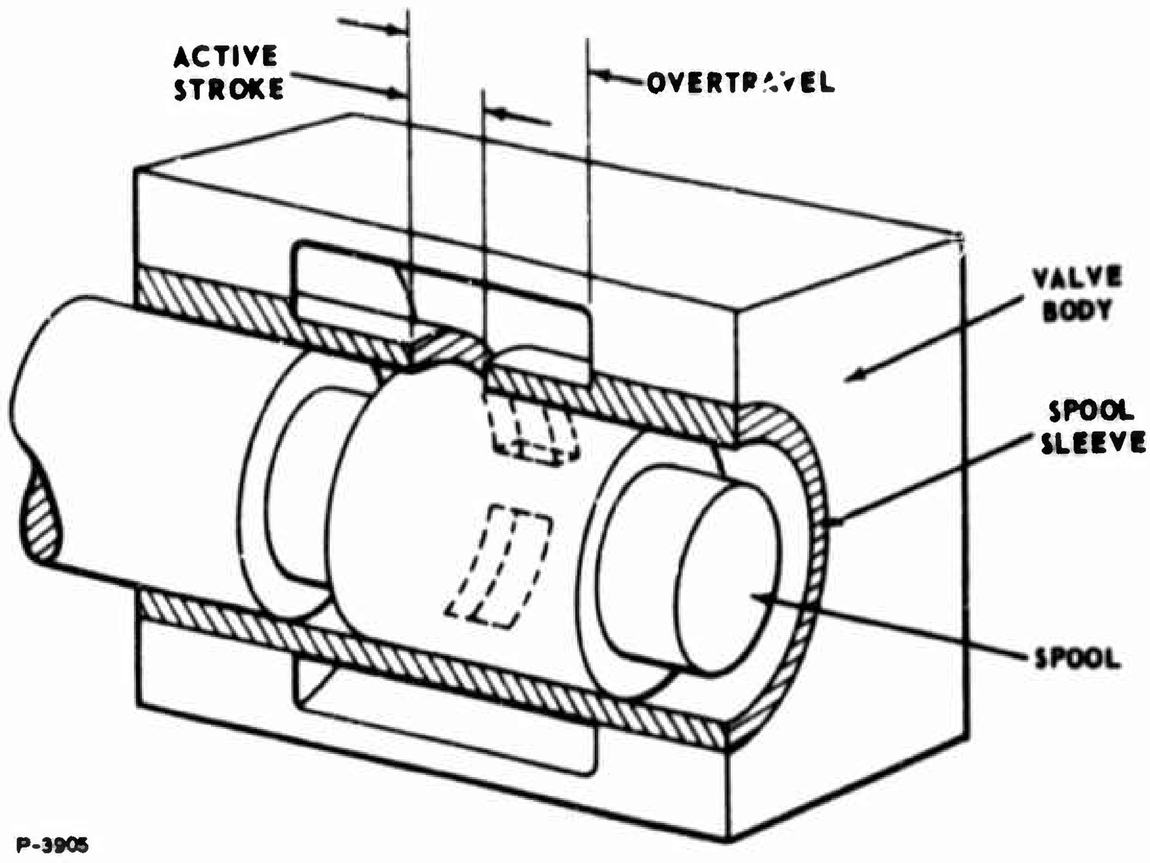


Figure 35 - Manual Servovalve Spool and Sleeve Design

The valve body assembly will mount directly to the Actuator-Manual Valve Latch. The supply flow will pass from the valve body through this latch into the power actuator through sliding seals between the latch and the actuator housing.

E. Clutch Mechanism

(1) Introduction

A number of methods are used to actuate and control clutches with the best choice depending on the application. The three basic types of actuation are: mechanical, electrical, and fluid (hydraulic or pneumatic). The mechanical system is ruled out in this application because of the complexity of the linkage required to actuate the clutch and the requirement for automatic operation. Most electrically actuated clutches are of the electromagnetic type, and because of the large forces involved to keep the clutch engaged, this type of clutch would be excessively heavy and the size would exceed the allowable envelope. A pneumatically actuated clutch is selected because of the requirement that the present hydraulic system not be modified and the fact that the DYNAVECTOR is a pneumatic actuator. Using a pneumatic clutch also reduces the weight of the clutch and provides for fail-safe operation as the clutch will automatically disengage when there is loss of pneumatic pressure, allowing hydraulic or mechanical control of the rudder.

The following clutch operational requirements are design objectives:

(a) The clutch must remain engaged under stall loading with a zero coefficient of friction.

(b) The clutch must disengage under stall loading with a coefficient of friction of 0.375 with pressure assist and 0.2 without pressure assist.

(2) Nomenclature

F_f = Friction force (lb)

F_k = Kickoff spring force (lb)

F_p = Piston force (lb)

F_s = Gear separating force (lb)

F_{ts} = Tension spring force (lb)

F_n = Normal force (lb)

F_t = Tangential force (lb)

ϕ = Pressure angle (deg)

T_o = Output torque (in-lb)

R_p = Pitch radius (in)

μ = Coefficient of friction

D_p = Pitch diameter (in)

b = Tooth length (in)

S_s = Shear stress (psi)

S_c = Compressive stress (psi)

A_t = Tooth area (in^2)

w = Washer width (in)

t_p = Tooth thickness at the pitch line (in)

n = Number of waves in wave washer

N = Number of teeth

A_m = Stress area (in^2)

d_w = Wire diameter (in)

t_w = Wall thickness (in)

D = Spring mean diameter (in)

ν = Poisson's ratio

d = Piston diameter (in)

ℓn = Natural logarithm

S_r = Radial stress (psi)

d_ℓ = Diameter of concentric ring on which face gear separating force acts (in)

s_t = Tangential Stress (psi)

P_s = Pressure (psia)

Δ = Deflection (in)

α = Constant dependent on ratio of outer radius to inner radius of a circular plate

E = Modulus of elasticity (psi)

Subscripts o and i refer to outside and inside respectively

Subscript 1 refers to face gear

(3) Clutch Design

(a) Design Description

The pneumatic tooth clutch, Figure 36, provides high torque capacity and positive engagement. It is an on-off device that consists of essentially three parts: splined output shaft of the actuator, a toothed piston having an internal spline mating it to the actuator output shaft, and a toothed output member connected to the rudder horn through a torque-limiting splined shaft.

When the volume between the actuator output and the piston is pressurized, and the volume between the piston and the output member is vented, the piston moves into engagement with the clutch output member, the teeth on the piston meshing with the teeth on the output member. Positive alignment is achieved by incorporation of a single point engagement design in the face gear teeth. When the engagement chamber is evacuated and the disengagement chamber pressurized, release plungers assist in separating the two clutch members forcing the piston back toward the actuator output shaft. The wave washer moves the piston toward the actuator output shaft to assure positive disengagement of the clutch and to minimize the possibility of accidental engagement due to vibrations or accelerations along the shaft axis.

(b) Force Analysis

Preliminary design layouts of the DYNAVECTOR actuator assembly in the F101B aircraft result in the maximum space envelope for the clutch as indicated by the clutch housing outline shown in Figure 36. To optimize the clutch package and to determine actuation

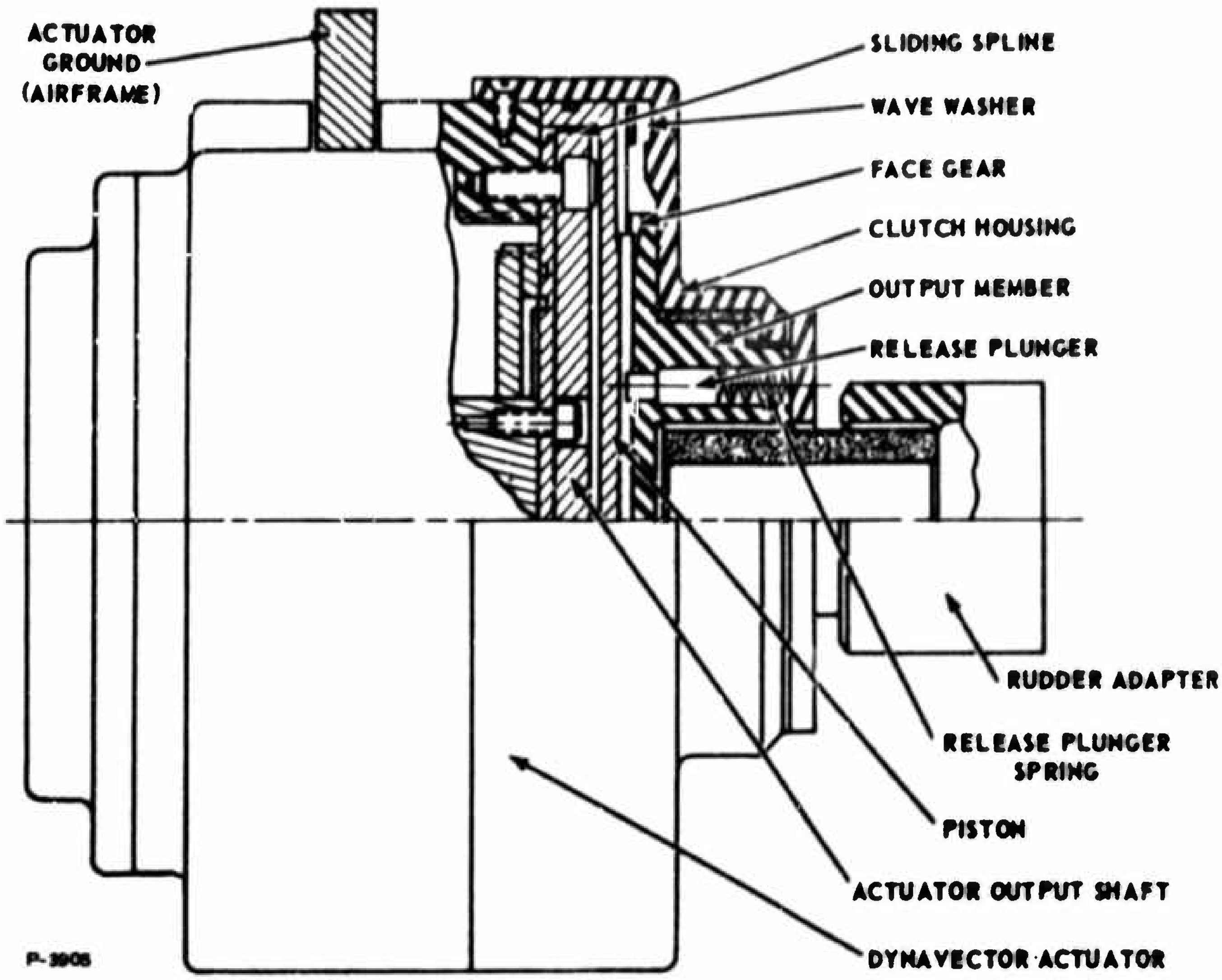


Figure 36 - Pneumatic Tooth Clutch

forces, a force analysis is conducted. Summing forces in the axial direction, the piston force, when the clutch is engaged, is expressed as

$$F_p = F_{s_1} + F_r + F_w - (F_{f_1} + F_f) \qquad (47)$$

The force required for disengagement without pressure assist, is obtained from equation (47) by setting $F_p = 0$.

$$F_{s_1} + F_r + F_w > F_{f_1} + F_f \qquad (48)$$

The friction forces (F_{f_1} and F_f) and the gear separating force (F_{s_1}) are derived from the following equations:

$$F_f = \mu F_n \tag{49}$$

$$F_n = \frac{F_t}{\cos \phi} = \frac{T_o}{R_p \cos \phi} \tag{50}$$

$$F_s = \frac{T_o}{R_p} \tan \phi \tag{51}$$

The sliding spline diameter is dictated by the actuator design while the face gear diameter is governed by the stresses in the output member. The sliding spline and face gear tooth form and pressure angle are identical to those used by Eclipse-Machine Division, Bendix Corporation, in their electromagnetic clutches. The design data is summarized in Table VI.

The piston force required to insure continuous engagement for a stall torque of 10,500 in-lbs is obtained from equation (47) for various face gear pressure angles and plotted for varying coefficients of friction in Figure 37. The force required to disengage the clutch is

Table VI - Pneumatic Clutch Design Data

Sliding Spline	Face Gear
Pressure angle = 30°	Pressure angle = 20°
Involute tooth form	Straight-sided tooth form
Pitch diameter = 5.833 in	Pitch diameter = 4.465 in (middle of tooth)
70 teeth	140 teeth
12/24 pitch	Tooth OD = 4.62 in
Addendum = 0.0416 in	Tooth ID = 4.31 in
Dedendum = 0.052 in	Tooth height (at OD) = 0.064/0.066 in
Tooth length = 0.340 in	Tooth spacing (@ OD tooth root) = 0.0275/0.0265

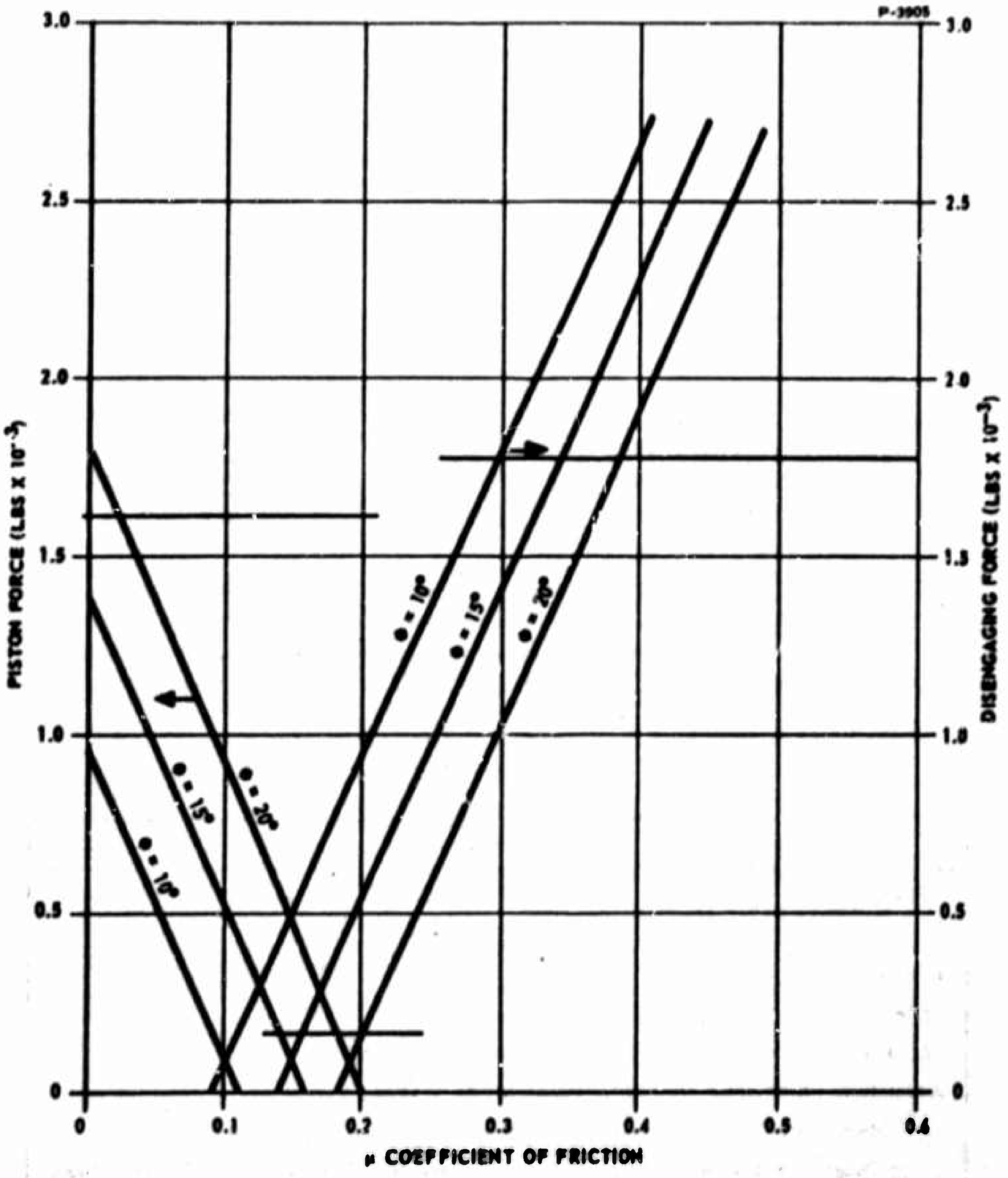


Figure 37 - Clutch Piston Force Requirement Versus Tooth Coefficient of Friction

obtained for various face gear pressure angles and coefficients of friction and also plotted in Figure 37. The coefficient of friction is assumed to be the same in the sliding spline and the face gear. The release spring force and the wave washer force at full engagement are assumed to be constants of 120 lbs and 55 lbs, respectively. The piston force required to insure clutch engagement for a pressure angle of 20 degrees under stall conditions and zero coefficient of friction is 1810 lbs, from Figure 37. The maximum piston diameter available is 6.40 in. which yields a pressure area of 32.2 sq. in. For a differential pressure of 50 psid, the maximum piston force is 1,610 lbs, as indicated by the horizontal line to the left in Figure 37. As is evident from the graph, this force will not keep the clutch engaged under stall conditions and zero friction. Since the DYNAVECTOR actuator is capable of 10,500 in-lbs output torque for a supply pressure of 50 psig, the clutch must remain engaged for a 50 psig supply pressure also. The piston diameter cannot be increased without exceeding the clutch envelope. Reducing the pressure angle of the face gear will reduce the separating force and, therefore the piston force required to keep the clutch engaged as shown in Figure 37. However, a decrease in the face gear pressure angle decreases the maximum coefficient of friction for which the clutch will disengage, as is evident from Figure 37. The design goal of fail-safe operation for loss of pressure to the actuator requires that the clutch disengage without pressure assist. The anticipated coefficient of friction for the candidate materials is 0.15 and does not vary significantly with time. To allow for all contingencies, a range in coefficient of friction of 0.03 to 0.20 is assumed. For this range of coefficients, the clutch will remain engaged and separate without pressure assist for a face gear pressure angle of 20 degrees and is capable of disengaging with pressure assist for a coefficient of friction up to 0.375.

(c) Stress Analysis

The spline tooth stress is determined from the following equations:

$$S_s = \frac{1.2732\ T_o}{(D_p^2)\ b} \tag{52}$$

$$S_c = \frac{2.5\ T_o}{(D_p^2)\ b} \tag{53}$$

which assume all teeth carrying the load. Assuming that only 25 percent of the teeth carry the load, the stresses on the spline are

$$S_s = \frac{1.2732\ (10{,}500)\ (4)}{(0.34)\ (5.833)^2} = 6{,}320 \text{ psi}$$

$$S_c = \frac{2.5\ \ (10{,}500)\ (4)}{(0.34)\ (5.833)^2} = 9{,}090 \text{ psi}$$

which is well below the yield strength of the material selected for the spline members - a nitrided steel.

The stresses in the face gear teeth are obtained from the following equations:

$$S_c = \frac{F_t}{A_t N} \qquad (54)$$

$$S_s = \frac{F_t}{t_p N b} \qquad (55)$$

Since the face gear teeth are cut on a radial line, the tooth is more highly stressed at the inner diameter of the tooth; therefore, the tangential force and the tooth area at this point are assumed for the whole tooth length when calculating the tooth stresses. Assuming that 25 percent of the teeth carry the load, the tooth stresses are

$$S_c = \frac{4860\ (4)}{(0.0101)\ (140)} = 13{,}770 \text{ psi}$$

$$S_s = \frac{4860\ (4)}{(0.05)\ (140)\ (0.155)} = 17{,}920 \text{ psi}$$

which are well below the yield strength of nitrided steel.

Stresses in the piston are due to tooth separating forces, torsional loads and pressure forces. For this preliminary design, the piston is assumed to consist of two parts; a cylinder with an internal spline and a disk with face gear teeth. The cylindrical portion of the piston is subjected to shear stress due to the torque and a tangential stress due to spline separating and pressure forces, which are determined from equations (56) and (57), respectively.

$$S_s = \frac{16\, T_o\, d_o}{\pi\, (d_o^4 - d_i^4)} \tag{56}$$

$$S_t = \frac{P_s\, d}{2\, t_w} = \frac{(F_s + P_s\, A_m)\, d}{2\, A_m\, t_w} \tag{57}$$

The area over which the spline separating force is assumed to act is at the root of the tooth which is

$$A_m = \pi\, (5.937)\, (0.470) = 8.76 \text{ in}^2$$

and the tangential stress from equation (57) is

$$S_t = \frac{2518\ (5.937)}{(2)\, (8.76)\, (0.2315)} = 3830 \text{ psi}$$

The shear stress for stall conditions from equation (56) is

$$S_s = \frac{16\, (10{,}500)\, (6.40)}{\pi\, (6.4^4 - 5.937^4)} = 726 \text{ psi}$$

The disk portion of the piston is subjected to face gear separating forces, pressure forces and torsional loads. The cylindrical portion of the piston is assumed not to affect the stresses in the disk. The force due to the release pins is assumed negligible. When the clutch is engaged, the stresses at the disk edge are obtained from equations (58) and (59)

$$S_r = S_{r_1} - S_{r_2} = \frac{3\, d^2 (\Delta P)}{16\, t_w^2} - \frac{3\, F_{s_1}}{2\, \pi\, t_w^2} \left[1 - \left(\frac{d_\ell}{d_o} \right)^2 \right] \qquad (58)$$

$$S_t = S_{t_1} + S_{t_2} + S_{t_3}$$

$$= S_{r_1} \nu + S_{r_2} \nu + \frac{F_t}{\pi\, d_\ell\, t_w} = S_{r_{max}} \nu + \frac{F_t}{\pi\, d_\ell\, t_w} \qquad (59)$$

When the clutch is disengaging with pressure assist and the face gear teeth are still in mesh, the radial stresses are additive and are obtained from equation (60) while the tangential stresses are the same as those obtained from equation (59)

$$S_r = S_{r_1} + S_{r_2} = \frac{3\, d^2 (\Delta P)}{16\, t_w^2} - \frac{3\, F_{s_1}}{2\, \pi\, t_w^2} \left[1 - \left(\frac{d_\ell}{d_o} \right)^2 \right] \qquad (60)$$

The deflections at the disk center must not be excessive and they are obtained for the engaged clutch and when the clutch is disengaging from equations (61) and (62), respectively.

$$\Delta = \frac{3\, d^4 (1 - \nu^2)(\Delta P)}{64\, E\, t_w^3} - \frac{3\, F_{s_1} (1 - \nu^2)}{8\, \pi\, E\, t_w^3} \left[\frac{1}{2} \left(d_o^2 - d_\ell^2 \right) - d_\ell^2 \,\ell n \left(\frac{d_o}{d_\ell} \right) \right] \qquad (61)$$

$$\Delta = \frac{3\, d^4 (1 - \nu^2)(\Delta P)}{64\, E\, t_w^3} - \frac{3\, F_{s_1} (1 - \nu^2)}{8\, \pi\, E\, t_w^3} \left[\frac{1}{2} \left(d_o^2 - d_\ell^2 \right) - d_\ell^2 \,\ell n \left(\frac{d_o}{d_\ell} \right) \right] \qquad (62)$$

The radial stresses at the edge and the disk deflection at the center are plotted for the assumed configuration in Figure 38 for the clutch in engagement and Figure 39 for the clutch disengaging and the face gear teeth still in mesh. The deflection should be kept small; however the weight of the piston must be kept minimum and, therefore a wall thickness of 0.20 inch is assumed.

The tangential or torsional stress for this wall thickness and F_t acting at the mean tooth diameter from equation (59) is

$$S_t = (17{,}250)\,(0.3) + \frac{4{,}710}{\pi\,(4.465)\,(0.2)} = 6.855 \text{ psi}$$

and the piston design is adequate.

The most highly stressed part of the output member of the clutch is the 0.200 inch thick disk from the face gear to the center hub which is subjected to torsional and gear separating forces. The critical area is at the hub where the stress due to gear separating forces is in bending, or a radial stress, and is expressed as

$$S_r = \frac{3\,F_{s_1}}{2\,\pi\,t_w^2} \left[\frac{2\,d_o^2\,(1+\nu)\ln\frac{d_o}{d_i} + (d_o^2 - d_i^2)\,(1-\nu)}{d_o^2\,(1+\nu) + d_i^2\,(1-\nu)} \right] \qquad (63)$$

Substituting the dimensions and loads into equation (63) and using Poissons ratio of 0.3 (for steel), the stress is

$$S_r = \frac{3\,(1642)}{2\,\pi\,(0.200)^2} \left[\frac{2\,(4.62)^2\,(1.3)\ln\frac{\overline{4.62}^2}{3} + (4.62 - 9)\,(0.7)}{(4.62)^2\,(1.3) + 9\,(0.7)} \right]$$

$$= 18{,}850 \text{ psi}$$

The shear stress due to the torsional load is also maximum at the hub and is expressed

$$S_s = \frac{F_{t_1}\,d_o}{\pi\,d_i^2\,t_w} \qquad (64)$$

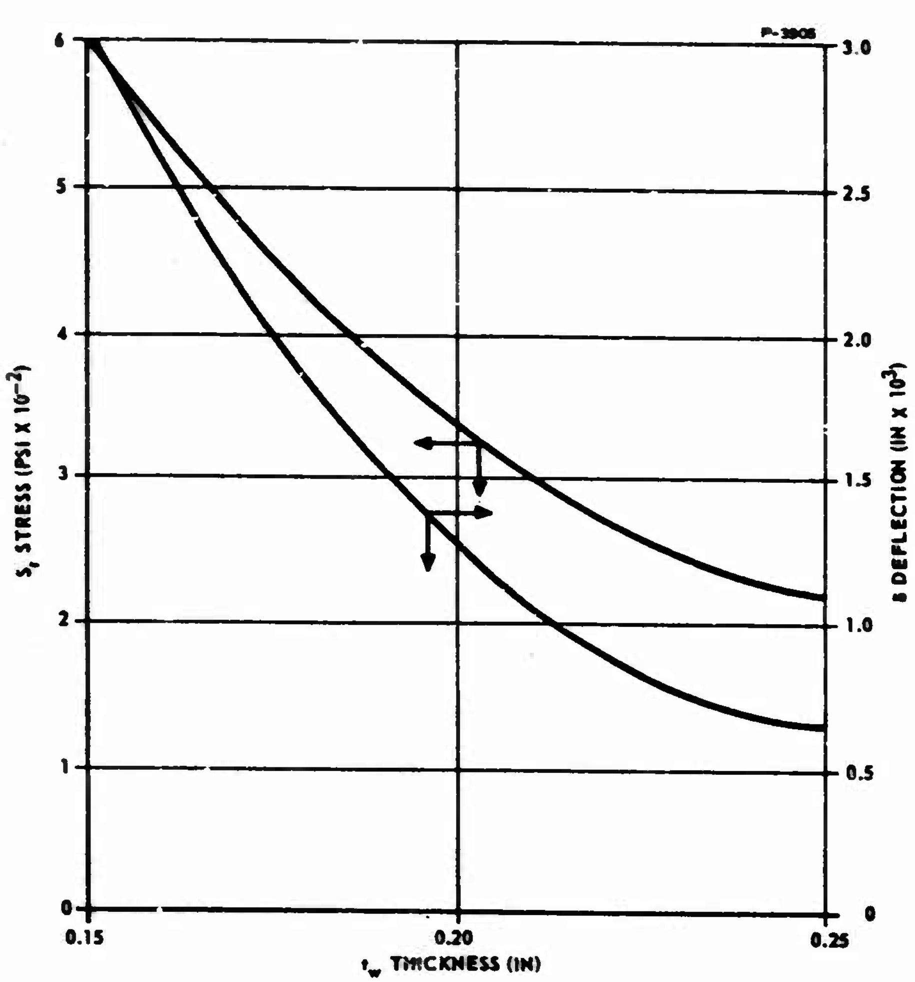


Figure 38 - Clutch Piston Radial Stress and Deflection Versus Wall Thickness (Clutch Engaged)

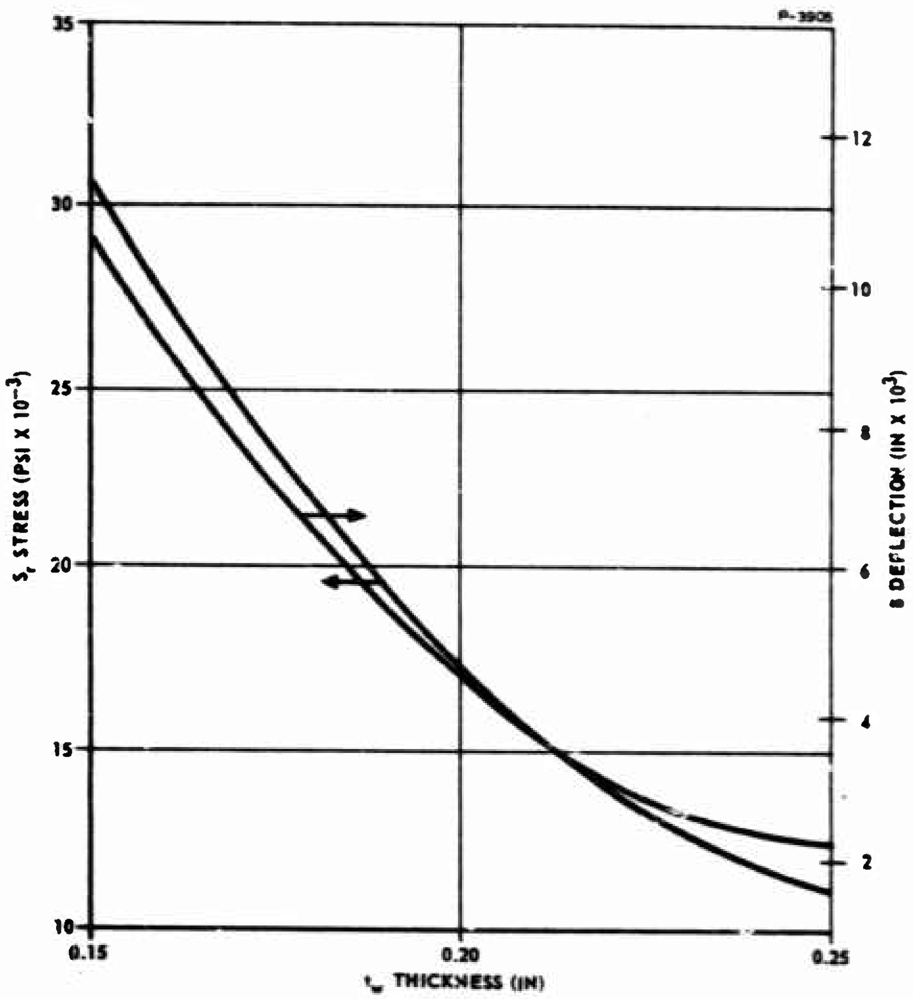


Figure 39 - Clutch Piston Radial Stress and Deflection Versus Wall Thickness (Clutch Disengaging)

$$S_s = \frac{(4520)\ (4.62)}{9\ \pi\ (0.2)} = 3690 \text{ psi}$$

Combining the radial stress, which is either compressive or tensile and the shear stress results in

$$S_{r_{max}} = \frac{18{,}850}{2} + \sqrt{\left(\frac{18{,}850}{2}\right)^2 + (3{,}690)^2}$$

$$= 19{,}535 \text{ psi}$$

$$S_{s_{max}} = \sqrt{\left(\frac{18{,}850}{2}\right)^2 + (3{,}690)^2}$$

$$= 10{,}110 \text{ psi}$$

The deflection of the face gear due to the gear separating forces is determined by the following equation

$$\Delta = \frac{\alpha\ F_{s_1}\ d_o^2}{4\ E\ t_w^3} \qquad (65)$$

where α is dependent on the ratio of the outer to the inner diameter and is equal to 0.025 for this case. For stall conditions and an E of 30×10^6 psi, the deflection at the face gear if unsupported is

$$\Delta = \frac{(0.025)\ (1642)\ (4.62)^2}{4\ (30 \times 10^6)\ (0.2)^3} = 0.000915 \text{ in.}$$

The wave washer is designed so that its output force when the clutch is engaged is 55 lbs. The stress and deflection equations for a wave washer are:

$$S = \frac{3\ \pi\ F\ D}{4\ w\ t_w^2\ n^2} \qquad (66)$$

$$\Delta = \frac{1.94\ F_o\ D^3}{E\ w\ t_w^3\ n^4} \tag{67}$$

The wave washer should not be deflected to its solid height, because a load at solid height cannot be held with any uniformity. A force of approximately twenty (20) lbs on the piston is desired when the clutch is completely disengaged to alleviate the problems of accidental engagement due to vibrations or accelerations. The desired spring rate is, therefore, approximately 390 lb/in. A wave washer with the following dimensions is selected

$$D = 6.0 \text{ in.}$$

$$w = 0.275 \text{ in.}$$

$$E = 30 \times 10^6 \text{ psi}$$

$$t_w = 0.0625 \text{ in.}$$

$$n = 3$$

The spring rate for the above wave washer is

$$\frac{F}{\Delta} = \frac{E\ w\ t_w^3\ n^4}{1.94\ D^3} = \frac{30 \times 10^6\ (0.275)\ (6.25)^3 \times 10^{-6}\ (3)^4}{1.94\ (6)^3}$$

$$= 391 \text{ lb/in}$$

which is the desired spring rate. The deflection at a force of 55 lbs is

$$\Delta = \frac{55}{391} = 0.1408 \text{ in.}$$

The pre-load deflection from free height is obtained by subtracting the stroke of the piston (0.090 in.) from the above deflection and is 0.0508 in. The pre-load force is

$$F = 0.0508 \times 391 = 19.85 \text{ lb}$$

which is the desired pre-load. The maximum stress for a solid height deflection 0.170 inches from equation (66) is

$$S = \frac{3\pi (0.17 \times 391)(6)}{4 (0.275)(6.25)^2\, 10^{-4}\, (3)^2}$$

$$= 97{,}500 \text{ psi}$$

which is satisfactory.

The helical coil release springs are standard springs manufactured by Eclipse Machine Division and have a spring rate of 50 lb/in. A release force of 15 lbs per spring is required to disengage the clutch in the fail-safe mode. The stress in the spring is expressed as

$$S = \frac{2.55\, F\, D}{d_w^3} \tag{68}$$

and is,

$$S = \frac{2.55 (15)(0.2475)}{(4.75)^3 \times 10^{-6}}$$

$$= 88{,}000 \text{ psi}$$

which is well below the endurance limit for the material used in the spring.

(d) Weight

The clutch assembly, as shown in Figure 36, has a design weight of eight (8) pounds. An electromagnetically actuated clutch of the equivalent torque capacity would weigh eighteen (18) pounds.

F. Automatic Actuator Servo Amplifier and Position Transducer

(1) Servo System

The automatic actuator servo loop consists of a fluid state servoamplifier, a fluid state position transducer, a servovalve, and the automatic actuator. The input or demand to the servo loop comes from the autopilot and is a position command signal. Figure 40 is a schematic of the servo circuit. The autopilot input signal at ± 2 psid pressure is summed with the position feedback signal in the summing amplifiers to produce a servo error signal. This error signal is amplified by a venjet amplifier circuit and used to drive the servovalve. The servovalve in turn operates the actuator which positions the position transducer on the output shaft to cancel out the error signal. Each stage operates in push-pull mode to eliminate pressure regulation requirements. The main 50 psig supply pressure is dropped through a series of orifices to provide the required pressure to each stage.

