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SUPPLEMENT 3

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SURVIVABLE FLIGHT CONTROL SYSTEM
INTERIM REPORT NO. 1
STUDIES, ANALYSES AND APPROACH

SUPPLEMENT FOR HYDRAULIC POWER AND ACTUATION STUDIES

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FOREWORD

This report was prepared by McDonnell Aircraft Company, St. Louis, Missouri, 63166, under Air Force Contract F33615-69-C-1827, PZ05, "Development and Flight Test Demonstration of a Survivable Flight Control System." This contracted effort comprises a major portion of development under the Air Force Systems Command Program No. 680J, "Survivable Flight Control System (SFCS)." The work was administered under the direction of the Air Force Flight Dynamics Laboratory, Wright-Patterson Air Force Base, Ohio, 45433, by Vernon R. Schmitt, Project Engineer.

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ABSTRACT

The Survivable Flight Control System (SFCS) Program is an advanced development program of which the principal objective is the development and flight test demonstration of an SFCS utilizing Fly-By-Wire and Integrated Actuator Package techniques. The studies and analyses conducted to date have sufficiently defined the system requirements to provide a definition of an approach to the implementation of the SFCS. The results of these studies and the definition of the approach are presented in the basic report. The details of the Control Criteria and Control Law Development studies are presented in report supplements 1 and 2, respectively.

The details of the Hydraulic Power and Actuation studies are reported in this supplement 3. The four major topics covered are:

- Survivable Stabilator Actuator Package (SSAP) Degree of Redundancy Trade Study. Nineteen configurations were examined, leading to selection of a full time duplex package with an aircraft central hydraulic system backing up each integrated hydraulic system. Two electric motors of approximately eight horsepower each will drive two soft cutoff pumps. The SSAP will have a dual tandem piston, and will be designed for a maximum steady state fluid temperature of 450°F.

- Hydraulic Power Supply Study. The SFCS requires addition of a fourth hydraulic system which will be based on the production F-4 APU.

- Fluid Selection Study. The SSAP and its backup central hydraulic systems cannot use MIL-H-5606 fluid, since it is limited to 275°F. MIL-H-83282, a new fire-resistant synthetic hydrocarbon fluid was selected. MIL-H-83282 is compatible with all the materials in the existing hydraulic systems of the F-4 test aircraft.

- Actuator Dynamic Analysis. The secondary actuator and the SSAP designs were analyzed to determine nuisance disconnect characteristics, stability, frequency response, and failure transients. A cross-element comparison technique has been selected for in-flight monitoring of both the quad-redundant electrohydraulic secondary actuators and the quad-redundant electromechanical secondary actuator of the SSAP.

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LIST OF ABBREVIATIONS AND SYMBOLS

ABBREVIATIONS:

ABS - Absolute

AC - Alternating Current

AFML - Air Force Materials Laboratory

AIR - Aerospace Information Report

Ampl - Amplifier

ANA - Army, Navy, Aeronautical

APU - Auxiliary Power Unit

ASTM - American Society for Testing & Material

Batt - Battery

Batt. Rly. - Battery Relay

BTU - British Thermal Units

CMD_1 - Command Signal to Element No.1

CMD_2 - Command Signal to Element No.2

CMD_3 - Command Signal to Element No.3

CMD_4 - Command Signal to Element No.4

CPS - Cycles Per Second

CSD - Constant Speed Drive

cis - Cubic Inches Per Second

cm - Centimeters

DC - Direct Current

DEMOD - Demodulator

DIFF - Differential Gear

db - Decibels

EDLC - Essential Direct Current Line Contactor

EGLC - Emergency Generator Line Contactor

E/H - Electrohydraulic

E/M - Electromechanical

Ess AC Bus - Essential Alternating Current Bus

Ess DC Bus - Essential Direct Current Bus

Ess T/R - Essential Transformer Rectifier

Ext. - Extend

FBW - Fly-By-Wire

ft - Foot

ft^2 - Square Feet

GCU - Ground Control Unit

Gen - Generator

gms - Grams

gpm, GPM - Gallons Per Minute

HDEG - Hydraulic Driven Electric Generator

HP - Horsepower

Hr - Hour

Hyd SOV - Hydraulic Shutoff Valve

Hz - Hertz

IAP - Integrated Actuator Package

ICAO - International Civil Aviation Organization

IFM - In-Flight Monitor

In - Inch

kg - Kilograms

l - Liters

Lbf - Pounds Force

Lbm - Pounds Mass

Lbs - Pounds

LH - Left Hand

LH AC Bus - Left Hand Alternating Current Bus

LH DC Bus - Left Hand Direct Current Bus

LH T/R - Left Hand Transformer Rectifier

LLC - Left Generator Line Contactor

LRU - Line Replaceable Unit

LVDT - Linear Variable Differential Transformer

Max - Maximum

MIM - Mechanical Isolation Mechanism

Min - Minutes

MOT - Motor

MP - Motor Pump

mm - Millimeters

P - Pressure

ΔP - Pressure Drop

PC - Power Control (Hydraulic System)

psi, PSI - Pounds Per Square Inch

R - Return

Ret - Retract

RH - Right Hand

RH AC Bus - Right Hand Alternating Current Bus

RH DC Bus - Right Hand Direct Current Bus

RH T/R - Right Hand Transformer Rectifier

RLC - Right Generator Line Contactor

RPM - Revolutions Per Minute

sec - Second In Time

SFCES - Survivable Flight Control Electronics Set

SFCS - Survivable Flight Control System

Sq - Square

SQRT - Square Root or $\sqrt{\ }$

SSAP - Survivable Stabilator Actuator Package

T - Temperature

Tach - Tachometer

TC - Bus Tie Contactor

T/R - Transformer Rectifier

UTIL - Utility Hydraulic System

VAC - Volts Alternating Current

VDC - Volts Direct Current

W - Weight

SYMBOLS:

A	Surface Actuator Piston Area
A_p	Actuator Active Piston Area
A_s	Surface Area of Integrated Actuator
A_1	Equal Area Piston Area, Retract Side
A_2	Equal Area Piston Area, Extend Side
A_3	Unequal Area Piston Area, Retract Side
A_4	Unequal Area Piston Area, Extend Side
B/P	Hydraulic By/Pass Valve
Br	Isothermal Tangent Bulk Modulus
B_s	Adiabatic Tangent Bulk Modulus
C_v	Specific Heat at Constant Volume
c	Velocity of Sound
D	Tube Diameter, Characteristic Dimension
d	Density of Fluid
E	MP_1, Input Signal
F	MP_2, Output Force
F_{AE}	Combined View Emissivity Factor
°F	Degrees Farenheit
Δ°F	Difference in Fluid Temperature
f	Actuator piston cycling frequency
G	MP_3
H	Feedback Gain, Normalized Feedback Gain, MP_4
HP_{aero}	Power Required to Move Aircraft Control Surface Load Against the Aerodynamic Load at That Flight Condition
HP_{aux}	Auxiliary Servo Pump Input

HP_{pi}	Pump Input Power
HP_{po}	Pump Output Power
h	Heat Transfer
hc	Convection Heat Transfer Coefficient
h_R	Radiation Heat Transfer Coefficient
h_T	Combined Heat Transfer Coefficient
I	PC-1
J	PC-2
K	Utility, A Generalized Constant, Normalized Amplifier Gain
k	Thermal Conductivity
L	Actuator Piston 1
L_{ni}	LVDT Null
M	Actuator Piston 2, Effective Mass of Linkage
Mi	Valve Null
N	Actuator Piston 3
N_a	Number of Active Elements Prior to Hardover Failure
N_R	Reynolds Number
O	Actuator Piston 4
P	Pressure, Differential Pressure
P_L	Pressure Differential
P_R	Return Pressure
P_r	Rated Pressure of Pump
P_S	Supply Pressure
P_s	Pressure Supplied by Pump
ΔP, Pi	Differential Pressure
ΔPs	Differential Pressure between Supply and Return

Q	Flow, Flow through MCV
Qr	Rated Flow of Pump
R	Gas Constant for Air, 53.35 (lb_f - ft/lbm - °R)
°R	Degrees Rankin
Si	Hydraulic Switching Valve
T	Temperature
Taw	Adiabatic Wall Temperature
Tc	Compartment Temperature °R
TP	Package Surface Temperature °R
t	Seconds
tc	Compartment Temperature °F, Package Surface Temperature °F
tr	Ram Air Temperature
UA	Overall Surface Conductance of Integrated Actuator Local Compartment
V, v	Velocity
W	Mass Flow Rate
X	Linear Amplitude
τ_{20V}	Time Constant: Servo Motor with 20 Volt Supply
$\tau_{28.3V}$	Time Constant: Servo Motor with 28.3 Volt Supply
θ_{M_1}	Angular Displacement: Element No. 1
θ_{M_2}	Angular Displacement: Element No. 2
θ_{M_3}	Angular Displacement: Element No. 3
θ_{M_4}	Angular Displacement: Element No. 4
δ	Actuator Attachment
δ_s	Stabilator Angular Displacement
ξ	Damping Ratio

Σ	Summation
ρ	Density
ω	Angular Rate
μ	Dynamic Viscosity
μ_s	Viscosity at the Tube Surface Temperature
ϕA, ϕB, ϕC	Three Phases of the Three-Phase Alternating Current Supply
λ	Failures per Million Flight Hours
f_{ACT}	Actuator Chamber

SECTION I

INTRODUCTION

The Survivable Flight Control System (SFCS) Program is a flight control advanced development program being conducted primarily by MCAIR under contract to the Air Force Flight Dynamics Laboratory (ADPO-680J). The principal objective of this program is the development and flight test demonstration on an F-4 aircraft of a Survivable Flight Control System utilizing Fly-By-Wire and Integrated Actuator Package techniques.

The SFCS program is being performed in two phases. Phase I has been completed and is described in Reference 1. The Phase II program and objectives are illustrated by Figure 1.

FBW control, as implied by its title, has no mechanical path from the pilot to the surface actuator. It incorporates aircraft motion sensors and provides for aircraft motion, rather than control surface position, to be the controlled variable. To be accepted by the aerospace industry as more than a research tool, the reliability of the FBW system must meet or exceed the reliability of the mechanical system it is replacing while showing advantages in other areas. The benefits foreseen for a FBW system may be summarized as:

- Enhanced Survivability
- Superior Aiming, Tracking, and Weapon Delivery

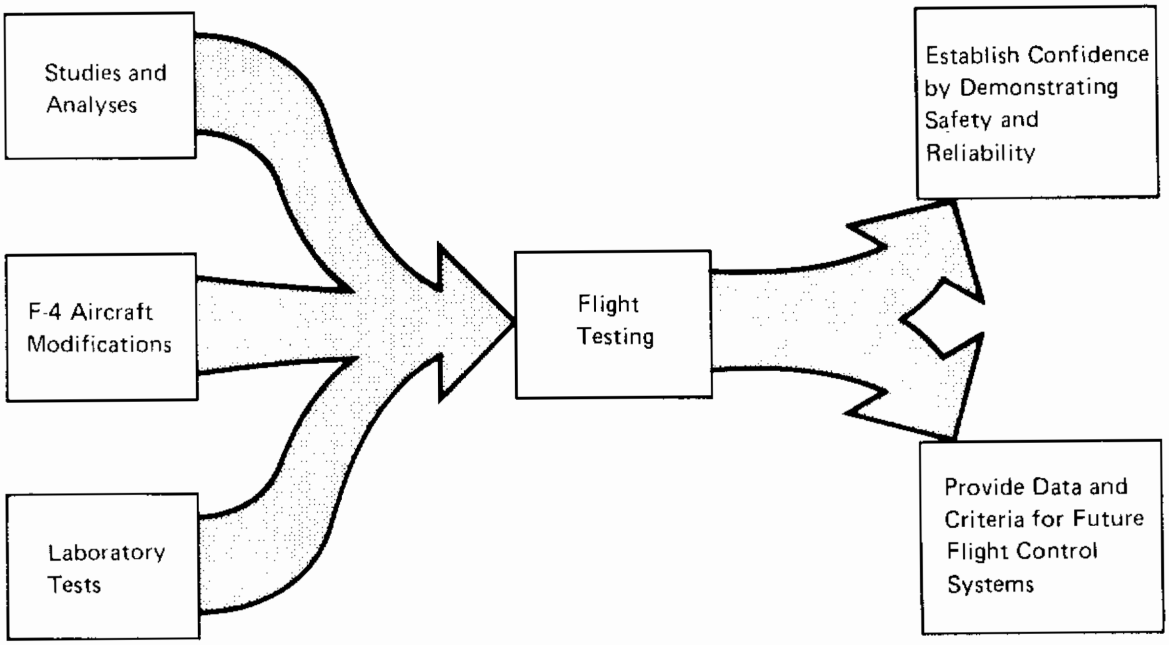

Studies and Analyses
F-4 Aircraft Modifications
Laboratory Tests
Flight Testing
Establish Confidence by Demonstrating Safety and Reliability
Provide Data and Criteria for Future Flight Control Systems

FIGURE 1

PHASE II - PROGRAM AND OBJECTIVES

F-4 WITH SURVIVABLE FLIGHT CONTROL SYSTEM

- Reduced Pilot Workload
- Flight Control Design and Installation Savings
- Decreased Cost of Ownership
- More Airframe Design Freedom

Having removed the mechanical control system from consideration, we now turn our attention to control surface actuation. Recent combat experience has shown that relatively minor damage, in the form of small arms fire, can result in aircraft loss due to loss of control. This is brought about by either hits in the hydraulic distribution system or by minor fires which in turn disrupt hydraulic system flow. By placing the hydraulic supply close to the surface actuator, we can reduce the probability of system damage due to minor combat damage. The ultimate in this concept is to integrate the hydraulic pump into the actuator package. If we also integrate the control device which converts FBW signals into flow commands for the main ram into the package we have an Integrated Actuator Package (IAP) using the concept of Power-By-Wire (PBW) to complement our FBW. A simplified form of IAP was flight tested as described in Reference 1.

Putting the two concepts of FBW and PBW together will result in a more survivable flight control system. The planned installation of the SFCS equipment in the F-4 test aircraft is shown in Figure 2.

The results of these studies and the definition of the approach are presented in the basic report. The details of the Hydraulic Power and Actuation studies are presented in this supplement. The details of the Control Law and Control Criteria studies are presented in additional supplements to this report.

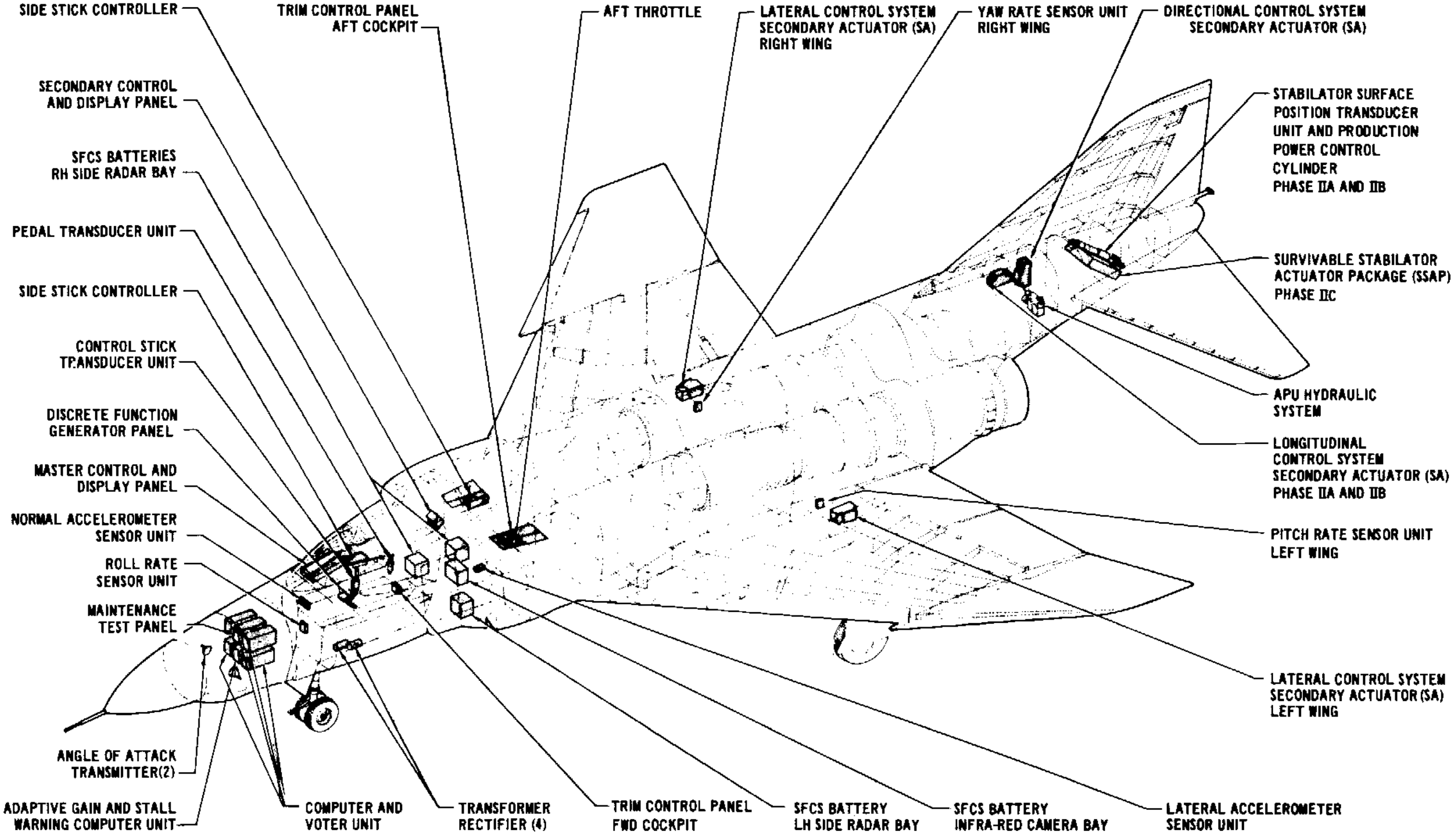

SIDE STICK CONTROLLER
TRIM CONTROL PANEL AFT COCKPIT
AFT THROTTLE
LATERAL CONTROL SYSTEM SECONDARY ACTUATOR (SA) RIGHT WING
YAW RATE SENSOR UNIT RIGHT WING
DIRECTIONAL CONTROL SYSTEM SECONDARY ACTUATOR (SA)
SECONDARY CONTROL AND DISPLAY PANEL
SFCS BATTERIES RH SIDE RADAR BAY
PEDAL TRANSDUCER UNIT
SIDE STICK CONTROLLER
CONTROL STICK TRANSDUCER UNIT
DISCRETE FUNCTION GENERATOR PANEL
MASTER CONTROL AND DISPLAY PANEL
NORMAL ACCELEROMETER SENSOR UNIT
ROLL RATE SENSOR UNIT
MAINTENANCE TEST PANEL
ANGLE OF ATTACK TRANSMITTER(2)
ADAPTIVE GAIN AND STALL WARNING COMPUTER UNIT
STABILATOR SURFACE POSITION TRANSDUCER UNIT AND PRODUCTION POWER CONTROL CYLINDER PHASE IIA AND IIB
SURVIVABLE STABILATOR ACTUATOR PACKAGE (SSAP) PHASE IIC
APU HYDRAULIC SYSTEM
LONGITUDINAL CONTROL SYSTEM SECONDARY ACTUATOR (SA) PHASE IIA AND IIB
PITCH RATE SENSOR UNIT LEFT WING
LATERAL CONTROL SYSTEM SECONDARY ACTUATOR (SA) LEFT WING
COMPUTER AND VOTER UNIT
TRANSFORMER RECTIFIER (4)
TRIM CONTROL PANEL FWD COCKPIT
SFCS BATTERY LH SIDE RADAR BAY
SFCS BATTERY INFRA-RED CAMERA BAY
LATERAL ACCELEROMETER SENSOR UNIT

FIGURE 2
SFCS EQUIPMENT LOCATION

SECTION II

SUMMARY

1. GENERAL

This supplement describes the studies which were made to:

- Determine the degree of redundancy for the Survivable Stabilator Actuator Package.
- Establish requirements of the hydraulic power supplies
- Analyse actuator dynamics
- Select the fluid to be used in SSAP and related aircraft hydraulic systems

In most cases data presented herein are submitted without reference to the source. Furthermore, the validity of data received from sources outside MCAIR has not been substantiated by test at MCAIR. Therefore, the designs, design techniques, and technical conclusions express MCAIR's best engineering judgments which consequently may or may not agree with designers, manufacturers or users of similar equipment.

2. SSAP DEGREE OF REDUNDANCY TRADE STUDY

The details of this study are presented in Section III of this report. A summary of the study is as follows:

a. The recommended level of redundancy for the SSAP is a full time duplex package with an aircraft central hydraulic system backing up each integrated hydraulic system.

b. The pump of the integrated hydraulic system should be of the soft cutoff type in order to meet the actuator stiffness requirements.

c. The package will be designed as a Class III Hydraulic System. It must be thermally improved to limit the fluid temperature to 450°F.

d. Only two sources of AC electrical power can be made available to the package. Each SSAP motor pump is driven from a separate generator.

e. The motor of the integrated hydraulic system cannot exceed 10 KVA, approximately 8 HP, due to limitations of starting current from the aircraft's main generators.

f. The main ram actuator must have dual tandem pistons without a tail rod to meet the structural constraints imposed by the test airplane.

3. HYDRAULIC POWER SUPPLY STUDY

The details of the study are presented in Section IV of this report. A summary of the study is as follows:

a. The production F-4 PC-1, PC-2, and Utility hydraulic systems have adequate flow and pressure capabilities to support the added hydraulic power requirements imposed by the SFCS secondary actuators and the MIM shift actuators.

b. A fourth hydraulic system is required.

c. The production F-4 APU will provide adequate flow and pressure for use in the fourth hydraulic system. In addition the reliability and thermal capabilities of the APU are adequate for the SFCS program.

d. MIL-H-83282, the high temperature fluid required in the PC-1 and PC-2 hydraulic systems for Phase IIC, will not affect the performance of present system components.

4. FLUID STUDY

The details of the study are presented in Section V of this report. Due to the high temperatures expected to be encountered in the SSAP, a Class III integrated hydraulic system is required. The production F-4, PC-1 and PC-2 hydraulic systems will be used as emergency back-up systems for the SSAP integrated hydraulic systems. Consequently, the fluid chosen must be compatible with both a Class III hydraulic system and the production F-4 hydraulic systems.

MIL-H-83282 fluid was selected to be used in the integrated systems of the SSAP and in the aircraft PC-1 and PC-2 systems. MIL-H-5606 fluid will be used in the aircraft Utility system and in the fourth hydraulic system. Consequently, the electrohydraulic secondary actuators will utilize both types of fluid.

5. ACTUATOR DYNAMIC ANALYSIS

The details of this analysis are presented in Section VI of this report. Analysis results are necessarily contingent on the present status of actuator design and the current definition of parameters. While no major problems were uncovered, it is expected that additional study and investigation will be required in specific areas. On the basis of present SSAP and secondary actuator configurations, the results of the analysis can be summarized as follows:

a. Secondary actuator nuisance disconnect characteristics are acceptable. However, a review of the analysis will be necessary when tolerances in the associated electronics are more fully defined. The status of SSAP nuisance disconnect characteristics is comparable.

b. A cross-element comparison monitoring technique has been selected for the SFCS IFM.

c. Both electrohydraulic and electromechanical quadruplex concepts are considered acceptable for the SFCS program and both will be used, the former on the secondary actuator and the latter in the SSAP.

d. The secondary actuator, SSAP surface actuator, and SSAP secondary actuator will be stable.

e. The electrohydraulic secondary actuator frequency response is adequate to permit proper SFCS operation.

f. The SSAP frequency response results indicate that the phase lag will be in excess of required limits.

g. Secondary actuator failure transients are acceptable. However, the effect of control linkage and surface actuator dynamic characteristics on failure transients should be investigated when the necessary information is available.

h. SSAP failure transients may be excessive on a third failure unless maximum rpm differences between servo motors can be reduced.

6. FUTURE STUDIES

The work reported herein represents a portion of the studies conducted to define the approach to mechanizing the hydraulic systems and actuators for the SFCS. As the hardware design and test progresses, the validity of some of the assumptions used herein will be demonstrated and empirical data will become available to define the hardware. The applicable portions of these and other SFCS studies will be updated and reported in future interim and/or final program reports.

SECTION III

SSAP DEGREE OF REDUNDANCY TRADE STUDY

1. GENERAL

A trade study was conducted specifically to determine the degree of redundancy to be specified for the Survivable Stabilator Actuator Package (SSAP) to be procured for use in the SFCS program. The results of this study indicated that the best redundancy choice for this program is a full time duplex package, i.e., a package with two integral motor-pump units, with an aircraft central hydraulic system back-up for each integral motor-pump unit. The configuration studies, method of analysis, and results of this trade study are presented herein.

In the initial stages of the study, both servo pump and soft cutoff pump concepts were considered. The servo pump concept was of special interest because of superior thermal characteristics. However, the capability of the servo pump concept to meet static and dynamic stiffness requirements comparable to those of the F-4 production actuator had not been demonstrated experimentally at the time this study was being conducted. Since that time, servo pump hardware has been built and tested, demonstrating significant potential for use in future integrated actuator package designs. See Reference 10 for details of this hardware development. Since working hardware was not available, the servo pump concept was considered too high a technical risk for the SFCS Program. The study effort, with the exception of the initial thermal evaluation of the servo pump concept, was based on the soft cutoff pump concept.

Thermal studies of the soft cutoff version of the various configurations indicate that these versions, if not equipped with specially tailored cooling provisions, will have a wide variation in operating temperatures, most of which are unacceptable. If designed to operate at a common fluid temperature, the soft cutoff versions will have heat transfer provisions of widely differing weights and capacities.

The study included the following types of redundancy:

- Full-time duplex with no back-up
- Full-time duplex with central hydraulic back-up
- Full-time triplex with no back-up
- Full-time triplex with central hydraulic back-up
- Full-time quadruplex with no back-up

The study also included redundancy concepts utilizing back-up through hydraulic switching valve arrangements, different combinations of actuator pistons and hydraulic systems, and a flow sharing concept.

2. BACKGROUND

The limitations of this trade study were established by ground rules and assumptions, the selection of functional and technical requirements, and a choice between pump concepts. Each of these categories is discussed in the following paragraphs.

a. Ground Rules and Assumptions

(1) A functional failure of a switching valve is such that neither of the incoming hydraulic sources can be used.

(2) An external seal loss, a switching valve leak, or a bypass valve leak cause the loss of all fluid from the channel in which the failure occurred and any back-up system for that channel.

(3) An open bypass valve failure connects the extend and retract chambers of the channel in which the failure occurred, preventing it from producing a force output.

(4) Aircraft AC electrical sources are assumed to be available as follows.

- AC Source 1 - connected to left hand AC bus
- AC Source 2 - connected to right hand AC bus
- AC Source 3 - hydraulic driven electric generator (Required only for MP_3)
- AC Source 4 - connected to left hand AC bus, switched to right hand bus if left hand bus fails (Required only for MP_4)

(5) Minimum dynamic stiffness after any two failures should be equivalent to that of the F-4 production actuator with one hydraulic system energized, 164,000 lbs/in.

(6) The minimum actuator force output after one failure should be 15,400 pounds; after any two failures 8,000 pounds.

(7) Motor pumps used as back-ups are off when no failure has occurred.

(8) One aircraft electrical bus cannot deliver sufficient power to start two motor pumps at over five horsepower each. One aircraft electrical bus can deliver sufficient power to...

- ...start one 8 horsepower motor pump as a maximum.
- ...sustain one operating 8 horsepower motor pump while starting one 5 horsepower motor pump as a maximum.

- ...start one 3 horsepower motor pump and start one 5 horsepower motor pump simultaneously as a maximum.
- ...sustain one operating 8 horsepower motor pump and one operating 5 horsepower motor pump, while starting one 5 horsepower motor pump.

(9) Hydraulic switching valves are automatic, one-way operating with reset capabilities, and are independent of relative hydraulic pressures in associated hydraulic systems.

(10) Electric motor driven pumps of 5 HP and up cannot be started on an HDEG (Hydraulic Driven Electric Generator) (See Paragraph 4b)

b. Functional and Technical Design Requirements

The functional and technical design requirements on which the trade study was based include:

- Performance (Dynamic stiffness and static force capability)
- Compatibility with aircraft electrical system
- Reliability
- Survivability
- Compatibility with two-fail-operate criteria
- Compatibility with central hydraulic systems
- Compatibility with the SFCS program schedule
- Maintainability
- Weight
- Steady State power requirements
- Envelope
- Safety
- Thermal considerations

c. Actuator Stall

Aerodynamic loads, for some flight and maneuver conditions, can exceed 8,000 pounds at the actuator output. The actuator output force after two failures must be 8,000 pounds minimum as stated in ground rule (6). Thus, actuator stall is a possibility. The potential stall condition is not considered an exception to the two-fail-operate criteria since degraded performance is permissible after two failures. The 8,000 pounds minimum value was chosen to provide flight safety for two nearly simultaneous supply failures at any flight condition.

As indicated above, aerodynamic loads, for some flight and maneuver conditions, will be of sufficient magnitude to stall the SSAP actuator for particular failure conditions. To avoid actuator back-off and loss of stiffness under stall loads, the installation of check valves in the SSAP actuators will be required. For actuators controlled by master control valves, the check valves are normally installed upstream of the master control valve in the pressure lines. Limiting the pressure between a check valve and master control valve by means of a pressure relief valve is not permissible, since it would defeat the purpose of the check valve.