(2) Servo Amplifier

The summing amplifiers, vortex valves, and venjets together represent the servo amplifier. The summing amplifier is a low gain

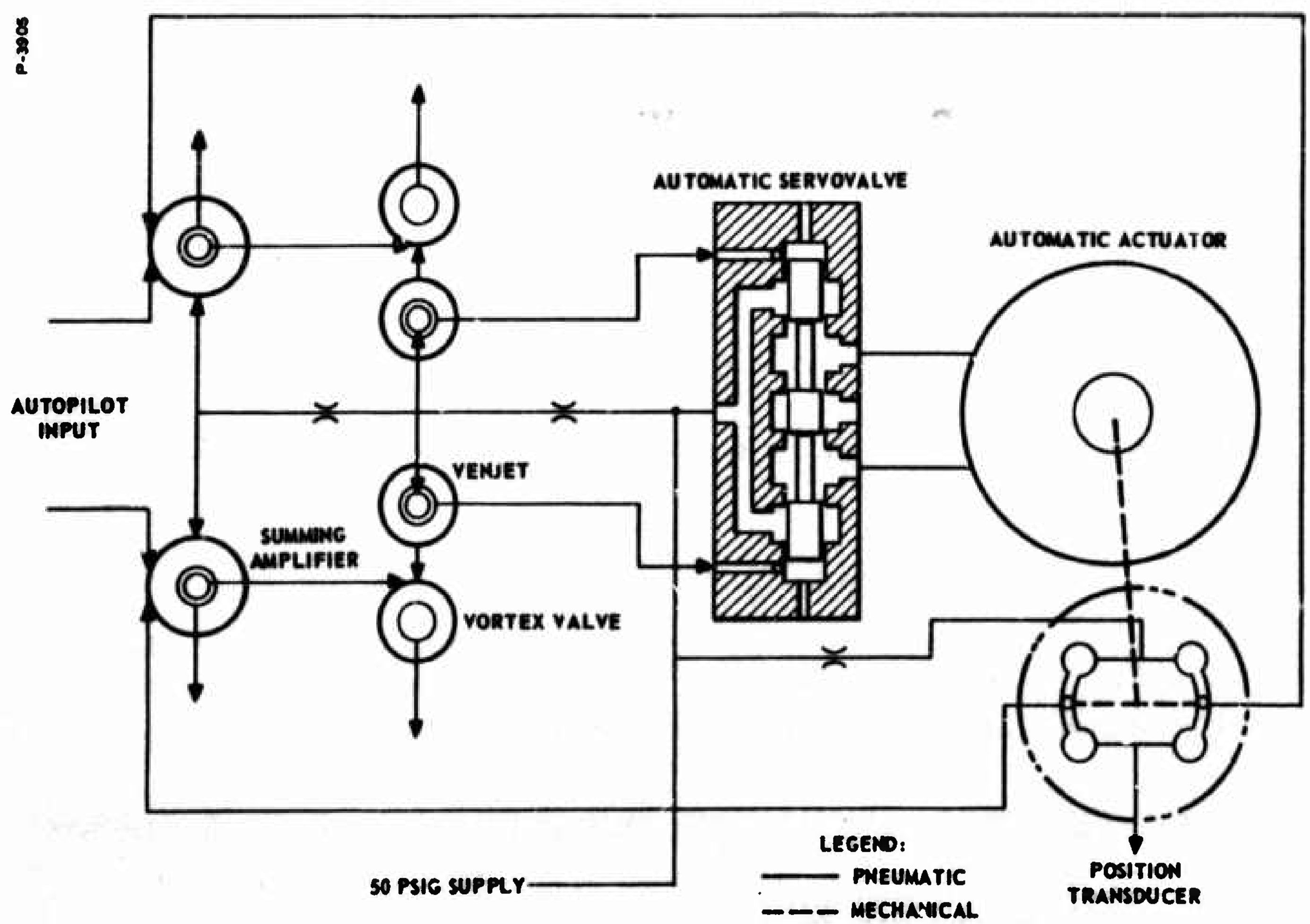


Figure 40 - Automatic Actuator Servosystem

vortex amplifier. It accepts the low level input and position feedback signals and preamplifies their sum to provide an error signal. As the input differential pressure swings positive the feedback differential pressure must become negative in order for the output differential to remain at zero.

Figure 41 is a design drawing of the summing amplifier. The vortex chamber diameter will be 1/4 inch. The unit is flange mounted and all ports are connected through this mounting flange. The amplifier itself will be a diffusion bonded assembly without internal seals or joints. The overall dimensions will be about 1/2 inch in diameter by 3/8 inch high. The material will be 440C stainless steel.

The error signal is amplified by a novel Bendix amplifier called a venjet or vented-jet amplifier. It utilizes a single power jet and receiver and is controlled by varying the pressure surrounding the power jet. Gains of up to 25 psi per psi input have been obtained making this device suitable for error signal amplification. A conventional vortex valve is used to control the vent pressure by throttling the vent flow. This venjet-vortex valve combination with give the highest power gain possible and therefore will also serve as the driver stage for the servovalve.

Figure 42 is a design drawing of the venjet-vortex valve circuit. It also is fabricated from 440C stainless steel and diffusion bonded together to eliminate seals and joints. The summing amplifier flange mounts to this assembly which in turn mounts to the actuator housing. The total amplifier weight is 2 oz.

(3) Position Transducer

The position transducer to be used in the automatic valve position control loop is a currently developed fluid state device.

The fluid state position transducer, provides a pressure signal that is a function of rotary position of the automatic actuator relative to the power actuator position. Figure 43 shows the transducer configuration designed for ± 5 degrees oscillation of the automatic actuator relative to the power actuator. The transducer body will be mounted to the power actuator output shaft with the converging variable area ducts, designed to create a linear pressure gradient mounted on the automatic actuator output member surface. Static pressure taps located on the power actuator surface remain stationary while the ducts rotate with the automatic actuator.

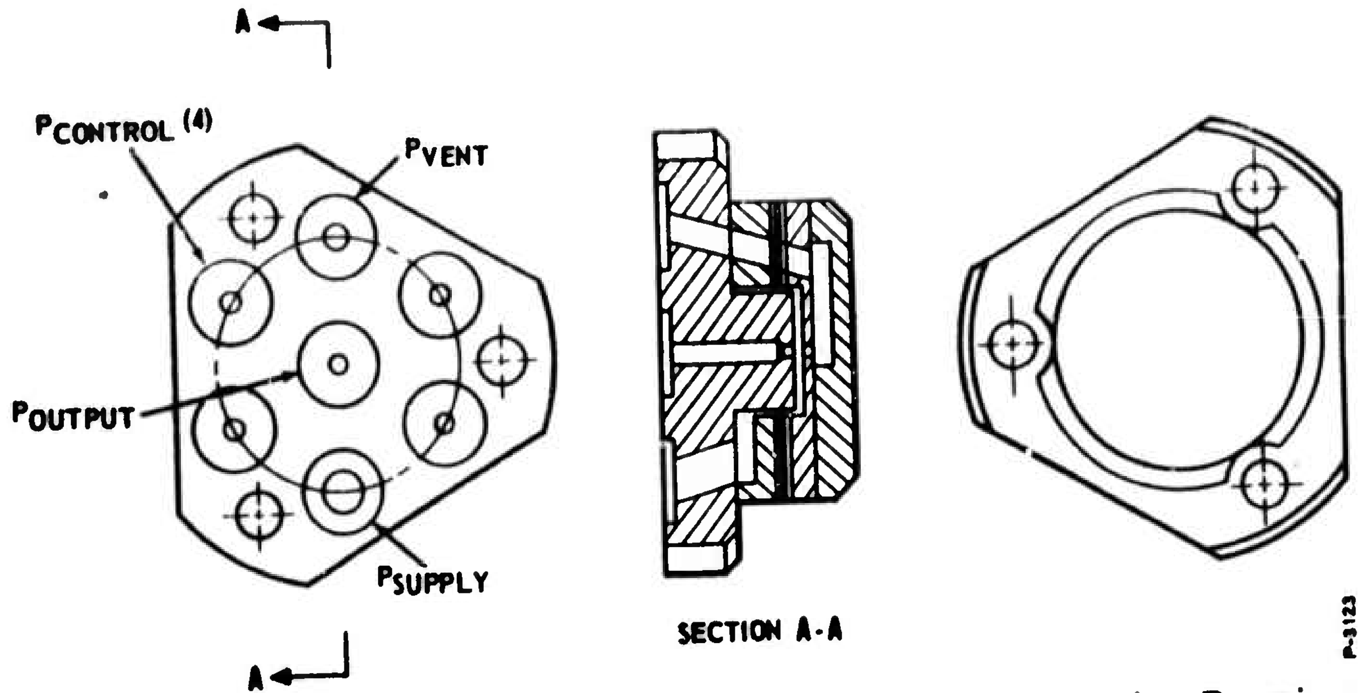


Figure 41 - Vortex Summation Amplifier Preliminary Design Drawing

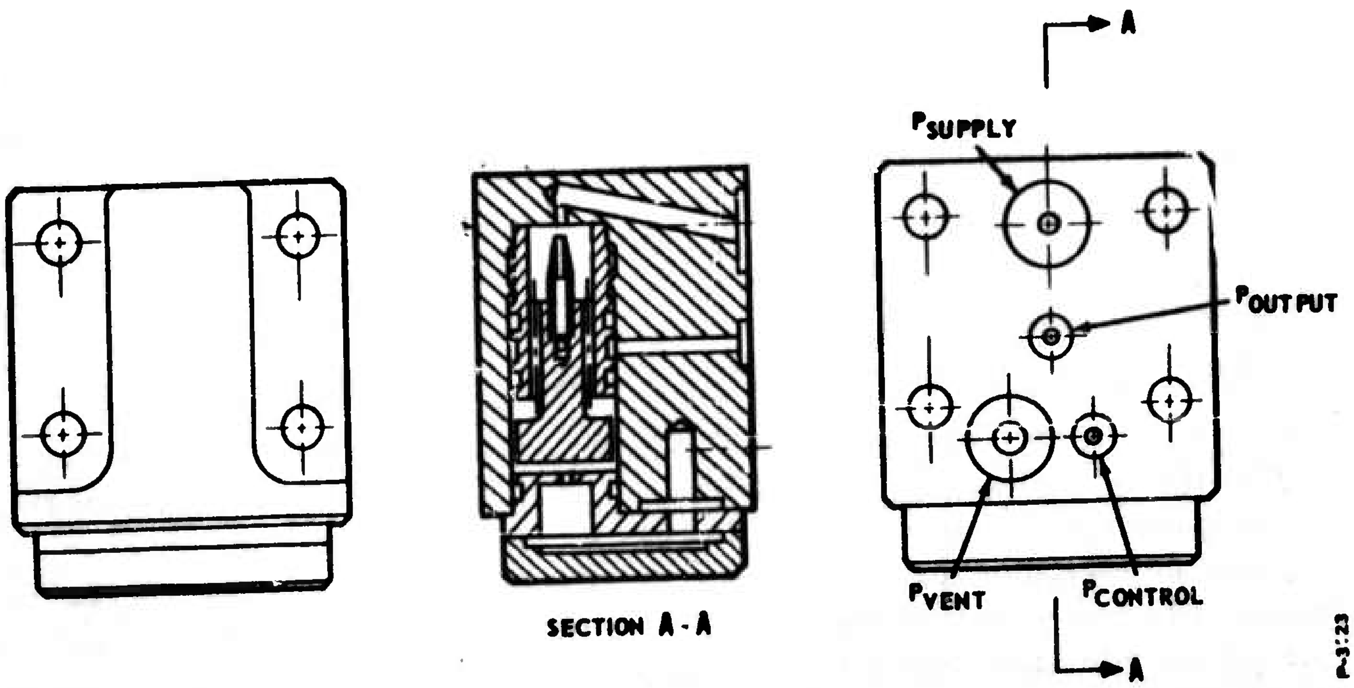


Figure 42 - Venjet-Vortex Valve Preliminary Design Drawing

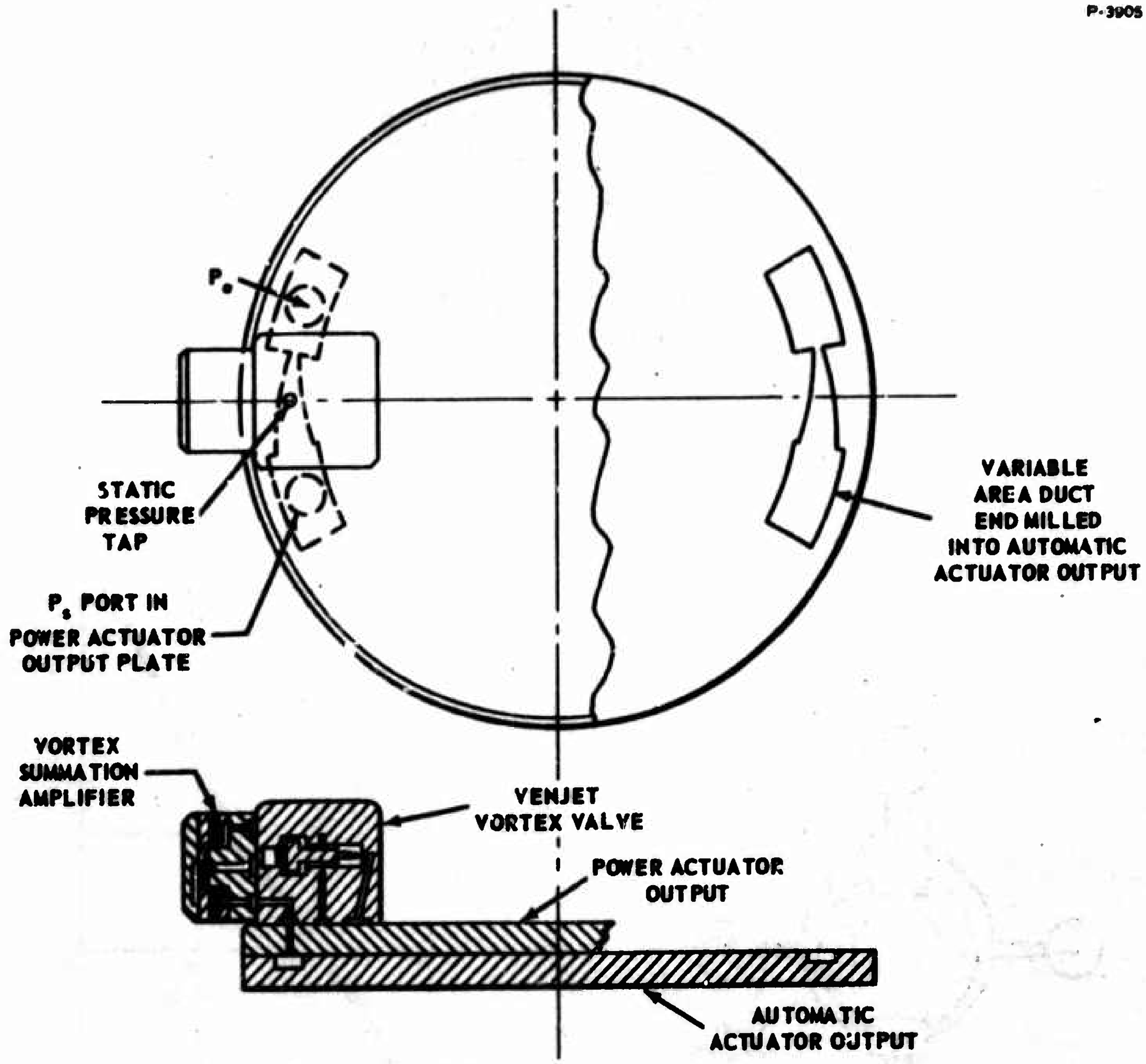


Figure 43 - Position Transducer Configuration

A ± 40° fluidic position transducer utilizing radial duct porting rather than surface porting has been developed and tested. An assembled view of this position transducer is shown in Figure 44 and a typical output curve is shown in Figure 45. The load flow-pressure curves are in actuality only the static orifice flow characteristics of the static taps.

Figure 44 - Assembled View of
Experimental Position Transducer

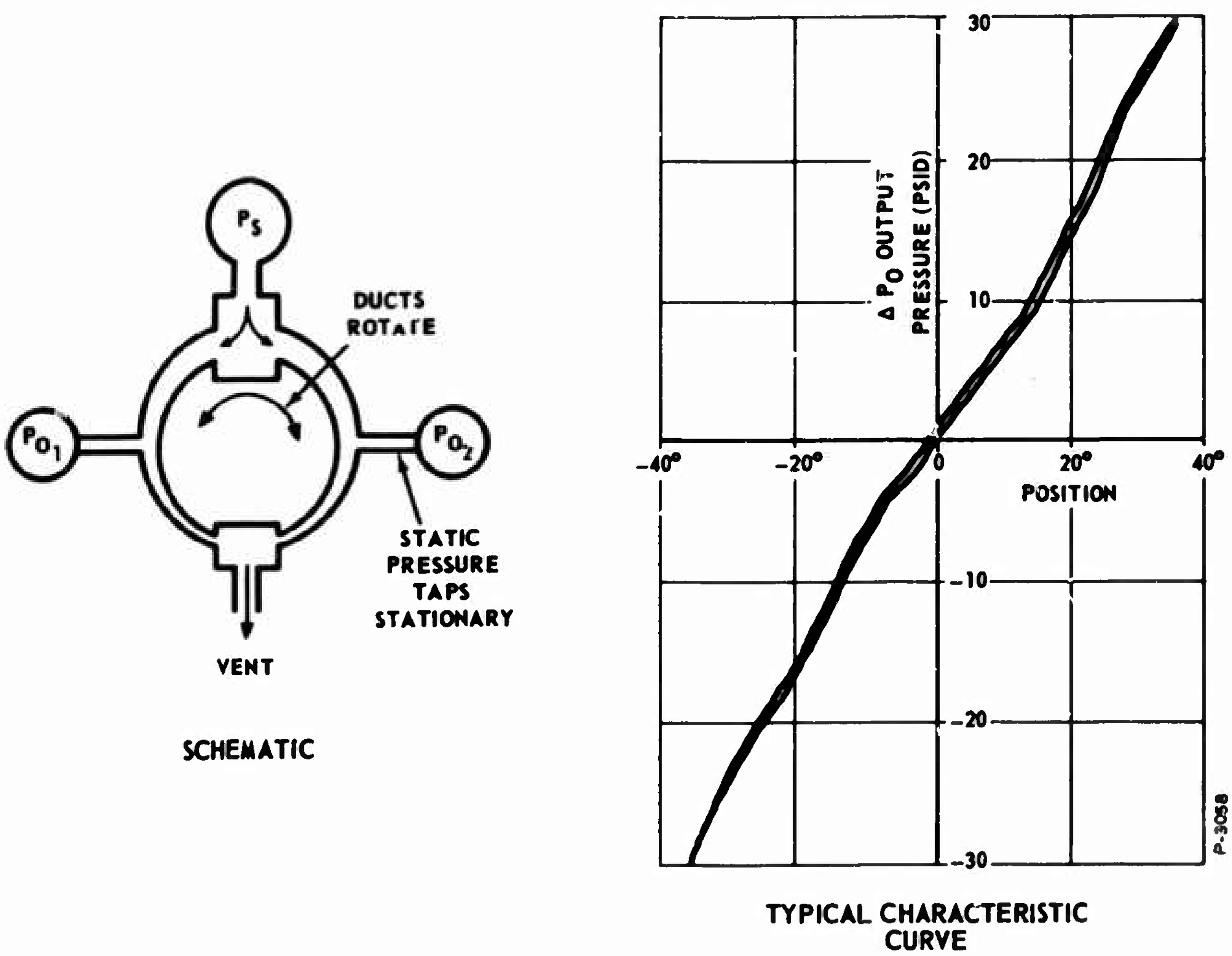


Figure 45 - Rotary Position Transducer Schematic and
Output Characteristic

Because of the simple construction and the absence of critical orifice sizes, the transducer will perform reliably without drift or permanent changes. The basic pressure relationships are not functions of temperature, thus, the device will operate over a temperature range limited only by structural strength. The output signal is easily trimmed to provide the desired output-input relationships and not subject to critical machining tolerances.

3. SERVO SYSTEM ANALYSIS

A. Torque-Speed Characteristics

The torque-speed characteristic of the actuator will be used to determine the dynamic response. A flow model of the actuator shown in Figure 46 was used to obtain this characteristic. The various symbols used are:

Q = Volumetric flow rate (in^3/sec)

P_s = Supply pressure (psia)

P_e = Ambient pressure (psia)

P_1 = Upstream motor pressure as measured at valve port (psia)

P_1' = Upstream motor pressure inside motor available for torque (psia)

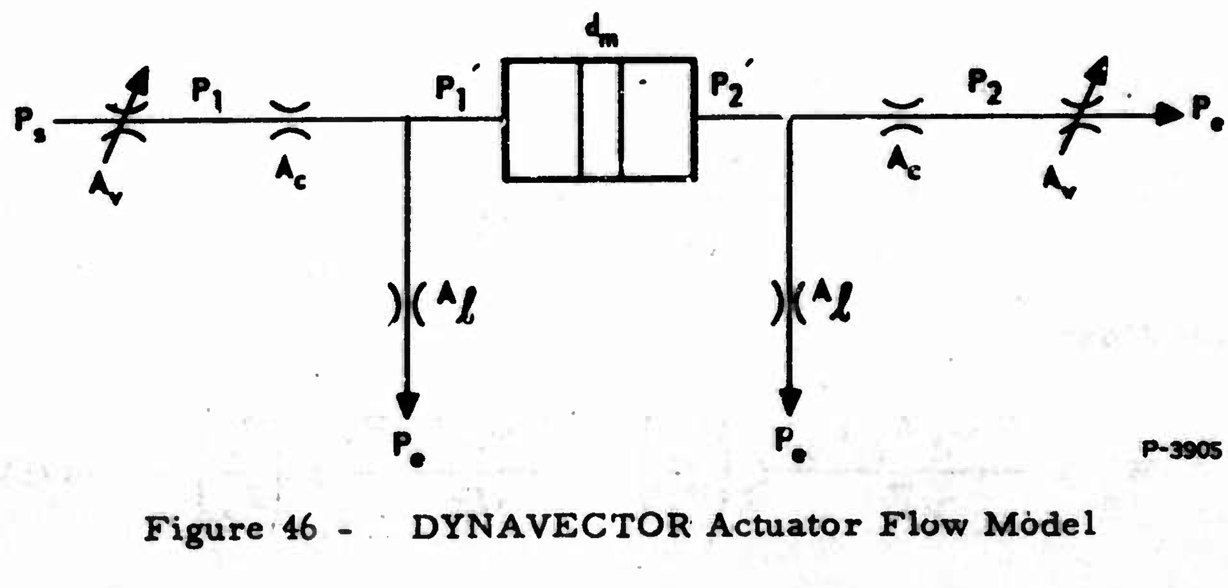


Figure 46 - DYNAVECTOR Actuator Flow Model

P_2 = Downstream motor pressure as measured at valve port (psia)

P_2' = Downstream motor pressure inside motor available for torque (psia)

A_v = Valve area

A_g = Commutation area

A_ℓ = Leakage area

The various flows are given by

Supply Flow:

$$W_s = \frac{C\,C_d\,A_v\,P_s}{\sqrt{T}}\,f_1\left(\frac{P_1}{P_s}\right) = \frac{C\,C_d\,A_c\,P_1}{\sqrt{T}}\,f_1\left(\frac{P_1'}{P_1}\right) \tag{69}$$

Upstream Leakage Flow:

$$W_{\ell_1} = \frac{C\,C_d\,A_\ell\,P_1'}{\sqrt{T}}\,f_1\left(\frac{P_e}{P_1'}\right) \tag{70}$$

Displacement Flow:

$$W = W_s - W_{\ell_1} \tag{71}$$

Downstream Leakage Flow:

$$W_{\ell_2} = \frac{C\,C_d\,A_e\,P_2'}{\sqrt{T}}\,f_1\left(\frac{P_e}{P_2'}\right) \tag{72}$$

Exhaust Flow:

$$W_e = \frac{C\,C_d\,A_c\,P_2'}{\sqrt{T}}\,f_1\left(\frac{P_2}{P_2'}\right) = \frac{C\,C_d\,A_v\,P_2}{\sqrt{T}}\,f_1\left(\frac{P_e}{P_2}\right) \tag{73}$$

P_2 = Downstream motor pressure as measured at valve port (psia)

P_2' = Downstream motor pressure inside motor available for torque (psia)

A_v = Valve area

A_g = Commutation area

A_ℓ = Leakage area

The various flows are given by

Supply Flow:

$$W_s = \frac{C\,C_d\,A_v\,P_s}{\sqrt{T}}\ f_1\left(\frac{P_1}{P_s}\right) = \frac{C\,C_d\,A_c\,P_1}{\sqrt{T}}\ f_1\left(\frac{P_1'}{P_1}\right) \tag{69}$$

Upstream Leakage Flow:

$$W_{\ell_1} = \frac{C\,C_d\,A_\ell\,P_1'}{\sqrt{T}}\ f_1\left(\frac{P_e}{P_1'}\right) \tag{70}$$

Displacement Flow:

$$W = W_s - W_{\ell_1} \tag{71}$$

Downstream Leakage Flow:

$$W_{\ell_2} = \frac{C\,C_d\,A_e\,P_2'}{\sqrt{T}}\ f_1\left(\frac{P_e}{P_2'}\right) \tag{72}$$

Exhaust Flow:

$$W_e = \frac{C\,C_d\,A_c\,P_2'}{\sqrt{T}}\ f_1\left(\frac{P_2}{P_2'}\right) = \frac{C\,C_d\,A_v\,P_2}{\sqrt{T}}\ f_1\left(\frac{P_e}{P_2}\right) \tag{73}$$

and from conservation of mass

$$W_e + W_{\ell_2} = W \quad (74)$$

The motor output torque is determined by the differential pressure $P_1' - P_2'$; input displacement, d_m in^3/rad; the gear ratio N'; and the efficiency η. Therefore,

$$T = (P_1' - P_2')\, d_m\, (N')\, \eta \text{ lb-ins} \quad (75)$$

Similarly the motor speed is a function of volumetric displacement flow and from the perfect gas law:

$$Q = \frac{W\,RT}{P_1'} \quad (76)$$

and if $\dot{\theta}$ (motor output speed) is to be given in degrees per second.

$$\dot{\theta} = \frac{360\ W\ RT}{2\,\pi\, d_m\, (P_1')\, N'} \quad (77)$$

Equations (69) thru (77) were programmed on a digital computer and simultaneously solved to determine torque and speed at various points between stall and full no load speed.

(1) Power Actuator

The torque-speed program inputs for the power actuator are:

P_s = 65 psia

P_e = 14.7 psia

C = 0.528

C_d = 0.80

η = 0.80

$T = 530°R$

$R = 640$

$k = 1.4$

$d_m = 0.755\ in^3/rad$

$A_\ell = 0.003\ in^2$

$A_c = 0.125\ in^2$

$A_v = 0.01,\ 0.02,\ 0.03,\ 0.04,\ (in^2)$

$N' = 370$

The results are shown in Table VII for the fully open valve case and the complete torque-speed curve is shown in Figure 47.

(2) Automatic Actuator

The program inputs for the automatic actuator are:

$P_s = 65$ psia

$P_e = 14.7$ psia

$C = 0.528$

$C_d = 0.80$

$\eta = 0.80$

$T = 530°R$

$R = 640$

$k = 1.4$

$d_m = 0.0352\ in^3/rad$

$A_e = 0.003\ in^2$

$A_c = 0.01\ in^2$

Table VII - Torque Speed Computer Program Results for Power Actuator

COMPUTER PROGRAM INPUTS						
$P_s = 65.0$	$P_e = 14.7$	$C = 0.528$	$C_d = 0.80$	$T = 530.0$		
$A_v = 0.04$	$A_l = 0.003$	$A_c = 0.125$	$A_e = 0.003$			
$D_m = 0.755$	$N' = 370.0$	$R = 640.0$	$k = 1.4$	$\eta = 0.8$		
$\dot{\theta}$ (deg/sec)	T (in-lbs)	W (lb/sec)	P1' (psia)	P2' (psia)	P1 (psia)	P2 (psia)
15.1	9587.3	0.01359	62.8	19.9	63.0	19.5
23.4	7620.8	0.02035	60.6	26.5	61.0	25.8
30.1	5717.8	0.02528	58.3	32.8	59.0	32.0
36.2	4086.9	0.02920	56.1	37.8	56.9	36.9
41.9	2653.1	0.03244	53.9	42.0	54.9	41.0
47.3	1367.8	0.03516	51.7	45.5	52.9	44.5
52.7	201.8	0.03747	49.5	48.5	50.9	47.4
58.1	- 863.5	0.03943	47.2	51.1	48.9	49.9
63.5	-1841.3	0.04109	45.0	53.2	46.9	52.0
69.1	-2740.9	0.04248	42.8	55.0	44.9	53.8

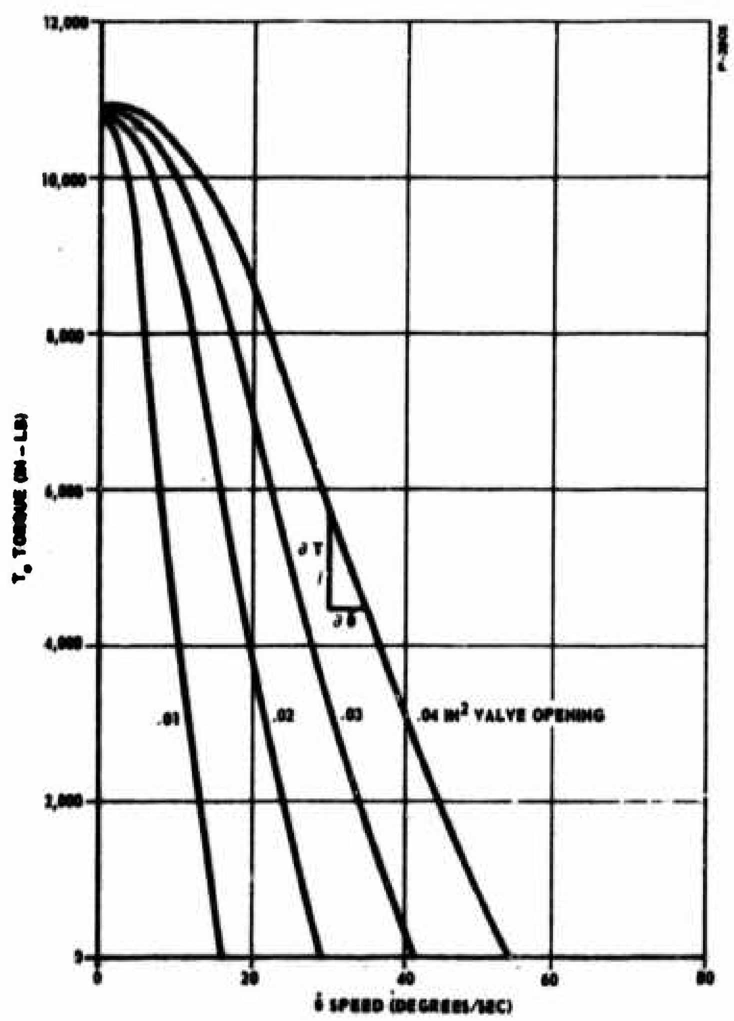


Figure 47 - Power Actuator Torque-Speed Characteristics

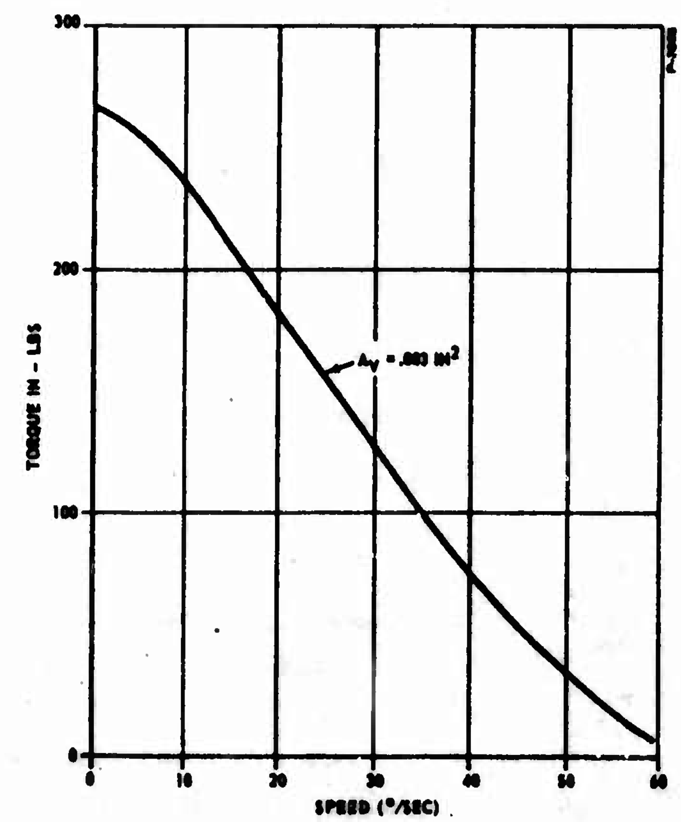


Figure 48 - Automatic Actuator Torque-Speed Characteristics

A_v = Variable

N' = 370

The torque speed characteristics are shown in Figure 48.

B. Inertia Loads

The total inertia loads are required for response calculations and are tabulated in Table VIII and IX.

C. Dynamic Response

(1) Power Actuator

The linear block diagram for the power actuator is given in Figure 49. The symbols used are defined as:

X = Pilot input (in)

Y_m = Manual valve spool position (in)

Z = Manual valve body position (in)

ℓ_1 = Input linkage length between manual valve and pilot input point (in)

ℓ_2 = Input linkage length between manual valve power actuator centerline (in)

θ' = Rudder position (rad)

$\dot{\theta}'$ = Rudder velocity (rad/sec)

β = Automatic actuator position (rad)

T_o = Power actuator torque (lb-in)

$\frac{\partial T_o}{\partial \dot{\theta}'}$ = Power actuator torque speed slope at operating point. (in-lb/sec)

$\frac{\partial T_o}{\partial Y_m}$ = Power actuator torque-valve position slope at operating point. (in-lb/in.)