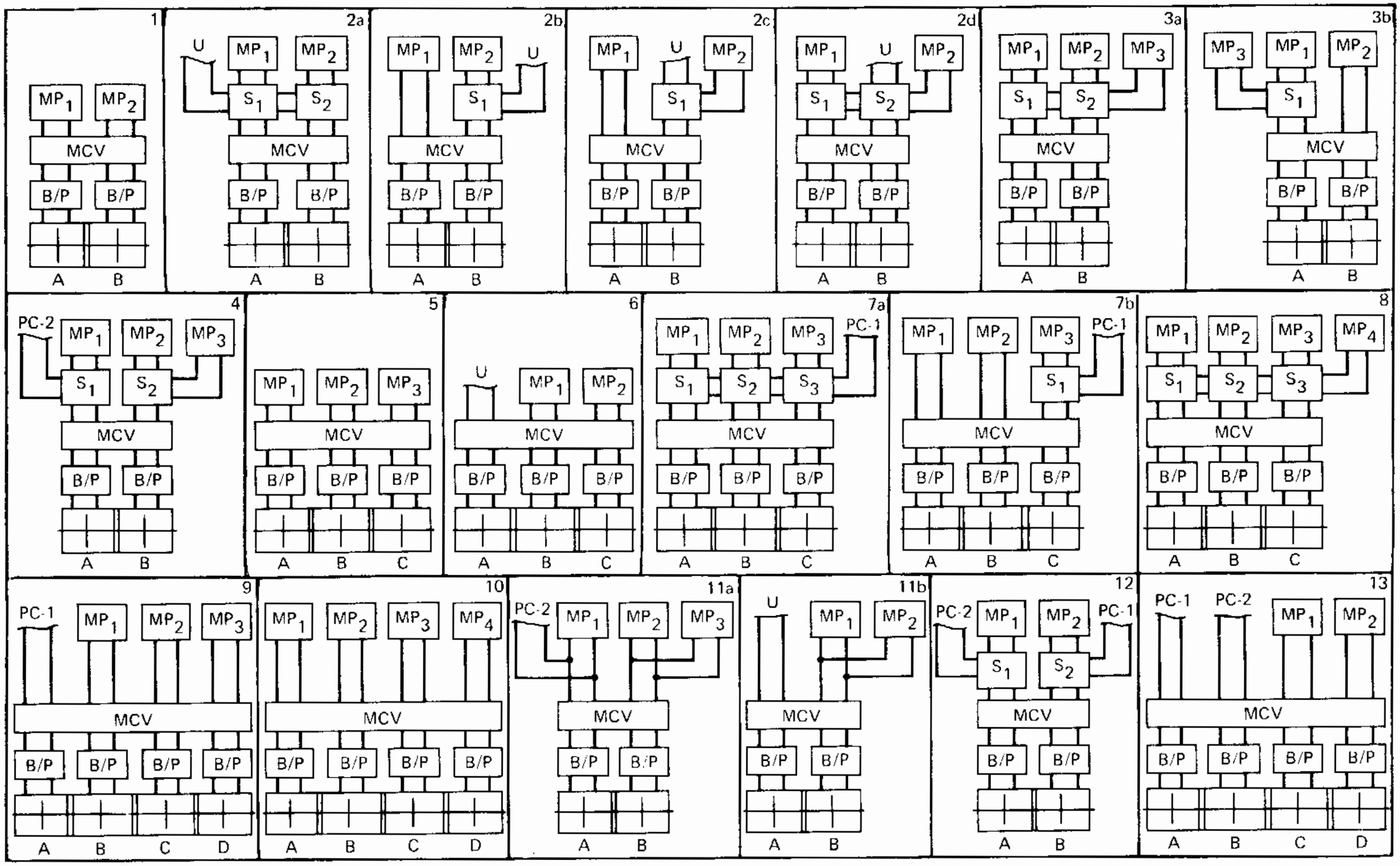
3. DISCUSSION

The initial tasks essential to the trade study included: the selection of configurations to be evaluated; establishing an aircraft electrical system consistent with the configurations being evaluated; and a brief description of the characteristics of each configuration. In providing a minimal description of the five original configurations, additional configurations appeared to be particularly adaptable to the F-4 aircraft, and these were added to the evaluation. It was also necessary to establish, for the purpose of the study, a third and fourth source of AC electrical power to accommodate the third and fourth motor pump featured in some configurations.

a. Selection of Configuration

The configurations selected for the trade study are those shown in Figure 3. All configurations shown are based on the soft cutoff pump concept equipped with master control valves. The configurations are categorized by number of integral motor pump units; e.g., a duplex actuator package has two motor pump units, whether they are used full time or in a back-up capacity. Where an actuator is described in terms of the number of channels it contains, the channels may be split into chambers for mechanization, with the characteristics given applying to the total of the individual chambers. For example, a dual channel actuator may have four chambers, two connected to each hydraulic supply.

In order to provide a basis for the evaluation of performance and electrical and thermal characteristics, piston areas were sized and pressures selected so that, for hydraulic source failures, the configuration would meet force and dynamic stiffness requirements. The hydraulic source failure considered aircraft hydraulic source failures as well as motor pump failures. The piston areas and pressures selected are shown in Table I.



A,B,C,D - Main Actuator Pistons
MCV - Master Control Valve
B/P - Bypass Valve
S - Switching Valve, Hydraulic

MP_1 - Motor Pump - L/H AC Bus
MP_2 - Motor Pump - R/H AC Bus
MP_3 - Motor Pump - Hydraulic (Utility) Driven Electric Generator
MP_4 - Motor Pump - L/H and R/H AC Buses

PC-1 - Central Hydraulic System (Primary 1)
PC-2 - Central Hydraulic System (Primary 2)
U - Central Hydraulic System (Utility)

FIGURE 3
SSAP CONFIGURATION SCHEMATICS

TABLE I
SSAP POWER REQUIREMENTS

Conf. No.	Piston Area (In.^{2})				Hydraulic Supply Pressure (PSI)							Electrical Power (HP)				Actuator Force Output (Minimum)	
	A	B	C	D	P_1	P_2	P_3	P_4	U	PC-1	PC-2	P_1	P_2	P_3	P_4	After 1 Source Failure	After 2 Source Failures
1	5.63	5.63			3,000	3,000						12	12			16,900	0
2-a	5.63	5.63			1,600	1,600			3,000			8	8			18,000	9,000
2-b	5.63	5.63			1,600	3,000			3,000			8	12			18,000	9,000
2-c	5.63	5.63			1,600	1,600			3,000			8	8			18,000	9,000
2-d	5.63	5.63			1,600	1,600			3,000			8	8			18,000	9,000
3-a	5.63	5.63			1,600	1,600	1,600					8	8	8		18,000	9,000
3-b	5.63	5.63			3,000	1,600	1,600					12	8	8		18,000	9,000
4	5.63	5.63			1,600	1,600	1,600				3,000	8	8	3		18,000	9,000
5	5.63	5.63	5.63		1,600	1,600	1,600					8	8	8		18,000	9,000
6	5.63	5.63	5.63		1,600	1,600			3,000			8	8			18,000	9,000
7-a	3.4	3.4	3.4		1,600	1,600	1,600			3,000		5	5	5		16,300	10,900
7-b	3.4	3.4	5.63		1,600	1,600	1,600			3,000		5	5	8		19,900	10,900
8	3.4	3.4	3.4	3.4	1,600	1,600	1,600	1,600				5	5	5	5	16,300	10,900
9	3.4	3.4	3.4	3.4	1,600	1,600	1,600			3,000		5	5	5		16,300	10,900
10	3.4	3.4	3.4	3.4	1,600	1,600	1,600	1,600				5	5	5	5	16,300	10,900
11-a	5.63	5.63			1,600	1,600	1,600				3,000	5	5	3		18,000	9,000
11-b	5.63	5.63			3,000	1,600			3,000			8	5			16,900	9,000
12	5.63	5.63			1,600	1,600				3,000	3,000	8	8			18,000	9,000
13	5.63	5.63	5.63	5.63	1,600	1,600				3,000	3,000	8	8			34,900	18,000

b. Aircraft Electrical System

For this study the electrical system schematic of Figure 4 shows the derivation of AC power for SSAP motor pumps and DC power for SFCES electronic equipment. AC power sources include the left hand bus, right hand bus, an HDEG, and a fourth source normally connected to the left hand bus which switches to the right hand bus when power on the left hand bus is lost. With the exception of the fourth AC source, motor pump loads are not interconnected between left and right hand buses. The HDEG and the fourth AC source are not available in the aircraft now, but were assumed for the purpose of the study.

c. Description of Configurations

(1) Configuration 1

Configuration 1 is a full-time duplex actuator package with no back-up. It consists of a dual channel actuator and two motor pump units, one for each channel of the actuator. Each motor pump produces 3,000 psi maximum pressure and 12.2 gpm maximum flow, each actuator piston has an area of 5.63 in^2. This combination is required to produce rated load after one failure.

(2) Configuration 2a

Configuration 2a is a full time duplex actuator package with a central hydraulic back-up. It consists of a dual channel actuator and two full time motor pump units, with two switching valves which can switch to the central hydraulic supply should either or both of the motor pump units fail. Each motor pump produces 1600 psi maximum pressure and 12.2 gpm maximum flow, and each actuator piston has an area of 5.63 in^2. After a source failure two systems remain in operation and thus the rated load can be produced with a lower pressure pump than with configuration 2b.

(3) Configuration 2b

Configuration 2b is the same as 2a except only one motor pump unit has a central hydraulic supply as back-up. The motor pump unit with the back-up produces 3,000 psi maximum pressure and 12.2 gpm maximum flow and the motor pump unit without the back-up produces 1600 psi maximum pressure and 12.2 gpm maximum flow. The difference in pressure is required to provide the full load requirements of the package when the motor pump unit without a back-up fails.

(4) Configuration 2c

Configuration 2c is a duplex actuator package which uses a central hydraulic supply full time. It consists of a dual

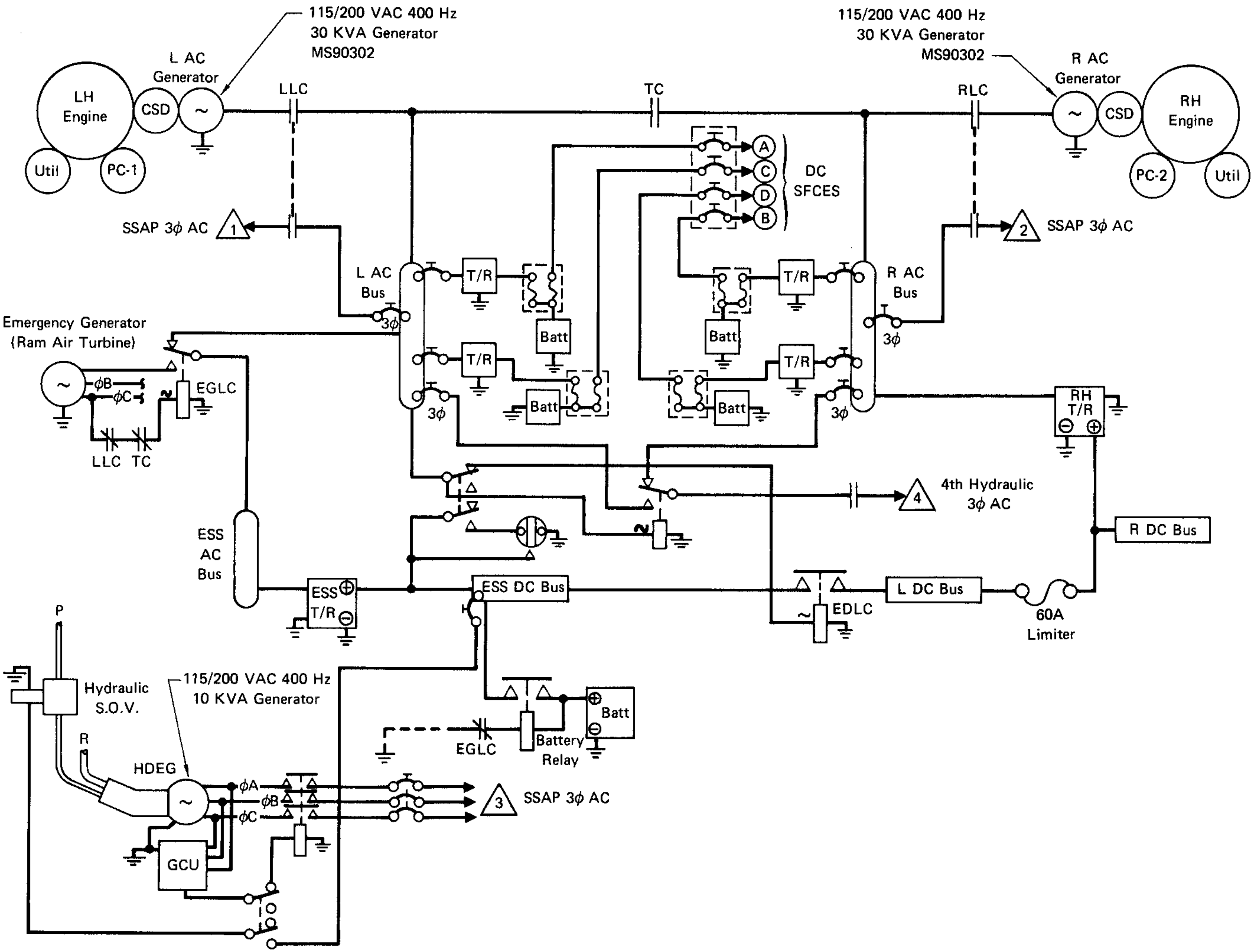


FIGURE 4
ELECTRICAL SCHEMATIC

channel actuator with one motor pump unit working into one channel and the central hydraulic supply working into the other channel. The second motor pump unit produces 1600 psi maximum pressure and 12.2 gpm maximum flow, and each piston has an area of 5.63 in^2. A hydraulic switching valve is used to switch the back-up supply on line and to switch a failed supply off.

(5) Configuration 2d

Configuration 2d is the same as 2c except one motor pump unit is used as a back-up for both the central hydraulic supply and the other motor pump unit. It is switched into the actuator channel whose supply fails first.

(6) Configuration 3a

Configuration 3a is a triplex actuator package. It consists of a dual channel actuator with two full-time motor pump units working into the actuator and a third motor pump used as a back-up for the other two units. Each motor pump unit produces 1600 psi maximum pressure and 12.2 maximum flow and each actuator piston has an area of 5.63 in^2. Hydraulic switching valves are used to switch the back-up supply on line and switch the first failed supply off.

(7) Configuration 3b

Configuration 3b is the same as 3a except the third motor pump unit is used to back-up only one of the active units. The motor pump unit with the back-up produces 3000 psi maximum pressure and 12.2 gpm maximum flow, the other two units produce 1600 psi maximum pressure and 12.2 gpm maximum flow. This difference in pressure is required to meet the load requirement in case the motor pump unit without a back-up is the first failure.

(8) Configuration 4

Configuration 4 is a triplex actuator package with central hydraulic back-up. It consists of a dual channel actuator with two full-time motor pump units working into the actuator. The third motor pump unit is used to back-up one of these units and the central hydraulic supply is used to back-up the other. Each of the full time motor pump units produces 1600 psi maximum pressure and 12.2 gpm maximum flow and the back-up motor pump unit produces 1600 psi maximum pressure and 4.8 gpm maximum flow. Each actuator piston has an area of 5.63 in^2. Hydraulic switching valves are used to switch a back-up supply on line and switch a failed supply off.

(9) Configuration 5

Configuration 5 is a full-time triplex actuator package. It consists of a triple channel actuator with three full-time motor pump units working into the actuator. Each motor pump unit produces 1600 psi maximum pressure and 12.2 gpm maximum flow, and each actuator piston has an area of 5.63 in^2. No hydraulic switching valves are used in this configuration.

(10) Configuration 6

Configuration 6 is a full-time duplex actuator package which uses a central hydraulic supply full-time. It consists of a triple channel actuator with two full-time motor pump units and a central hydraulic supply working into the actuator. Each motor pump unit produces 1600 psi maximum pressure and 12.2 gpm maximum flow, and each actuator piston has an area of 5.63 in^2. No hydraulic switching valves are used in this configuration.

(11) Configuration 7a

Configuration 7a is a full-time triplex actuator package with central hydraulic back-up. It consists of a triple channel actuator with three full-time motor pump units working into the actuator and a central hydraulic supply used as a back-up for all three units. Each motor pump unit produces 1600 psi maximum pressure and 7.4 gpm maximum flow, and each actuator piston has an area of 3.4 in^2. Hydraulic switching valves are used to switch the back-up supply on line and switch a failed supply off.

(12) Configuration 7b

Configuration 7b is the same as 7a except only one motor pump unit has the central hydraulic supply as a back-up. The motor pump unit with the back-up is working into a channel with a piston area of 5.63 in^2. This motor pump unit produces 1600 psi maximum pressure and 12.2 gpm maximum flow. The increase in piston area and flow is required to meet the stiffness and force requirements when the two motor pump units without a back-up fail.