Table VIII - Power Actuator Inertias

Member	Inertia	Gear Ratio From Member to Output Shaft	Inertia at Output Shaft (In-Lb-Sec^2)
Gear-Active			
Rotating	0.0552	20	22.10
Eccentric	0.000166	370	22.70
Gear-Balance			
Rotating	0.0328	20	13.11
Eccentric	0.000166	370	22.70
Housing	0.053	1	0.53
Rudder	2.33	1	2.33
		Power Actuator Total	83.47

P-3905

Table IX - Automatic Actuator Inertias

Member	Inertia (In-Lb-Sec^2)	Gear Ratio From Member to Output Shaft	Inertia at Output Shaft (In-Lb-Sec^2)
Ring Gear			
Rotating	0.00051	20	0.2040
Eccentric	0.00000222	370	0.3020
Output Shaft	0.0030	1	0.0030
Link and Manual Valve Spool	0.0060	1/4	0.0004
		Automatic Actuator Total	0.5094

P-3905

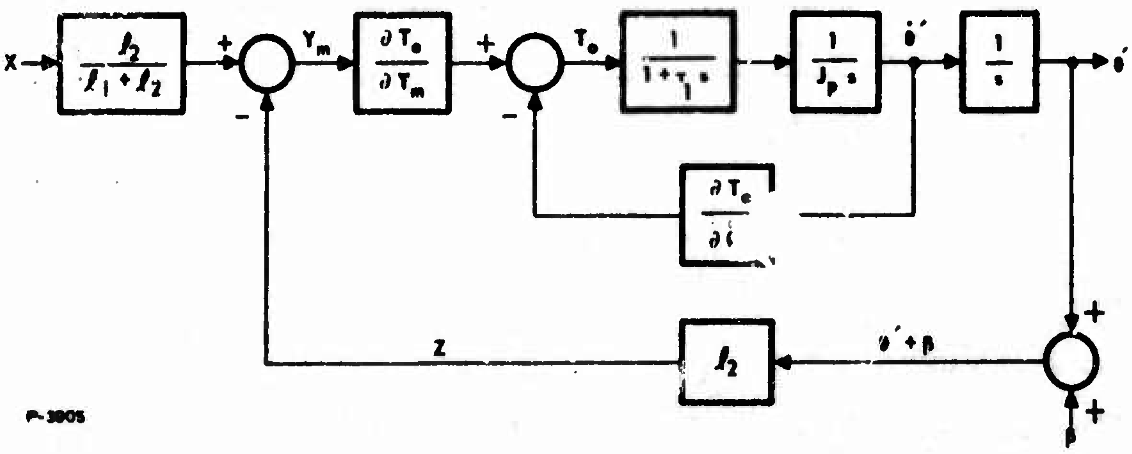


Figure 49 - Power Actuator Block Diagram

J_p = Power actuator total inertia (in-lb-sec^2)

τ_1 = Compressibility time constant (sec)

$$\tau_1 = \frac{V \dfrac{\partial T_o}{\partial \dot{\theta}'}}{(d_m)^2 \, k \, P_o \, R^2} \tag{78}$$

where

P_o = average quiescent actuator pressure at operating point

The block diagram reduces to a third order characteristic equation given by:

$$\frac{\theta'}{X} = \frac{1}{\tau_1 J_p s^3 + J_p s^2 + \dfrac{\partial T_o}{\partial \theta'} s + \dfrac{\partial T_o}{\partial Y_m} \left(\dfrac{\partial Z}{\partial \dot{\theta}'}\right)\left(\ell_2\right)} \tag{79}$$

where the static stiffness K_o is given by

$$K_o = \left(\frac{\partial T_o}{\partial Y_m}\right)\left(\ell_2\right) \tag{80}$$

This equation may be normalized into

$$\frac{\theta}{X} = \frac{K_o}{\dfrac{S^3}{\beta w_{ns}^3} + \dfrac{\alpha S^2}{\beta w_{ns}^2} + \dfrac{S}{\beta w_{ns}} + 1} \tag{81}$$

where:

$$w_{ns}^2 = \frac{\dfrac{\partial T_o}{\partial \dot{\theta}'}}{J_p \tau_1} \tag{82}$$

$$\frac{\alpha}{\beta} = \frac{J_p \left(w_{ns}\right)^2}{K_o} \tag{83}$$

$$\beta = \frac{K_o}{\frac{\partial T_o}{\partial \dot\theta'} \left(w_{ns}\right)} \tag{84}$$

From Figure 47 picking an operating point at 5000 in-lbs the torque-speed characteristic $\partial T_o / \partial \dot\theta$ is $15{,}200 \frac{\text{in-lbs}}{\text{rad/sec}}$. The inertia J_p from Table VIII is 83.47 in-lb-sec^2 and the pressurization time constant τ_1 is given by equation (78)

$$\tau_1 = \frac{3\ (15{,}200)}{\left[0.755\ (370)\right]^2 (1.4)\ (40)} = 0.0104 \text{ sec}$$

Therefore, from equation (82)

$$w_{ns}^2 = \frac{15{,}200}{(83.5)\ (0.0104)} = 17{,}500$$

$$w_{ns} = 132 \text{ rad/sec} = 21 \text{ cps}$$

From equation (83)

$$\frac{\alpha}{\beta} = \frac{83.5\ (17{,}500)}{K_o}$$

$$\beta = \frac{K_o}{15{,}200\ (132)}$$

By setting the gain K_c at 700,000 in-lb/rad, α = 0.73 and β = 0.35. This results in a power actuator response characteristic as shown in Figure 50.

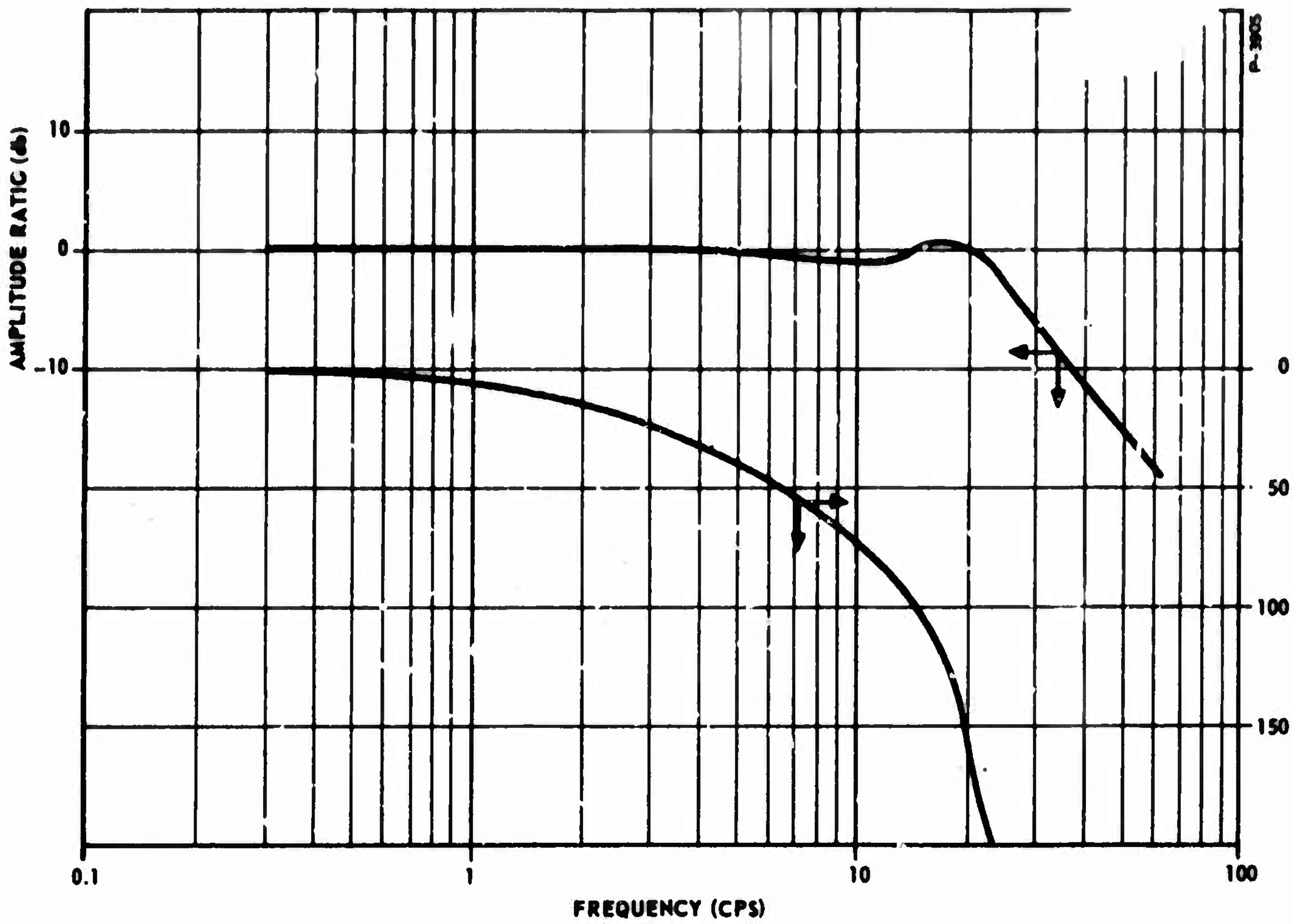


Figure 50 - Power Actuator Frequency Response to Mechanical Linkage Inputs

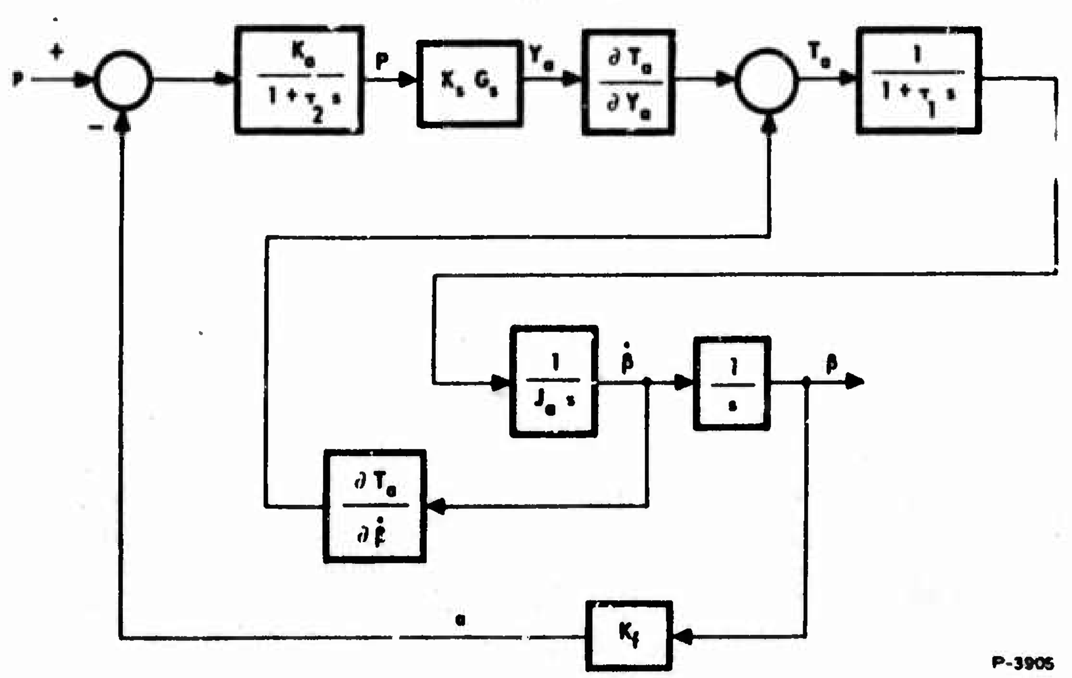


Figure 51 - Automatic Actuator Block Diagram

(2) Automatic Actuator

The linear block diagram for the automatic actuator is given in Figure 51.

The nomenclature for this block diagram is as follows:

P = Amplifier output (psid)

a = Feedback pot output (psid)

Y_a = Automatic valve position (in)

p = Auto pilot input (psid)

K_a = Amplifier gain (psi/psi)

K_s = Servovalve gain (in/psi)

K_f = Feedback point gain (psi/rad)

G_s = Servovalve transfer function

τ_2 = Amplifier time constant (sec)

τ_1 = Pressurization time constant (sec)

J_a = Automatic actuator moment of inertia (in-lb-sec^2)

β = Automatic actuator position (rad)

$\dot{\beta}$ = Automatic actuator velocity (rad/sec)

T_a = Automatic actuator torque (in-lb)

$\frac{\partial T_a}{\partial Y_a}$ = Automatic actuator torque gain at operating point $\frac{\text{in-lb}}{\text{in.}}$

$\frac{\partial T_a}{\partial \dot{\beta}}$ = Automatic actuator torque-speed slope at operating point $\frac{\text{in-lb}}{\text{rad/sec}}$

The servovalve transfer function is given by,

$$\frac{Y_a}{S} = \frac{K_v}{\dfrac{S^3}{\beta\, w_{nv}} + \dfrac{a\, S^2}{\beta\, w_{nv}^2} + \dfrac{S}{\beta\, w_{nv}} + 1} \qquad (85)$$

and,

$$w_{nv}^{\;2} = \frac{2\, A^2\, k\, P_o}{MV} = \text{[illegible]00 cps} \qquad (86)$$

where:

M = Spool mass (lb-sec^2/in)

V = End chamber volume (in^3)

k = 1.4 for air

P_o = End chamber quiescent pressure

A = Spool end area

also,

a = 1.0

β = 0.50

The block diagram of Figure 51 may be reduced to the form shown in Figure 52.

From Figure 48, $\partial T_a / \partial \dot{\beta} = 900 \dfrac{\text{in-lbs}}{\text{rad-sec}}$ and τ_1 is calculated to be 0.009 seconds. Setting the amplifier time constant τ_2 at 8 cps, the closed loop response of the automatic actuator was obtained and is shown in Figure 53. The required gain $K_a\, K_s\, \dfrac{\partial T_a}{\partial Y_a}\, K_f$ is 30,000 in-lbs/rad.

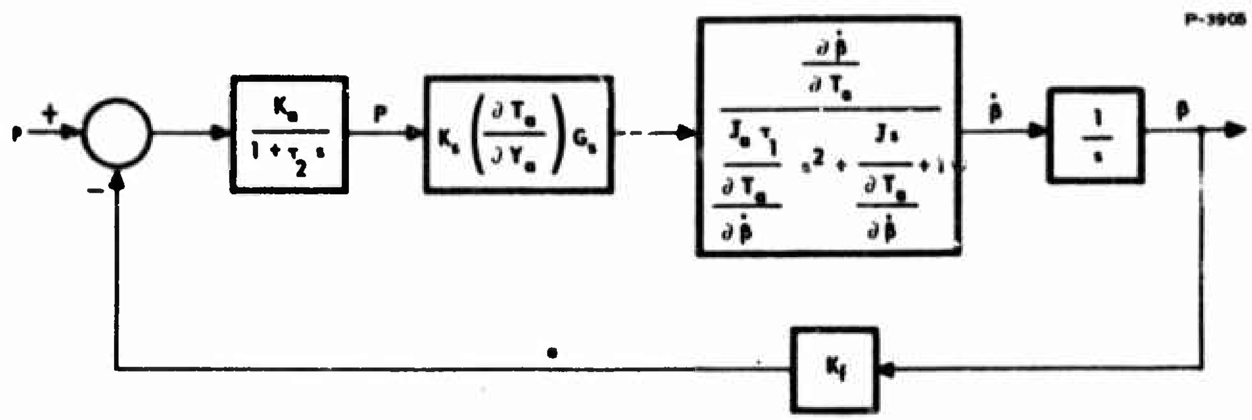


Figure 52 - Automatic Actuator Block Diagram (Reduced Form)

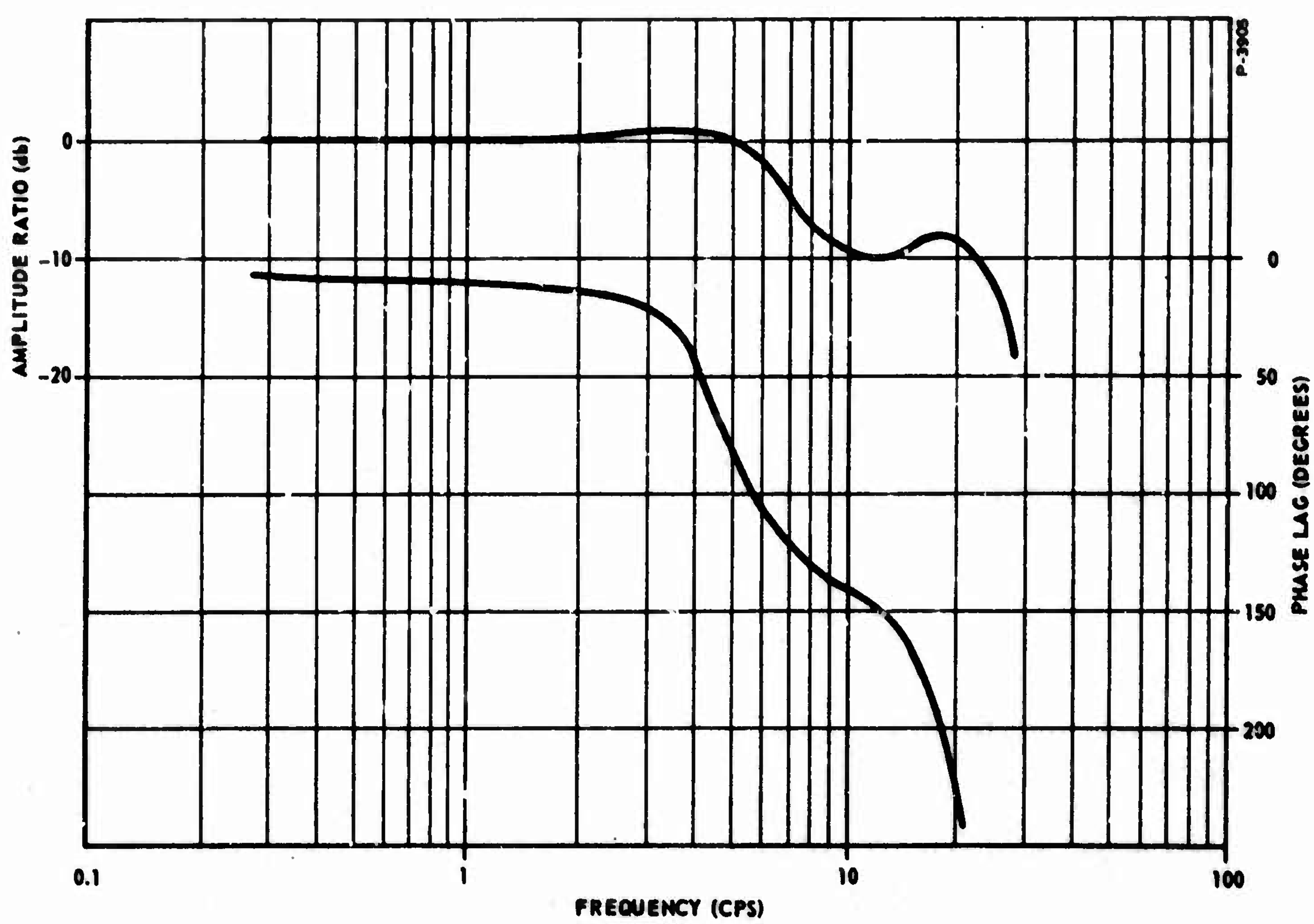


Figure 53 - Automatic Actuator Frequency Response to Autopilot Inputs

(3) Servomechanical System

The total system response to autopilot inputs, p, is the sum of the two response curves of Figures 50 and 53 and is shown in Figure 54 superimposed on the required response characteristic.

D. Gain Requirements

(1) Power Actuator

The input linkage geometry is such that the ratio of input linkage lengths is given by

$$\frac{\ell_1}{\ell_1 + \ell_2} = \frac{4}{5} \text{ and } \ell_2 = 0.975 \text{ inches}$$

Since,

$$K_o = \left(\frac{\partial T_o}{\partial Y_m}\right) \left(\ell_2\right) = 700{,}000 \ \frac{\text{in-lb}}{\text{rad}}$$

$$\frac{\partial T_o}{\partial Y_m} = \frac{700{,}000}{0.975} = 718{,}000 \text{ in-lbs/in.}$$

From Figure 47 the torque gain at the operating point is about 20,000 in-lbs for full valve stroke thus, the required stroke is given by:

$$Y_{max} = \frac{20{,}000}{718{,}000} = 0.028 \text{ inches}$$

Therefore, the valve must be fully open in 0.028 inches or 0.112 inches of pilot input.

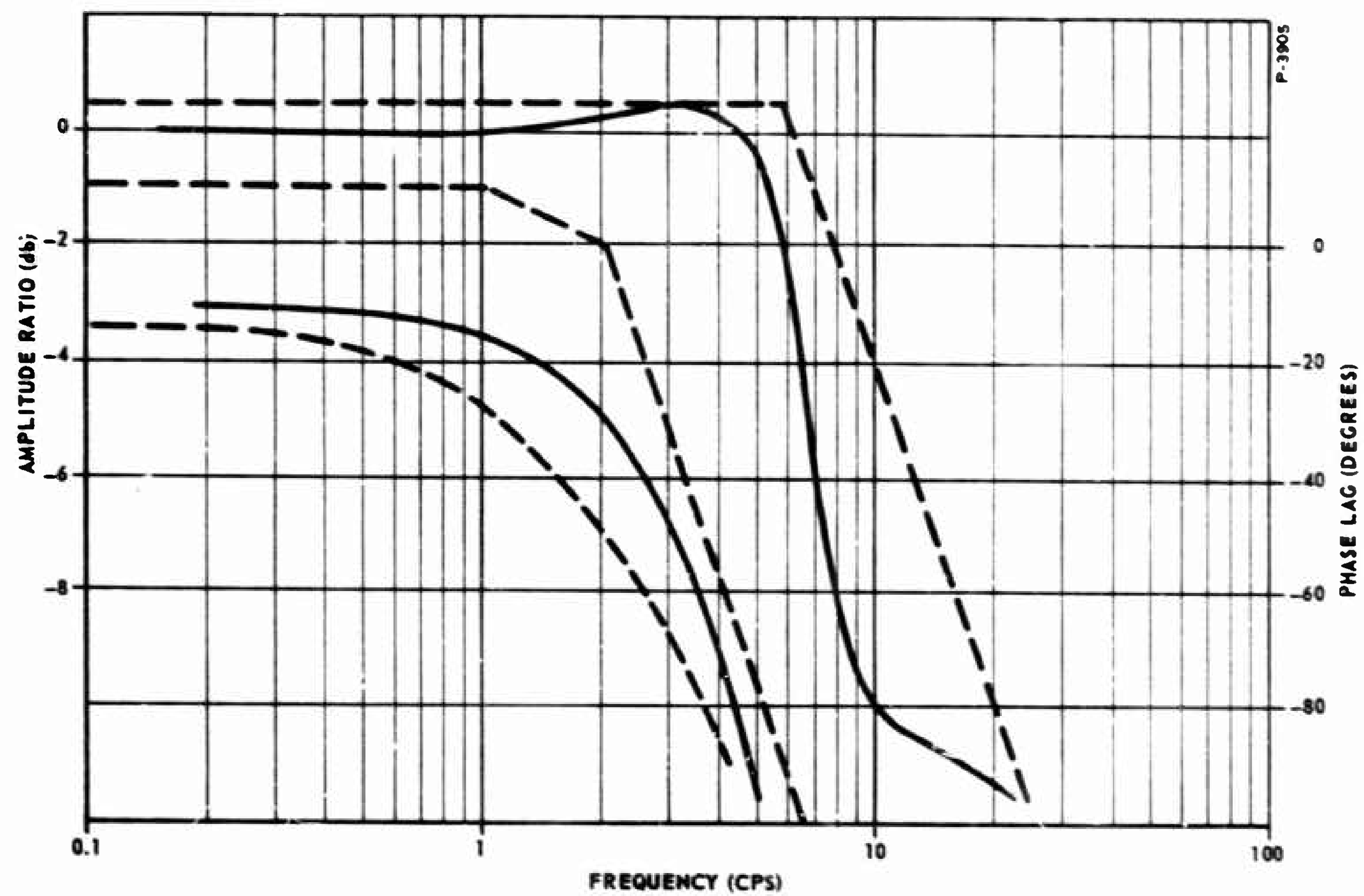


Figure 54 - Complete Actuator Response to Autopilot Inputs

(2) Automatic Actuator

Assuming a valve gain K_s of $\frac{1}{600}$ in/psi and a feedback gain K_f of 60 psi/rad along with a torque gain of 60,000 in-lbs/in. the required amplifier gain is calculated from:

$$K_a K_s \frac{\partial T_a}{\partial Y_a} K_f = 30{,}000 \frac{\text{in-lbs}}{\text{rad}}$$

$$K_a = \frac{30{,}000}{\frac{1}{600}(60{,}000)(60)} = 5.0 \text{ psi/psi}$$

4. INSTALLATION REQUIREMENTS

For a flight evaluation test program the pneumatic DYNAVECTOR rudder actuator would be mounted concentric to the rudder axis and coupled to the rudder horn assembly by a pneumatic clutch. This mounting configuration would allow the DYNAVECTOR actuator to operate in parallel with the existing hydraulic power cylinder and damper assemblies. The aircraft pilot would have the option of any of the following operative modes with such a dual actuator system during the evaluation tests:

- Hydraulic system operative.

Pneumatic system decoupled from rudder and inoperative.

- Hydraulic system operative.

Pneumatic system decoupled from rudder but operative. Output of decoupled pneumatic system monitored to provide correlation of pneumatic output to pneumatic and hydraulic systems commands and hydraulic system output.

- Hydraulic system inoperative. Pneumatic system coupled to rudder and operative. Hydraulic system may be reactuated and pneumatic system decoupled from rudder at any time in event of pneumatic system malfunction.

The pneumatic DYNAVECTOR actuator would be coupled to an adapter bracket mounted to the rudder horn by a fail safe pneumatic clutch. The clutch is spring loaded out of engagement, and engaged by pressurizing with 50 psig compressor bleed air.

Manual input commands from the pilot controls are accomplished by a linkage coupling between the DYNAVECTOR actuator manual valve and the existing bell crank used for manual inputs to the hydraulic power actuator. Automatic mode yaw damper inputs are provided by a pneumatic signal from the yaw damper system to the automatic valve mounted to the DYNAVECTOR actuator rotary housing.

Figure 2 shows a side view sectional drawing of the F101B aircraft tail assembly in the area of the hydraulic power actuator cylinder between fuselage stations F.S. 832.36 and F.S. 801.00. The DYNAVECTOR actuator mounting structure would be attached to the bulkheads at stations F.S. 832.36 and F.S. 814.90.

The linkage for aircraft pilot manual commands would be attached to the bell crank located at elevation W.L. 96.78, fuselage station F.S. 799.00.

Figure 3 shows a view looking up the rudder axis, the location of the hydraulic power and damper actuators and the concentric mounting of the DYNAVECTOR actuator.

Figure 4 shows a view forward along the aircraft longitudinal axis at fuselage station F.S. 832.36.

Figure 5 is a sectional view forward normal to the rudder axis at elevation W.L. 90.62 and station F.S. 831.346.

The modifications required of the F101B aircraft for the installation of the DYNAVECTOR actuator are as follows:

A. Bulkhead Attachment

(Ref. Figures 2 and 3)

Provide required mounting attachments to bulkheads at F.S. 832.36 and F.S. 814.90 for attachment of DYNAVECTOR actuator support structure.

B. Rudder Horn Attachment

(Ref. Figure 5)

Mount rudder horn adapter to rudder horn for pneumatic clutch output member.

C. Fuel Vent Line Valve Hydraulic Line

(Ref. Figures 2 and 4)

Reroute hydraulic line to fuel vent line valve at F.S. 832.36 to eliminate interference with DYNAVECTOR actuator assembly.

D. Manual Input Linkage Attachment

(Ref. Figure 2)

Provide mounting for linkage ycke at linkage input point on integrated hydraulic power cylinder.

E. Power Actuator Hydraulic Lines

(Ref. Figure 2)

Reroute hydraulic line to power actuator to eliminate interference with DYNAVECTOR actuator assembly.

F. Pneumatic Power Supply and Exhaust Attachments

Provide power supply line attachments to jet engine compressor bleed point and actuator exhaust port for venting outside of aircraft fuselage if hot exhaust is not permissible in tail section.

G. Hydraulic Power Cylinder Disengagement

A two position, spring-solenoid valve would be required in the pressure supply line to the hydraulic power cylinder. In the solenoid "on" position supply pressure would be ported to the detent control cylinder shown in Figure 2, and the cylinder supply line would be vented to tank. The cylinder by-pass valve would open as in the event of loss of utility hydraulic system pressure, thereby allowing fluid to pass from one side of the power cylinder piston to the other, permitting pneumatic system operation. The lock-up mechanism locking the pilot input to the power cylinder housing would normally become engaged with such a loss of system pressure. Lock-up would necessarily have to be prevented during pneumatic operation or else the rudder horn would be mechanically linked to the pilot manual command linkage. Lock-up would be prevented by a hydraulic signal to a separate piston and cylinder assembly (Detent control cylinder) that would act as a stop thereby preventing lock-up detent engagement. Upon reversion to the hydrualic mode, the lock mechanism stop assembly would be spring loaded out of engagement and thus not interfere with normal lock-up procedure in the event of actual loss of utility power supply.

5. POWER CONSUMPTION STUDY

A. Summary

The objective of this analysis is to predict and compare the fuel consumption rate of three motor-actuators for a given flight duty cycle. The actuators to be considered are gear, vane and DYNA-VECTOR actuator.

The gear and vane motors must be coupled to a transmission whereas the DYNAVECTOR actuator is an integrated motor-transmission machine.

The technique used in predicting the fuel consumption rate is:

(1) Select the best performance data available of each of the selected motor-actuators.

(2) Normalize and plot the specific fuel consumption data with respect to normalized speed. (reference paragraphs B & E)

(3) Establish the torque speed profile required for steady state, stall and cyclic conditions as specified in paragraphs C and D.

(4) Predict the stall and zero load consumption. (reference paragraph D) This is required to establish the "end points" of the fuel consumption curves.

(5) Plot fuel consumption rate versus flight time for the duty cycle of paragraphs C and D.

The conclusions of this power consumption study are that the average consumption rate for the duty cycle (reference paragraph C) based on a 240 minute flight at 530°R gas temperatures is:

Gear Motor	0.011 lb/sec
Vane Motor	0.006 lb/sec
DYNAVECTOR Actuator	0.024 lb/sec

These results are based on optimized consumption data for the gear and vane motors and the current status of DYNAVECTOR actuator development based on actual DYNAVECTOR actuator test results. It is predicted that with continued development of the DYNA-VECTOR actuator, consumption requirements equivalent to the best gear motor characteristics should be attainable. Based on DYNAVECTOR actuator fuel consumption equivalent to gear motor characteristics, the predicted maximum fuel consumption at 910°R gas temperatures for the flightworthy DYNAVECTOR rudder actuator would not exceed 0.018 lb/sec and the mission average consumption would be 0.0043 lb/sec.

B. Normalized Specific Fuel Consumption

Previous analysis had concluded that the gear and vane motor (with transmissions) are the best standard units available for use in the rudder actuator design concept. To gain comparative data, gear, vane and DYNAVECTOR actuator motors were analyzed for performance requirements of paragraph C.

The best available fuel consumption data of each of the motors was normalized as follows:

For any percent of zero load speed:

$$\left[RT\ (SFC)\right]\ \text{Dimensionless} = \left[RT_{in}\right]\left[SFC\ \frac{LB}{HP\ HR}\right]\left[K\right] \qquad (87)$$

$$SFC\ \frac{LB}{HP\ HR} = \frac{SFC\ \frac{LB}{HP\text{-}HR}\ \text{Measured}}{\eta\ \text{Transmission Efficiency}}$$

where:

$R = \frac{In - lb}{Lb\ °R}$ - Gas constant of fluid being normalized

$T = °R$ - Fluid temperature

$K = \frac{1}{23.8\ 10^{6}} = 0.042\ 10^{-6}\ \frac{HP\text{-}HR}{LB\text{-}IN}$ correction factor to convert RT (SFC) into a dimensionless number.

η = Transmission efficiency equal to 1.0 in this analysis.

A normalized specific fuel consumption curve for each of the three motors is given in Figure 55.

C. Rudder Actuator Duty Cycle

An assumed duty cycle has been derived for the DYNAVECTOR rudder actuator application. This duty cycle defines the rudder actuator load-speed requirements for a four hour flight mission as summarized in the following table.

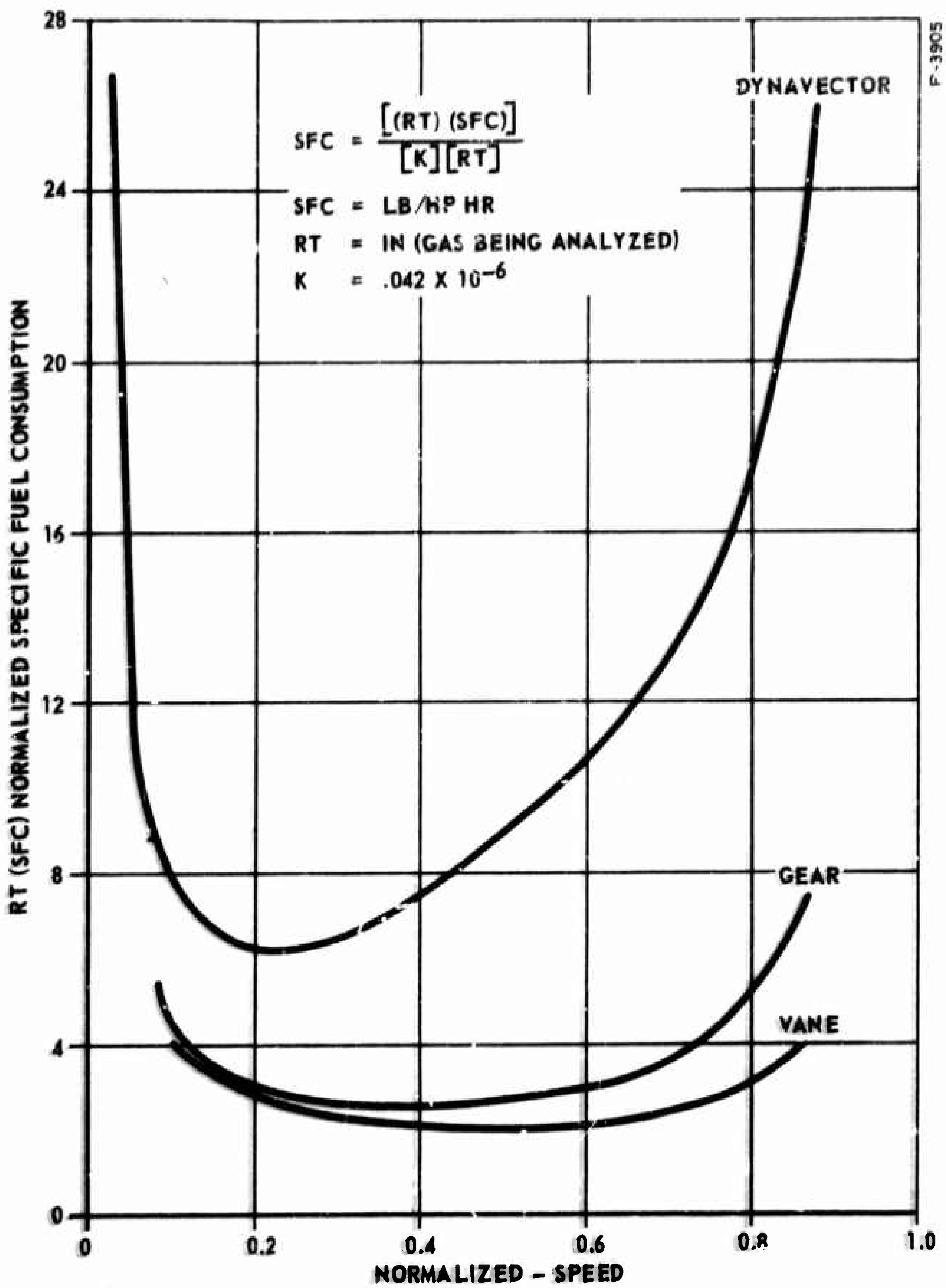


Figure 55 - Normalized Specific Fuel Consumption

Table X - Flight Mission Pneumatic Rudder Actuator Duty Cycle

STALL CONDITIONS		
Stall Torque (lb-ins)	Rudder Position (degrees)	Duration of Stall (minutes)
5,400	20	4
4,860	20	4
2,430	10	10
1,620	10	10
810	5	30

CYCLIC CONDITIONS			
Amplitude (± degrees)	Frequency (cps)	Torque Variation (lb-ins)	Time (minutes)
20	0.435	0 to ±4,860	84
5	0.871	0 to ± 810	35
0.8	2.18	0	63

D. Steady-State Fuel Consumption

The normalized specific fuel consumption curves based on actual test data for the gear, vane and DYNAVECTOR actuator systems described have been used to predict the steady-state fuel consumption of these systems when subjected to the horsepower requirements of the rudder actuator application.

Steady-state performance is based on a linear torque-speed load line between a stall torque of 5,400 lb-ins and a zero-load speed of 60 degrees per second.

Predicted fuel consumption values in the maximum output horsepower range are based on the SFC curves in Figure 55 for the gear, vane and DYNAVECTOR actuator systems as shown in Figures 56, 57 and 58 respectively. These values are indicated as "SFC Test Data" points.

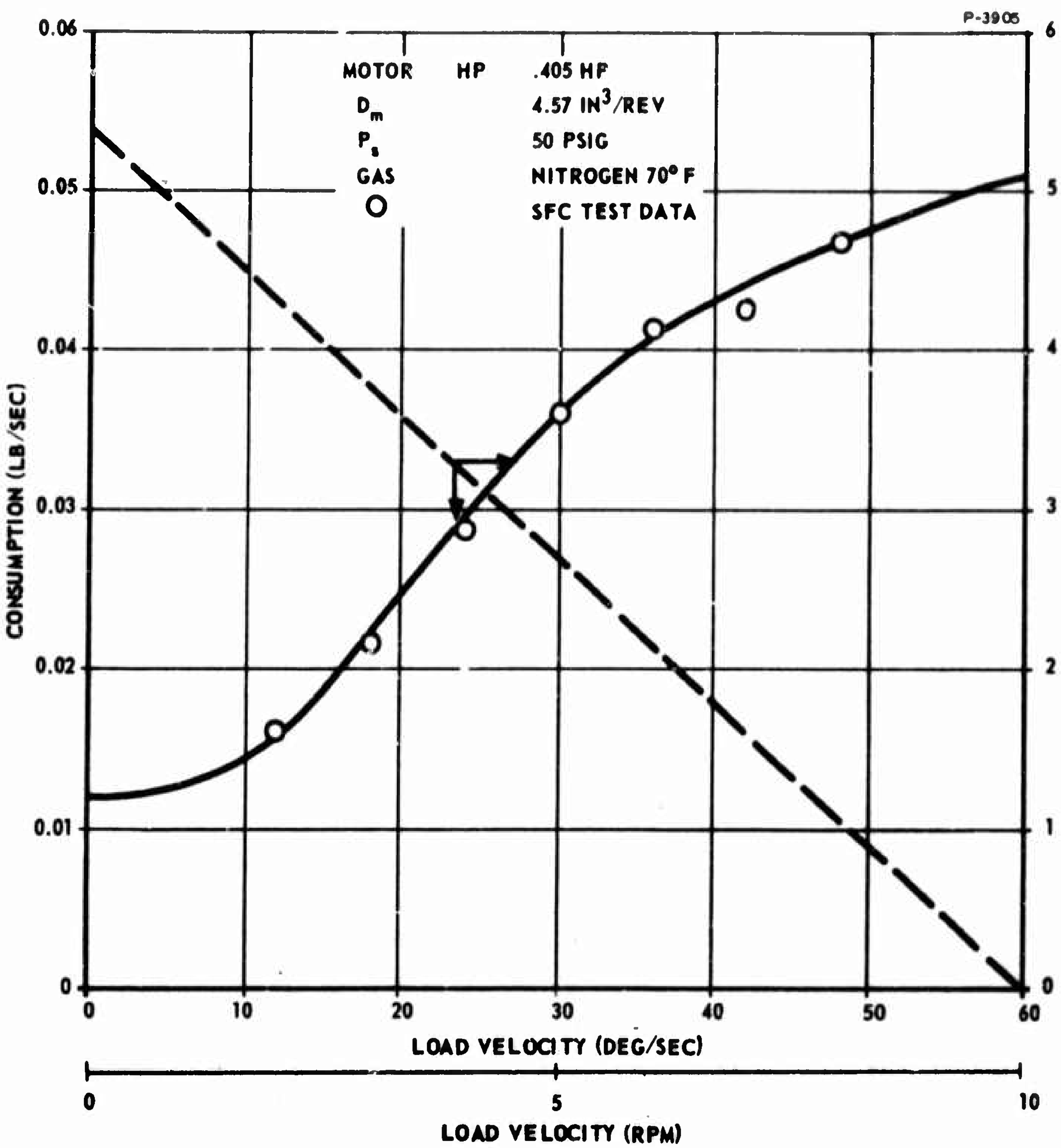


Figure 56 - Rudder Actuator Application DYNAVECTOR Steady-State Fuel Consumption

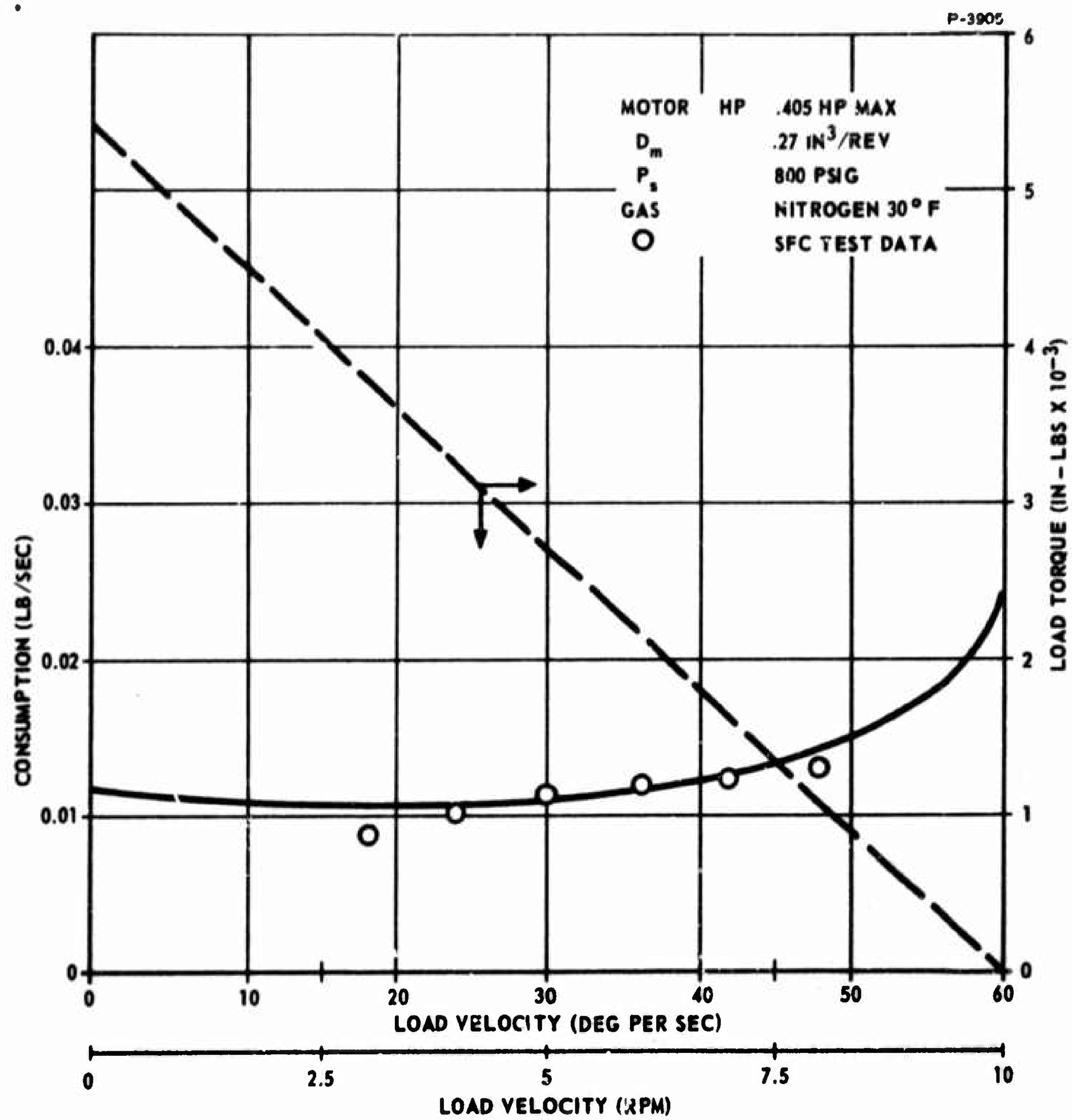


Figure 57 - Rudder Actuator Application Gear Motor Steady-State Fuel Consumption

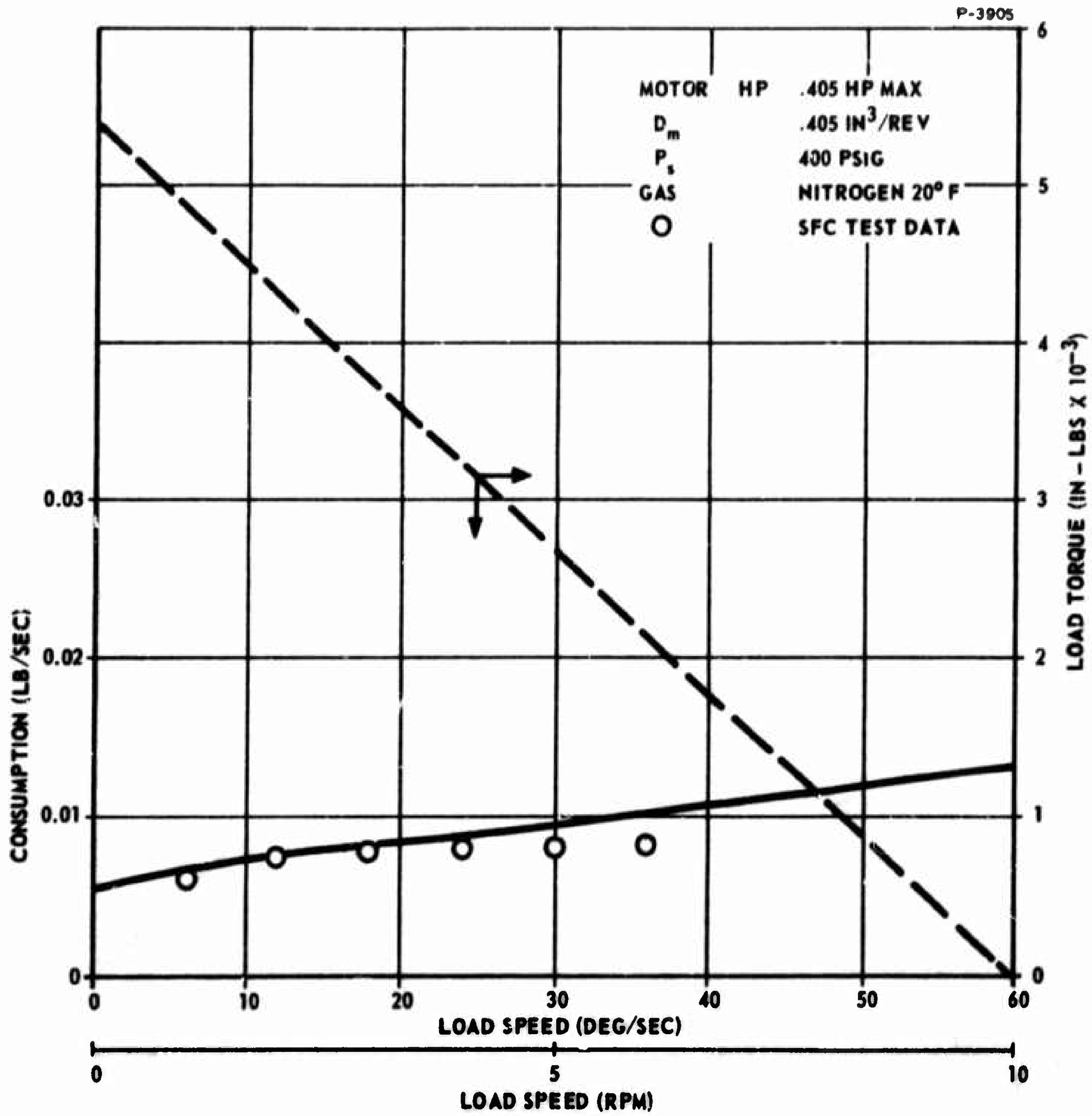


Figure 58 - Rudder Actuator Application Vane Motor Steady-State Fuel Consumption

The predicted fuel consumption requirements for the zero horsepower conditions of stall and zero-load speed cannot be calculated from the normalized SFC data of Figure 55. Realistic estimates may be made however by redesigning the subject test motors to meet the rudder application horsepower requirements as described below.

(1) <u>Gear Motor Stall and Zero-Load Fuel Consumption</u>

The pneumatic gear motor analyzed has a displacement of 3.2 in^3/rev. Performance data for this motor when operated with a 800 psig, 30°F nitrogen supply are as follows:

Maximum horsepower	=	4.8
Stall torque - T_s	=	275 lb-ins
Zero-Load speed - N_m	=	5,000 rpm
Stall flow - W_a	=	0.137 lbs/sec
Zero-Load flow - W_b	=	0.30 lbs/sec

The torque-speed characteristic for this motor is nearly linear with a peak horsepower of 4.8. The peak horsepower point for sizing the rudder actuator is 0.405 based on a linear characteristic from a load stall torque of 10,200 lb-ins and a zero-load speed of 10 rpm. To allow a prediction of the stall and zero-load consumption of a resized gear motor capable of 0.405 peak power, it has been assumed the motor displacement must be reduced by the power ratio. Therefore, the motor stall torque will be reduced by the displacement ratio change, assuming operation at 800 psig supply, and the motor zero-load speed will remain at 5,000 rpm.

Therefore, the resized gear motor stall torque may be given by

$$\frac{T_s}{275} = \frac{0.405}{4.8} \tag{88}$$

$$T_s = 23.2 \text{ lb-ins}$$

The resized gear motor displacement will be

$$D_m = \frac{0.405}{4.8}(3.2) = 0.27 \text{ in}^3/\text{rev} \tag{89}$$

Assuming the stall leakage is directly proportional to the displacement, the stall leakage of a 0.405 hp gear motor at 800 psig nitrogen supply will be

$$W_a = 0.137 \left(\frac{0.27}{3.2}\right) = 0.0115 \text{ lbs/sec} \qquad (90)$$

The zero-load consumption for the 4.8 hp gear motor can be stated by the equation

$$W_b = K_d (N_m)(D_m) \qquad (91)$$

$$0.3 \text{ lbs/sec} = K_d (5{,}000)(3.2) \qquad (92)$$

Likewise, the resized gear motor zero-load consumption relationship would be

$$W_b = K_d (5{,}000)(0.27) \qquad (93)$$

The constant K_d would be identical for equations (92) and (93), assuming identical gas density conditions.

The resized gear motor zero-load speed consumption would be

$$W_b = \frac{(5{,}000)(0.27)}{(5{,}000)(3.2)} (0.30) = 0.0254 \text{ lbs/sec} \qquad (94)$$

(2) <u>Vane Motor Stall and Zero-Load Fuel Consumption</u>

The vane motor analyzed has a displacement of 6.0 in^3/rev. Performance data for this motor when operated at a 400 psig 20°F nitrogen supply are as follows:

$$HP_{max} = 6.0$$
$$T_s = 185 \text{ lb-ins}$$
$$N_m = 5{,}100 \text{ rpm}$$
$$W_a = 0.083 \text{ lbs/sec}$$
$$W_b = 0.195 \text{ lbs/sec}$$

The required vane motor displacement for a 0.405 peak horsepower application may be given by equation (89) as follows

$$D_m = \left(\frac{0.405}{6.0}\right)(6.0) = 0.405 \text{ in}^3\text{/rev.} \qquad (95)$$

Assuming the stall leakage is directly proportional to the displacement, the stall leakage of a 0.405 horsepower vane motor with a displacement of 0.405 in^3/rev would be

$$W_a = 0.083 \left(\frac{0.405}{6.0}\right) = 0.0056 \text{ lbs/sec} \qquad (96)$$

The zero-load consumption is:

$$W_b = \frac{(5,100)(0.405)}{(5,100)(6.0)} (0.195) = 0.0132 \text{ lbs/sec} \qquad (97)$$

(3) Dynavector Actuator Stall and Zero-Load Fuel Consumption

The DYNAVECTOR actuator tested has a displacement of 2.62 in^3/rev. Performance data for this actuator when operated with a 400 psig, 70°F nitrogen supply are as follows:

Maximum Horsepower	=	0.94 hp
Stall Torque - T_s	=	1,850 lb-ins
Zero-Load Speed - N_m	=	205 rpm
Stall Flow - W_a	=	0.055 lb/sec
Zero-Load Flow - W_b	=	0.15 lb/sec
Transmission Ratio	=	15:1

The rudder actuator DYNAVECTOR actuator has a displacement of 4.57 in^3/rev and a ratio of 370:1. The stall torque of a 370:1 DYNAVECTOR actuator at 50 psig supply will be

$$T_s = 1,850 \left(\frac{50}{400}\right) \left(\frac{4.57}{2.62}\right) \left(\frac{370}{15}\right) = 10,000 \text{ lb-ins} \qquad (98)$$

Assuming the stall leakage is directly proportional to the displacement and supply pressure, the stall leakage for the 4.57 in^3/rev DYNAVECTOR actuator would be

$$W_a = 0.055 \left(\frac{50}{400}\right) \left(\frac{4.57}{2.62}\right) = 0.012 \text{ lb/sec} \qquad (99)$$

The zero-load speed fuel consumption may be estimated by calculating the displacement flow of 50 psig supply air at 70°F with the DYNAVECTOR actuator at an output zero-load speed of 10 rpm.

$$W_b = \gamma (D_m)(N_m) \qquad (100)$$

where

$$\gamma = \text{weight density} = \frac{P}{RT} = \frac{65}{53.3(530)(12)} = 1.81 \times 10^{-4} \text{ lb/in}^3$$

$$W_b = 1.81 \times 10^{-4} (4.57) \left(\frac{10 \times 370}{60}\right) = 0.051 \text{ lb/sec}$$

For a gas supply at 600°F, the zero-load mass flow rate will be

$$W_b = \left(\frac{560}{1,060}\right)(0.051) = 0.027 \text{ lb/sec} \qquad (101)$$

The method of calculating the zero-load flow consumption by the displacement equation may be confirmed by calculating the flow value for the 2.62 in^3/rev DYNAVECTOR actuator tested. Assuming air/nitrogen at 400 psig and 70°F the weight density is

$$\gamma = \frac{P}{RT} = \frac{415}{53.3(530)(12)} = 12.2 \times 10^{-4} \text{ lb/in}^3 \qquad (102)$$

The mass flow at 205 rpm output with a transmission ratio of 15:1 is

$$W_b = \gamma (D_m)(N_m) = 12.2 \times 10^{-4} (2.62) \left(\frac{15 \times 205}{60}\right) = 0.165 \text{ lb/sec}$$

This calculated mass flow is within 7 percent of the actual recorded test value of 0.155 lb/sec.

(4) Fuel Consumption During Power Transmission

The normalized specific fuel consumption curves of Figure 55 were used to compute the fuel consumption requirements during power transmission. The intermediate full consumption points, indicated as "SFC data" in Figures 56, 57 and 58 were calculated as follows. For any given speed point (taken as a percent of zero load speed), the weight flow rate may be found by taking the product of the load horsepower and the SFC value from Figure 55.

$$W = [SFC]\ [HP] \text{ lbs/HR}$$

$$W = [RT\,(SFC)]\ [HP] \left[\frac{1}{60\,K\,RT}\right] \text{lbs/min} \qquad (103)$$

where:

$\left[RT\ (SFC)\right]$ = Dimensionless Number at any percent of Zero Load Speed

HP = Horsepower - Determined from the steady-state torque speed curve

$\frac{1}{60\ K}$ = 3.96×10^5 lbs-in/HP-min units correction factor (Figure 55)

Equation (103) becomes

$$W = RT\ (SFC) \left[HP\right] \left[\frac{3.96 \times 10^5}{RT}\right] \text{lb/min}$$

where

R = in-lbs/lbs-°R - gas constant of fluid being used

T = °R fluid temperature

The normalized specific fuel consumption curve (Figure 55) cannot be used for stall and zero load values since the resulting product of equation (103) would be zero.

E. Duty Cycle Fuel Consumption

The duty cycle defined in paragraph C above is comprised of both stall and cyclic conditions. Stall load consumption has been assumed to be a direct ratio of the stall load function directly proportional to the stall load magnitude. The cyclic consumption, however, consists of both stall and power transmission flow conditions. Calculations for the cyclic consumption therefore consider both stall and SFC consumption conditions.

(1) Stall Load Fuel Consumption

The stall load fuel consumption requirements for the DYNAVECTOR actuator, gear and vane motor systems has been found as defined in paragraph D above at a load torque value of 5400 lb-in. Assuming the stall consumption at any other stall torque load value is directly proportional to the torque magnitude, the consumption for the torque values specified in the duty cycle may be found. As an example, the DYNAVECTOR actuator predicted fuel consumption demand at a stall torque of 2430 lb-in is

$$W_a = \frac{2430}{5400}(0.012) = 0.0054 \text{ lbs/sec}$$

Figure 59 shows graphically the stall consumption requirements for the duty cycle stall conditions for each of the three actuation systems analyzed. The average consumption rates for the duration of stall load conditions, 58 minutes is summarized as follows:

$$\text{DYNAVECTOR Actuator} - W_a = \frac{14.09}{58} = 0.243 \text{ lb/min} = 0.0041 \text{ lb/sec}$$

$$\text{Gear Motor} - W_a = \frac{13.50}{58} = 0.233 \text{ lb/min} = 0.0039 \text{ lb/sec}$$

$$\text{Vane Motor} - W_a = \frac{6.43}{58} = 0.111 \text{ lb/min} = 0.0019 \text{ lb/sec}$$

(2) Cyclic Fuel Consumption

The predicted steady-state fuel consumption curves (Figures 56, 57 and 58) were "re-normalized" as shown in Figure 60 using the techniques described in paragraph B above. The re-normalized specific fuel consumption curve incorporates the assumptions described in paragraph D(4).

An example solution for calculating the cyclic fuel consumption for the DYNAVECTOR actuator is given as follows:

Assume the rudder conditions are as follows;

amplitude	±20 degrees
frequency	0.435 cps
load torque	±4860 lb-in

The rudder displacement is given by

$$\theta' = \theta'_o \cos(\omega t) \quad 0 < \omega t < \pi$$

where

θ' = rudder position for any angular displacement (ωt)

θ'_o = initial rudder amplitude - 20°

ωt = angular displacement of the forcing frequency

where

$\omega = 2 \pi f t = 2.74 t$

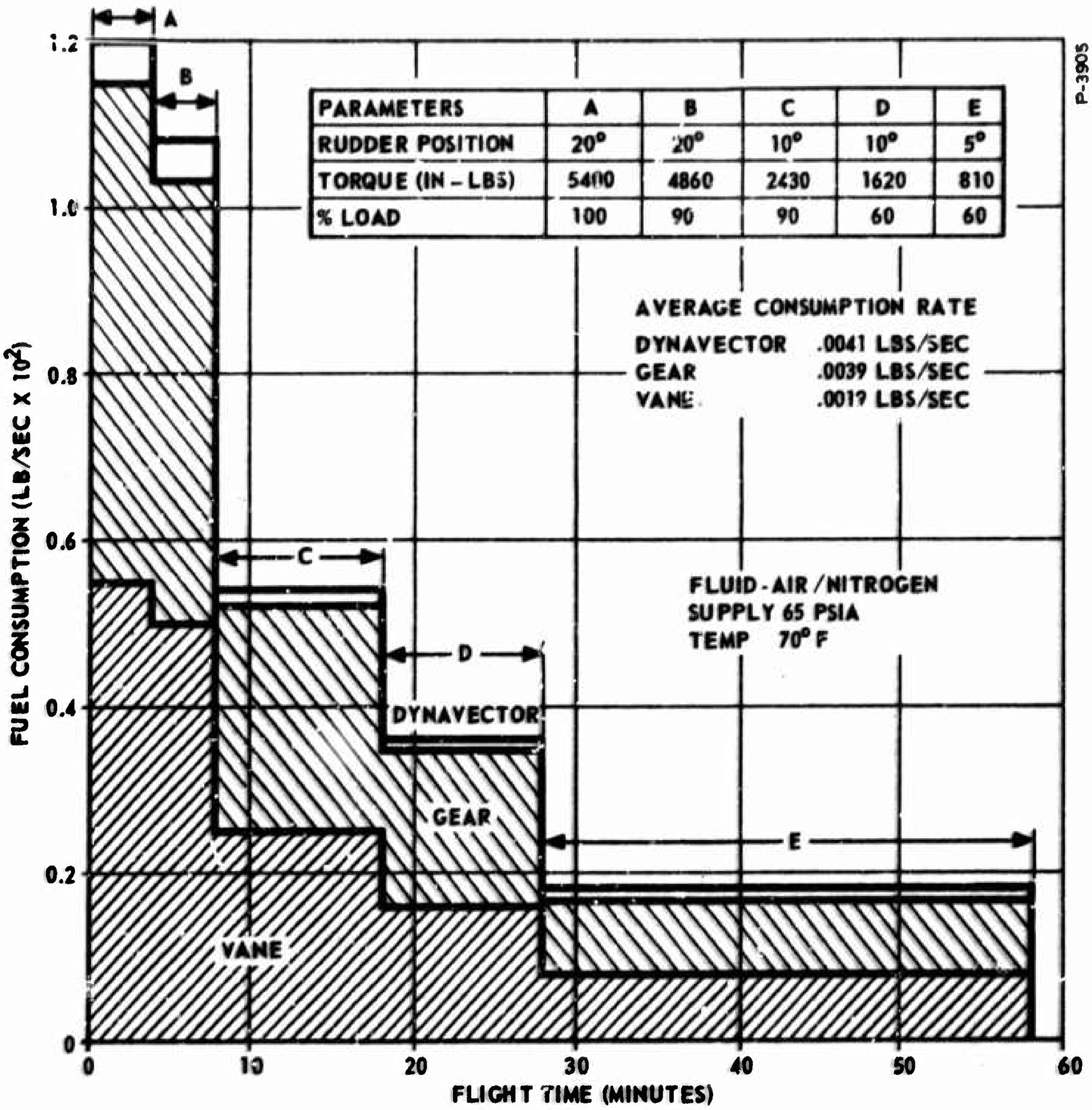


Figure 59 - Duty Cycle Stall Load Fuel Consumption

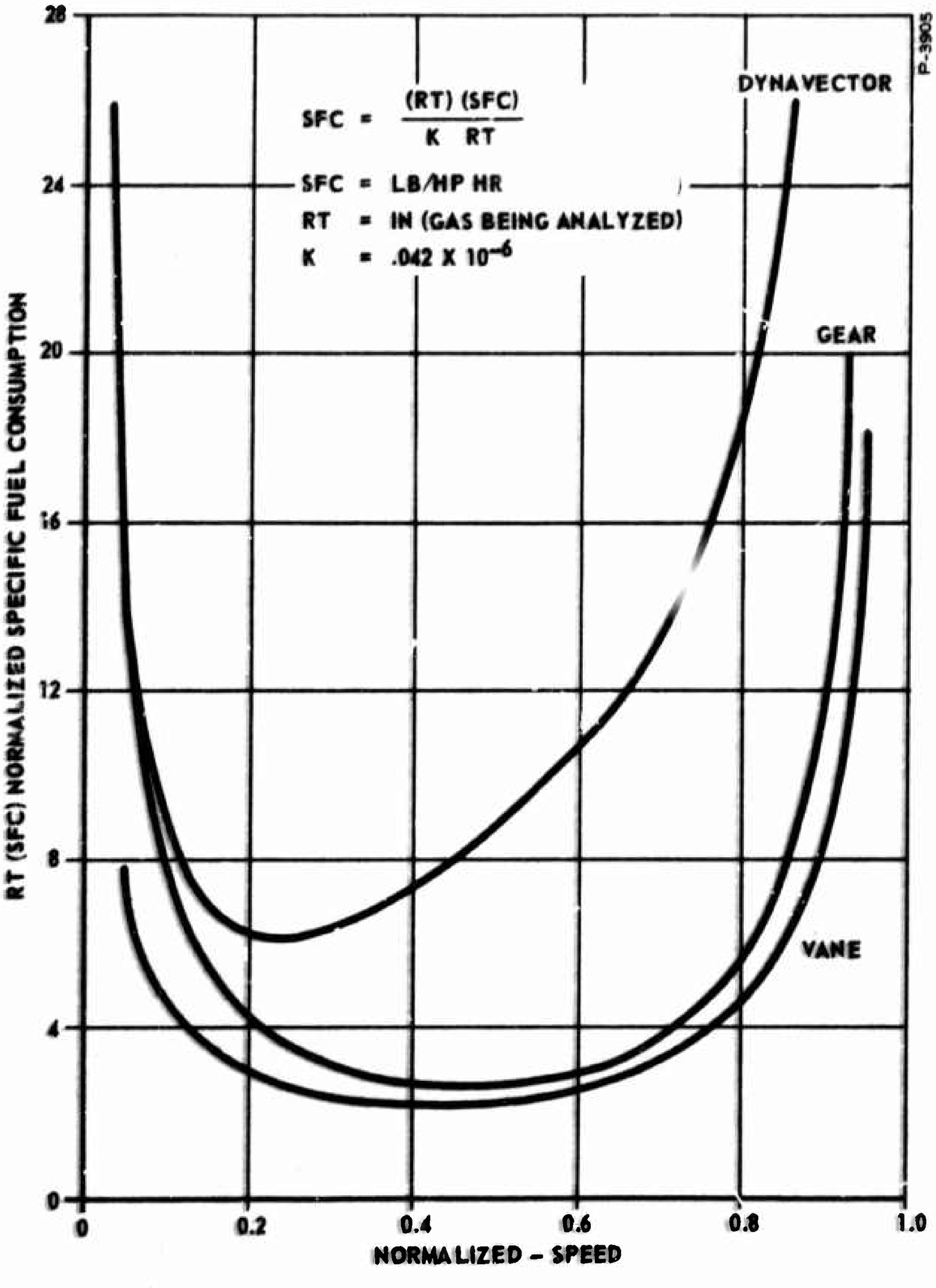


Figure 60 - Normalized Specific Fuel Consumption

Rudder-Angular Velocity:

$$\dot{\theta}' = \theta'_o \omega \sin(\omega t)$$

$$\dot{\theta}' = 54.8 \sin(\omega t) \quad 0 < \omega t < \pi$$

where:

$$\ddot{\theta}'_{max} = 54.8 \text{ deg/sec at rudder null } (0° \text{ displacement})$$

Load Torque:

$$4{,}860 \cos(\omega t) \quad 0 < \omega t < \pi/2$$

The stall consumption may be calculated by

$$W_a = \text{Stall Consumption} = \frac{4{,}860}{5{,}400}(0.012) = 0.0108 \text{ lbs/sec}$$

where

0.012 is the steady-state stall consumption at 5,400 in-lbs load torque

$$W_b = \text{zero load consumption} = \frac{54.8}{60}(0.051) = 0.0465 \text{ lbs/sec}$$

where

54.8 = zero load rudder velocity at 0.435 cps - deg/sec

60 = zero load rudder velocity at steady-state conditions - deg/sec

0.051 = zero load steady-state fuel consumption - lbs/sec

The power transmission consumption values are found by:

$$W = \left[SFC\right] \cdot \left[HP\right] \text{ lbs/sec}$$

where

SFC = specific fuel consumption - lbs/HP-sec

$$= \frac{\left[RT\,(SFC)\right]}{(0.0148)(3{,}600)} = \frac{\left[RT\,(SFC)\right]}{53.3}$$

$\left[RT\,(SFC)\right]$ = normalized specific fuel consumption (from Figure 60) at a percent of zero load speed.