(13) Configuration 8

Configuration 8 is a quadruplex actuator package. It consists of a triple channel actuator with three full-time motor pump units working into the actuator and a fourth motor pump unit serving as backup for the other three units. Each motor pump produces 1600 psi maximum pressure and 7.4 gpm maximum flow, and each actuator piston has an area of 3.4 in^2. Hydraulic switching valves are used to switch the back-up supply on line and switch the first failed supply off.

(14) Configuration 9

Configuration 9 is a full-time triplex actuator package which uses a central hydraulic supply full-time. It consists of a quadruple channel actuator with three full-time motor pump units and a central hydraulic supply working into the actuator. Each motor pump produces 1600 psi maximum pressure and 7.4 gpm maximum flow, and each actuator piston has an area of 3.4 in^2. No hydraulic switching valves are used in this configuration.

(15) Configuration 10

Configuration 10 is a full-time quadruplex actuator package. It consists of a quadruple channel actuator with four full-time motor pump units working into the actuator. Each motor pump unit produces 1600 psi maximum pressure and 7.4 gpm maximum flow, and each actuator piston has an area of 34. in^2. No hydraulic switching valves are used in this configuration.

(16) Configuration 11a

Configuration 11a is a triplex actuator package which uses a central hydraulic supply full-time in a flow sharing method. It consists of a dual channel actuator with three full-time motor pump units and a central hydraulic supply. Two of the motor pump units are working into one channel and the remaining motor pump and the central hydraulic supply are working into the other channel. The motor pump unit working with central hydraulics and one of the other units produce 1600 psi maximum pressure and 7.4 gpm maximum flow. The remaining motor pump unit produces 1600 psi maximum pressure and 4.8 gpm maximum flow. Each actuator piston has an area of 5.63 in^2. No hydraulic switching valves are used in this configuration, all hydraulic supplies operate full-time and share the flow requirements.

(17) Configuration 11b

Configuration 11b is a duplex actuator package which uses a central hydraulic supply full-time. It consists of a dual channel actuator with a central hydraulic supply working into one channel and two full-time motor pump units sharing the flow into the other channel. One of the motor pumps produces 3000 psi maximum pressure and 7.4 gpm maximum flow, the other motor pump produces 1600 psi maximum pressure and 7.4 gpm maximum flow. Each actuator piston has an area of 5.63 in^2. This combination is required to produce rated load if the central hydraulic supply is the first failure. No hydraulic switching valves are used in this configuration.

(18) Configuration 12

Configuration 12 is a full-time duplex actuator package which uses two central hydraulic supplies as back-up. It consists of a dual channel actuator and two full-time motor pump units with each unit backed up by a central hydraulic supply. Each motor pump unit produces 1600 psi maximum pressure and 12.2 gpm maximum flow, and each actuator piston has an area of 5.63 in^2. Hydraulic switching valves are used to switch a back-up supply on line and switch a failed supply off.

(19) Configuration 13

Configuration 13 is a full-time duplex actuator package which also uses two central hydraulic systems full-time. It consists of a quadruple channel actuator and a quadruplex master control valve. Two of the four channels are supplied by two full-time motor pump units while the remaining two channels are supplied by two central hydraulic systems. Each motor pump unit produces 1600 psi maximum pressure and 12.2 gpm maximum flow, and each of the four actuator pistons has an area of 5.62 in^2. Hydraulic switching valves are not required for this configuration.

4. RESULTS

The results of the trade study are summarized in Table II in a comparison matrix of design approaches. A brief discussion of each category and the basis of the ratings for that category is contained in the ensuing paragraphs.

a. Performance

This category is based on dynamic spring rate requirements and force output capability. The criteria used are consistent with what is required in the SSAP procurement specification, which are that after any two failures the dynamic stiffness shall be no less than that of the F-4 production stabilator actuator with one hydraulic system operating, 0.164×10^6 lb/in, and the force output capability shall be no less than 8,000 pounds. These are difficult requirements to meet since an engine failure accompanied by a generator failure can result in the loss of three hydraulic systems. Because of the critical nature of these requirements, all configurations which failed to meet them were rated unacceptable.

b. Compatibility With Aircraft Electrical System

The electrical portion of the trade study was based on two considerations:

- o an analysis of all failures relating to the electrical system components including all random failures and dependent failures which occur as a result thereof.

TABLE II
COMPARISON MATRIX OF SSAP DESIGN APPROACHES

	Wt Fac-tor	Configuration																		
		1	2a	2b	2c	2d	3a	3b	4	5	6	7a	7b	8	9	10	11a	11b	12	13
1) Performance	10	D	D	D	D	D	D	D	A	A	A	D	D	D	D	D	A	D	A	A
2) Compatibility with Aircraft Electrical System	10	D	A	D	A	A	D	D	A	D	A	D	D	D	D	D	A	A	A	A
3) Reliability	10	D	A	A	A	A	D	D	A	D	A	A	A	D	A	D	A	A	A	A
4) Envelope	10	—	—	—	—	—	—	—	A	—	B	—	—	—	—	—	A	—	A	C
5) Thermal Considerations	5	C	B	C	B	B	B	C	C	C	B	B	C	B	B	C	A	B	B	B
6) Survivability	5	C	C	C	C	C	C	B	A	B	B	A	A	A	A	A	A	B	B	A
7) Safety	3	C	C	C	C	C	C	C	C	C	C	C	C	C	C	C	C	C	A	A
8) Compatibility with Two-Fail-Operate Criteria	3	D	C	C	C	C	C	C	A	A	A	A	A	A	A	A	A	C	A	A
9) Compatibility with SFCS Program Schedule	3	A	A	A	A	A	B	B	B	B	B	B	B	C	B	C	B	A	A	A
10) Compatibility with Central Hydraulic Systems	2	A	C	C	C	C	A	A	C	A	B	C	C	A	B	A	C	B	C	B
11) Maintainability	2	B	B	B	B	B	B	B	B	C	B	C	C	C	C	C	C	C	B	C
12) Weight	1	B	B	B	B	B	C	C	C	C	C	C	C	C	C	C	B	A	B	C
13) Steady State Power Requirements	1	D	B	C	B	B	B	C	B	B	B	B	C	B	B	B	B	B	B	B
Total	65	D	D	D	D	D	D	D	168	D	159	D	D	D	D	D	176	D	177	161

Ratings:
A - Excellent, meets all requirements (3 points)
B - Satisfactory, involves only minor compromise (2 points)
C - Acceptable, involves significant compromises (1 point)
D - Unacceptable
— - Not rated

- compatibility with the aircraft power generating and distribution system capacities.

(1) Assumptions

The evaluation was based on the following assumptions:

(a) Loads on main buses of aircraft prior to addition of any SFCS equipment are equal to the basic RF-4C minus the loads associated with the forward looking radar (1.35 KVA) and the high frequency communication system (0.98 KVA). The forward looking radar and high frequency communication systems will not be installed on the SFCS aircraft. The basic load of 18.9 KVA minus 2.33 KVA is 16.57 KVA. This total load includes an essential bus load of 2.2 KVA which is to be increased by 10% due to possible SFCS requirements. The total AC load on the aircraft buses (prior to the addition of the Motor Pump packages and the DC conversion equipment and batteries) is assumed to be:

$16.57 + 0.1(2.2) = 16.57 + 0.22 = 16.79$ KVA
Total Load ≈ 16.8 KVA

(b) Based upon past experience with the F-4, partially Phase I data, the following duty cycle has been estimated as a suitable criterion to define operation of all motor pump packages:

full load - 20% of total time; Eff. = 0.82, PF = 0.78
3/4 load - 80% of total time; Eff. = 0.82, PF = 0.71
Starting KVA = 5 (full load KVA)

(c) The application of the above duty cycle to representative motor characteristics results in the following:

12 HP Motor: -

Average load = 12 KVA $\approx$ 86% of full load
Starting load $\approx$ 70 KVA

8 HP Motor: -

Average load = 8 KVA $\approx$ 86% of full load
Starting load = 46 KVA

5 HP Motor: -

Average load = 5 KVA $\approx$ 86% of full load
Starting load = 29 KVA

(d) The available off-the-shelf HDEG's include a unit which is composed of a hydraulic motor driving a 10 KVA generator. The 10 KVA generator has the usual overload capacities of 15 KVA (150%) for 2 minutes and 20 KVA (200%) for 5 seconds. However, the hydraulic motor in this package is not designed to drive the generator to these overload capacities. The Supplier of this unit feels that the unit could barely start an electric motor which imposed a starting (inrush) load of approximately 62 amperes per phase. This is the expected inrush current of a 3.25-3.50 HP motor. The unit, therefore, has insufficient capacity to start a 5, 8, or 12 horsepower motor. Any of the candidate configurations using a HDEG are, therefore, considered to be incompatible with the aircraft electrical system with the exception of Configurations 4 and 11a.

(e) Any of the candidate configurations which, under motor pump starting conditions, cause the total load on either of the main generators to exceed 60 KVA are considered to be incompatible with the aircraft electrical system. The aircraft main generators are rated at 60 KVA for 5 seconds and, during single generator operation, the application of loads exceeding this value could conceivably cause the loss of the generator. The loss of the second generator would result in a total loss of AC power.

(f) The DC sources for the SFCES are assumed to represent an average load of 2.76 KVA to the aircraft AC buses.

(2) Analysis

The candidate configurations were subjected to two separate analyses:

(a) Truth charts were made to establish the various failure modes of each configuration. These truth charts were used to determine the degree of redundancy of each candidate configuration. Causes for rejection of a particular configuration were based on:

- o Failure of SSAP caused by two or less random electrical failures. Exception: A double engine failure requires control system recovery actions by the pilot which are identical to those required in any F-4 aircraft under the same failure modes.

(b) Each candidate configuration was subjected to a power capacity and loading analysis to determine the compatibility of the system with the aircraft electrical generating and distribution system. Causes for rejection of a particular configuration were based on:

- o Use of HDEG driven motor greater than 3 horsepower
- o Inrush load exceeding 60 KVA

Configurations 2a, 2c, 2d, 4, 6, 11a, 11b, 12, and 13 appear to be satisfactory SSAP candidates based on the aircraft electrical system compatibility criteria.

c. Reliability

In evaluating reliability, each configuration was analyzed to determine the number of failures per million flight hours (λ = failures per million flight hours) for that configuration. The resulting failure rate includes all electronic, electrical, hydraulic, mechanical and engine failures which could cause aircraft loss due to loss of pitch axis control. The failure rate data used in the analysis, except for the master control valve which is calculated, were derived primarily from F-4 experience and supplier proposal reliability estimates and are shown in Table III. This calculated requirement was applied to any failure, or combination of failures, which would result in a catastrophic failure of the aircraft. MCAIR failure data do not identify a MCV failure in approximately 3.25 million F-4 flying hours.

TABLE III
EQUIPMENT FAILURE RATES

	Failures per 10^6 Hours
Electronics (Per Channel, Except Servo Amps)	275.0
Servo Amps	15.0
Aircraft AC Electrical Supplies (Single)	115.0
(Dual)	14.0
Utility Hydraulics and HDEG	160.0
Battery	70.0
Secondary Actuator (Per Channel)	250.0
Engine Flameout – No Restart	30.0
Electric Motor	2.5
Hydraulic Pump	20.0
Motor-Pump-Reservoir, etc.	25.0
Master-Control Valve and Linkages	0.05
Cylinder Sections (Cracks, Leaks)	7.0
Switching Valve (Pressure Loss)	3.5
(No Switch)	2.0
Primary Hydraulics	30.0
Utility Hydraulics	143.0

The ratings used to assess relative reliability were based on the reliability goal of 0.23 failures per million flight hours for the entire SFCS. An allocation of 0.116 failures per million flight hours (half the SFCS failure rate) to the pitch axis is allowable since the lateral and directional axes are redundant and only one must function to return and land the test aircraft. Moreover, if each axis had an inherent failure rate of 0.23 failures per million flight hours, the failure rate for the entire SFCS would be:

$$(2.3 \times 10^{-7}) + (2.3 \times 10^{-7})(2.3 \times 10^{-7}) =$$

$$(2.3 \times 10^{-7}) + (5.29 \times 10^{-14}) = 0.23 \times 10^{-6}$$

Therefore, a pitch axis failure rate of 0.20 failures per million flight hours can be considered acceptable.

Thus, the ratings used to assess relative reliability were:

A - Excellent, meets all requirements ($\lambda < 0.116$)
B - Satisfactory, involves minor compromise ($0.116 < \lambda < 0.160$)
C - Acceptable, involves significant compromise ($0.160 < \lambda < 0.200$)
D - Unacceptable ($\lambda > 0.200$)

The trade study reliability calculations were based on all failures and failure combinations which cause loss of pitch axis control. This is illustrated in Table IV for Configuration (1) which shows four failure combinations which result in the loss of pitch axis control. The highest failure rate of the four failure combinations is the 14 failures per million flight hours for the dual electrical supply. This failure alone would result in an unacceptable rating. The failure rate which results from the other failure combinations are relatively low.

TABLE IV
CONFIGURATION 1 TOTAL FAILURES

Failure or Failure Combination	Failure Rate x 10^6
(1) MCV	0.05
(2) Dual Electrical Supply	14.00
(3) Single Electrical Supply or Engine in Combination with MPI, MP2, Cylinder Section A or Cylinder Section B. (115 + 30) (25 + 25 + 7 + 7) = (145) (64) =	0.00928
(4) MPI or Cylinder Section A in Combination with MP2 or Cylinder Section B. (25 + 7) (25 +7) = (32) (32) =	0.001025
Total Failure Rate Per Million Hours	14.060304

The results of the reliability evaluation are shown in Table V with most of the configurations having approximate failure rates of either 14 or 0.05 failures per million flight hours. Those configurations with a failure rate of 14 failures per million flight hours would be completely disabled by a dual aircraft electrical supply loss. The remaining combinations show a failure rate per million flight hours of 0.05, which is the minimum value assigned to the MCV, or a value slightly higher which reflects the effect of additional failure combinations.

TABLE V
CONFIGURATION FAILURE RATES

Configuration	Losses/10^6 Hours
1	14.060
2a	0.056
2b	0.053
2c	0.053
2d	0.054
3a	14.054
3b	14.052
4	0.051
5	14.050
6	0.051
7a	0.051
7b	0.051
8	14.050
9	0.050
10	14.050
11a	0.053
11b	0.059
12	0.051
13	0.050

d. Survivability

In evaluating the survivability of each configuration considered in the trade study, the number of killing hit combinations were enumerated and compared to the total hit combinations possible. This was done for discrete numbers of hits on the system; specifically one, two, and three hits. Seventeen components were considered in these combinations, the components being:

(1) L/H Engine

(2) R/H Engine

(3) L/H Generator

(4) R/H Generator

(5) PC-1 Hydraulics

(6) PC-2 Hydraulics

(7) Utility Hydraulics

(8) HDEG

(9) Motor Pump No.1	(13) Actuator Chamber No.1
(10) Motor Pump No.2	(14) Actuator Chamber No.2
(11) Motor Pump No.3	(15) Actuator Chamber No.3
(12) Motor Pump No.4	(16) Actuator Chamber No.4
	(17) Actuator Attachment

The particular configuration determines which of these components are considered concurrently. A hit on any component will kill or disable that component. Multiple hits on components were considered possible, but the first hit caused component kill. Configurations with fewer killing hit combinations, in proportion to the total hit combinations possible, were considered superior. The survivability ratings of the trade study configurations are included on the Comparison Matrix of Design Approaches, Table II. Further survivability data are presented in Appendix I.

e. Compatibility With Two-Fail Operate Criteria

This requirement is based on a failure effects analysis which includes single and dual failures. In some instances a single failure results in the loss of more than one hydraulic source and a dual failure results in the loss of more than two hydraulic sources. For the failures considered, any configuration which met the two-fail-operate requirement was given an A rating. Lower ratings were given for the remaining configurations, depending on the number of failures for which the requirement was not met. There are certain conditions for which the full two-fail-operate criteria must be violated for the SFCS program, due to the use of an existing airframe, lack of required technology or the economics of an R&D program. Driving a slab stabilator requires use of a single horn and single attach fitting. Use of more than two coupled master control valves has been found to be beyond the scope of machining precision available today. These are examples of areas in which violation of the two-fail-operate criteria does not eliminate a configuration from consideration.

f. Compatibility With SFCS Program Schedule

This category represents an evaluation of producibility and procuribility in terms of their effect on the SFCS program schedule and incorporates an appropriate penalty for paper concepts such as those involving three or four motor pump units. Penalties were assessed for each motor pump unit, triple or quadruple master control valves, the requirement for an HDEG and the requirement for a fourth AC electrical source.

g. Compatibility With Central Hydraulic Systems

Where fluids are mixed by switching, it is obvious that different fluids cannot be used. Hence, a system which is backed up by a central hydraulic supply cannot be allowed to exceed the temperature limitations of the aircraft central hydraulic supply fluid. A system operating at high temperatures could be filled with a high temperature fluid to eliminate many undesirable effects of high temperature on the fluid if the fluid is not mixed with a central hydraulic supply. However, conduction throughout the package could still locally expose the central system to an excessive temperature.