The value:

$$\frac{1}{0.0148} = \frac{1}{\dfrac{RT}{23.8 \times 10^6}} \text{ (gas being analyzed) HP-HR/lbs-in}$$

is a correction factor to convert dimensionless RT(SFC) of Figure 60 to specific fuel consumption - lbs/HP-HR

$$\frac{1}{3{,}600} = \text{correction factor to convert hours to seconds.}$$

<u>For $0 < \omega t < \pi/2$ (Figure 61)</u>

Solve for:

$$W = \left[SFC\right] \cdot \left[HP\right] \text{ lbs/sec} \quad 0 < \omega t < \pi/2$$

Select a rudder velocity of 21 deg/sec:

$$\text{percent of zero load speed} = \frac{21}{54.8} = 38.5 \text{ percent}$$

at 38.5 percent zero load speed;

$$RT\ (SFC) = 7.1 \text{ (reference Figure 60)}$$

$$\text{Load HP} = \frac{T_1 \times N_1}{63{,}000}$$

where

$$T_1 = 4{,}860 \cos (\omega t) \text{ lb-in}$$

$$N_1 = \frac{\text{deg/sec}}{6} \text{ RPM}$$

$$\omega t = \pi/8 \text{ at } \theta' = 21 \text{ deg/sec}$$

using:

$$\dot{\theta}' = 54.8 \sin (\omega t)$$

$$\text{Load HP} = \frac{(4{,}860 \cos 22.5°)(\frac{21}{6})}{63{,}000} = 0.25$$

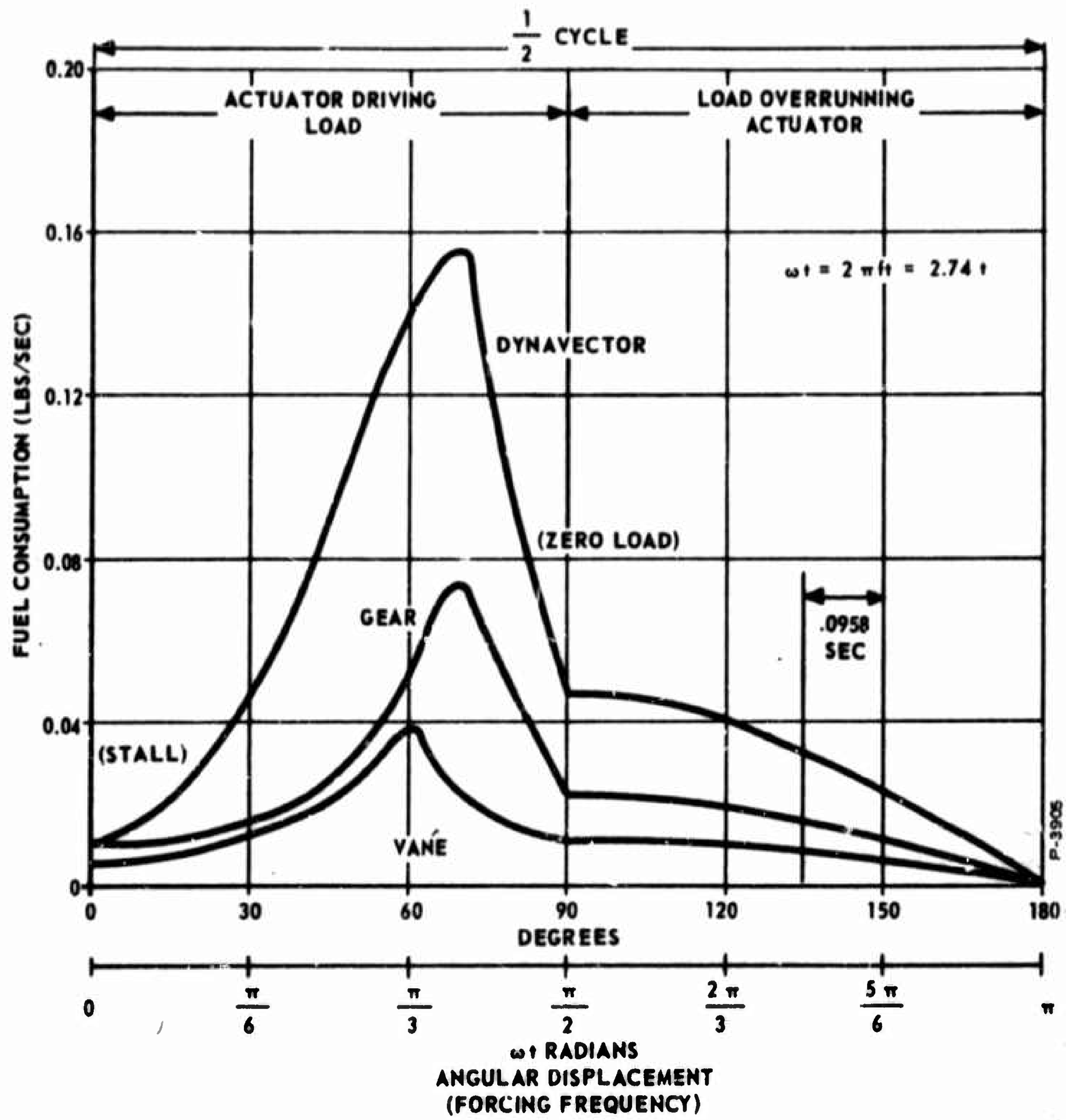


Figure 61 - Cyclic Fuel Consumption-Cyclic Frequency 0.435 cps

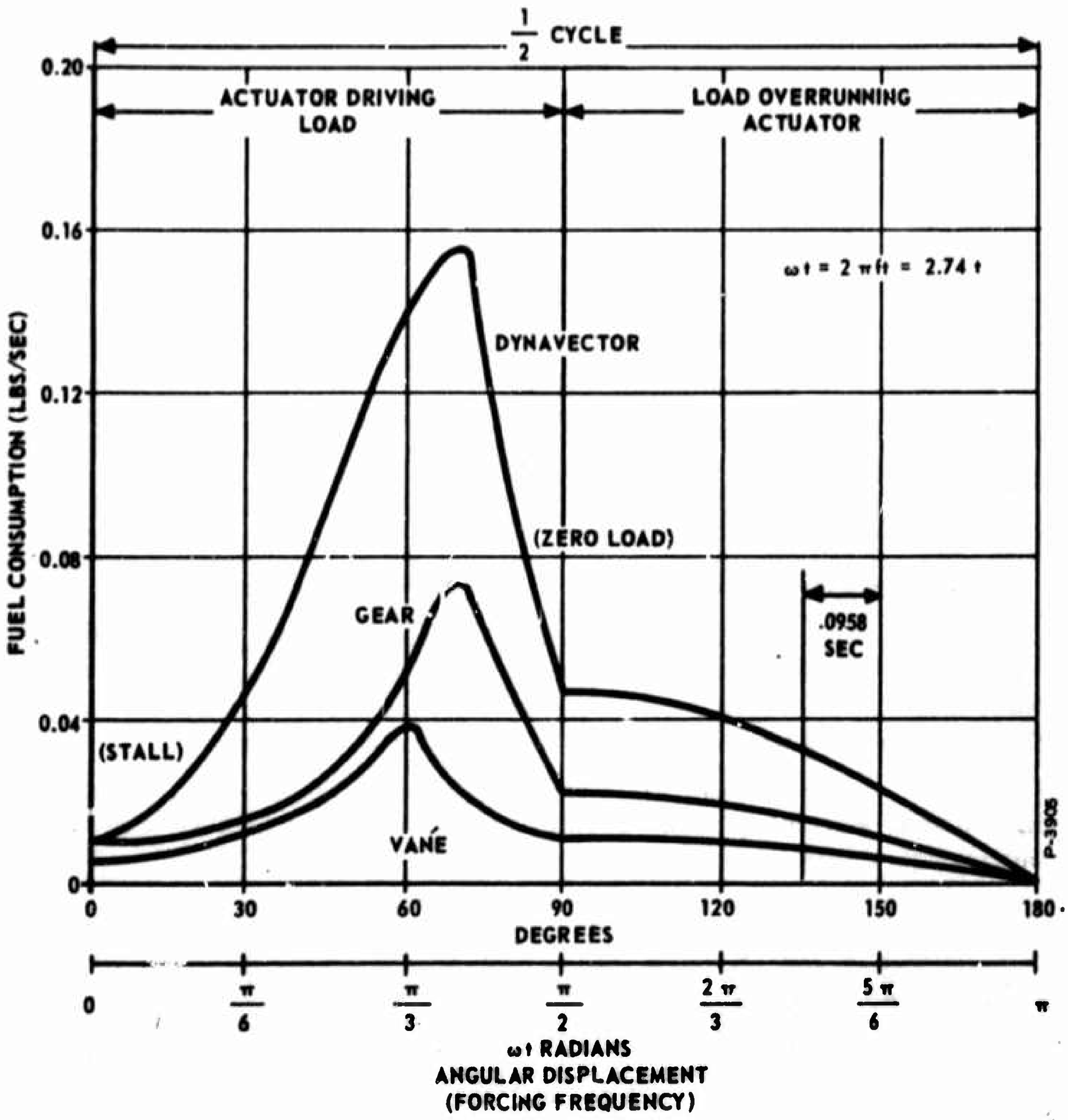


Figure 61 - Cyclic Fuel Consumption-Cyclic Frequency 0.435 cps

therefore:

$$W = \left[SFC\right] \left[HP\right]$$

$$= \frac{\left[RT\ (SFC)\right]}{53.3}\ HP$$

$$W = \frac{7.1}{53.3} \cdot 0.25 = 0.0333 \text{ lbs/sec.}$$

<u>For $\pi/2 < \omega t < \pi$ (Figure 61)</u>

The fuel consumption was computed assuming:

(1) An overrunning torque is acting on the rudder.

(2) During this quarter cycle the motor requires displacement consumption only.

Fuel Consumption $= W_b \times \sin(\omega t)$

where

$W_b = 0.0465$ lbs/sec

ωt = angular displacement of the forcing frequency.

The results of this analysis for each of the motors considered are shown in Figure 61 for a cyclic frequency of 0.435 cps. Figure 62 presents the instantaneous cyclic fuel consumption for a cyclic frequency of 0.871 cps and Figure 63 for a cyclic frequency of 2.18 cps.

The average value for the half cycle shown was determined as follows: Assuming cyclic conditions at a frequency of 0.435 cps the average flow rate is given by

$$W_{average} = \frac{1}{2}\left[\sum_{\omega t = 0}^{\omega t = \pi/2} \frac{W_1 t_1 + W_2 t_2 \ldots W_n t_n}{\sum_1^n t} + 0.707\ W_b\right] \tag{104}$$

where

W_1 = average fuel consumption for interval of time t_1 taken from Figure 61

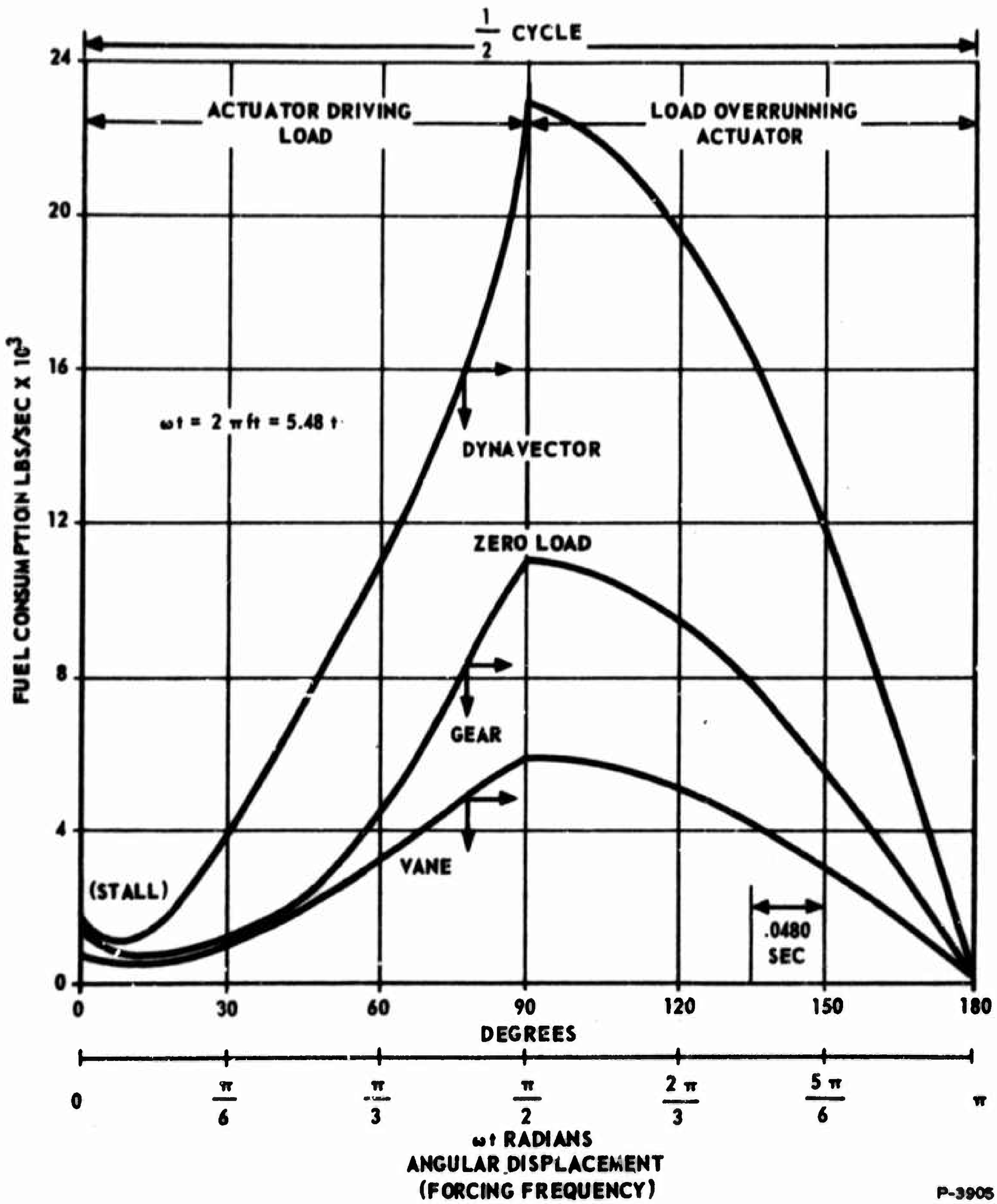


Figure 62 - Cyclic Fuel Consumption-Cycle Frequency 0.871 cps

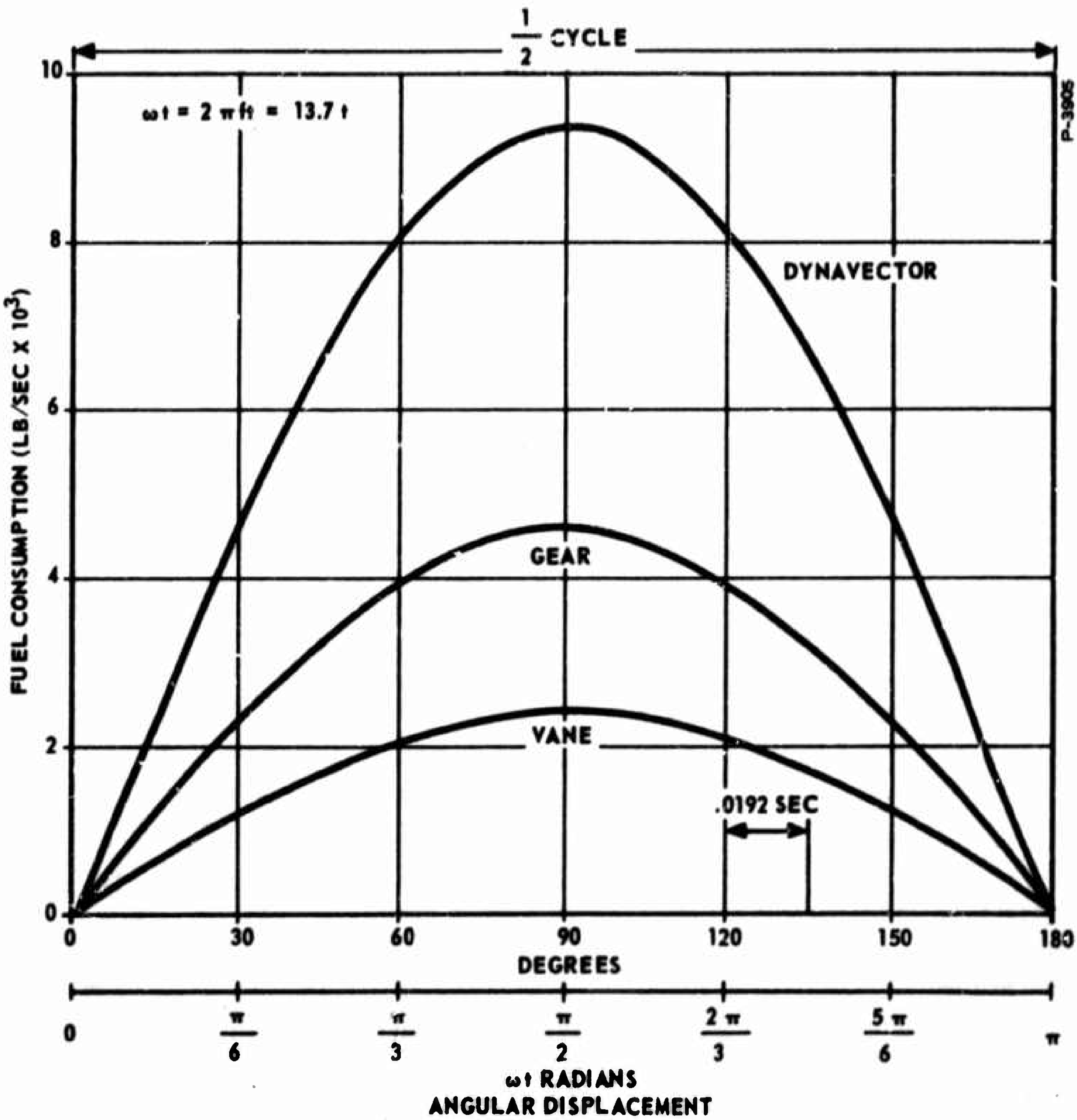


Figure 63 - Cyclic Fuel Consumption-Cycle Frequency 2.18 cps

$$\sum_{1}^{n} t = \text{total time for 1/4 cycle at 0.435 cps} = 0.575 \text{ sec}$$

0.707 = average value of an area under a sinusoidal curve of the form - W sin (ωt)

W_b = zero load consumption = 0.047 lbs/sec

A solution of equation (104) gives

$$W_{avg} = \frac{1}{2}\left[\frac{t}{6t}\left[0.464\right] + 0.0327\right] = 0.055 \text{ lbs/sec.}$$

where

t = 0.0958 seconds

The gear and vane motor consumption rates were averaged in the same manner for a frequency of 0.435 cps and are:

Gear motor - W_{avg} = 0.0234 lbs/sec

Vane motor - W_{avg} = 0.0129 lbs/sec

For a cyclic frequency of 0.871 cps

DYNAVECTOR actuator - W_{avg} = 0.0124 lbs/sec

Gear motor - W_{avg} = 0.0058 lbs/sec

Vane motor - W_{avg} = 0.0034 lbs/sec

For a cyclic frequency of 2.18 cps the average consumption rate is:

DYNAVECTOR actuator - W_{avg} = 0 + 0.707 (0.00935) = 0.0066 lbs/sec

Gear motor - W_{avg} = 0 + 0.707 (0.0046) = 0.0033 lbs/sec

Vane motor - W_{avg} = 0 + 0.707 (0.0024) = 0.0017 lbs/sec

Figure 64 graphically summarizes the duty cycle cyclic fuel consumption rates for each of the three actuation systems analyzed.

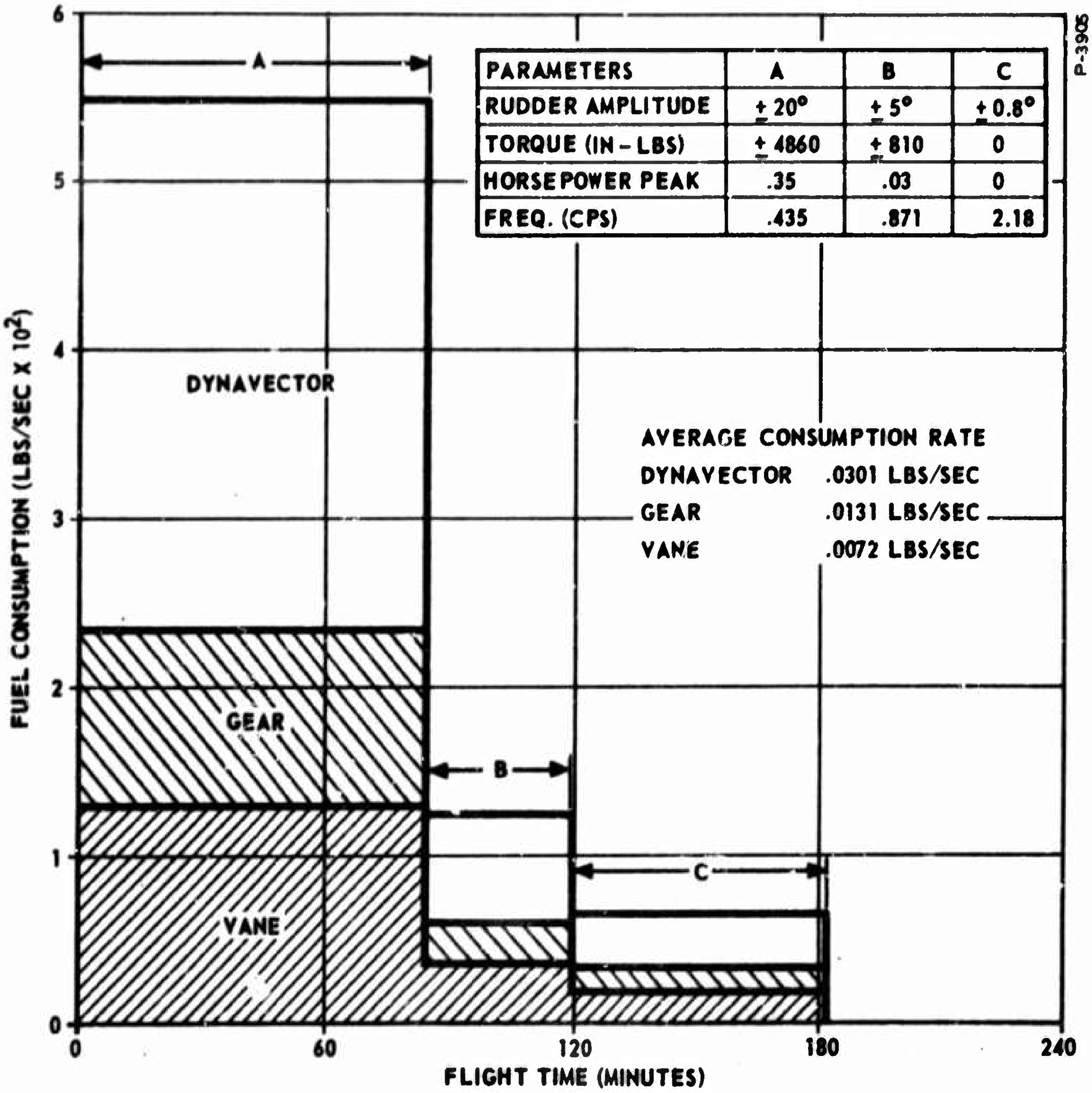


Figure 64 - Duty Cycle Cyclic Fuel Consumption

The average fuel consumption for the entire 4 hour mission may be found by summing Figures 59 and 64 and are as follows:

DYNAVECTOR actuator - W_{avg} = 0.024 lbs/sec

Gear motor - W_{avg} = 0.011 lbs/sec

Vane motor - W_{avg} = 0.006 lbs/sec

6. FAILURE MODE AND RELIABILITY ANALYSES

A. Pneumo-Mechanical Rudder Servomechanism Degraded Performance

The reliability block diagram of Figure 65 indicates that all major components of the pneumatic DYNAVECTOR rudder actuator assembly are considered to be in series; therefore, each component must function properly to attain proper rudder control in the pneumatic mode. A discussion of the degraded mode performance of the pneumatic system is presented below to define the extent and manner of component failures which may be tolerated during flight tests until return to hydraulic system becomes mandatory. It should be noted that several of the series components shown in Figure 65 perform checkout or monitoring functions only. These components would be omitted in a final flight qualified primary system where parallel installation with a hydraulic system was not a requirement. The list of these components includes:

- Two-position hydraulic solenoid valve
- Detent control cylinder
- Input linkage microswitch
- Pneumatic supply solenoid valve
- Clutch supply solenoid valve
- Actuator - rudder interlock valve
- Pneumatic clutch
- Torsional shear section

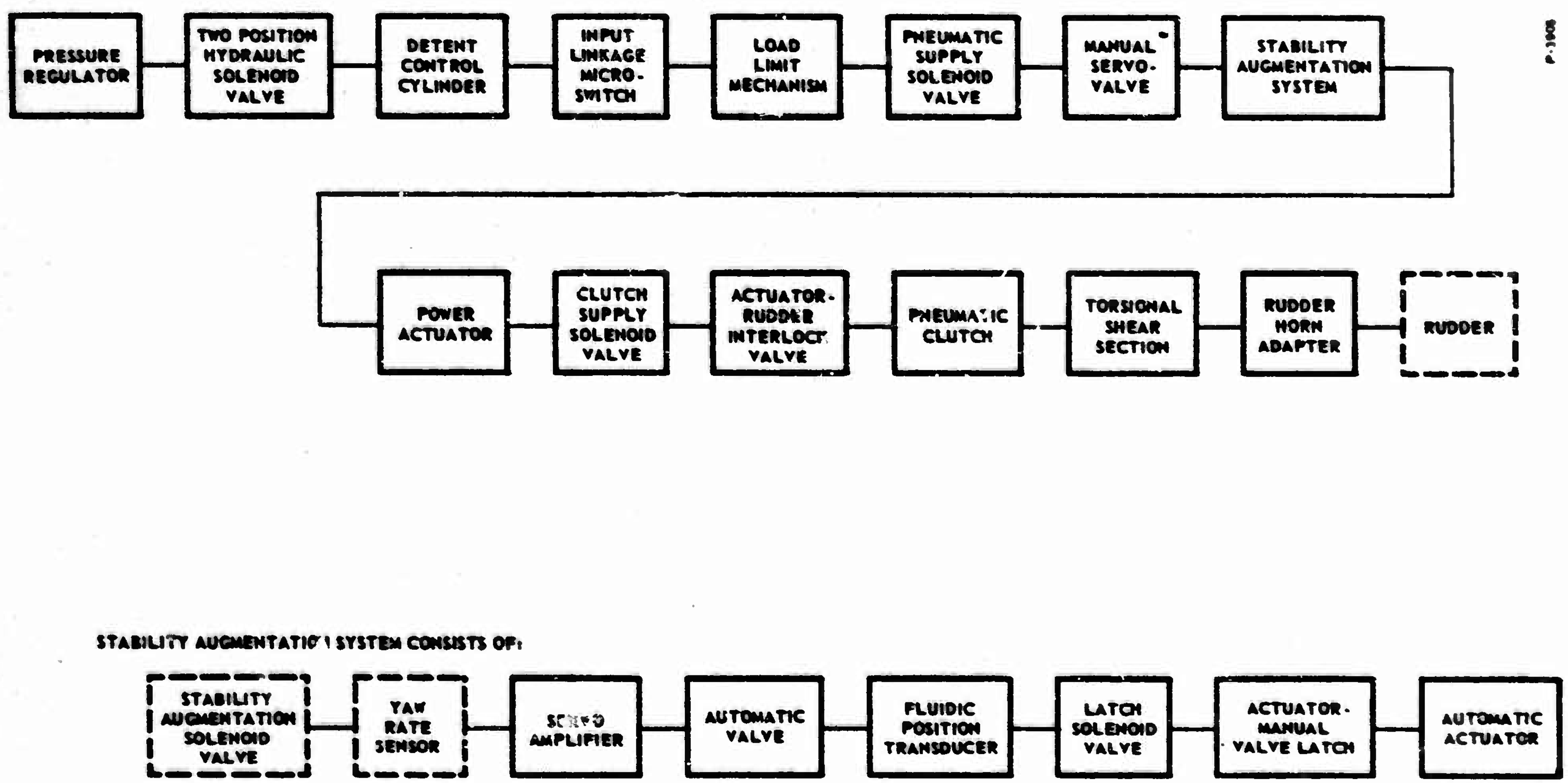


Figure 65 - Pneumo-Mechanical Rudder Control Servomechanism Reliability Block Diagram

(1) Two-Position Hydraulic Solenoid Valve

Failure of the solenoid to displace the spring positioned spool would prevent hydraulic system shutdown and subsequent pneumatic system operation.

(2) Detent Control Cylinder

Failure of the cylinder to actuate against the detent lever spring in response to actuation of the two position hydraulic solenoid valve would produce a condition equivalent to utility hydraulic system failure. The integrated hydraulic power cylinder would act as a solid link between the pilot input linkage and the rudder, thus allowing the rudder to be moved directly by pilot effort.

(3) Input Linkage Switch

Failure of the input linkage switch to actuate upon proper phasing of the pneumatic and hydraulic linkage command positions would be equivalent to an error in the pneumatic system command position and switch over from hydraulics to pneumatics could not occur.

(4) Load Limit Mechanism

Failure of the load limit mechanism may occur so as to cause the mechanism linkage to act either as an open link or a rigid link. If the spring or structural linkage members should fracture, the mechanism would function as an open link and not transmit pilot commands to the pneumatic system manual valve. Upon such a failure, the input linkage microswitch would be forced out of detent since the pneumatic and hydraulic commands would no longer be properly phased, and reversion to hydraulic mode would occur.

In the event the load limit mechanism fails as a locked-up member, the linkage shear section would shear under linkage input force exerted by the rudder pedal control system and the displacement of the hydraulic linkage system would, therefore, not be mechanically limited to the manual valve and linkage stroke freedom which is equivalent to approximately ± 9 degrees of rudder motion.

(5) Pneumatic Supply Solenoid Valve

Failure of the pneumatic supply solenoid valve in the normal off spring loaded position would prevent the pilot from engaging the pneumatic system or from disengaging the hydraulic system.

(6) Manual Servovalve

Failure of the manual servovalve to stroke would cause the load limit linkage microswitch to lose detent position thereby de-energizing the pneumatic supply solenoid valve and the two-position hydraulic solenoid valve, which would return rudder control immediately to hydraulics.

(7) Power Actuator

In the event the power actuator fails, either in a locked-up mode or free running, reversion to hydraulic system rudder control may be accomplished as defined in (6) "Manual Servovalve" above.

(8) Clutch Supply Solenoid Valve

The clutch supply valve is a two-position normally disengaged valve. In the spring position, pneumatic static pressure is always ported to the clutch piston so as to maintain the clutch in a disengaged position. In the event the valve fails in this position, engagement of the clutch to the rudder is prevented.

If the valve fails in the energized position, the clutch would remain engaged to the rudder. If a double failure condition occurs whereby the power actuator system is inoperable, reversion to hydraulic control would occur. Disengagement of the pneumatic system from the rudder would be produced by torquing the shear section of the clutch output with the hydraulic actuator in the normal operative mode.

(9) Actuator - Rudder Interlock Valve

The actuator-rudder interlock valve is a sliding surface valve port that assures proper alignment of the pneumatic actuator output to the rudder position. A failure condition of this valve, whereby pneumatic flow is restricted from the clutch, would prevent clutch engagement to the rudder thereby assuring proper hydraulic control.

(10) Pneumatic Clutch

Failure of the clutch in the disengaged position will eliminate the option of pneumatic rudder control but will not affect normal hydraulic control.

A failure in the engaged position will require a shearing of the clutch output shaft shear section when hydraulic control override occurs.

(11) Stability Augmentation System

The components of the stability augmentation system are shown in series in Figure 65. Both the stability augmentation solenoid valve and yaw rate sensor and signal processor are external to the pneumo-mechanical rudder servomechanism. In the event any of the stability augmentation components become inoperative, stability augmentation operation is non-functional but manual power operation remains operative. The normal position of the actuator-manual valve latch is in manual power mode. Upon latch pneumatic pressurization, the latch becomes locked into stability augmentation/auto pilot mode. Therefore, the predominate failure mode is in the manual power mode condition. In the event failure occurs with the latch in the stability augmentation condition, and only manual power mode is desired, the stability augmentation may be shut down and normal manual power mode operation will still be feasible since the latch locks the manual valve body to the power actuator output shaft by an intermediate member, the automatic actuator output pawl.

B. Mathematical Model and Reliability Analyses

(1) The four hour flight mission duty cycle defined in Section II, paragraph 2 of this report, has been utilized in this failure mode and reliability analysis. Based on this duty cycle, an estimation has been made of the percentage of time each of the four actuator operative modes occurs during a four hour flight time. This estimated percentage distribution is as follows:

Mode	Mission Time
Manual Power Mode	55 Percent
Manual power mode with stability augmentation in monitor	15 Percent
Manual power mode with stability augmentation operative	15 Percent
Autopilot operation	15 Percent

Table XI - Mathematical Model of DYNAVECTOR Rudder Actuator Components

Actuator Component	Failure Probability Summation
(a) Valve Latch Assembly	$\sum_{i=1}^{i=4} \lambda_{ai}$
(b) Pressure Regulator	$\sum_{i=1}^{i=3} \lambda_{bi}$
(c) Power Actuator	$\sum_{i=1}^{i=6} \lambda_{ci}$
(d) Automatic Actuator	$\sum_{i=1}^{i=6} \lambda_{di}$
(e) Pneumatic Clutch	$\sum_{i=1}^{i=8} \lambda_{ei}$
(f) Automatic Servovalve	$\sum_{i=1}^{i=3} \lambda_{fi}$
(g) Detent Control Assembly	$\sum_{i=1}^{i=3} \lambda_{gi}$
(h) Linkage Lever & Manual Assembly	$\sum_{i=1}^{i=10} \lambda_{hi}$
(j) Fluidic Position Transducer	$\sum_{i=1}^{i=2} \lambda_{ji}$
(k) Load Limit Mechanism	$\sum_{i=1}^{i=4} \lambda_{ki}$
(l) Automatic Servovalve Amplifier	$\sum_{i=1}^{i=2} \lambda_{li}$
(m) Actuator-Rudder Interlock Valve	λm

The DYNAVECTOR Rudder Actuator Assembly Block Diagram of Figure 65 and Mathematical Model of Table XI are based upon mission success.

The basic approach to the analysis of the DYNAVECTOR Servomechanism is made from the part failure point of view. The procedure presented herein is limited to components which are essentially assemblies capable of subdivision into parts for analytical purposes.

The Reliability Block Diagram is a representation of the functional relationship of the part to the overall DYNAVECTOR Actuator assembly. Failure mode worksheets, presented in Appendix C, define the components of each major subassembly shown in Figure 65. The pertinent failure modes of each of these components are defined on these worksheets as are the applicable failure rates and operational time requirements.

Table XII - Preliminary Reliability Prediction of DYNAVECTOR Rudder Actuator Components

3000 Hour Design Life	$\Sigma\lambda$ Failure Rate PPMH	t Operational Time, Hours	$\Sigma\lambda t$	$R = e^{-\Sigma\lambda t}$ Reliability
(a) Valve Latch Assembly	12.6	1350	0.017	0.98309
(b) Pressure Regulator	20.0	3000	0.060	0.94176
(c) Power Actuator	23.0	3000	0.069	0.93332
(d) Automatic Actuator	15.5	1350	0.0209	0.97931
(e) Pneumatic Clutch	----	----	0.036	0.96464
(f) Automatic Servovalve	20.0	1350	0.027	0.97336
(g) Detent Control Assembly	7.0	5/6	5.8×10^{-6}	0.999994
(h) Linkage Lever and Manual Assembly	17.0	3000	0.051	0.95027
(j) Fluidic Position Transducer	0.01	1350	13.5×10^{-6}	0.999986
(k) Load Limit Mechanism	6.70	3000	0.0201	0.98010
(l) Automatic Servovalve Amplifier	4.0	1350	0.0054	0.99461
(m) Actuator-Rudder Interlock Valve	6.0	1500	0.009	0.99104
(n) Hydraulic, Pneumatic and Clutch Supply Solenoid Valves	30.0	5/6	25×10^{-6}	0.99997
(o) Latch Solenoid Valve	10.0	1/2	5×10^{-6}	0.99999

The results of the reliability analysis are shown in Table XII. The tabulated reliability values are based on a 3000 hour design life requirement. The power actuator design life is 3000 hours whereas the automatic actuator and associated stability augmentation/autopilot components have a design life of 45 percent of the mission design life or 1350 hours. The reliability values for the detent control cylinder assembly and all solenoid valves are based on the actuation time required for 3000 switching cycles. From Table XII it can be seen that the major gains in reliability for the 3000 hour design life should be directed toward the linkage lever and manual servovalve assembly, the pneumatic clutch, power actuator and pressure regulator.

The estimated reliability for the complete DYNAVECTOR Rudder Actuator Assembly based on these reliability numbers for the duty cycle of the four hour flight mission is 0.9995.

SECTION IV

CONCLUSIONS

The feasibility of a flightworthy low pressure pneumo-mechanical servomechanism capable of controlling an aircraft control surface has been established by this design study.

The design and performance characteristics of the current DYNAVECTOR actuator design described in Section III of this report represents the present status of the DYNAVECTOR actuator development program. These performance parameters may be improved with the continued technological advancement of the DYNAVECTOR actuator concept. DYNAVECTOR actuator performance improvement would be expected in the following parameters:

- o Torque to weight ratio
- o Torque squared to inertia ratio
- o Specific fuel consumption
- o Horsepower to weight ratio
- o Volume
- o Reliability

1. Summary of Qualified DYNAVECTOR Actuator Performance Parameters

Based on the results of this study and the experience the Bendix Corporation has had in the development of pneumatic flight control systems in general, a realistic estimate of the performance parameters for a flight qualified DYNAVECTOR rudder actuator system may be made at this time. The DYNAVECTOR actuator servomechanism system could be designed to duplicate the performance of the integrated hydraulic rudder power cylinder assembly which consists of three major subassemblies; a cylinder assembly, a control assembly, and an electrohydraulic servovalve assembly. The estimated DYNAVECTOR actuator performance parameters are summarized in Table XIII.