MCAIR recognizes the problems involved in mixing dissimilar hydraulic fluids at high differential temperatures in the SSAP design. Therefore the same type high temperature hydraulic fluid, MIL-H-83282, will be used in both the central and SSAP integral hydraulic systems. The hydraulic fluid selection study is presented in Section V.

The ratings used to assess the configurations compatibility with the central hydraulic system were:

A - No central hydraulic systems used.
B - Central hydraulic systems used but not mixed with the local systems.
C - Central hydraulic systems and local systems are mixed.

h. Maintainability

The maintainability ratings are based on the inherent complexity and operating characteristics of each configuration as they effect support requirements. Maintainability is a design area where some compromise between optimum maintainability characteristics and program economics must be made. For this reason, the rating of excellent (A) and unacceptable (D) were not applied in the maintainability ratings.

i. Weight

The weights of the following components are estimates based on the Simplex packages and APU weights which are 90 lbs. and 25 lbs. respectively; the master control valve is considered to be part of the surface actuator.

COMPONENT	WEIGHT IN POUNDS
Surface Actuator	
Dual	50
Triple	100
Quadruple	130
Soft Cutoff Motor Pump Reservoirs	
5 HP	30
8 HP	42
12 HP	55

COMPONENT	WEIGHT IN POUNDS
Soft Cutoff Flow Sharing Motor Pump Reservoirs	
2 x 5 HP	48
1 x 5 HP + 1 x 8 HP	55
2 x 8 HP	63
Secondary Actuator	20
Switching Valve	3

Configuration	1	2a	2b	2c	2d	3a	3b	4	5	6
Weight	180	160	170	157	160	202	212	190	246	204

Configuration	7a	7b	8	9	10	11a	11b	12	13
Weight	219	225	249	240	270	148	125	160	234

The ratings used to assess relative weight were:

A $W < 135$
B $135 < W < 185$
C $185 < W$

j. Maximum Steady State Power Requirements

The steady state electrical power input is a guide to the general efficiency of the package. The results are based on soft cutoff pump systems with the following supply capability.

% of Max. Hinge Moment	100	50	100	50	50	100
% of Max. Velocity	100	100	50	50	50	50
Steady State HP	3.7	2.1	2.1	1.5	.6	.6

k. Safety

Based on 22,158 production flights at MCAIR by all model F-4's hydraulic discrepancy reports are as follows:

System	PC-1	PC-2	Utility
No.of Reports	146	137	735

These discrepancies are not always failures but do include leakage, malfunctions, or discrepancies of associated equipment not identified. Hydraulic leaks have caused radar antenna problems and were coded to the radar system vice the utility hydraulic system. This indicates that the total utility hydraulic system discrepancies may be higher than stated above. The contamination level of the utility system is higher because of the many components in the system and this also contributes to the higher failure rate.

Because of the high failure rates of the utility hydraulic system all configurations utilizing the utility system as a primary or back-up system were rated lower than those utilizing PC-1 and PC-2 hydraulic systems. This lower rating applied whether the utility system is used as a primary or back-up system.

As a result, Configurations 12 and 13 are rated above configurations 2a through 11b. Configuration (1) is rated low because it does not have a back-up system.

1. Envelope

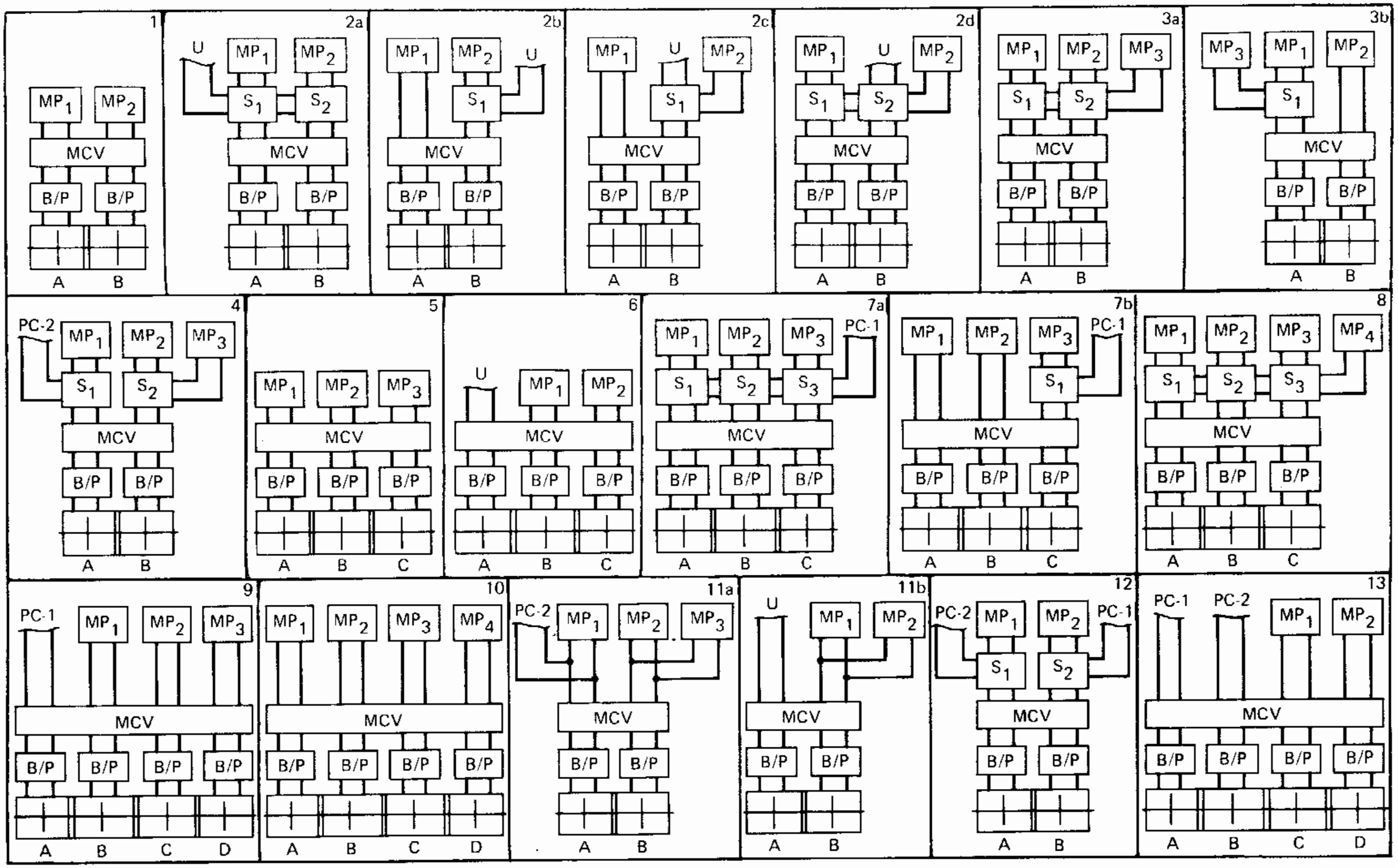
Early envelope studies for the SSAP indicated clearly the necessity of having the package body fixed to structure and its piston moving with the surface. To have provided adequate clearance for a moving body package dragging 22 wire bundles and 6 hydraulic lines over a 10.17 inch stroke would have resulted in modifications to major structural members in the aft fuselage.

In this trade study, a study of potential envelope problems was not initiated until late; consequently, the envelope investigation was restricted to the configurations shown on Figure 5, which had not been eliminated due to other considerations. Thus, only configurations 4, 6, 11a, 12, and 13 of Figure 5 were considered. Of these configurations, 4, 11a, and 12 have only two pistons and do not present an envelope problem. Therefore, actuator layouts were made for triple and quadruple piston configurations only as shown in the ensuing list. No layouts are shown for the dual tandem piston configurations since they are essentially identical to the actuator layout of Figure 6 except that only one dual piston in tandem is required. It should be noted that the triple piston actuator layouts of Figures 6, 7 and 8 correspond to configuration number 6 of Figure 5. The quadruple piston actuator layout of Figure 9 corresponds to configuration 13 of Figure 5. For configuration 12 a parallel actuator layout can be used without presenting an envelope problem. Figure 10 represents the available envelope.

FIGURE	MOTOR PUMPS	PISTONS	RELATIVE RATING
-	Duplex	Dual Tandem	A
4	Duplex	Triple Tandem	D
5	Duplex	Triple Parallel	B
6	Duplex	Triple Parallel	C
7	Duplex	Quadruple Parallel	C
8	Available Envelope		

Actuators shown in Figures 6 through 9 require a yoke to link the multiple rod ends together since the aircraft installation cannot tolerate more than two pistons in line. The combination of multiple tandem pistons and rod-end yoke presents a particular problem because the yoke wipes through a bulkhead at F.S.600 in Figure 6. Because of the rework required, this actuator layout is considered economically unacceptable. The use of a parallel piston actuator layout without a tail pipe avoids this problem since it moves the rod-end yoke well away from the bulkhead.

The difference between the actuator layouts of Figure 7 and 8 is in the arrangement of the motor pumps relative to the line-of-action



A,B,C,D - Main Actuator Pistons
MCV - Master Control Valve
B/P - Bypass Valve
S - Switching Valve, Hydraulic

MP_1 - Motor Pump - L/H AC Bus
MP_2 - Motor Pump - R/H AC Bus
MP_3 - Motor Pump - Hydraulic (Utility) Driven Electric Generator
MP_4 - Motor Pump - L/H and R/H AC Buses

PC-1 - Central Hydraulic System (Primary 1)
PC-2 - Central Hydraulic System (Primary 2)
U - Central Hydraulic System (Utility)

FIGURE 5
SSAP CONFIGURATION SCHEMATICS

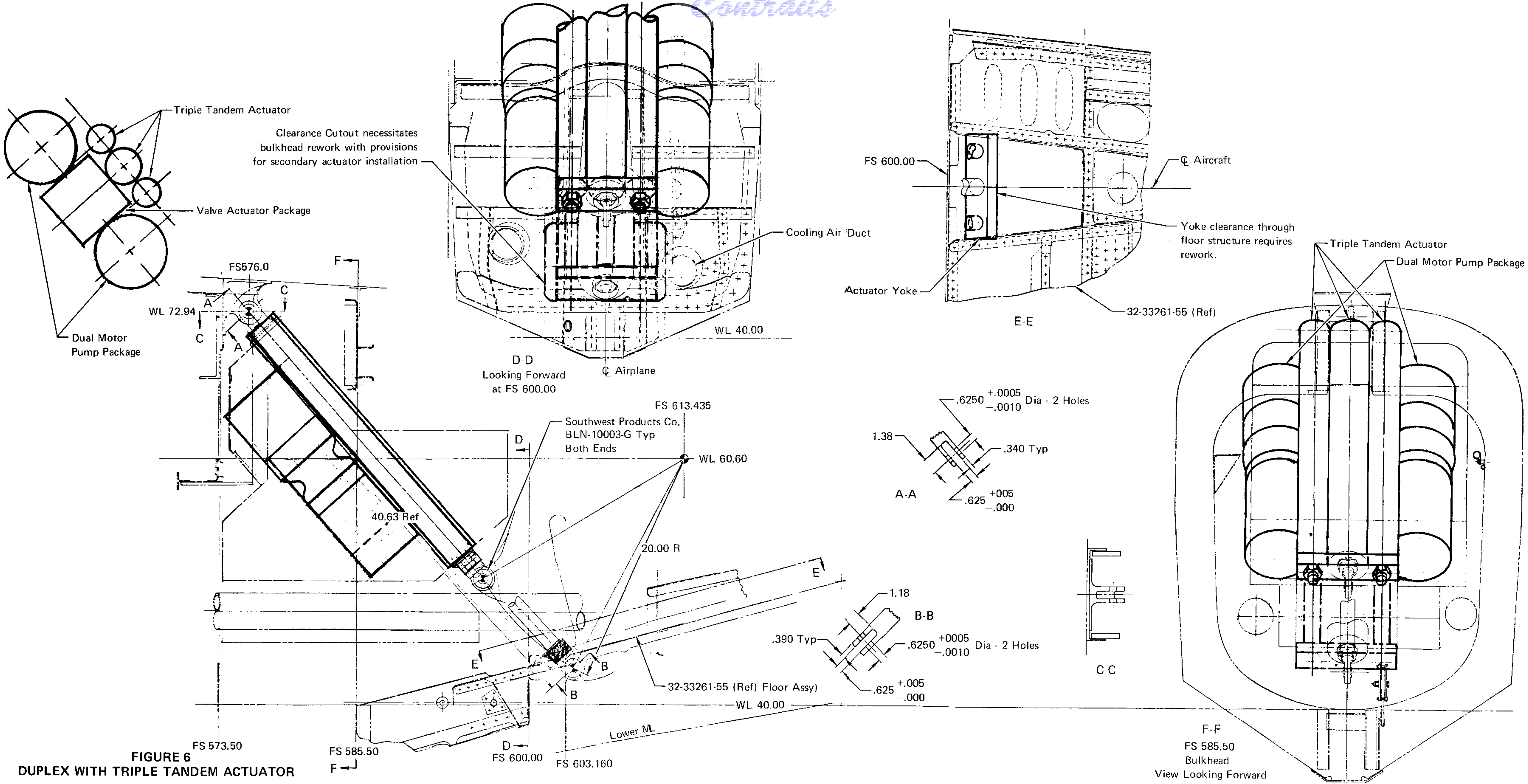


FIGURE 6
DUPLEX WITH TRIPLE TANDEM ACTUATOR
(Configuration 6)

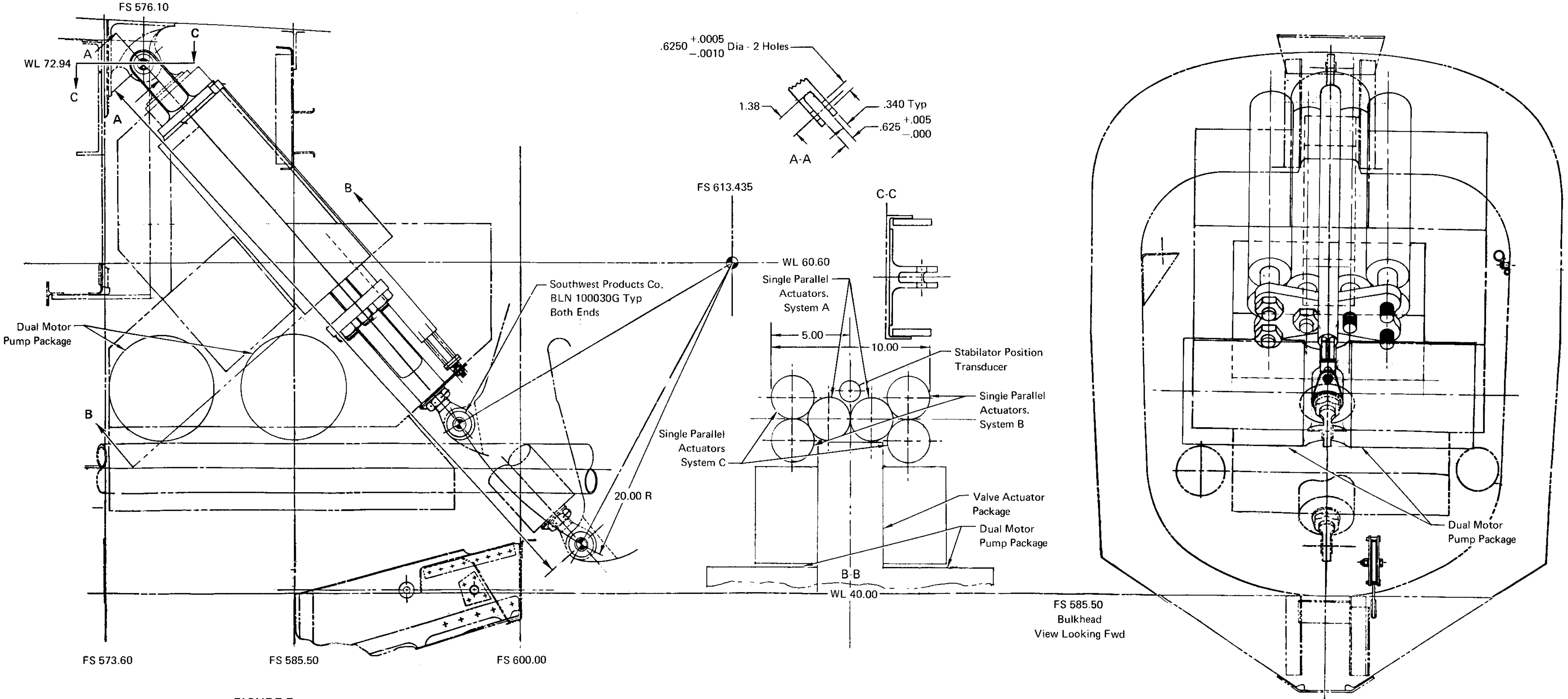


FIGURE 7
DUPLEX WITH TRIPLE PARALLEL ACTUATOR
(Configuration 6)

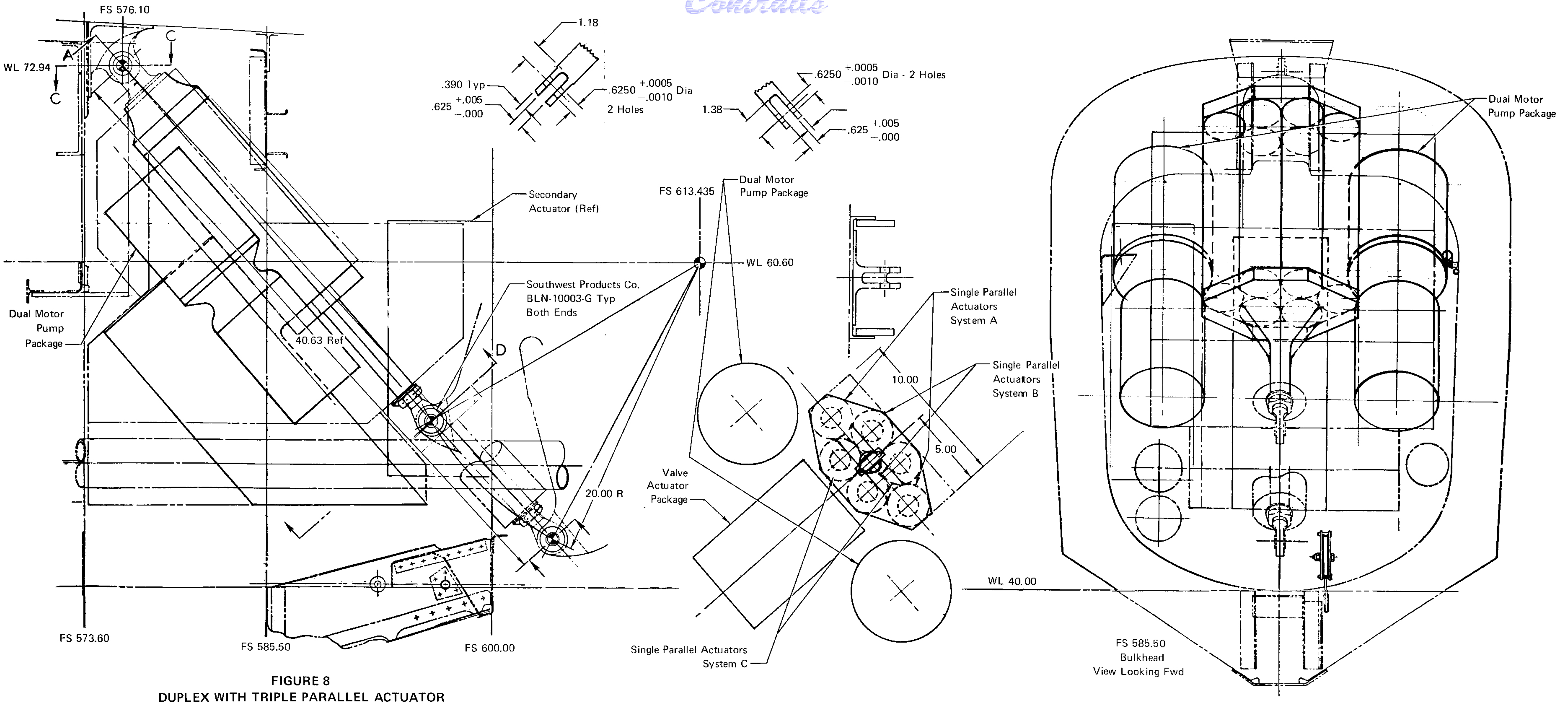


FIGURE 8
DUPLEX WITH TRIPLE PARALLEL ACTUATOR
(Configuration 6)

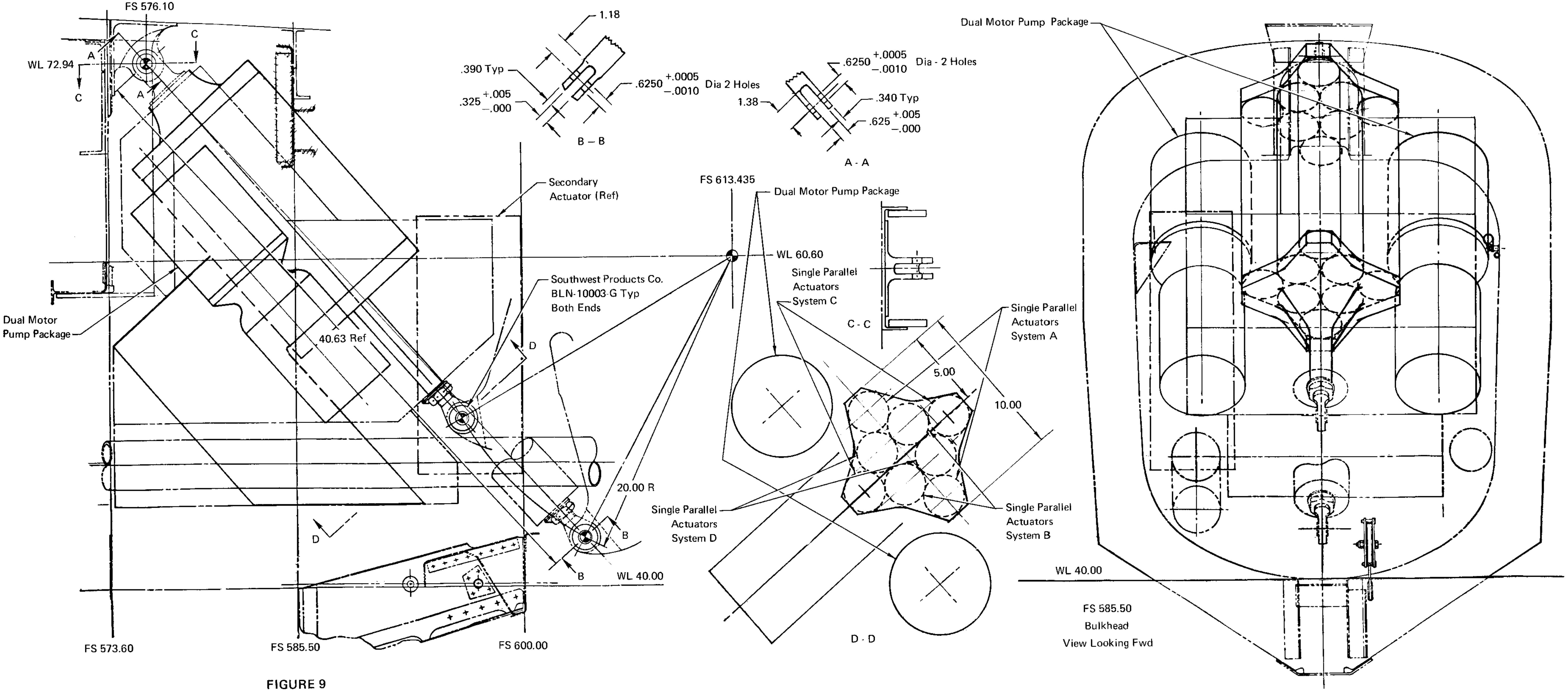

FS 576.10
WL 72.94
A
A
C
C
1.18
.390 Typ
.325 +.005 −.000
.6250 +.0005 −.0010 Dia 2 Holes
B – B
.6250 +.0005 −.0010 Dia - 2 Holes
1.38
.340 Typ
.625 +.005 −.000
A - A
Dual Motor Pump Package
Secondary Actuator (Ref)
FS 613.435
Dual Motor Pump Package
WL 60.60
Single Parallel Actuators System C
C - C
Southwest Products Co. BLN-10003-G Typ Both Ends
Dual Motor Pump Package
40.63 Ref
D
D
Single Parallel Actuators System A
5.00
10.00
20.00 R
B
B
WL 40.00
Single Parallel Actuators System D
Single Parallel Actuators System B
D - D
WL 40.00
FS 585.50 Bulkhead View Looking Fwd
FS 573.60
FS 585.50
FS 600.00

FIGURE 9
DUPLEX WITH QUADRUPLE PARALLEL ACTUATOR
(Configuration 13)

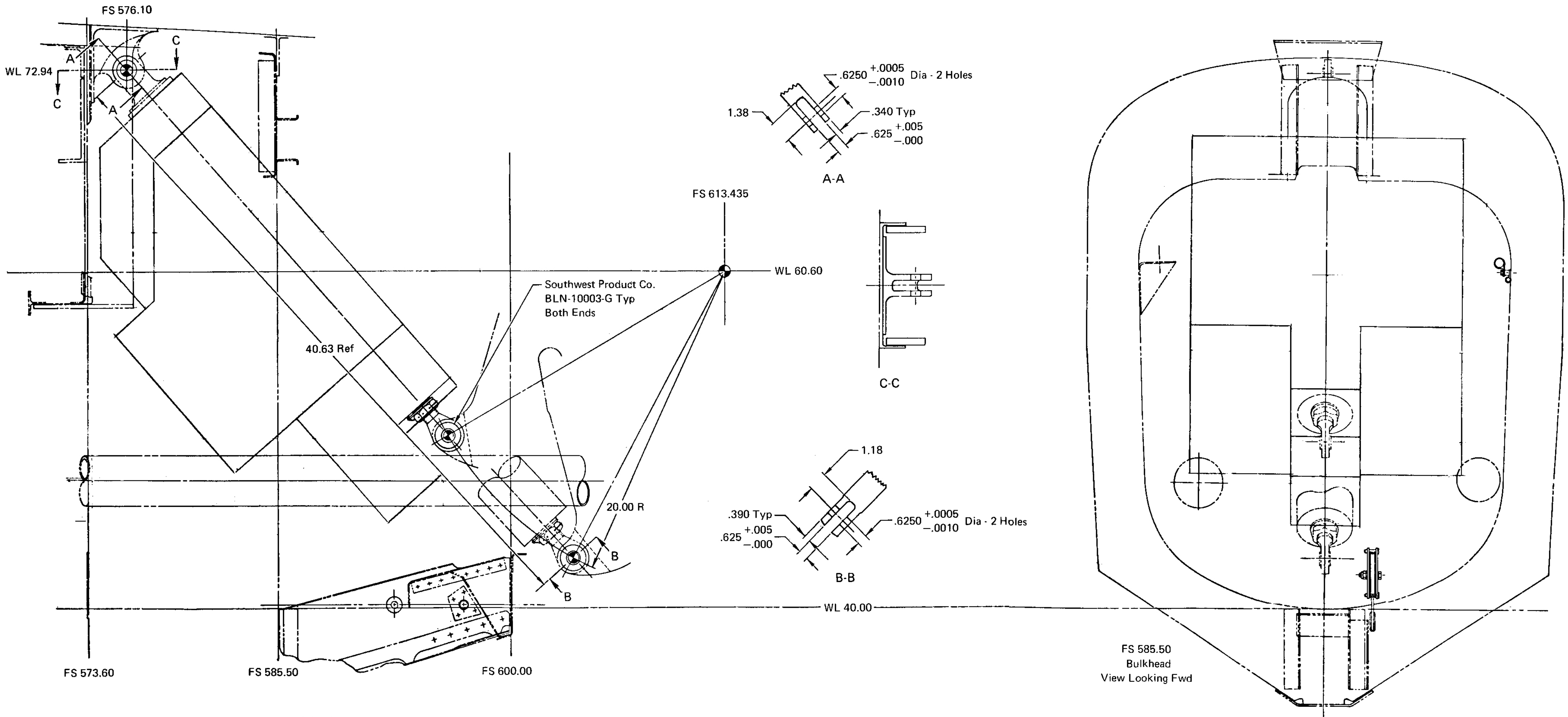


FIGURE 10
AVAILABLE ENVELOPE

of the actuator. In Figure 7, the motor pumps are arranged perpendicular to the actuator line-of-action while in Figure 8 the motor pumps are arranged parallel to the actuator line-of-action. The arrangement of Figure 7 is preferred because it makes better utilization of the available space. The quadruple-parallel actuator layout of Figure 9 is rated the same as Figure 8 actuator layout because the difference in terms of envelope is not considered significant.

The relative ratings listed above were shown to distinguish between the various actuator layouts considered. Of the three triple-piston actuator layouts investigated, only the actuator layout of Figure 7 is considered in the comparison matrix of Table II.

m. Thermal Considerations

(1) General

Thermal considerations are a primary concern in IAP (Integrated Actuator Package) design. All designs submitted in response to the SSAP Procurement Specification were carefully evaluated with respect to thermal characteristics. Conclusive analytical thermal comparisons can be made between the various configurations under consideration only after a thermal model of each configuration is devised. Thermal model characteristics are determined by the design detail of the configurations. The lack of empirical design detail made it necessary to devise the thermal models from package design detail contrived in the thermal analysis itself. The many variables involved in devising the models, the number of configurations, and the brevity of the study all combine to make difficult a thermal optimization of each configuration. Moreover, the variables leading to the optimization of each configuration have a potentially different impact on the aircraft installation and therefore this impact must be studied. The rationale used in devising the thermal models is presented in Appendix II.