Table XIII - Flight Qualified DYNAVECTOR Rudder Actuator Performance Parameters

Installation	Concentric to rudder axis (reference Figures 2-5)
Weight	14 pounds
Volume	(reference Figure 6)
Specific Fuel Consumption (rated horsepower)	0.02 lb/sec-hp
Stall Torque	10200 ± 500 in-lb
Maximum Velocity	60 deg/sec
Maximum Acceleration	150 deg/sec^2
Supply Pressure	50 to 200 psig
Gas Temperature	100°F to 450°F
Altitude	Sea level to 50,000 feet
Ambient Temperature	-65°F to 270°F
Life	3000 hours
Duty Cycle	Reference DS-742 (Section 4.3.3)
4 Hour Mission Average Fuel Consumption at 450°F	0.0043 lb/sec
Cyclic Maximum Fuel Consumption	0.0080 lb/sec
Stall Maximum Fuel Consumption	0.0088 lb/sec
Frequency Response and Phase Shift	(reference Figure 12)
Actuator Rated Horsepower Capability	0.405 hp.
Load Spring Rate	270 lb-in/deg

2. Flight Qualified DYNAVECTOR Actuator Fuel Consumption

The fuel consumption trade-off study presented in Section III of this report has been based on actual test data recorded for the types of motors described. It should be noted that the gear and vane motor test results represent the best fuel consumption test results available after an extensive review of the subject motors. The DYNAVECTOR actuator test results were obtained on the first fluid DYNAVECTOR actuator assembly ever built for performance evaluation. Prior DYNAVECTOR actuator assemblies were plastic shop air penumatic actuators built for concept demonstration purposes. The DYNAVECTOR actuator tested at 400 psig air was model PL-015-U2 intended

Table XIV - Fuel Consumption Requirements of Current Design and Flight Qualified Design DYNAVECTOR Actuators

Power Supply: 50 psig air, 450°F

Actuator Design: Displacement, Motor 4.57 in^3/rev; Transmission Ratio 370:1; Torque Capacity 10,200 in-lb; No-Load Speed 60 deg/sec

Duty Cycle Fuel Concumption Requirements	Current DYNAVECTOR Actuator Consumption (lb/sec)	Flight Qualified DYNAVECTOR Actuator Consumption (lb/sec)
(1) Stall Load (in-lb)		
5400	0.0091	0.0080
4860	0.0082	0.0079
2430	0.0041	0.0040
1620	0.0027	0.0026
810	0.0014	0.0013
(2) Cyclic Conditions		
$\theta_0 \pm 20°$, 0.435 cps, Torque 0 to ±4860	0.0420	0.0080
$\theta_0 \pm 5°$, 0.871 cps Torque 0 to ±810 in-lb	0.0095	0.0027
$\theta_0 \pm 0.8°$, 2.18 cps No-Load	0.0050	0.0015
(3) Four Hour Mission		
Average Fuel Consumption (lb/sec)	0.0180	0.0043

P-3905

for use at 1000°F gas and ambient conditions. The actuator has been successfully operated at these elevated temperature conditions but has not yet been completely optimized with respect to fuel consumption performance. The actuator hardware configuration has not been altered appreciably during the test program except for minor modifications to the porting areas. The initial optimization techniques employed on this model have, however, resulted in a 20 to 60 percent reduction in the fuel consumption requirements for this model.

Continued technological advancement of the DYNAVECTOR actuator concept will result in an improvement in the DYNAVECTOR actuator torque-to-weight and weight-to-horsepower ratios and fuel consumption values. The optimized DYNAVECTOR actuator design would have a fuel consumption requirement compatible with current state-of-the-art vane motor requirements. Table XIV summarizes the current DYNAVECTOR actuator fuel consumption requirements for the rudder actuator duty cycle as compared to the anticipated fuel requirements for a developed and flight qualified DYNAVECTOR actuator.

APPENDIX A

PRELIMINARY DESIGN AND PERFORMANCE SPECIFICATIONS

PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-742	C

ENGINEERING SPECIFICATION

TITLE	DATE
PRELIMINARY PNEUMATIC RUDDER CONTROL ACTUATOR DESIGN SPECIFICATION	28 February 1966

1.0 DESCRIPTION

1.1 This design specification covers the requirements for a low-pressure high response pneumatic rudder control actuator consisting of a rotary power boost actuator and corresponding manually operated servovalve, an automatic control input servovalve, and a linkage summation system to correlate the automatic control actuator output to the desired manual command position. The unit shall have an equal output in each direction with the capability of operation in three different modes: (1) manual operation similar to that of a conventional hydraulic power control unit, (2) operation equivalent to that of a normal autopilot series servo plus power cylinder, and (3) power off operation where the unit shall follow the hydraulic or manual control system.

1.2 A series servo is a position type of pneumatic servo mechanism which adds pneumatic signals from a yaw sensor to those provided by the pilot in such a manner that the rudder can be moved independently of the rudder pedals. The summation of these signals causes a summed output of the power actuator.

2.0 DESIGN REQUIREMENTS

2.1 Material and Workmanship

2.1.1 Materials and workmanship shall be as stated in Specification MIL-P-8564, MIL-P-5518C, and MIL-E-5400.

2.1.2 Metals

All metals used in construction shall be corrosion resistant and/or suitably protected to resist corrosion during the expected service and storage life of the unit. The use of dissimilar materials shall be avoided whenever practicable.

2.1.3 Weight of Materials

Materials of the lightest possible weight, consistent with the service and strength requirements, shall be used.

2.1.4 Non-Standard Material Approval

Approval for the use of non-standard parts and materials shall be in accordance with Specification MIL-P-8564.

PREPARED BY	CHECKED BY	APPROVED BY
D H Kerska D. H. Kerska	R Read	R Read

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2.1.5 Strength

The following safety factors shall be applied to the stress analysis of the unit:

(a) Yield Strength = 1.3 x Design Yield Strength

(b) Ultimate Strength = 1.5 x Design Ultimate Strength

2.2 Dimension

The unit shall conform to the envelope drawing 2161066 and the design data noted thereon.

2.3 Installation

The pneumatic rudder control actuator shall be installed in parallel to the present hydraulic rudder control system per the installation drawing 2160873 and shall incorporate design features such that the following operational modes are possible.

(a) Disengagement of the pneumatic control actuator package when operation in the hydraulic mode is commanded.

(b) Disengagement when a hard over signal in the automatic mode is experienced by the pneumatic servo package.

(c) No occurence of transients when switching modes of operation, (e.g., pneumatic to hydraulic and hydrauli to pneumatic) and the capability of the passive control system to follow the active control system.

2.4 Weight

The pneumatic control actuator shall have a weight goal of twenty (20) pounds.

2.5 Instrumentation

Instrumentation shall be incorporated into the actuator design to monitor the following parameters:

(a) Automatic servovalve input signal

(b) Manual servovalve position

(c) Manual servovalve input signal

(d) Servomotor position and velocity

(e) Servoactuator position and velocity

2.6 Marking of Ports

All ports shall be durably and legibly marked as to function. Declamanias shall not be used. Single letters, such as "P" for pressure, will not be acceptable.

3.0 OPERATIONAL REQUIREMENTS

3.1 Environmental Operating Conditions

The actuator system shall be designed to operate under the following ground and/or flight conditions:

(a) Temperature - Gas temperature of 100° to 450°F
Ambient temperature of -65° to 270°F.

(b) Supply Pressure - Supply pressure shall vary from 50 to 200 psig. If it is necessary to operate at a fixed pressure level, consideration shall be given to implementing an accumulator and conventional pressure regulation subsystem.

(c) Altitude - Sea level to 50,000 feet.

(d) Flight Acceleration Loads - The unit shall be structurally able to withstand without failure a 17.0 g ultimate acceleration in any direction and shall operate satisfactorily without malfunction under a 12.0 g acceleration in any direction.

3.2 Torque-Speed Requirements

The pneumatic control actuator shall be capable of operation in accordance with the curve of Figure A with a supply pressure of 50 psi.

3.3 Output Motion

The output motion of the control actuator shall be consistent with the following requirements:

(a) Manual Operation Output - ± 20 degrees with ± 1 degree accuracy. Angular velocity 60 deg/sec.

(b) Automatic Operation - ± 5 degrees with ± 0.25 degree positional accuracy. Angular velocity of 60 deg/sec.

(c) Maximum surface deflection velocity shall be 60 deg/sec for both manual and automatic operation.

(d) Maximum surface deflection acceleration shall be 150 deg/sec^2 for both manual and automatic operation.

3.4 Output Torque

The control actuator shall deliver a rotary output stall torque of 10200 ± 500 in-lb.

3.5 Chatter and Instability

The unit shall operate smoothly without sustained chatter or instability under all operating conditions.

3.6 Dynamic Response

The unit, under a linear spring load as shown in Figure B, shall operate within the limits specified in Figure C. Load inertia is negligible.

3.7 Duty Cycle

The unit shall be capable of withstanding a duty cycle of 3000 hours. (reference Section 4.3.3)

4.0 QUALIFICATION REQUIREMENTS

4.1 Data Required

The following data shall be supplied as part of the qualification test:

(a) Qualification Test Procedure - Five (5) copies of the proposed Qualification Test Procedure shall be submitted to the Contracting Agency prior to testing.

(b) Bearing Failure - A history of bearing failure and malfunctions during qualification testing shall be provided.

(c) Stress Report - Two (2) copies of a complete stress analysis of the unit shall be provided.

(d) Effects of Flight Inertia Loads - Two (2) copies of an analysis of the unit considering the effects of "g" loading as specified in Paragraph 3.1 (d) shall be submitted.

(e) Clearance Analysis - An analysis of clearances, certifying that binding will not occur at extreme temperatures with worst tolerance build-up shall be supplied.

(f) Flow Data - The unit flow shall be recorded for all qualification tests and such record shall be recorded in the Test Log.

(g) Test Log - An accurate record of the qualification tests conducted per the Qualification Test Procedure of Paragraph 4.1 (a) shall be kept in a log book.

4.2 Test Conditions

4.2.1 Environment

The qualification tests shall be accomplished with the unit subjected to the environmental conditions specified in Paragraph 3.1 (a) and (b) as defined below.

4.2.2 Valve Operation

Operation of the control valve shall be accomplished manually in all of the tests unless specifically noted otherwise.

4.3 Qualification Tests

4.3.1 Flow

The unit flow shall not exceed 0.024 lbs/sec while operating with a gas supply temperature in the range 70°F to 450°F.

4.3.2 Reversion to Hydraulic Mode

The unit shall be checked for satisfactory reversion to hydraulic operation upon command or loss of pneumatic supply pressure per Paragraph 2.3 (a).

4.3.3 Life Cycle Endurance Tests

4.3.3.1 Room Temperature Test - The actuator shall be subjected to the following load-speed conditions with a room temperature air supply pressure of 50 [illegible]:

Mode	Amplitude (degrees)	Frequency (cps)	Number of Cycles	Linear Load Variation (lb-in)
Manual	20	0.435	125,000	No Load
	20	0.435	2,500	0-4860
	10	0.615	7,500	0-2430
	10	0.615	7,500	0-1620
	5	0.871	25,000	0-810
Automatic	5	0.871	75,000	No Load
	2	1.37	175,000	No Load
	0.8	2.18	500,000	No Load
	5	0.871	2,500	4100 - 6700
	2	1.372	2,500	4900 - 5900

4.3.3.2 Rapid Warm-Up - With the actuator at a cold soak temperature of -65°F, an air supply at 450°F and 50 psig shall be provided to the actuator. The load-speed conditions defined in 4.3.3.1 shall be repeated to simulate an actuator cold start and hot test run. Ambient temperature shall be 270°F during hot test.

4.3.3.3 Room Temperature and Warm-Up Recycle - Following test 4.3.3.2 repeat test 4.3.3.1 once, followed by test 4.3.3.2 to be repeated once.

4.3.4 Frequency Response

The actuator shall be checked for frequency response when subjected to the spring rate load characteristic as defined in Figure B. Tests shall be conducted with 50 psig air supply at both room temperature and at 450°F.

4.3.5 Chatter and Instability

The unit shall operate smoothly without chatter or instability under all conditions during the qualification tests specified.

4.3.6 Humidity

The unit shall be subjected to the Humidity Tests of MIL-E-5272, Procedure I.

4.3.7 Vibration

The unit shall be vibrated in accordance with MIL-E-5272, Procedure I.

5.0 APPLICABLE DOCUMENTS AND DRAWINGS

5.1 Applicable Documents

The following documents of the issue in effect on date of contract form a part of this specification to the extent specified herein:

MIL-P-8564	Pneumatic System Components, Aeronautical General Specification for
MIL-P-5518C	Pneumatic Systems, Aircraft; Design, Installation, and Data Requirements for
MIL-E-5400	Electronic Equipment - Airborne General Specification for
MIL-E-5272C	Environmental Testing, Aeronautical and Associated Equipment, General Specification for

PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-742	a

ENGINEERING SPECIFICATION

TITLE	DATE
PRELIMINARY PNEUMATIC RUDDER CONTROL ACTUATOR DESIGN SPECIFICATION	28 February 1966

MIL-S-4040C Solenoid, Electrical, General Specification for

MIL-A-8629 (AER) Airplane Strength and Rigidity

MIL-P-5514B Packings, Installation and Gland Design of Aircraft Hydraulic and Pneumatic

5.2 Applicable Drawings

The following drawings, incorporating the revisions noted, shall form a part of this specification to the extent specified herein:

BRLD 2160873 Layout, F101BAirplane Rudder Control System, Power Cylinder Linkage

BRLD 2161066 F101B Rudder Dynavector Actuator

PREPARED BY	CHECKED BY	APPROVED BY
D.H. Kerska D. H. Kerska		R Ross

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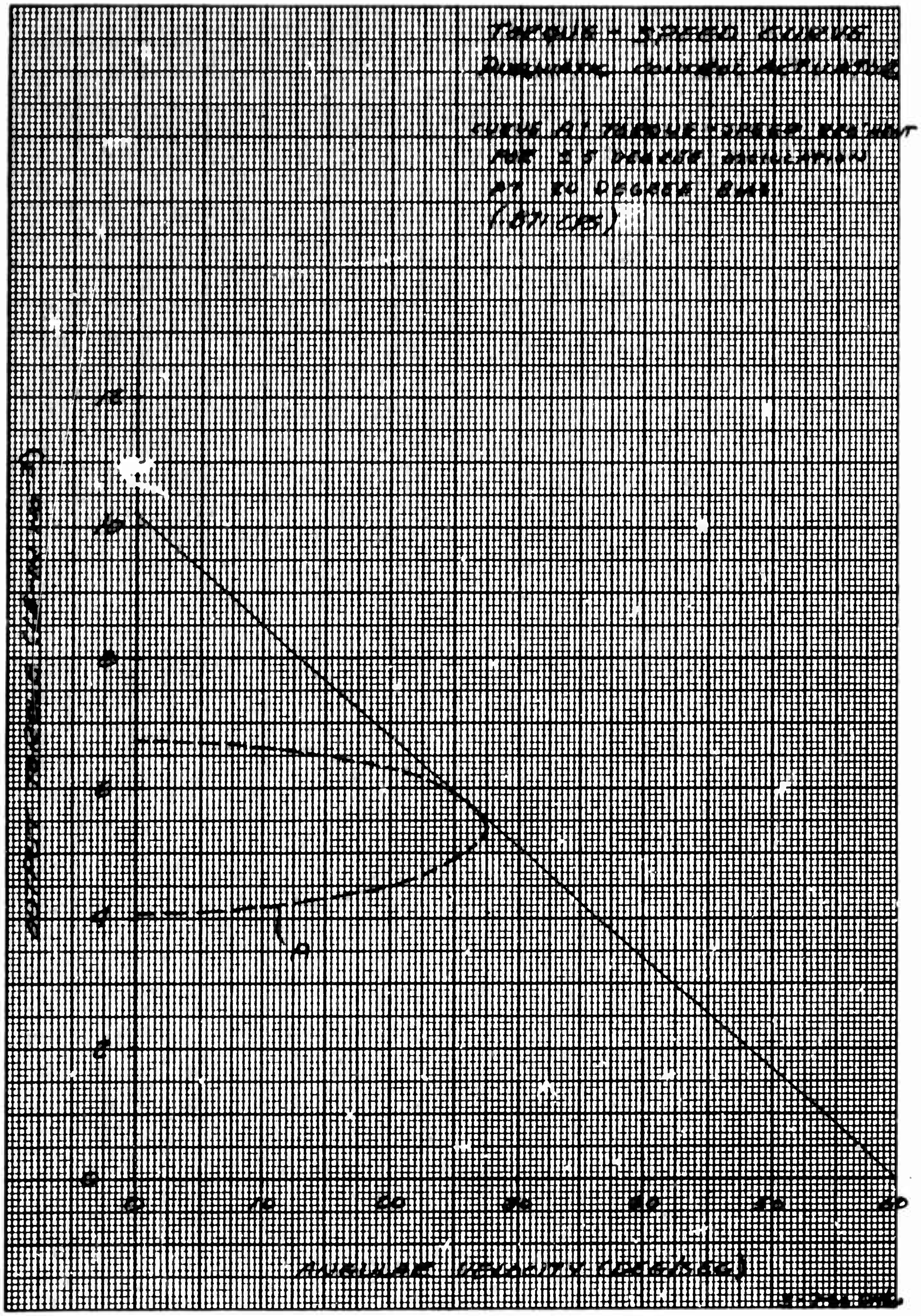

DS 742 . Figure A

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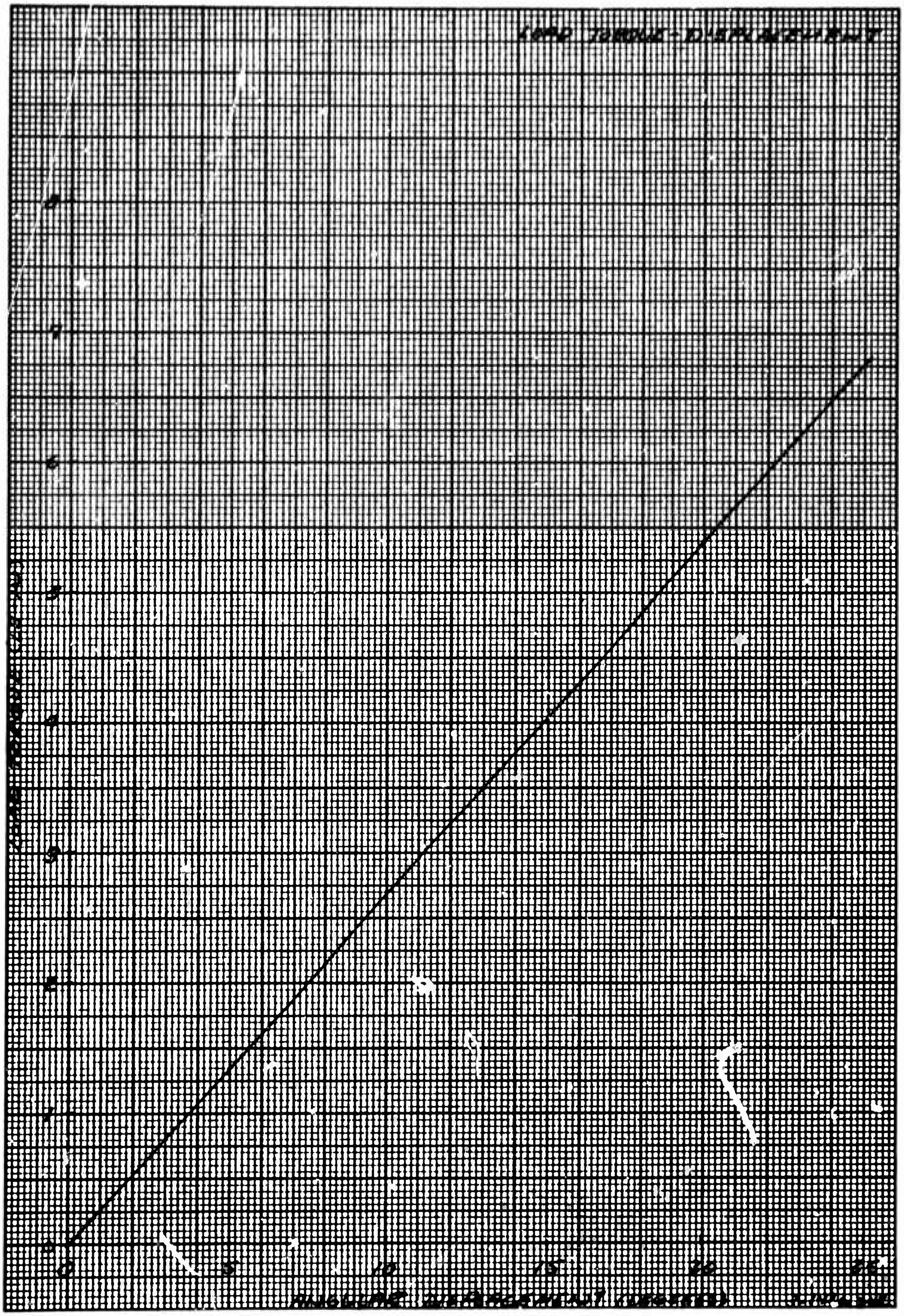
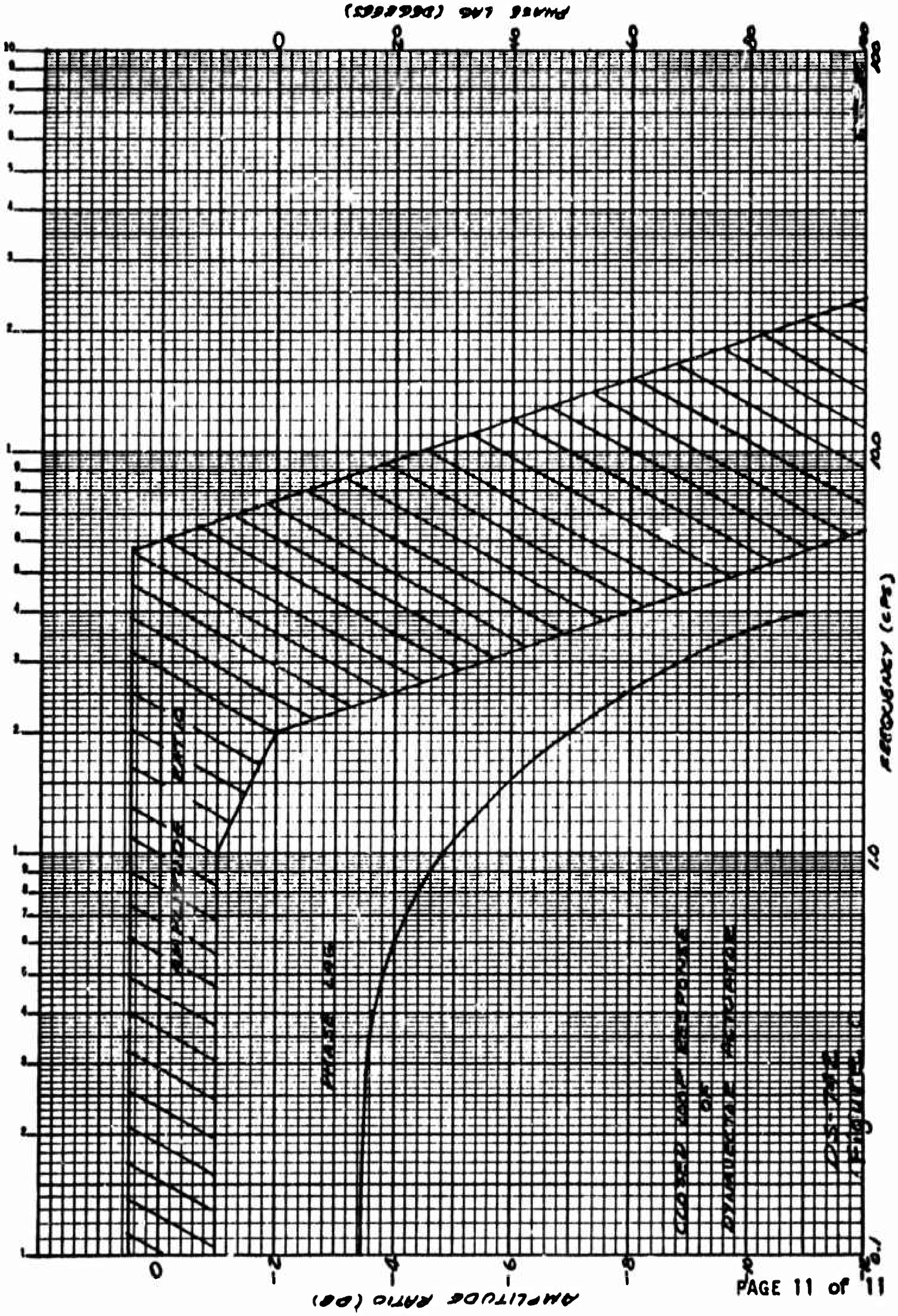


DS 742 Figure 8


PHASE LAG (DEGREES)
FREQUENCY (CPS)
AMPLITUDE RATIO (DB)
AMPLITUDE RATIO
PHASE LAG

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PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-311		11272	DS - 743	A

ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Rudder Control Actuator Available Power Supply	March 15, 1966

1.0 GENERAL

This specification defines the available pneumatic power supply for the Pneumo-Mechanical Servomechanism applicable to F101B aircraft rudder control. The power is derived from bleed air of the compressor section of the Pratt and Whitney JT3 turbojet engines which is also the power supply for the cockpit pressurization and air-conditioning system. This specification is to serve as information for the preliminary design of the pneumatic rudder control actuator.

2.0 POWER SUPPLY CHARACTERISTICS

2.1 Supply Pressure

The pneumatic rudder control actuator shall be capable of performance as specified in DS-742 with a supply pressure variation of 50 to 200 psig. Based on the engine compressor discharge characteristics in Figures A and C the availability of this supply pressure is limited to normal engine operation to an altitude of approximately 40,000 feet at flight speeds greater than Mach 1.0 and for throttled down or idling engine operation to 10,000 feet. The pneumatic control actuator can operate with supply pressures less than 50 psig; however, degraded performance can be expected.

2.2 Supply Temperature

The temperature of the supply gas may vary from 100°F to 450°F as indicated in the flight envelope summary presented in Figure C. The apparent discrepancy between the data presented in Figure C and Figure B is due to different locations in the engine bleed air duct. The curves of Figure B are included to allow extrapolation for aircraft operating conditions not presented in the flight envelope summary of Figure C.

2.3 Supply Flow

The maximum available supply flow is obtained from the table in Figure C. This varies from a minimum of 0.5 lb/sec for the aircraft in the descent mode at an altitude of 30,000 feet to 3.5 lb/sec for sea level take-off. The primary function of the engine bleed air is to provide flow for cockpit pressurization and air-conditioning system; therefore, actuator flow requirements must be kept to a minimum.

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JJKerska		RG Read

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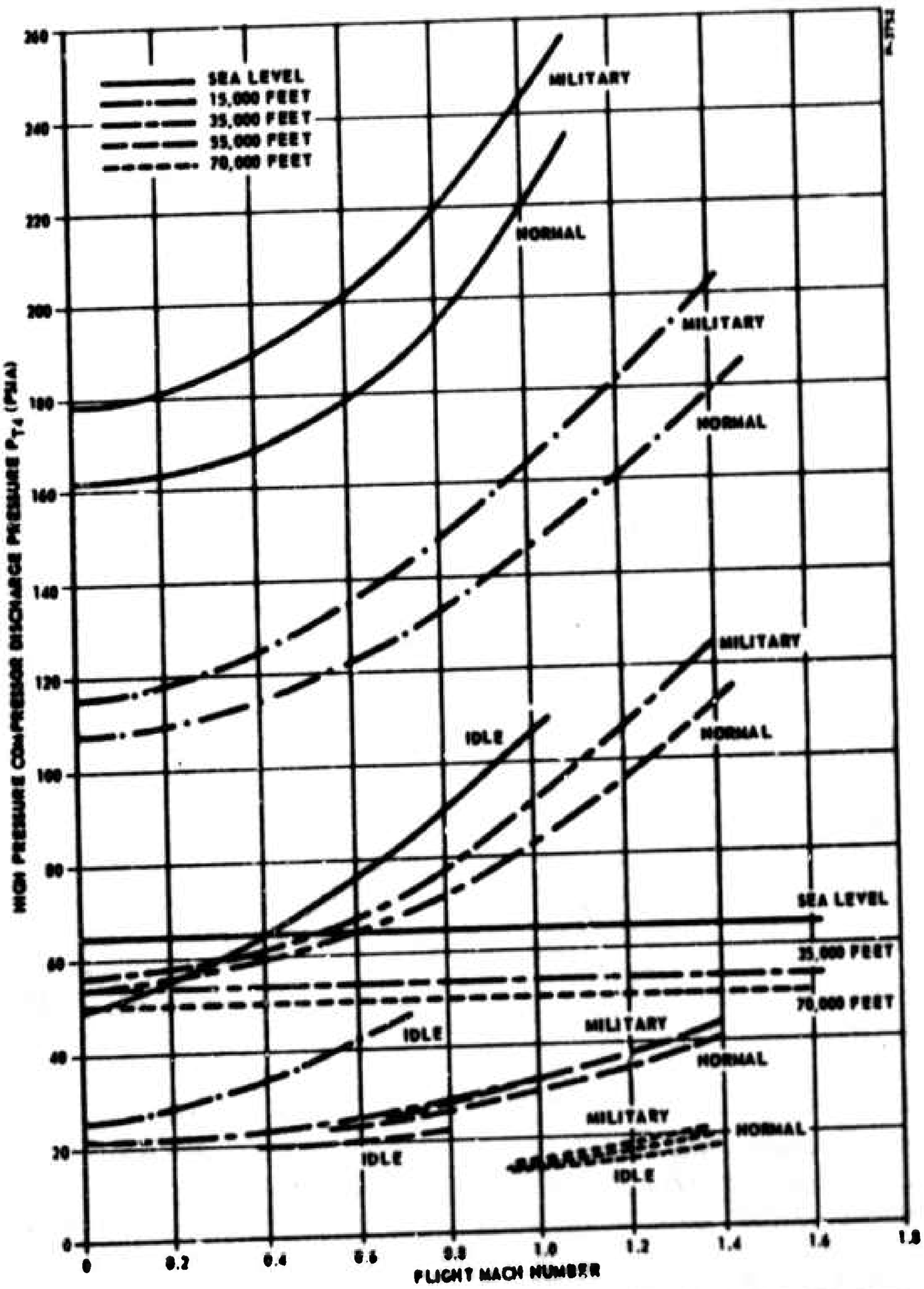
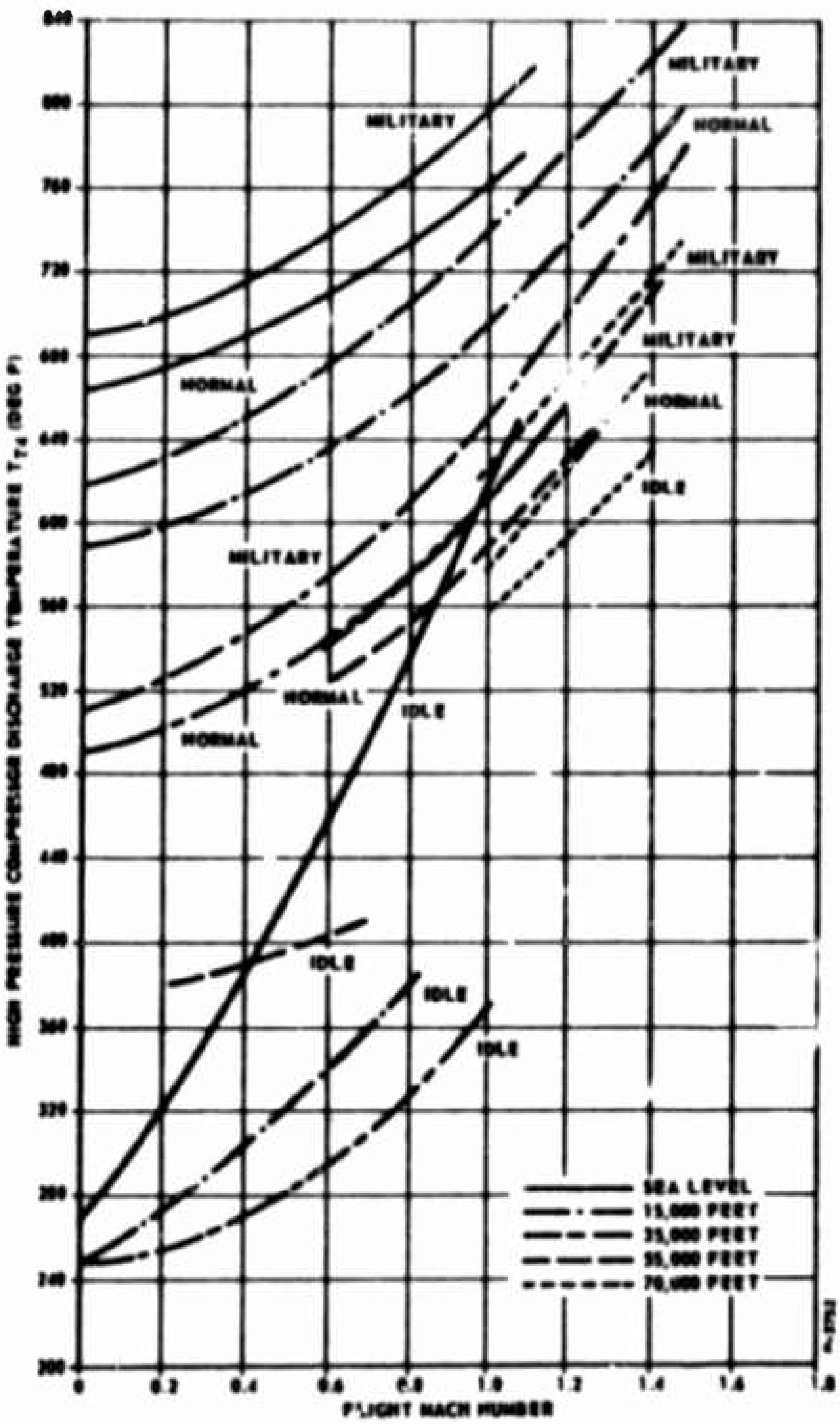
FIGURE A



Compressor Discharge Pressure Versus Flight Mach Number for Pratt and Whitney JT3 Engine

FIGURE B



Compressor Discharge Temperature Versus Flight Mach Number for Pratt and Whitney JT3 Engine

PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
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ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Rudder Control Actuator Available Power Supply	March 15, 1966

Flight Condition	Alt. (ft)	Mach No.	Bleed Air			
			P_s (psig)	P_s (psia)	t (^oF)	w (lb/sec)
Take-off	0	0	108.3	123	< 100	3.5
Climb	10,000	0.50	86.89	97	< 100	3.1
Climb	15,000	0.50	82.71	91	< 100	2.4
Climb	30,000	0.71	61.63	66	145	1.8
Level	40,000	1.05	52.28	55	595	1.2
	38,800	1.03	55.12	59	530	1.2
Level	[illegible],600	0.87	41.24	44	486	0.9
Descent	30,000	0.86	21.63	26	490	0.5
Descent	16,200	0.62	52.11	60	< 100	1.2
Landing Approach	1,000	< 0.40	37.81	52	< 100	0.8

P_s - Static bleed pressure

<u>Flight Envelope Compressor Bleed Air Characteristics</u>

FIGURE C

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APPENDIX B

FLIGHTWORTHY SYSTEM SPECIFICATIONS

PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-747	

ENGINEERING SPECIFICATION

TITLE FLIGHTWORTHY PNEUMATIC DYNAVECTOR* RUDDER ACTUATOR

DATE May 15, 1966

1.0 DESCRIPTION

This design specification covers the requirements for a flightworthy low-pressure high response pneumatic rudder control actuator having an equal output in each direction and with the capability of operation in four different modes.

(a). Manual operation similar to that of a conventional integrated hydraulic power cylinder.
(b). Operation equivalent to that of a normal autopilot series servo plus power cylinder.
(c). Manual operation with stability augmentation in monitor.
(d). Manual operation with stability augmentation operative.

The pneumatic actuator in conformance to the specification requirements consists of:

(a). Load limit mechanism
(b). Manual valve lever
(c). Power actuator
(d). Single point engagement pneumatic clutch
(e). Rudder horn adapter
(f). Actuator-rudder interlock valve
(g). Actuator-manual valve latch
(h). Automatic valve
(i). Automatic valve amplifier
(j). Automatic actuator
(k). Fluidic position transducer
(l). Clutch, power supply, and latch switches
(m). Miscellaneous monitoring instrumentation

The design and performance characteristics of the above actuator are summarized in Table I.

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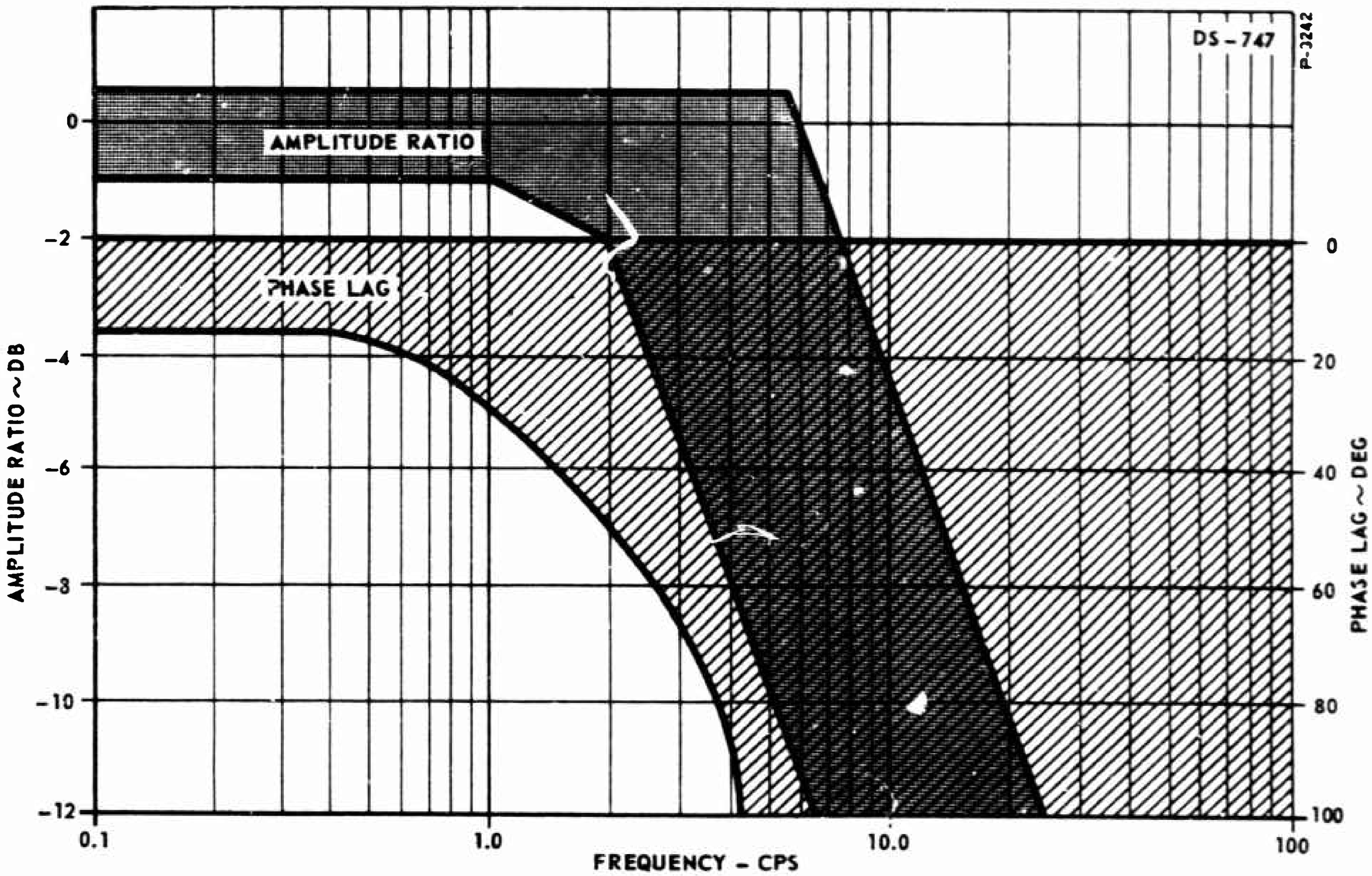

Figure C - Closed Loop Response of Servo Assembly

PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-747	

ENGINEERING SPECIFICATION

TITLE FLIGHTWORTHY PNEUMATIC DYNAVECTOR RUDDER ACTUATOR

DATE May 15, 1966

TABLE I

DYNAVECTOR Model	PH-370-B1
Stall Torque	10,200 in-lbs
No-Load Speed	60 deg/sec
Maximum Horsepower	0.405 hp
Weight	14 lbs
Design Life	3,000 hours
Maximum Fuel Consumption @ 450°F	0.018 lbs/sec
Supply Pressure	50 psig
Gas Temperature	100°F to 450°F
Manual Input	± 1.65 in. (± 25 deg rudder motion)
Automatic Input	± 2 psid @ 15 psig
Automatic Output (Relative to Manual Position)	± 5 deg
Power Actuator Response to Manual Inputs (-3 db point)	25 cps
Power Actuator Response to Stability Augmentation (-3 db point)	6.2 cps

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D. H. Kerska D. H. Kerska		RG Read

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2.0 DESIGN REQUIREMENTS

2.1 Material and Workmanship

2.1.1 Materials and workmanship shall be as stated in Specification MIL-T-8564C, MIL-P-5518C, and MIL-E-5400.

2.1.2 Metals

All metals used in construction shall be corrosion resistant and/or suitably protected to resist corrosion during the expected service and storage life of the unit. The use of dissimilar materials shall be avoided whenever practicable.

2.1.3 Weight of Materials

Materials of the lightest possible weight, consistent with the service and strength requirements, shall be used.

2.1.4 Non-Standard Material Approval

Approval for the use of non-standard parts and materials shall be in accordance with Specification MIL-T-8564C.

2.1.5 Strength

The following safety factors shall be applied to the stress analysis of the unit:

(a) Yield Strength = 1.3 x Design Yield Strength
(b) Ultimate Strength = 1.5 x Design Ultimate Strength

2.2 Dimension

The unit shall conform to the envelope drawing 2162309 and the design data noted thereon.

2.3 Installation

The pneumatic rudder control actuator shall be installed in parallel to the present hydraulic rudder control system per the installation drawing 2161693 and shall incorporate design features such that the following operational modes are possible.

(a) Disengagement of the pneumatic control actuator package when operation in the hydraulic mode is commanded.

(b) Disengagement when a hard over signal in the automatic mode is experienced by the pneumatic servo package.

(c) No occurrence of transients when switching modes of operation, (e.g., pneumatic to hydraulic and hydraulic to pneumatic) and the capability of the passive control system to follow the active control system.

2.4 Weight

The pneumatic control actuator weight shall not exceed fourteen (14) pounds. The control actuator consists of the power actuator, automatic actuator, manual and automatic servovalves, the fluidic transducer and flueric servoamplifiers and the input linkage lever.

2.5 Instrumentation

Instrumentation shall be incorporated into the actuator design to monitor the following parameters:

(a) Power actuator position relative to ground.
(b) Automatic actuator position relative to power actuator.
(c) Manual valve body relative to power actuator output.
(d) Input linkage lever relative to power actuator output.
(e) Automatic valve spool position relative to automatic valve body.
(f) Clutch engagement and disengagement positions.

2.6 Marking of Ports

All ports shall be durably and legibly marked as to function. Decalcomanias shall not be used. Single letters, such as "P" for pressure, will not be acceptable.

PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-747	

ENGINEERING SPECIFICATION

TITLE	DATE
FLIGHTWORTHY PNEUMATIC DYNAVECTOR RUDDER ACTUATOR	May 15, 1966

3.0 OPERATIONAL REQUIREMENTS

3.1 Environmental Operating Conditions

The actuator system shall be designed to operate under the following ground and/or flight conditions:

(a) Temperature - Gas Temperature of 100° to 450°F
Ambient temperature of -65° to 270°F.

(b) Supply Pressure - Supply pressure shall vary from 50 to 200 psig. If it is necessary to operate at a fixed pressure level, consideration shall be given to implementing an accumulator and conventional pressure regulation subsystem.

(c) Altitude - Sea level to 50,000 feet.

(d) Flight Acceleration Loads - The unit shall be structurally able to withstand without failure a 17.0 g ultimate acceleration in any direction and shall operate satisfactorily without malfunction under a 12.0 g acceleration in any direction.

3.2 Torque-Speed Requirements

The pneumatic control actuator shall be capable of operation in accordance with the curve of Figure A with a supply pressure of 50 psig.

3.3 Output Motion

The output motion of the control actuator shall be consistent with the following requirements:

(a) Manual Operation Output - ± 20 degrees with ± 1 degree accuracy. Angular velocity 60 deg/sec.

(b) Automatic Operation - ± 5 degrees with ± 0.25 degree positional accuracy. Angular velocity of 60 deg/sec.

(c) Maximum surface deflection velocity shall be 60 deg/sec for both manual and automatic operation.

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(d) Maximum surface deflection acceleration shall be 150 deg/sec^2 for both manual and automatic operation.

3.4 Output Torque

The control actuator shall deliver a rotary output stall torque of 10,200 ± 500 in-lbs.

3.5 Chatter and Instability

The unit shall operate smoothly without sustained chatter or instability under all operating conditions.

3.6 Dynamic Response

The unit, under a linear spring load as shown in Figure B, shall operate within the limits specified in Figure C. Load inertia is negligible.

3.7 Duty Cycle

The unit shall have a design life of 3,000 hours. A typical four-hour flight envelope which the unit must be capable of withstanding is shown in Tables II and III.

Table II - Stall Torque During Four-Hour Flight

Stall Torque (in-lbs)	% Load	Amplitude (degrees)	Duration (minutes)
5,400	100	20	4
4,860	90	20	4
2,430	90	10	10
1,620	60	10	10
810	60	5	30
		Total Time	58 minutes

Table III - Oscillatory Conditions During Four-Hour Flight

Mode	Amplitude (± degrees)	Frequency (cps)	Torque Variation (lb-in)	Time (minutes)
Automatic	5	0.871	0 to 810	35
Automatic	0.8	2.18	0	63
	Total Time in Automatic Mode			98 minutes
Manual	20	0.435	0 to 4,860	84 minutes

4.0 QUALIFICATION REQUIREMENTS

4.1 Data Required

The following data shall be supplied as part of the qualification test:

(a) Qualification Test Procedure - Five (5) copies of the proposed Qualification Test Procedure shall be submitted to the Contracting Agency prior to testing.

(b) Bearing Failure - A history of bearing failure and malfunctions during qualification testing shall be provided.

(c) Stress Report - Two (2) copies of a stress analysis of the unit shall be provided.

(d) Effects of Flight Inertia Loads - Two (2) copies of an analysis of the unit considering the effects of "g" loading as specified in Paragraph 3.1 (d) shall be submitted.

(e) Clearance Analysis - An analysis of clearances, certifying that binding will not occur at extreme temperatures with worst tolerance build-up shall be supplied.

4.2 Test Conditions

4.2.1 Environment

The qualification tests shall be accomplished with the unit subjected to the environmental conditions specified in Specification PS-412.

4.2.2 Valve Operation

Operation of the control valve shall be accomplished manually in all of the tests unless specifically noted otherwise.

4.3 Qualification Tests

The unit shall be subjected to the functional and endurance tests as specified in Specification PS-412.

5.0 APPLICABLE DOCUMENTS AND DRAWINGS

5.1 Applicable Documents

The following documents of the issue in effect on date of contract form a part of this specification to the extent specified herein:

MIL-P-8564C	Pneumatic System Components, Aeronautical General Specification for
MIL-P-5518C	Pneumatic Systems, Aircraft; Design, Installation, and Data Requirements for
MIL-E-5400	Electronic Equipment - Airborne General Specification for
MIL-E-5272C	Environmental Testing, Aeronautical and Associated Equipment, General Specification for
MIL-S-4040C	Solenoid, Electrical, General Specification for
MIL-A-8629 (AER)	Airplane Strength and Rigidity
MIL-P-5514B	Packings, Installation and Gland Design and Aircraft Hydraulic and Pneumatic

5.2 Applicable Drawings

The following drawings, incorporating the revisions noted, shall form a part of this specification to the extent specified herein:

BRLD 2161698 Layout, F101B Airplane Rudder Control System, Power Cylinder Linkage

BRLD 2162309 F101B Rudder DYNAVECTOR Actuator

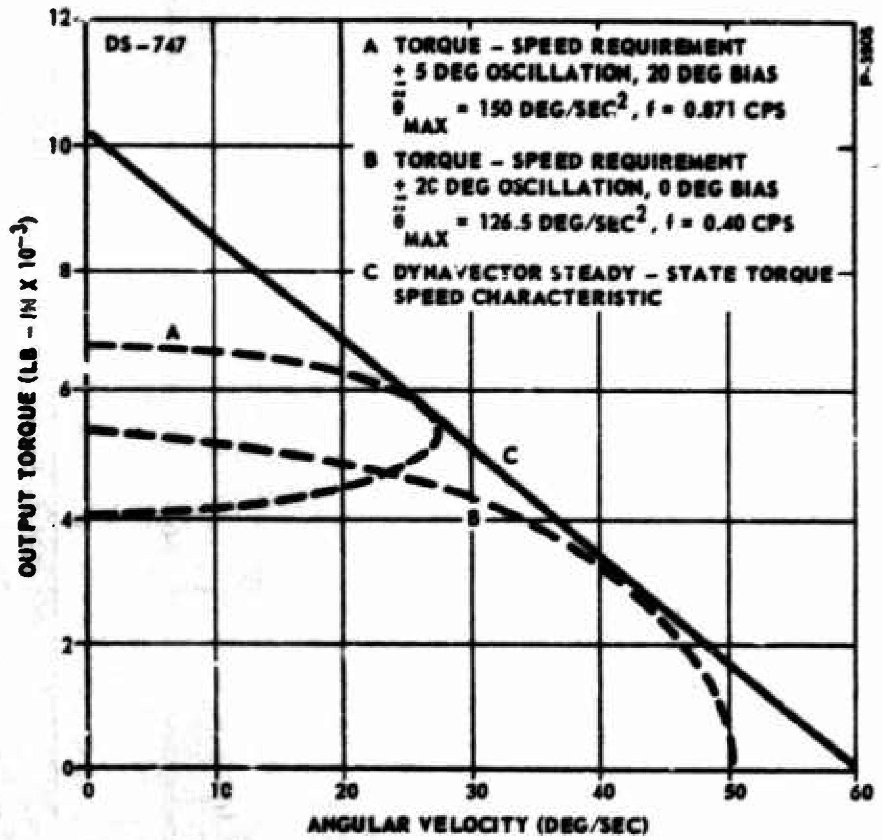

Figure A- Torque-Speed Capability Pneumatic DYNAVECTOR Actuator

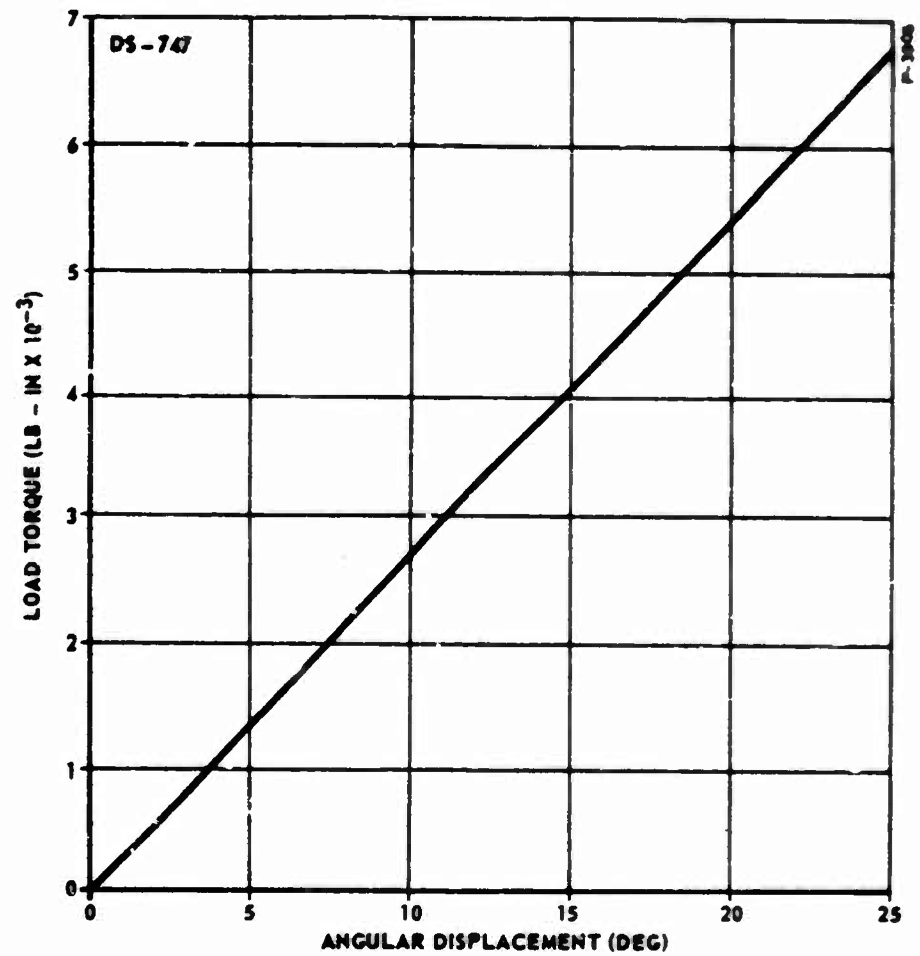

Figure B- Load Torque-Displacement Curve

PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-748	

ENGINEERING SPECIFICATION

TITLE	DATE
Manual Servovalve Design Specification	May 15, 1966

1.0 DESCRIPTION

This design specification presents the requirements for a manual servovalve for use in a pneumatic rudder control system described in Specification DS-747. The valve shall accept a command input by direct mechanical linkage to the pilot and a mechanical command from the stability augmentation/autopilot system.

2.0 DESIGN REQUIREMENTS

2.1 Material and Workmanship

2.1.1 Materials and workm. ship shall be stated in Specifications MIL-P-8564C, MIL-F-5518C, and MIL-E-54.0.

2.1.2 Metals

All metals used in construction shall be corrosion resistant and/or suitably protected to resist corrosion during the expected service and storage life of the unit. The use of dissimilar materials shall be avoided whenever practicable.

2.1.3 Weight of Materials

Materials of the lightest possible weight, consistent with the service and strength requirements, shall be used.

2.1.4 Non-Standard Material Approval

Approval for the use of non-standard parts and materials shall be in accordance with Specification MIL-P-8564C.

2.1.5 Strength

The following safety factors shall be applied to the stress analysis of the unit:

(a) Yield Strength = 1.3 x Design Yield Strength

(b) Ultimate Strength = 1.5 x Design Ultimate Strength

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2.2 Size

The unit shall conform to the dimensional envelope of drawing 2162309 and the design data noted thereon.

2.3 Installation

The manual servovalve shall be installed per drawing 2162309 and shall be adaptable to the corresponding power actuator. Installation must be performed in such a way as to meet the following requirements:

(a) The force applied by the pilot or automatic valve to the manual valve as seen by the manual valve to obtain 1.0 in/sec output velocity shall not exceed 0.5 pound.

(b) In manual mode, the manual servovalve shall be capable of controlling the power actuator output displacement at amplitudes up to ± 27 degrees. When responding to automatic mode commands the manual servovalve shall limit the power actuator output to an amplitude of ± 5 degrees.

(c) The manual valve shall be the controlling valve of the servomotor/actuator assembly.

2.4 Weight

The manual servovalve shall have a weight goal of 0.9 pound.

2.5 Instrumentation

The valve shall be compatible with suitable instrumentation to monitor valve spool position and input signal.

3.0 PERFORMANCE REQUIREMENTS

3.1 Environmental Operating Conditions

The manual servovalve shall be designed to operate under the following ground and/or flight conditions:

(a) Temperature - Gas temperature of 100°F to 450°F
Ambient temperature of -65°F to 270°F

(b) Working Fluid - The working fluid shall be air.

(c) Supply Pressure - 50 psig ± 5 psig.

(d) Altitude - Sea level to 50,000 feet.

(e) Flight Acceleration Loads - The unit shall be able to withstand, without failure, a 17.0g ultimate acceleration force in any direction and shall operate satisfactorily without malfunction under 12.0g load acceleration.

3.2 Input Characteristics

(a) Signal - Position Demand.

(b) Rated Input - ± 0.020 in. + ± 0.180 lost motion.

(c) Maximum Input Force - 1.0 ± 0.1 lbs at spool centerline.

3.3 Output Characteristics

(a) Blocked Port Pressure Gain - 10 psi per percent of rated input

(b) Maximum Blocked Port Pressure Differential - (min.) 48 psid at 50 psig supply pressure.

(c) Rated No Load Flow - 0.048 ± 0.004 lbs/sec at 50 psig supply pressure, (70°F gas temperature).

3.4 Input-Output Characteristics

(a) Hysteresis - ± 3 percent.

(b) Linearity - ± 5 percent.

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DH Kaska		RG Read

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PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-749	

ENGINEERING SPECIFICATION

TITLE	DATE
Automatic Servovalve Design Specification	May 15, 1966

1.0 DESCRIPTION

This design specification presents the requirements for an automatic servovalve for use in a pneumatic rudder control system described in Specification DS-747. The unit shall accept an input from a fluid interaction amplifier and shall be consistent with the requirements set forth in this specification.

2.0 DESIGN REQUIREMENTS

2.1 Material and Workmanship

2.1.1 Materials and workmanship shall be as stated in Specifications MIL-P-8564C, MIL-P-5518C, and MIL-E-5400.

2.1.2 Metals

All metals used in construction shall be corrosion resistant and/or suitably protected to resist corrosion during the expected service and storage life of the unit. The use of dissimilar materials shall be avoided whenever practicable.

2.1.3 Weight of Materials

Materials of the lightest possible weight, consistent with the service and strength requirements, shall be used.

2.1.4 Non-Standard Material Approval

Approval for the use of non-standard parts and materials shall be in accordance with Specification MIL-P-8564C.

2.1.5 Strength

The following safety factors shall be applied to the stress analysis of the unit:

(a) Yield Strength = 1.3 x Design Yield Strength

(b) Ultimate Strength = 1.5 x Design Ultimate Strength

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2.2 Size

The unit shall conform to the envelope shown on drawing 2162309 and the design data noted thereon.

2.3 Installation

The automatic servovalve shall be installed per drawing 2162309 and shall be compatible with the corresponding automatic actuator.

2.4 Weight

The automatic servovalve shall have a weight goal of 0.5 pound.

2.5 Instrumentation

The automatic servovalve shall be provided with suitable instrumentation to monitor the input signal to the valve and the valve spool position.

3.0 PERFORMANCE REQUIREMENTS

3.1 Environmental Operating Conditions

The automatic servovalve shall be designed to operate under the following ground and/or flight conditions:

(a) Temperature - Gas temperature of 100°F to 450°F.
Ambient temperature of -65°F to 270°F.

(b) Working Fluid - The working fluid shall be air.

(c) Supply Pressure Range - 50 psig ± 5 psig.

(d) Altitude - Sea Level to 50,000 feet.

(e) Flight Acceleration Loads - The unit shall be able to withstand, without failure, a 17.0g ultimate acceleration in any direction and shall operate satisfactorily without malfunction under a 12.0g load acceleration in any direction.

(f) Natural Frequency - The automatic servovalve shall have a natural frequency of 30 cps. minimum.

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2835-3110		11272	DS-749	

ENGINEERING SPECIFICATION

TITLE	DATE
Automatic Servovalve Design Specification	May 15, 1966

3.2 Input Characteristics

(a) Signal - Differential pressure ±5 psid

(b) Quiescent Pressure - 40 psia ± 5 psi

(c) Rated Flow - 0.0002 lbs/sec

3.3 Output Characteristics

(a) Blocked Part Pressure Gain - 10 psi per percent of rated input.

(b) Maximum Blocked Part Pressure Differential (min) - 40 psid.

(c) Rated No Load Flow - 0.0015 ± 0.0004 lbs/sec

3.4 Input - Output Characteristics

(a) Hysteresis - The valve shall have a hysteresis limit of 1 percent.

(b) Linearity - ±5 percent

(c) Frequency Response

(a) -3db Amplitude - 80 cps

(b) 90° Phase Shift - 50 cps

(d) Resolution - 1 percent rated input.

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PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-750	

ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Tooth Clutch Design Specification	May 15, 1966

1.0 Description

This design specification presents the requirements for a pneumatic tooth clutch for use in the pneumatic rudder control system described in Specification DS-747. The clutch will provide for disengagement of the pneumatic servo-mechanism when operation in the hydraulic mode is commanded and shall be consistent with the requirements set forth in this specification.

2.0 Design Requirements

2.1 Material and Workmanship

2.1.1 Materials and workmanship shall be as stated in Specifications MIL-P-8564C, MIL-P-5518C, and MIL-E-5400.

2.1.2 Metals

All metals used in construction shall be corrosion resistant and/or suitably protected to resist corrosion during the expected service and storage life of the unit. The use of dissimilar materials shall be avoided whenever practicable.

2.1.3 Weight of Materials

Materials of the lightest possible weight, consistent with the service and strength requirements, shall be used.

2.1.4 Non-Standard Material Approval

Approval for the use of non-standard parts and materials shall be in accordance with Specification MIL-P-8564C.

2.1.5 Strength

The following safety factors shall be applied to the stress analysis of the unit:

(a) Yield Strength = 1.3 x Design Yield Strength

(b) Ultimate Strength = 1.5 x Design Ultimate Strength

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D. H. Kerska		

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PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-750	

ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Tooth Clutch Design Specification	May 15, 1966

2.1.6 Bearings and Seals

Bearings and seals incorporated into the clutch design shall be capable of operation under the environmental conditions set forth in Section 3.1 of this specification.

2.2 Size and Weight

The unit shall conform to the dimensional envelope indicated on drawing 2162309 and the design data noted thereon and have a weight goal of 8 pounds.

2.3 Installation

The pneumatic tooth clutch shall be installed per the installation drawing 2161693.

2.4 Instrumentation

The pneumatic clutch shall be provided with instrumentation to indicate engagement and disengagement.

3.0 Performance Requirements

3.1 Environmental Operating Conditions

The clutch shall be designed to operate under the following ground and/or flight conditions:

(a) Temperature - Gas temperature of 100° to 450°F

(b) Supply Gas - Air

(c) Supply Pressure - 50 psig minimum

(d) Altitude - Sea level to 50,000 feet

(e) Flight Acceleration Loads

The unit shall be structurally able to withstand without failure a 17.0 g ultimate acceleration force in any direction and shall operate satisfactorily without malfunction under a 12.0 g acceleration force in any direction.

PREPARED BY	CHECKED BY	APPROVED BY
D. H. Kerska D. H. Kerska		R. G. Ross

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PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-750	

ENGINEERING SPECIFICATION

TITLE	DATE
Pneumatic Tooth Clutch Design Specification	May 15, 1966

3.2 Operational Requirements

The clutch shall incorporate design features such that:

(a) The pneumatic servomechanism can be disengaged when operation in the hydraulic mode is commanded.

(b) The pneumatic servomechanism can be disengaged when a hard-over signal is experienced by the pneumatic servo package in the automatic mode.

(c) Clutch engagement can occur only when the pneumatic servomechanism output position is in phase with rudder position.

(d) The clutch will disengage mechanically when loss of supply pressure occurs.

3.3 Torque Capacity - 10,500 in-lb

3.4 Power Capacity - 1.54 hp

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PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-751	

ENGINEERING SPECIFICATION

TITLE	DATE
Position Transducer Design Specification	May 15, 1966

1.0 Description

This design specification presents the requirements for a pneumatic position transducer for use in a pneumatic rudder control system described in Specification DS-747. The position transducer will be used in the automatic valve position control loop and shall conform to the requirements set forth in this specification.

2.0 Design Requirements

2.1 Material and Workmanship

2.1.1 Materials and workmanship shall be as stated in Specifications MIL-P-8564C, MIL-P-5518C, and MIL-E-5400.

2.1.2 Metals

All metals used in construction shall be corrosion resistant and/or suitably protected to resist corrosion during the expected service and storage life of the unit. The use of dissimilar materials shall be avoided whenever practicable.

2.1.3 Weight of Materials

Materials of the lightest possible weight, consistent with the service and strength requirements, shall be used.

2.1.4 Non-Standard Material Approval

Approval for the use of non-standard parts and materials shall be in accordance with Specification MIL-P-8564C.

2.1.5 Strength

The following safety factors shall be applied to the stress analysis of the unit:

(a) Yield Strength = 1.3 x Design Yield Strength

(b) Ultimate Strength = 1.5 x Design Ultimate Strength

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2.2 Dimension

The unit shall be of minimum size capable of performance per 3.2, 3.3, and 3.4.

2.3 Installation

The transducer shall be designed directly into the actuator housing and output shaft and does not add weight to the unit.

3.0 Performance Requirements

3.1 Environmental Operating Conditions

The position transducer shall be designed to operate under the following ground and/or flight conditions:

(a) Temperature - Gas temperature of 100° to 450°F

(b) Working Fluid - The working fluid shall be air.

(c) Supply Pressure - 50 psig ± 5 psig

(d) Altitude - Sea level to 50,000 feet.

(e) Flight Acceleration Loads - The unit shall be structurally able to withstand without failure a 17.0 g ultimate acceleration in any direction and shall operate satisfactorily without malfunction under a 12.0 g acceleration in any direction.

3.2 Input Characteristics

(a) Signal - Angular Position

(b) Rated Input - ± 5°

3.3 Output Characteristics

(a) Output Pressure - 20 psia ± 5 psi

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ENGINEERING SPECIFICATION

TITLE	DATE
Position Transducer Design Specification	May 15, 1966

3.3 Output Characteristics (Continued)

(b) Output Sensitivity - 1 psid ± .1 psi per degree of rotation

(c) Average Output Flow - 0.0001 lbs/sec

3.4 Input-Output Characteristics

(a) Linearity - ± 5%

(b) Resolution - 1%

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PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
2835-3110		11272	DS-752	

ENGINEERING SPECIFICATION

TITLE	DATE
Servo Amplifier Design Specification	May 15, 1966

1.0 DESCRIPTION

This design specification presents the requirements for a fluid interaction servo amplifier for use in a pneumatic rudder control system described in Specification DS-747. The amplifier is to consist of a proportional vortex summing amplifier with isolated inputs consisting of position, demand, and amplifier feedback signals and a jet type amplifier that raises amplifier output pressure level and allows it to be fed back for gain adjustment and possible servo compensation.

2.0 DESIGN REQUIREMENTS

2.1 Material and Workmanship

2.1.1 Materials and workmanship shall be as stated in Specifications MIL-P-8564C, MIL-P-5518C, and MIL-E-5400.

2.1.2 Metals

All metals used in construction shall be corrosion resistant and/or suitably protected to resist corrosion during the expected service and storage life of the unit. The use of dissimilar materials shall be avoided whenever practicable.

2.1.3 Weight of Materials

Materials of the lightest possible weight, consistent with the service and strength requirements, shall be used.

2.1.4 Non-Standard Material Approval

Approval for the use of non-standard parts and materials shall be in accordance with Specification MIL-P-8564C.

2.1.5 Strength

The following safety factors shall be applied to the stress analysis of the unit:

(a) Yield Strength = 1.3 x Design Yield Strength
(b) Ultimate Strength = 1.5 x Design Ultimate Strength

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PROJECT NO.	THE BENDIX CORPORATION RESEARCH LABORATORIES DIVISION SOUTHFIELD, MICHIGAN	CODE IDENT.	SPECIFICATION NO.	REV.
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ENGINEERING SPECIFICATION

TITLE	DATE
Servo Amplifier Design Specification	May 15, 1966

2.2 Size and Weight

The unit shall conform to an envelope dimension of 0.5 in square by 0.75 in and shall have a weight goal of 1.0 ounce.

3.0 PERFORMANCE REQUIREMENTS

3.1 Environmental Operating Conditions

The amplifier shall be designed to operate under the following ground and/or flight conditions:

(a) Temperature - Gas temperature of 100°F to 450°F
Ambient temperature of -65°F to 270°F

(b) Working Fluid - The working fluid shall be air.

3.2 Input Characteristics

(a) Input No. 1 -

(1) Input Signal - Differential Pressure

(2) Input Pressure - ± 2 psid

(3) Input Quiescent Pressure - 20 psia ± 5 psi

(4) Input Impedance Characteristics - The unit shall conform to the input characteristics of Figure 1.

(b) Input No. 2 -

(1) Input Signal - Differential Pressure

(2) Input Pressure - ± 2 psid

(3) Input Quiescent Pressure - 20 psia ± 5 psi

(4) Input Impedance Characteristics - The unit shall conform to the input characteristics of Figure 1.

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3.3 Output Characteristics

(a) Output Signal - Differential Pressure

(b) Output Pressure - ±5 psid minimum

(c) Output Quiescent Pressure - 40 psia ± 5 psi

(d) Output Impedance Characteristics - The unit shall conform to the output characteristics of Figure 2.

(e) Absolute Pressure Gain - 5 psi/psi ± 50 percent

3.4 Linearity

±5 percent.

3.5 Frequency-Response

The response of the unit shall be within the range indicated in Figure 3.