(2) Scope of Considerations

The cooling of a particular package depends upon the effective surface area of the package, local radiation and convection rates, per unit area, the hydraulic and electrical heat dissipation, the capacity of heat exchangers or other supplemental heat transfer provisions, and the capacity, temperature, and delivery scheme of the ram airflow or other heat sink provided.

(a) Effective Surface Area

The effective surface area depends, first, upon what the actual surface area is, and secondly, upon what the temperature gradients are along the surface. The temperature gradients in turn are dependent upon the relative thermal remoteness of the various portions of the package surface from the internal heat sources. The thermal remoteness is a function of the internal distribution of power losses and the internal heat flow paths. The internal heat flow paths are not only influenced by the physical dimensions and thermal conductances of various supporting members or package protuberances, but also by the internal physical plumbing arrangement and associated hydraulic flow rates.

(b) Local Radiation and Convection Rates

The local radiation heat transfer rate is influenced by the thermal emissivities of the package and aircraft surfaces and the view factors characterizing the installation. Surface convection rates are significantly influenced by local air velocities. Air velocities, in turn, are influenced by ventilation airflow rates, use, location and capacity of fans, considerations of special shrouding, and physical limitations on where air may be ducted for spot cooling.

(c) Hydraulic and Electrical Heat Dissipation

The heat dissipated by a particular package is the cooling load at steady state conditions. It comprises the power losses of the hydraulic pump and the associated flow circuits and those of the electric motor. The hydraulic pump losses are determined by pump efficiency and are influenced by pump flow rate and the supply and discharge pressures. In a valve controlled soft cutoff arrangement, all of the pump input power becomes heat unless work is being done in moving the surface. The electric motor power losses for analytical purposes may be considered to vary as a function of its efficiency and output rating.

(d) Supplemental Heat Transfer Provisions

The capacity required of the heat exchangers or other supplemental heat transfer provisions for a package is equal to the portion of the heat dissipated by the package that, at the operating temperature selected, cannot be dissipated from the surface of the hydraulic components themselves. The size of a heat exchanger cooled by compartment air is influenced by the temperature of the compartment. The compartment temperature is, of course, a function of the temperature and velocity of the ram airflow provided.

Therefore, along with a suitable means of package heat dissipation, a method of removing the dissipated heat from the compartment without overheating the aircraft structure, the SSAP electric motors, or surrounding equipment must be established. The attendant concerns of scoop design, airflow distribution, airflow exhaust porting and related factors must be considered.

(e) Heat Sink Provisions

The capacity of the heat sink available to the package influences what the size of the heat exchanger or other heat transfer provisions must be to attain the selected operating temperature. Tradeoffs are required to establish whether the heat exchanger should be enlarged or the capacity of the heat sink increased. The capacity of the heat sink also determines what temperature the package ambient will attain.

For analytical purposes, the temperature difference between configurations can be greatly reduced by arbitrarily increasing without regard to envisioned installation constraints and other specific limits, the ventilating air mass flow rate, the finned cooling areas, the local air velocity, etc. The temperature difference can be essentially eliminated by adding individually sized heat exchangers to the various configurations. All these increases produce some penalty to the aircraft. Selecting the most satisfactory fluid operating temperature for a configuration involves multiple tradeoffs. Ideally, the benefits gained by lowering the fluid operating temperature in specific increments must be compared to the penalties incurred by employing the associated cooling provisions or techniques. In this trade study simplifying assumptions are made to facilitate completion. For example, the weight penalty associated with heat transfer improvements are based upon a linear increase in heat exchanger or fin weight with increasing heat transfer capacity, rather than upon an optimized heat exchanger design for each of the 34 thermal models. These simplifications are not considered to invalidate the conclusions drawn herein.

(3) Approach

The thermal trade study included the following tasks:

(a) Translating the configuration into a thermally complete design by analytically sizing the package.

(b) Calculating the heat generated.

(c) Determining unimproved operating temperatures and the need for improvement.

(d) Determining the improvements required to operate at a given fluid temperature.

(e) Calculating the potential requirements for increased ventilation.

(f) Assessing the feasibility of achieving state-of-the-art operating temperatures.

(g) Devising and implementing an overall thermal rating and integrating with the total rating scheme.

The environmental conditions and duty cycles used in evaluating thermal considerations are based on the thermal evaluation conditions presented in Table VI.

TABLE VI
THERMAL EVALUATION CONDITIONS

Parameter		Parameter Value	
		High Altitude Soak Flight Condition	Transient Flight Condition
Mach Number		1.46	2.4
Altitude	(Ft)	50,000	50,000
Design Day	—	ICAO Standard	ANA Cold
Time at Flight Condition	(Min)	Continuous	4.5
Ram Air Inlet Temperature	(°F)	95	318
Ram Air Flow Rate	(Lb/Min)	13	30
Actuator Trim Load	(Lb)	5000	6600
Gust Loading Duty Cycle			
Actuator Amplitude (Peak to Peak)	(In.)	1.2	0.86
Frequency	(Hz)	0.8	1
Loading Period	(Min)	1	1
Time Between Periods	(Min)	5	5

The comparative thermal condition of the packages at the high altitude soak condition is considered a bellwether of comparative package thermal conditions throughout the flight profile. Thus the heat dissipation and baseline unimproved thermal environment for each package were determined at the high altitude soak flight condition by utilizing the gust loading duty cycle and the ram air flow rates listed. The unimproved operating temperatures and the specific heat transfer improvement requirements used in this study were calculated for this condition. An estimate was also made of the packages' heat storage capacity and their ability to withstand the high temperature transient flight condition listed.

(4) Calculated Operating Temperatures

The unimproved fluid operating temperature is a function of the heat dissipated per unit area from the hydraulic circuits and the compartment temperature to which the heat is dissipated. The compartment temperature is a function of the sum of the heat dissipated by both the electrical and hydraulic components.

Fluid operating temperatures for thermally unimproved soft cutoff configurations are as follows:

CONFIGURATION	1	2a	2b	2c	2d	3a	3b	4	5	6
TEMPERATURE-F°	800	585	700	596	585	588	700	584	600	615

CONFIGURATION	7a	7b	8	9	10	11a	11b	12	13
TEMPERATURE-F°	525	585	525	545	585	485	585	584	526

Although servo controlled pump configurations were eliminated as discussed previously, preliminary thermal analyses were conducted for comparison.

The unimproved servo controlled version of a configuration can be expected, with a modest duty cycle, to operate more than 100°F cooler than the soft cutoff version. The servo controlled versions are also less susceptible to overheat due to high duty cycles.

The primary meaning of the temperature figures is signified by their being designated "unimproved" package operating fluid temperatures. This designation indicates that the temperatures pertain to configurations built without including heat transfer improvements specially tailored to limit temperature to a specific level. The high temperatures shown are considered valid indicators of the need for heat transfer improvement. Operating temperatures as high as these "unimproved" values are not to be expected in configuration designs wherein package thermal characteristics and heat transfer improvements receive the proper design emphasis. Incorporating the proper heat transfer improvements, such as heat exchangers, into the packages to reduce their fluid temperature is considered to be more feasible for the SFCS program than acquiring a hydraulic fluid and compatible seals capable of withstanding the high temperatures predicted for unimproved packages. Accordingly, the thermal ratings assigned the configurations are based on reducing steady state fluid temperatures to 350°F or 450°F.

The servo pump versions investigated during this study exhibit thermal advantages over the soft cutoff pump versions partially due to the sizing techniques used for servo pumps. The servo pumps were initially sized on the basis that the actuator piston areas would need to be double those of the soft cutoff pump version in order to provide the required actuator stiffness. As a general rule, then, the pump flow rates of the servo pumps used in the thermal analysis were twice or nearly twice the flow rates used in the thermal analysis of the soft cutoff pumps. If analytical reasons or design innovations are found such that servo pump actuators would have the same stiffness with the same piston areas as the soft cutoff pump actuators, this would increase even more the thermal advantage held by the servo pump versions. Since servo pump stiffness data were not available at the time of this study, only the soft cutoff pump is considered in the thermal ratings.

(5) Thermal Rating of Configurations

(a) Definition of Rating System

A - Baseline aircraft ventilation provisions and reasonable SSAP self-contained heat transfer provisions result in a moderate operating fluid temperature of 350°F or less steady state.

B - Aircraft ventilation provisions and/or SSAP self-contained heat transfer provisions moderately exceed the desirable levels of "A" to achieve a steady state fluid temperature of 350°F or less.

C - Aircraft ventilation provisions and/or SSAP self-contained heat transfer provisions significantly exceed baseline and desirable levels of "A". The steady state fluid temperature may go to 450°F.

D - Unacceptable, aircraft or SSAP heat transfer provisions required are so extensive and/or operating temperatures so high as to constitute a major threat to the SFCS program schedule, and/or package reliability, and anticipated program costs.

To employ a "D" rating in this study without the expenditure of an excessive amount of supporting analytical effort requires the use of many essentially subjective judgements. While some of the configurations investigated appear to necessitate or be better suited to nonpreferred and/or somewhat costly heat transfer techniques, such as component heat exchangers, it is misleading to dismiss them as "unacceptable". Some of the nonpreferred techniques that might be employed as a "last resort" for configurations having a poor thermal rating are:

- Redundant oil-to-air heat exchangers
- Package shrouding to achieve a high air velocity over a large part of the package surface
- Redundant heat transport loops to convey heat to a remotely located heat exchanger where space is more readily available. A special case is redundant heat pipes to convey heat from heat sources to finned heat exchange stubs in individual fan and/or ram air ducts.
- Techniques in the categories of "Artificial" cooling which are prohibited by definition of the program objectives. Included in this category are ram air expanders and similar techniques which employ a refrigeration effect.

For even the worst configuration examined, it appears possible to achieve state-of-the-art temperature levels. However, this achievement is attainable through the use of relatively heavy and/or destructible heat dissipation equipment and undesirably large ventilation airflow increases. The penalties that such extensive cooling provisions impose on the performance of the F-4 test aircraft do not appear significant; however, it is believed that the performance penalty would be a major consideration in evaluating the suitability of the SSAP configurations for a future development aircraft with high performance goals. In the case of the SFCS program, the major detrimental consideration appears to be theoretically reduced survivability, increased vulnerability to handling damage, decreased reliability, increased program costs, and/or undesirable diverting of the program technical effort and funds to design, installation and testing of the cooling provisions and interfacing hardware (or the risk of such). It is desirable to promote for this application package thermal features that are consistent both with the SFCS program goals and potential future applications of integrated actuator packages.

(b) Ratings Assigned

The thermal ratings for each configuration are provided as shown in Table VII. The thermal ratings allow higher fluid temperatures and nonpreferred heat transfer techniques to be rated "C". Thermal ratings are integrated in the Comparison Matrix of SSAP Design Approaches, Table II.

TABLE VII
THERMAL RATING SUMMARY

Configuration	1	2a	2b	2c	2d	3a	3b	4	5	6
Overall Thermal Rating	C	B	C	B	B	B	C	C	C	B

Configuration	7a	7b	8	9	10	11a	11b	12	13
Overall Thermal Rating	B	C	B	B	C	A	B	B	B

n. Other Considerations

The comparison matrix does not include cost considerations. A preliminary attempt was made to estimate relative costs on the basis of system complexity. A comparison of these estimates with actuator cost estimates from manufacturers indicated little correlation, however, so cost estimates were not included in the comparison matrix.

5. SELECTION OF THE DESIGN APPROACH

The results of evaluations by the comparison matrix indicate, as shown in Table II, that all configurations except 4, 6, 11a, 12 and 13 are unacceptable in one or more categories. The ratings for these five configurations are nearly equal. The consideration of cost and technical risk are included at this point to assist in the selection. Configuration 4 and 11a require a third motor-pump unit and a HDEG, and therefore, would cost more than configuration 12. Configurations 6 and 13 would incure more technical risk, in developing a triple or quadruple control valve and actuator, than configuration 12 which must only develop the hydraulic switching valves.

In the absence of compelling factors not uncovered in this limited trade study, configuration 12 is considered to represent the lowest cost and the least technical risk. The use of two central hydraulic systems in configuration 12 gives the increased safety necessary for successful completion of the SFCS program but in no way compromises the future use of IAPs where central hydraulic system(s) back-up may or may not be desired, depending on the overall aircraft layout, mission, and design philosophy. Therefore, configuration 12 was recommended for the SSAP. This recommendation was subsequently approved by the Government, and is the basis of the current SSAP design.

SECTION IV

HYDRAULIC POWER SUPPLY STUDY

1. GENERAL

This study was conducted to establish the requirements for the SFCS hydraulic power supplies. To fulfill the two fail operate requirements of the SFCS it will be necessary to use a fourth hydraulic system in the test airplane, in addition to the three existing systems which are PC-1, PC-2, and utility. The study included the following:

- System requirements for the secondary actuators; Mechanical Isolation Mechanism (MIM) shift actuator; rudder surface actuator back-up hydraulic supply; and the Survivable Stabilator Actuator Package (SSAP) back-up supply
- Analysis of the effect of the additional power requirements on the test aircraft's central hydraulic systems
- Analysis of the power requirements for the fourth hydraulic system
- Reliability analysis of the effect of the fourth hydraulic system on the test aircraft
- Thermal analysis of the fourth hydraulic system

The effect of the added hydraulic power requirements for the SFCS on the three existing aircraft systems is negligible. The fourth hydraulic system powered by an electric motor driven pump is adequate to supply hydraulic power to one channel of the secondary actuators, and to serve as a back-up hydraulic power source for the rudder surface actuator.

The use of MIL-H-83282 (MLO 68-5) hydraulic fluid in PC-1 and PC-2 hydraulic systems will not significantly affect the performance of either the secondary actuators or the surface actuators. A test performed at MCAIR, under Air Force contract number F33657-69-C-0861 and reported in Reference 1, showed an insigificant difference in actuator performance when using MIL-H-83282 and MIL-H-5606 fluid in the F-4 Iron Bird.

2. SYSTEM REQUIREMENTS

a. System Configurations

The hydraulic system schematics for the basic F-4E and the SFCS program are depicted in Figures 11 through 14. Figure 11 is that for the F-4E. Figures 12, 13 and 14 are schematics for Phase IIA, IIB, and IIC, respectively. The utility hydraulic system is not shown in detail, due to its complexity.