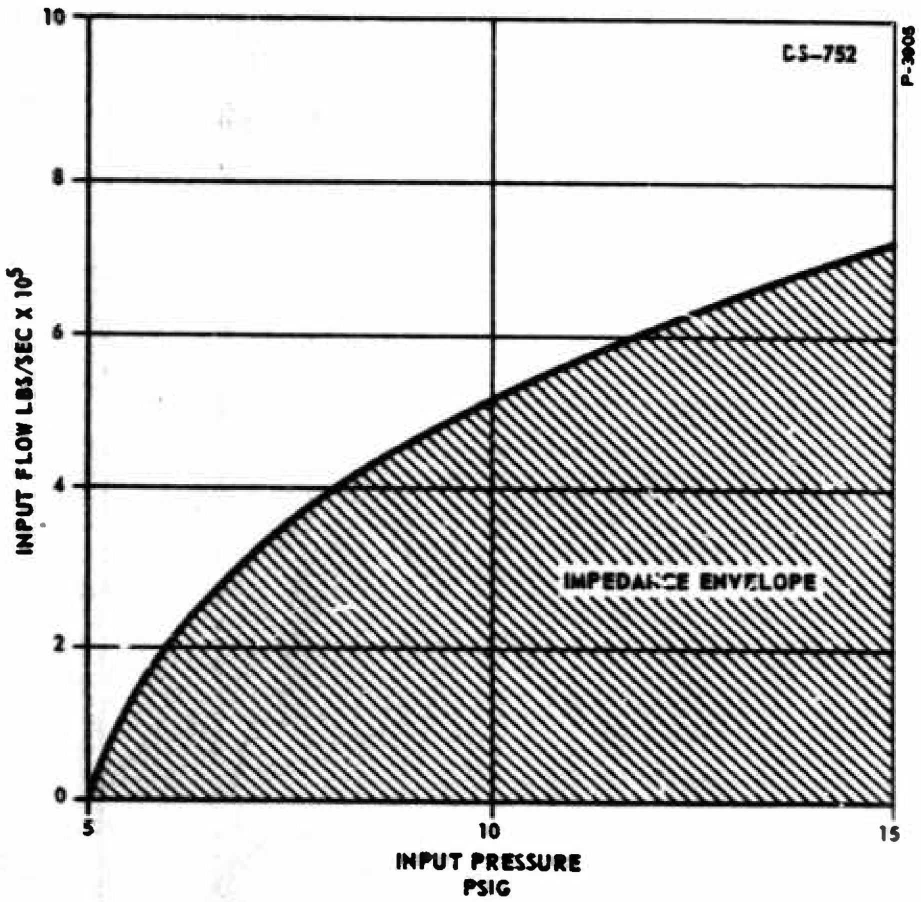

Figure 1 - Input No.s 1 and 2, Impedance Characteristics for One Side of Push-Pull Inputs

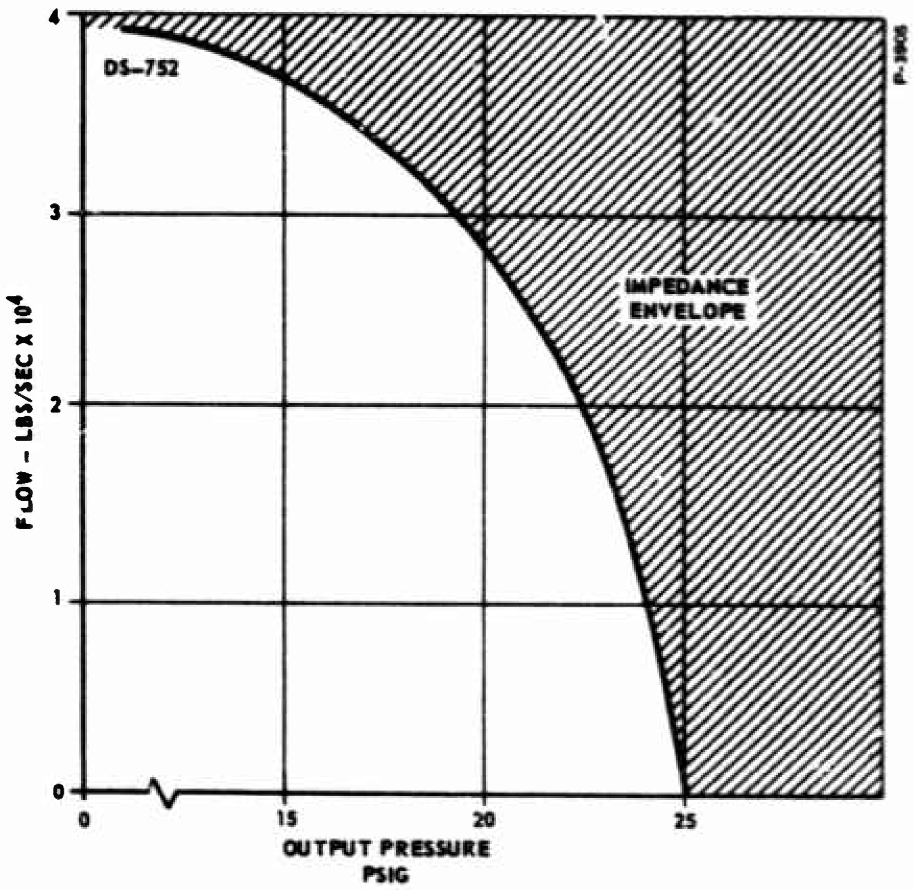

Figure 2 - Output Impedance Envelope for Zero Psid ± 0.5 Psi Input Signal

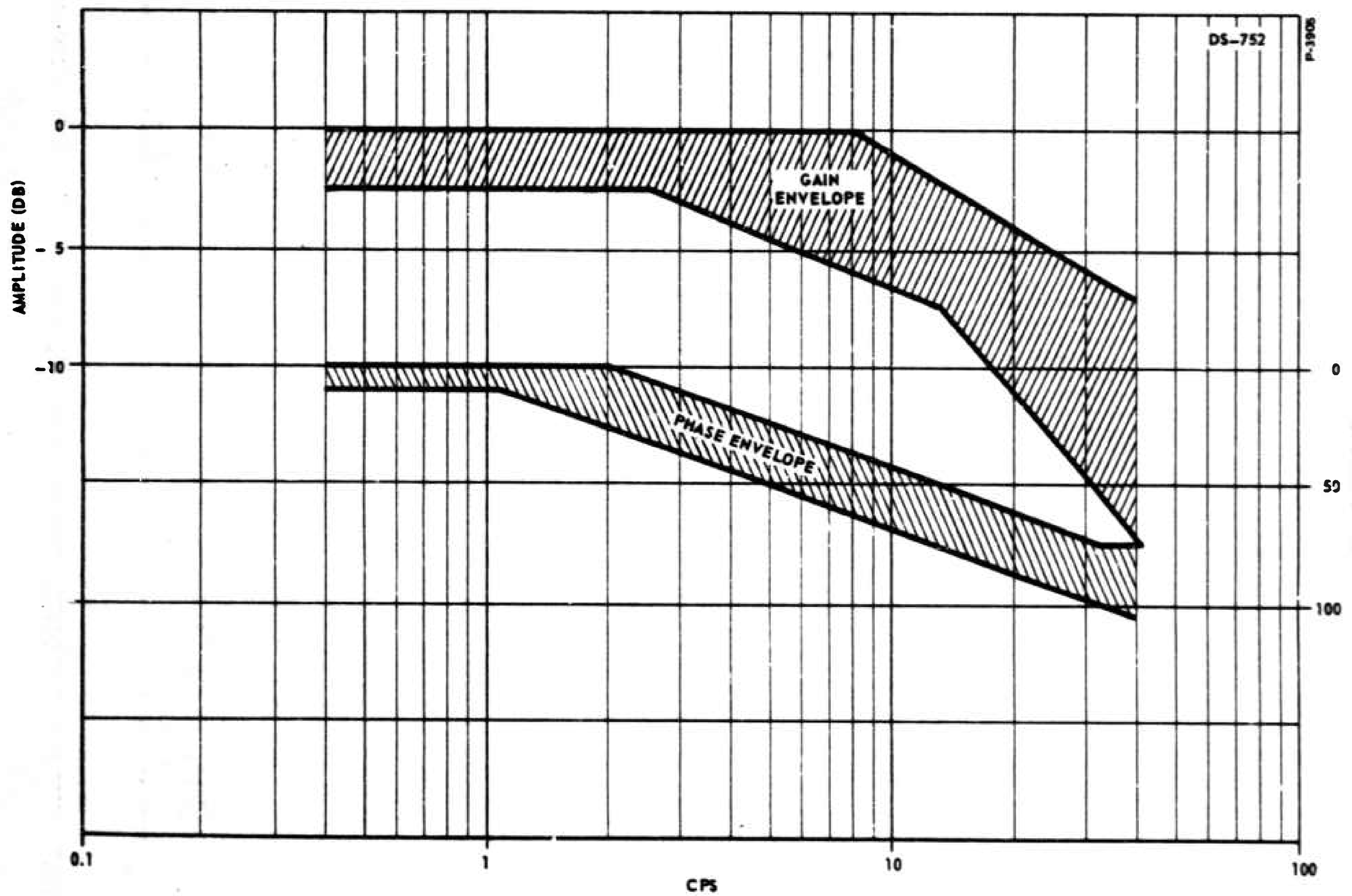

Figure 3 - Amplitude Frequency Response Reference Input ± 0.2 Psid

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ENGINEERING SPECIFICATION

TITLE	DATE
POWER SUPPLY REGULATOR DESIGN SPECIFICATION	May 15, 1966

1.0 DESCRIPTION

This design specification presents the requirements for a power supply regulator for use in a pneumatic rudder control system described in Specification DS-747. The regulator shall be designed to maintain a constant supply pressure to provide a constant maximum actuator output force.

2.0 DESIGN REQUIREMENTS

2.1 Material and Workmanship

2.1.1 Materials and workmanship shall be as stated in Specifications MIL-P-8564C, MIL-P-5518C, and MIL-E-5400.

2.1.2 Metals

All metals used in construction shall be corrosion resistant and/or suitably protected to resist corrosion during the expected service and storage life of the unit. The use of dissimilar materials shall be avoided whenever practicable.

2.1.3 Weight of Materials

Materials of the lightest possible weight, consistent with the service and strength requirements, shall be used.

2.1.4 Non-Standard Material Approval

Approval for the use of non-standard parts and materials shall be in accordance with Specification MIL-P-8564C.

2.1.5 Strength

The following safety factors shall be applied to the stress analysis of the unit:

(a) Yield Strength = 1.3 x Design Yield Strength

(b) Ultimate Strength = 1.5 x Design Ultimate Strength

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2.2 Size

The unit shall be of minimum size and weight consistent with aircraft design.

2.3 Installation

The regulator shall be installed per the installation drawing 2161698 and shall be capable of controlling supply pressure to the pneumatic servomechanism.

2.4 Instrumentation

Suitable instrumentation shall be provided to monitor regulator output.

3.0 PERFORMANCE REQUIREMENTS

3.1 Environmental Operating Conditions

The regulator shall be designed to operate under the following ground and/or flight conditions:

(a) Temperature - Gas temperature of 100° to $450^\circ F$
Ambient temperature of -65° to $270^\circ F$

(b) Working Fluid - The working fluid shall be air.

(c) Supply Pressure Range - 50 to 200 psig

(d) Ambient Pressure Range - 14.7 to 1.69 psia

(e) Flight Acceleration Loads - The unit shall be able to withstand, without failure, a 17.0 g ultimate acceleration force in any direction and shall operate satisfactorily without malfunction under a 12.0 g load in any direction.

3.2 Output Characteristics

(a) Regulated Output Pressure Range - 50 ± 5 psig
(b) Output Flow Range ($70^\circ F$) - 0.1 lbs/sec to 0.01 lbs/sec

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2835-3110		11272	DS-754	

ENGINEERING SPECIFICATION

TITLE	DATE
Linkage and Installation Design Specification	May 15, 1966

1.0 DESCRIPTION

This design specification presents the linkage and installation requirements for a pneumatic rudder control system described in Specification DS-747.

2.0 DESIGN REQUIREMENTS

2.1 Linkage Requirements

All linkage of the control unit and of the control unit to input and output members of the aircraft shall be consistent with the following requirements:

(a) Linkage members shall operate in an environmental temperature range of -65°F to 270°F.

(b) Materials used in linkages shall be capable of transmitting forces specific to each linking member without:

(1) undue stress or strain on linkage or system components.

(2) excessive friction or interference.

(c) All linkage systems shall be consistent with the drawing 2161698.

2.2 Installation Requirements

Installation shall be performed per the installation drawing 2161698. Care should be taken that all fastening and/or locking materials are functional at all environmental conditions of the unit.

The entire rudder control system shall be so designed as to allow for installation through the compartment doors, 54 and 56, shown on Macdonnell Construction Drawing 20-34204.

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ENGINEERING SPECIFICATION

TITLE	DATE
PRELIMINARY QUALIFICATION TEST PLAN: 10,200 In-Lb PNEUMATIC DYNAVECTOR ACTUATOR	May 15, 1966

1.0 PURPOSE OF TEST PLAN

This preliminary test plan defines the functional check-out tests and life endurance tests to be imposed on a rotary pneumatic 10,200 in-lb torque capacity DYNAVECTOR. The functional check-out tests will be conducted both prior to life testing and after actuator refurbishment at the conclusion of the life test before delivery.

2.0 LOG BOOK DOCUMENTATION

An equipment log shall be compiled for the DYNAVECTOR Actuator and shall contain the following items:

2.1 Title Page: Log Book Rotary Pneumatic DYNAVECTOR Model ______. Serial Number __________, BRLD Project _________, Wright-Patterson Air Force Base Contract ______________.

2.2 Table of Contents

2.3 Section I, Equipment Complement

Specify the following as applicable documents:

(a) DYNAVECTOR Assembly Drawing __________.
(b) Check-Out Test Fixture Schematic and Equipment Lists.
(c) Life Test Fixture Schematic and Equipment Lists.
(d) Preliminary Design Specification.

2.4 Section II, Inspection Reports

All inspection reports pertaining to the actuator assembly deliverable hardware are to be compiled in this section. If any part initially rejected by inspection is used in the actuator assembly, a copy of the report defining part disposition and statement of subsequent acceptance (rework, waiver, etc.) is to be incorporated in Section II.

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ENGINEERING SPECIFICATION

TITLE	DATE
PRELIMINARY QUALIFICATION TEST PLAN: 10,200 In-Lb PNEUMATIC DYNAVECTOR ACTUATOR	May 15, 1966

2.5 Section III, Calibration Data

State test equipment used and calibration history of said equipment.

2.6 Section IV, Functional Check-Out Test Data

A compilation of all check-out data prior to life tests and observations [illegible] st articles and test stand functioning shall be entered in this Section.

2.7 Section V, Life Test Data

A compilation of all life test data accumulated and observations of test articles and test stand functioning shall be entered in this Section.

2.8 Section VI, Refurbished Assembly Functional Check-Out Test Data

A compilation of all check-out data recorded on the refurbished unit prior to shipment to the customer and observations of said unit and test stand functioning shall be entered in this Section.

2.9 Section VII, Engineering Release Notices, Change Notices, and E.I. Forms

Copies of all drawing release notices, change notices and engineering instruction forms of deliverable hardware are to be included in this Section.

2.10 Section VIII, Functional Check-Out Operating Events

Record each time deliverable hardware is tested during check-out tests prior to life tests and define adjustments or modifications required to operate deliverable hardware and/or test equipment (repairs, rework, inspections and maintenance). Test data is to be recorded in Section IV.

2.11 Section IX, Life Test Operating Events

Record sequence of life tests and define adjustments or modifications required to operate deliverable hardware and/or test equipment (repairs, rework, inspection, and maintenance). Test data is to be recorded in Section V.

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2.12 Section X, Refurbished Assembly Functional Check-Out Ope ting Events

Record each time deliverable hardware is tested and define adjustments or modifications required to operate deliverable hardware and/or test equipment (repairs, rework, inspection, and maintenance). Test data is to be recorded in Section VI.

2.13 Section XI, Failure Reports

A copy of BRLD failure data report BC-RLD-47 shall be filed in this section in the event of catastrophic or degradation failure of deliverable hardware during any deliverable hardware tests.

3.0 TEST CONDITIONS

3.1 Test Media

Tests shall be performed with air or nitrogen except that burst pressure tests may be performed with a liquid.

3.2 Temperatures

All room temperature tests shall be conducted with the ambient and inlet test media temperatures between 70°F and 120°F. The temperature tests shall be conducted within the following temperature ranges:

Gas Temperature	100°F to 450°F
Ambient Temperature	-65°F to 275°F

3.3 Filtration

The test media shall be continuously filtered through a filter element which is equivalent to a 10-micron nominal standard filter element. The filter and element shall be satisfactory for the temperature range encountered and cleaned or changed regularly to avoid clogging.

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ENGINEERING SPECIFICATION

TITLE	DATE
PRELIMINARY QUALIFICATION TEST PLAN: 10,200 In-Lb PNEUMATIC DYNAVECTOR ACTUATOR	May 15, 1966

4.0 TEST REQUIREMENTS

All of the tests except the vibration and shock test will be conducted with a load test fixture capable of producing a spring-rate characteristic load of 270 in-lbs/degree with a stall torque capability of 10,700 in-lbs.

4.1 Functional Check-Out Tests

All tests are to be conducted under room temperature conditions unless specified otherwise.

4.1.1 Examination of Product

Each component shall be carefully examined to determine conformance to the requirements of MIL-P-8564C for design, weight, workmanship, marking, conformance to applicable drawings and for any visible defects.

4.1.2 Break-In Run

The break-in run shall be for a duration of one (1) hour minimum. Shaft torque loading may be constant or vary sinusoidally so as to impose a peak actuator pressure differential of 25% maximum required pressure differential to produce 10,200 in-lbs torque. Shaft speed may be constant at a minimum of 60 degrees per second or vary sinusoidally from zero to 60 degrees per second maximum.

After the break-in run, the actuator shall be disassembled and examined. If all parts are in acceptable condition, the actuator shall be reassembled and tests continued per 4.1.3. If working parts require replacement, the actuator shall be reassembled using the replacement parts, and the break-in run and subsequent disassembly, examination, and reassembly repeated.

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4.1.3 Proof Pressure Test

A proof pressure of 300 psi shall be applied after the servomechanism has been at a soak temperature of 275°F for a period of 72 hours for at least two (2) successive times and held two (2) minutes for each pressure application. The unit shall be operated in its normal function between applications of the test pressure with ambient temperature of 275°F and test media temperature of 450°F. There shall be no evidence of excessive external leakage, excessive distortion, or permanent set.

4.1.4 Extreme Temperature Functioning Tests

4.1.4.1 Low Temperature

The unit shall be connected to a 50 psig supply pressure and maintained at a temperature not warmer than -65°F for 4 hours after the temperature has stabilized at -65°F. After this period the servomechanism shall be actuated at least 10 times. Variation in torque-speed performance shall not exceed ± 10% of design value. Increase in temperature during the test owing to operation is permitted.

4.1.4.2 Intermediate Temperatures

Immediately following the low temperature test (4.1.4.1) the test arrangement shall be warmed rapidly to a temperature of 275°F.. While the temperature is being raised, the servomechanism shall be actuated at maximum increments of 70°F to determine satisfactory operation throughout the temperature range. These check tests shall be made without waiting for the temperature of the entire servomechanism to stabilize.

4.1.4.3 High Temperature Test

The temperature shall be maintained at 275°F for a length of time sufficient to allow all parts of the servomechanism to obtain the temperature. The servomechanism shall then be actuated at least ten (10) times at an ambient temperature of 275°F and test media temperature of 450°F. Variation in torque-speed performance shall not exceed ± 10% of the design value.

4.1.4.4 Reversion to Hydraulic Mode

The servomechanism shall be checked for satisfactory reversion to hydraulic operation upon command or loss of pneumatic supply pressure per DS-747 during the above room temperature, low temperature and high temperature tests. This test is to include actuation of the pneumatic tooth clutch.

4.1.5 Frequency Response Tests

Frequency response tests shall be conducted under no-load conditions (actuator decoupled from load mechanism) and against a torque load (actuator driving load mechanism spring rate and inertia about a zero deflection null).

During all tests the actuator output shaft speed shall be velocity limited to 60 degrees per second. Tests shall be conducted to establish the frequency to effect a 90 degree phase shift and amplitude decay of 3 db.

4.2 Endurance Tests

4.2.1 High Temperature Test

The servomechanism shall be subjected to sinusoidal operation in accordance with the following schedule (Table I) with ambient temperature at 275°F, test media temperature of 450°F, and a supply pressure of 50 psig. After conclusion of the above test, soak the unit at 275°F for two (2) hours. Pressure is to be maintained during the first hour and reduced to approximately zero psi for the second hour.

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ENGINEERING SPECIFICATION

TITLE PRELIMINARY QUALIFICATION TEST PLAN:
10,200 In-Lb PNEUMATIC DYNAVECTOR ACTUATOR

DATE May 15, 1966

TABLE I

Mode	Amplitude (degrees)	Frequency (cps)	Number of Cycles	Linear Load Variation (lb-in)
Manual	20	0.435	125,000	No Load
	20	0.435	2,500	0-4860
	10	0.615	7,500	0-2430
	10	0.615	7,500	0-1620
	5	0.871	25,000	0-810
Automatic	5	0.871	75,000	No Load
	2	1.37	175,000	No Load
	0.8	2.18	500,000	No Load
	5	0.871	2,500	4100-6700
	2	1.372	2,500	4900-5900

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4.2.2 Low Temperature and Rapid Warm-Up Test

Repeat the low temperature test specified in 4.1.4.1 at -65°F and the intermediate and high temperature tests specified in 4.1.4.2 and 4.1.4.3 at 70°F to 120°F.

4.2.3 High Temperature Test

Repeat the high temperature test as specified in 4.2.1 except increase the number of cycles by a factor of three (3). At the conclusion of this test, repeat the low temperature test (4.1.4.1). At the conclusion of this test, the unit shall operate satisfactorily and leakage shall not be excessive. The unit shall be disassembled and carefully inspected; there shall be no evidence of excessive wear in any part.

4.3 Vibration Test

The unit shall be attached to a rigid fixture capable of transmitting the vibration conditions specified herein. Attachment of the unit to the fixture shall be made through the service mounting which represents dynamically the most adverse service mounting possible. The unit shall be subjected to no-load operation including clutch actuation during the vibration test. Each resonant and cycling period shall be divided into two (2) parts; the first part being conducted at room temperature and the second part at an ambient temperature of 275°F and a test media temperature of 450°F. Tests shall be conducted under both the resonance and cycling conditions specified herein. The amplitude of applied vibration shall be monitored on the test fixture near the unit mounting points. At the end of the test, the unit shall be subjected to room temperature function test.

TABLE II

Vibration Test Schedule

(Times shown refer to one axis of vibration)

Number of resonances	0	1	2	3	4
Total vibration time at resonance*	-	30 Min	1 hr	1-1/2 hr	2 hr
Cycling time	3 hr	2-1/2 hr	2 hr	1-1/2 hr	1 hr

* 30 minutes at each resonance

4.3.1 Resonance Tests

Resonant modes of the unit shall be determined by varying the frequency of applied vibration slowly through the specified range at vibratory accelerations not exceeding those shown in Figure 1. Individual resonance surveys shall be conducted with vibration applied along each axis of any set of three mutually perpendicular axes of the unit. The unit shall be vibrated at the indicated resonant conditions for the periods shown in the Vibration Test Schedule (Table II) and with the applied double amplitudes of vibratory accelerations in Figure 1. These periods of vibration shall be accomplished with vibration applied along each of the three mutually perpendicular axes of vibration. When more than one resonance is encountered with vibration applied along any one axis, each resonance shall be sustained for the period shown in the applicable portion of the vibration test schedule. If more than four resonances are encountered with vibration applied along any one axis, the four most severe resonances shall be chosen for test.

4.3.2 Cycling Tests

The unit shall be vibrated under the cycling conditions specified herein for the applicable periods listed in the vibration schedule (Table II). The frequency shall be cycled between 5 and 500 cycles per second at an applied double amplitude of 0.036 inches or an applied acceleration of ± 10 g whichever is the lower value. The rate of change of frequency shall be logarithmic, and such that 15 minutes are required to proceed from 5 to 500 back to 5 cps. When there is no provision for logarithmic cycling, a linear rate of frequency change may be used.

4.4 Humidity Test

Moisture resistance shall be established by humidity test Procedure I of Specification MIL-E-5272. At the conclusion of this test, the unit shall operate normally through twenty-five (25) cycles.

4.5 Failure of Parts

If during the design integrity program (qualification testing) the test is terminated because of a part failure, the actuator shall be replaced or repaired using a redesigned part(s) or in the case of faulty material or workmanship the procuring authority may authorize the installation of a part(s) of the original design and the defect overcome.

The program shall be considered complete when all of the parts within the actuator have been completed without failure and the requirements of the program as specified in the applicable model specification. Should the actuator test be continued from the point of failure with repaired or replaced parts, subsequent failure of parts that have successfully completed the total endurance requirements will not be considered cause for rejection.

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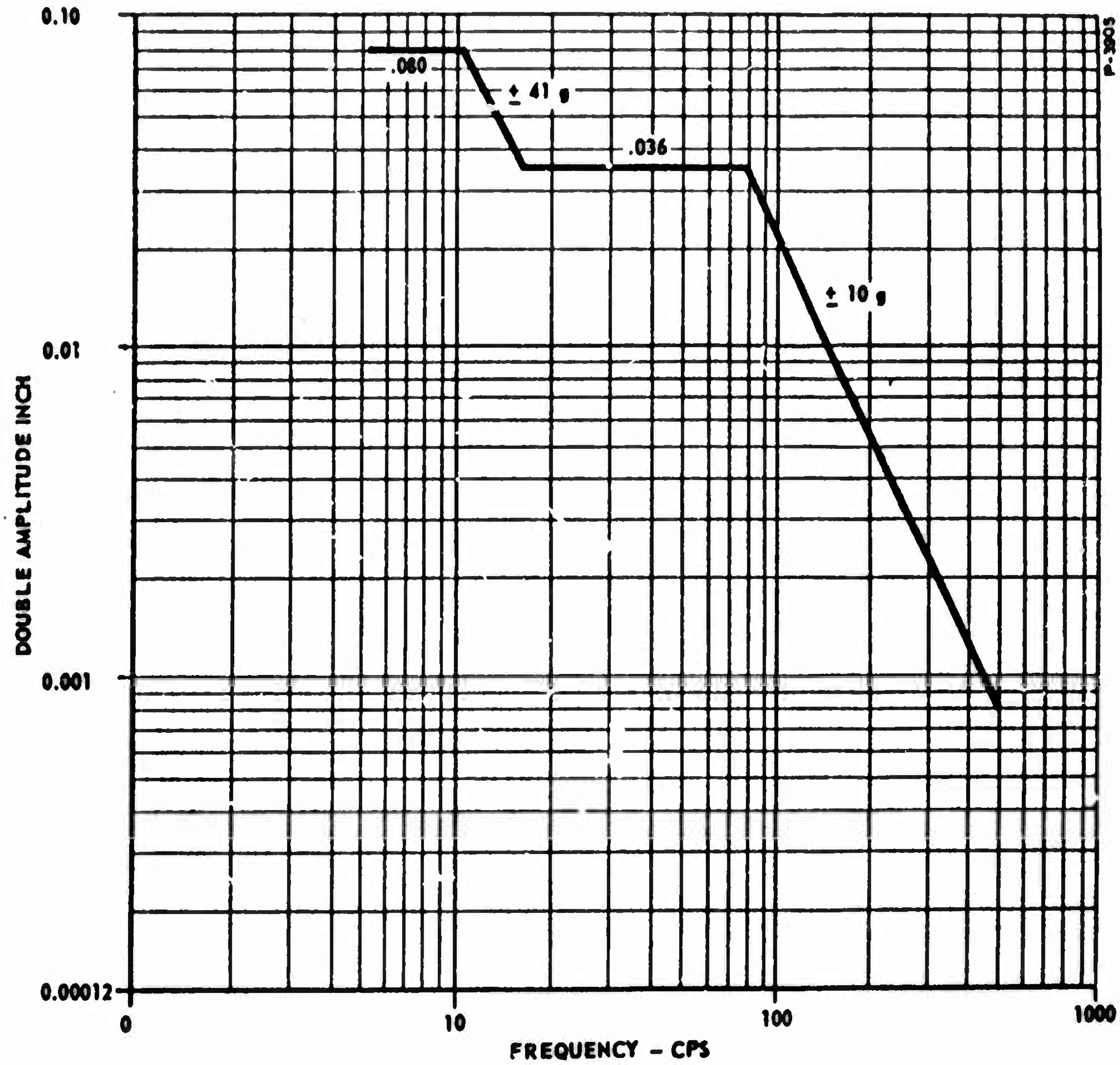
ENGINEERING SPECIFICATION

TITLE Preliminary Qualification Test Plan:
10,200 In-Lb Pneumatic Dynavector Actuator

DATE May 15, 1966

FIGURE 1 - RANGE CURVE FOR VIBRATION TEST



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APPENDIX C

FAILURE MODE ANALYSIS WORK SHEETS

FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Automatic Actuator*

*t = 1350 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY X 10^{-6}
Vanes (8)	Increase leakage, reduce performance	0.50	Wear	8	4.00
Vane Springs (8)	Individual failure - reduce actuator performance All fail-actuator inoperative	0.50	Fatique, bending	8	4.00
Bearing (1)	Wear-no effect. Seizure-actuator inoperative. Galling reduce performance slightly	1.00	Wear, galling, seizure	1	1.00
Transmission (Epicyclic Unbalanced)	Wear-increased backlash, increased leakage. Bending, shear - reduce actuator performance, actuator inoperative if large number of teeth fail	5.00	Wear, bending, shear	2 Pass	5.00
End Plates (2)	Increased leakage - reduced actuator performance	0.50	Wear	2	1.00
Output Arm	No output to actuator manual valve latch	0.50	Bending, shear	1	.50
				Total	15.50

BC/RLD 78

FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Power Actuator *

*t = 3000 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY X 10^{-6}
Bearings (2)	Galling-little effect, decrease efficiency slightly. Seizure-power actuator inoperative Compressive-little effect	1.00	Galling, seizure, compressive	2	2.00
Vane Springs (8)	Individual failure-reduce performance. Total failure-possible power actuator inoperative (no displacement chambers.)	0.50	Fatique, bending	8	4.00
Vanes (8)	No effect, slight possibility of cross-chamber leakage	0.50	Wear, bending	8	4.00
Thrust Bearings (4)	Wear-no effect. Seizure actuator inoperative	0.50	Wear, compressive, galling, seizure	4	2.00
Transmission	Wear-increase backlash. Bending, shear-effect output performance. Seizure - actuator inoperative	10.00	Bending, shear, galling wear.	3 Pass	10.00
End Plates (2)	Wear-increase leakage, reduce performance Galling-reduce performance. Seizure-actuator inoperative.	0.50	Wear, galling, seizure	2	1.00
				Total	23.00

BC/RLD 78

FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Pneumatic Clutch

$^{*}t$ = 3000 hours

$^{**}t$ = 5/6 hour

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY X 10^{-6}
Release Springs (8)**	Individual failure-no effect if in pressure disengage mode; no pressure disengage-no effect if coefficient of friction is < 0.175 or loading < stall.	0.30	Fatigue, Shearing	8	2.40
Release Plungers(8)**	Shear failure - possible malfunction of clutch because of loose parts. Wear failure - reduce release load - same effect as spring failure.	0.50	Bending, shear, wear	8	4.00
Wave Washer (1)**	Piston could move with acceleration or vibration in axial direction - possible accidental engage of clutch if unpressurized.	0.01	Bending	1	0.01
Sliding Spline (1)*	Bending failure remote, seizure or galling - clutch inoperative; wear-create backlash.	0.50	Wear, bending, galling, seizure	1	.50
Piston(1)**	Galling or seizure - clutch inoperative. Wear-excesive leakage-slow operation. Bending-yield center portion-clutch inoperative - either cannot engage or disengage if permanent set occurs.	1.00	Wear, bending, galling, seizure	1	1.00
Output Shaft*	Clutch inoperative if in shear. Bending failure remote housing support when loaded.	0.50	Bending, shear	1	0.50

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FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Pneumatic Clutch (Continued)

*t = 3000 hours
**t = 5/6 hour

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY $\times 10^{-6}$
Face Gear*	Bending or shear - possible clutch unable to engage if most of teeth fail; no effect if flow fail. Wear-excessive backlash seizure - lock clutch engaged.	2.00	Bending, shear, wear, seizure	1 Pass	2.00
Piston Seal**	Slow operation, if excesive leakage clutch unable to engage and stay engaged.	2.00	Leakage	1	2.00

BC/RLD 78

FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Actuator-Rudder Interlock Valve*

*t = 1500 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY x 10^{-6}
Actuator-Rudder Interlock Valve	No Effect	6.0	Wear, Leakage	1	6.00

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FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Load Limit Mechanism *

*t = 3000 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY x 10^{-6}
Negator springs (2) 0.003 x 0.250 spring steel	(1) Loss of linkage to manual valve	0.50	Fatigue	2	1.00
Ball Spline Saginaw No. 0375-3-0156	(2) More input force required to stroke linkage	2.00	Galling	1	2.00
	(3) Excessive Input force transmitted to manual valve		Seizure		
Lower Support Bearings (2)	(4) Same as item (2)		Galling		
	(5) Prevent force transmittal to manual valve	1.00	Seizure	2	2.00
Micro switch " x E" (Sealed)	Prevent Clutch Engagement	1.70	Loss of Signal	1	1.70
		6.20			6.70

BC/RLD 78

FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Automatic Servovalve Amplifier*

*t = 1350 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY $\times 10^{-6}$
Vortex Summation	Bias servoamplifier - incorrect signal to servovalve	2.00	Contamination	1.0	2.00
Venjet-Vortex Valve	Bias servoamplifier - incorrect signal to servovalve	2.00	Contamination	1.0	2.00
				Total	4.00

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FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Linkage lever and Manual Valve Assembly *

* t = 3000 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY $\times 10^{-6}$
Bearing-spherical rod end	(1) Increased input lever force required to stroke valve	0.50	Galling	1	.50
	(2) Valve spool will not respond to input command		Seizure		
Lever-Input Linkage Control	(3) Same as item (2)	0.50	Structural Failure	1	.50
	(4) Same as item (1)		Galling at Clutch Pivot		
	(5) Same as item (2)		Seizure at Clutch Pivot		
Lever Pivot Pin	(6) Same as item (1)	0.50	Galling	1	.50
	(7) Same as item (2)		Seizure		
Pins(2) Spool Driving	(8) Valve spool will function normally. Pin is retained	0.50	Shear-(One pin)	2	1.00
	(9) Same as item (2)		Shear-(Two pins)		
End Cap (2)	(10) Loss of supply fluid and/or pressure	0.250	Flexural Failure	2	.50
Seal "O" Ring (2) (End Cap)	(11) Same as item (10)	1.00	Hardening and/or extrusion	2	2.00

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FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Linkage Lever and Manual Valve Assembly* (Continued)

*t = 3000 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY X 10^{-6}
Spool	(12) Same as item (1)	8.00	Galling	1	8.00
	(13) Same as item (2)		Seizure		
Spool-Orifice Flow Areas (2)	(14) Leakage flow to/from activator	1.00	Erosion of metering edges	2	2.00
Sleeve/spool Sleeve/body fit	(15) Same as (10)		Excessive Clearances		
Valve Body	(16) Same as items (1) and (2)	1.00	Bore Deformation		1.00
Flex Pivot Bearing	(17) Same as item (2)	1.00	Fatigue		1.00
					17.00

SC/RLO '8

FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Pressure Regulator *

*t = 3000 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY $\times 10^{-6}$
Diaphragm	No control on output pressure	12.00	Rupture Leakage	1	12.00
Pintle	No output pressure control	6.00	Seizure	1	6.00
Spring	No output pressure	2.00	Fatique, Shear	1	2.00
					20.00

DC/RLD 70

FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Detent Control Cylinder Assembly*

*t = ?/6 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY $\times 10^{-6}$
Detent Control Cylinder	(1) Sliding piston will not stroke detent lever	2.00	Distortion	1	2.00
Detent Control Sliding Piston	(2) Increased control pressure required to stroke piston	4.00	Galling	1	4.00
	(3) Same as item (1)		Seizure		
Seals-Sliding "O" Ring (2)	(4) Same as item (2)	1.00	Hardening (Aging)	1	1.0
	(5) Loss of Fluid and/or control pressure		Extrusion		
		7.00			7.0

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FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Fluidic Position Transducer*

*t = 1350 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Mode	Assumed Failure Rate	QUANTITY	FAILURE PROBABILITY X 10^{-6}
Variable Area Duct in Automatic Activator Output Plate Pressure Tap Orifice in Power Actuator Output Member		Orifice Constriction	Total.01		.01

BC/RLD 74

FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Solenoid Valves

*t = 5/6 hours **t = 1/2 hour

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY $\times 10^{-6}$
Hydraulic Supply*	Open Coil-Lack of actuation	10.0	Valve in-Operative	3	30.0
Pneumatic Supply*	Hydraulic System still active				
Clutch Supply*					
Actuator-Manual Valve Latch**	Open coil-prevents Stability Augmentation	10.0	Valve in-operative	1	10.0

BC/RLD 78

FAILURE MODE AND EFFECTS ANALYSIS WORKSHEET

DWG. OR SK. REFERENCE NO.	ASSEMBLY NAME
2162309	Actuator Manual Valve Latch*

*t = 1350 hours

PART OR COMPONENT	EFFECT ON ASSEMBLY	Assumed Failure Rate	Assumed Failure Mode	QUANTITY	FAILURE PROBABILITY $\times 10^{-6}$
Body-Cylinder Bore (2)	(1) Sliding plunger will not engage/ disengage automatic actuator output.	1.00	Distortion	2	2.00
Spring (2)	(2) Manual Valve would not re-center when cover rotates	0.30	Fatigue (Loss of Load)	2	.60
Plunger-Sliding (2)	(3) Same as Item (1) & (2)		Seizure		
	(4) More pressure required to stroke plunger to engage/disengage automatic actuator output.	1.00	Galling	2	2.00
Seals-Sliding "O" Ring (4)	(5) Same as Item (4)	2.00	Hardening (Aging)		
	(6) Loss of Supply Fluid and/or Pressure		Extrusion	4	8.00
		4.30		10	12.60

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