(1) The Phase IIA configuration differs from the F-4E configuration as follows:

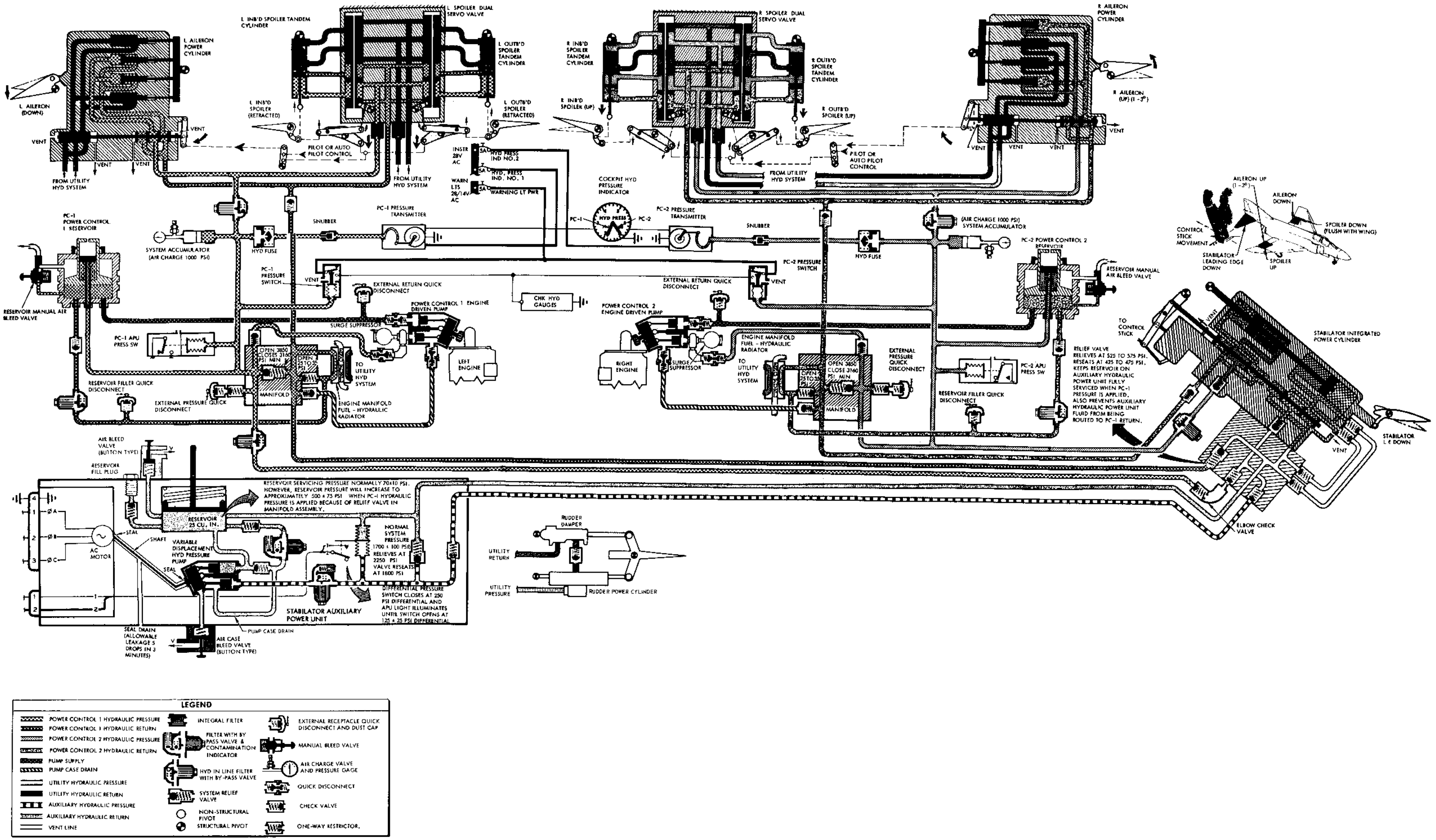

L INB'D SPOILER TANDEM CYLINDER
L SPOILER DUAL SERVO VALVE
L AILERON POWER CYLINDER
L OUTB'D SPOILER TANDEM CYLINDER
L AILERON (DOWN)
L INB'D SPOILER (RETRACTED)
L OUTB'D SPOILER (RETRACTED)
VENT
VENT
VENT
VENT
FROM UTILITY HYD SYSTEM
PILOT OR AUTO PILOT CONTROL
FROM UTILITY HYD SYSTEM
INSTR 28V AC
5A
HYD PRESS IND NO.2
5A
HYD. PRESS IND. NO. 1
WARN LTS 28/14V AC
5A
WARNING LT PWR
R INB'D SPOILER TANDEM CYLINDER
R SPOILER DUAL SERVO VALVE
R OUTB'D SPOILER TANDEM CYLINDER
R AILERON POWER CYLINDER
R INB'D SPOILER (UP)
R OUTB'D SPOILER (UP)
R AILERON (UP) (1 - 3°)
VENT
VENT
VENT
VENT
PILOT OR AUTO PILOT CONTROL
FROM UTILITY HYD SYSTEM
COCKPIT HYD PRESSURE INDICATOR
PC-1
HYD PRESS
PC-2
PC-1 POWER CONTROL 1 RESERVOIR
SYSTEM ACCUMULATOR (AIR CHARGE 1000 PSI)
HYD FUSE
SNUBBER
PC-1 PRESSURE TRANSMITTER
PC-2 PRESSURE TRANSMITTER
SNUBBER
HYD FUSE
(AIR CHARGE 1000 PSI) SYSTEM ACCUMULATOR
PC-2 POWER CONTROL 2 RESERVOIR
AILERON UP (1 - 3°)
AILERON DOWN
CONTROL STICK MOVEMENT
SPOILER DOWN (FLUSH WITH WING)
STABILATOR LEADING EDGE DOWN
SPOILER UP
RESERVOIR MANUAL AIR BLEED VALVE
PC-1 PRESSURE SWITCH
VENT
EXTERNAL RETURN QUICK DISCONNECT
CHK HYD GAUGES
PC-2 PRESSURE SWITCH
VENT
EXTERNAL RETURN QUICK DISCONNECT
RESERVOIR MANUAL AIR BLEED VALVE
POWER CONTROL 1 ENGINE DRIVEN PUMP
SURGE SUPPRESSOR
PC-1 APU PRESS SW
OPEN 3850 CLOSES 3160 PSI MIN
OPEN 25 TO 35 PSI
TO UTILITY HYD SYSTEM
LEFT ENGINE
MANIFOLD
RESERVOIR FILLER QUICK DISCONNECT
EXTERNAL PRESSURE QUICK DISCONNECT
ENGINE MANIFOLD FUEL - HYDRAULIC RADIATOR
POWER CONTROL 2 ENGINE DRIVEN PUMP
RIGHT ENGINE
SURGE SUPPRESSOR
ENGINE MANIFOLD FUEL - HYDRAULIC RADIATOR
TO UTILITY HYD SYSTEM
OPEN 25 TO 35 PSI
OPEN 3850 CLOSE 3160 PSI MIN
MANIFOLD
EXTERNAL PRESSURE QUICK DISCONNECT
PC-2 APU PRESS SW
RESERVOIR FILLER QUICK DISCONNECT
TO CONTROL STICK
RELIEF VALVE RELIEVES AT 525 TO 575 PSI. RESEATS AT 425 TO 475 PSI. KEEPS RESERVOIR ON AUXILIARY HYDRAULIC POWER UNIT FULLY SERVICED WHEN PC-1 PRESSURE IS APPLIED. ALSO PREVENTS AUXILIARY HYDRAULIC POWER UNIT FLUID FROM BEING ROUTED TO PC-1 RETURN.
VENT
STABILATOR INTEGRATED POWER CYLINDER
VENT
STABILATOR L E DOWN
ELBOW CHECK VALVE
AIR BLEED VALVE (BUTTON TYPE)
V
RESERVOIR FILL PLUG
RESERVOIR SERVICING PRESSURE NORMALLY 70±10 PSI. HOWEVER, RESERVOIR PRESSURE WILL INCREASE TO APPROXIMATELY 500 ± 75 PSI WHEN PC-1 HYDRAULIC PRESSURE IS APPLIED BECAUSE OF RELIEF VALVE IN MANIFOLD ASSEMBLY.
4
1
ØA
2
ØB
3
ØC
AC MOTOR
SEAL
SHAFT
RESERVOIR 25 CU. IN.
VARIABLE DISPLACEMENT HYD PRESSURE PUMP
SEAL
1
2
NORMAL SYSTEM PRESSURE 1700 ± 100 PSI) RELIEVES AT 2250 PSI VALVE RESEATS AT 1800 PSI
DIFFERENTIAL PRESSURE SWITCH CLOSES AT 250 PSI DIFFERENTIAL AND APU LIGHT ILLUMINATES UNTIL SWITCH OPENS AT 125 ± 25 PSI DIFFERENTIAL
STABILATOR AUXILIARY POWER UNIT
1
2
1
2
SEAL DRAIN (ALLOWABLE LEAKAGE 5 DROPS IN 3 MINUTES)
V
AIR CASE BLEED VALVE (BUTTON TYPE)
PUMP CASE DRAIN
RUDDER DAMPER
UTILITY RETURN
UTILITY PRESSURE
RUDDER POWER CYLINDER
LEGEND
POWER CONTROL 1 HYDRAULIC PRESSURE
POWER CONTROL 1 HYDRAULIC RETURN
POWER CONTROL 2 HYDRAULIC PRESSURE
POWER CONTROL 2 HYDRAULIC RETURN
PUMP SUPPLY
PUMP CASE DRAIN
UTILITY HYDRAULIC PRESSURE
UTILITY HYDRAULIC RETURN
AUXILIARY HYDRAULIC PRESSURE
AUXILIARY HYDRAULIC RETURN
VENT LINE
INTEGRAL FILTER
FILTER WITH BY PASS VALVE & CONTAMINATION INDICATOR
HYD IN LINE FILTER WITH BY-PASS VALVE
SYSTEM RELIEF VALVE
NON-STRUCTURAL PIVOT
STRUCTURAL PIVOT
EXTERNAL RECEPTACLE QUICK DISCONNECT AND DUST CAP
MANUAL BLEED VALVE
AIR CHARGE VALVE AND PRESSURE GAGE
QUICK DISCONNECT
CHECK VALVE
ONE-WAY RESTRICTOR.

FIGURE 11
F-4E FLIGHT CONTROL HYDRAULIC SYSTEM

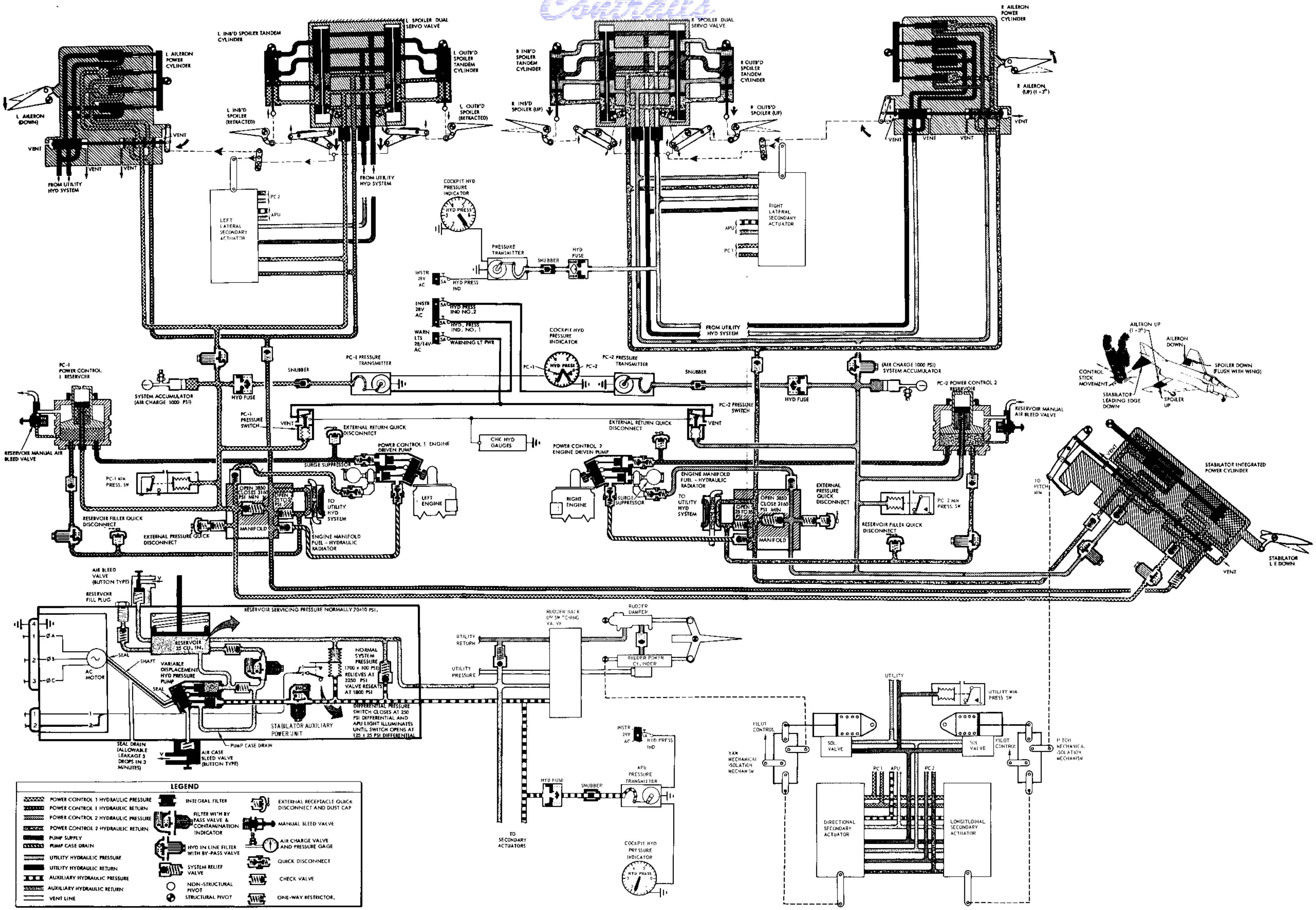

L AILERON POWER CYLINDER
L AILERON (DOWN)
VENT
FROM UTILITY HYD SYSTEM
L INB'D SPOILER TANDEM CYLINDER
L SPOILER DUAL SERVO VALVE
L OUTB'D SPOILER TANDEM CYLINDER
L INB'D SPOILER (RETRACTED)
L OUTB'D SPOILER (RETRACTED)
LEFT LATERAL SECONDARY ACTUATOR
PC 2
APU
R INB'D SPOILER TANDEM CYLINDER
R SPOILER DUAL SERVO VALVE
R OUTB'D SPOILER TANDEM CYLINDER
R INB'D SPOILER (UP)
R OUTB'D SPOILER (UP)
R AILERON POWER CYLINDER
R AILERON (UP) (1 - 3°)
RIGHT LATERAL SECONDARY ACTUATOR
PC 1
COCKPIT HYD PRESSURE INDICATOR
HYD PRESS
PRESSURE TRANSMITTER
SNUBBER
HYD FUSE
INSTR 28V AC
HYD PRESS IND
HYD PRESS IND NO. 2
HYD. PRESS IND. NO. 1
WARN LTS 28/14V AC
WARNING LT PWR
SA
PC-1 POWER CONTROL 1 RESERVOIR
RESERVOIR MANUAL AIR BLEED VALVE
SYSTEM ACCUMULATOR (AIR CHARGE 1000 PSI)
PC-1 PRESSURE TRANSMITTER
PC-1
PC-2
PC-2 PRESSURE TRANSMITTER
(AIR CHARGE 1000 PSI) SYSTEM ACCUMULATOR
PC-2 POWER CONTROL 2 RESERVOIR
PC-1 PRESSURE SWITCH
PC-2 PRESSURE SWITCH
EXTERNAL RETURN QUICK DISCONNECT
CHK HYD GAUGES
POWER CONTROL 1 ENGINE DRIVEN PUMP
POWER CONTROL 2 ENGINE DRIVEN PUMP
SURGE SUPPRESSOR
PC-1 MIN PRESS. SW
PC 2 MIN PRESS. SW
OPEN 3850 CLOSES 3160 PSI MIN
OPEN 25 TO 35 PSI
MANIFOLD
TO UTILITY HYD SYSTEM
LEFT ENGINE
RIGHT ENGINE
ENGINE MANIFOLD FUEL - HYDRAULIC RADIATOR
RESERVOIR FILLER QUICK DISCONNECT
EXTERNAL PRESSURE QUICK DISCONNECT
CONTROL STICK MOVEMENT
AILERON UP (1 - 3°)
AILERON DOWN
SPOILER DOWN (FLUSH WITH WING)
STABILATOR LEADING EDGE DOWN
SPOILER UP
STABILATOR INTEGRATED POWER CYLINDER
TO PITCH MIM
STABILATOR L E DOWN
AIR BLEED VALVE (BUTTON TYPE)
RESERVOIR FILL PLUG
RESERVOIR SERVICING PRESSURE NORMALLY 70±10 PSI.
RESERVOIR 25 CU. IN.
ØA
ØB
ØC
AC MOTOR
SEAL
SHAFT
VARIABLE DISPLACEMENT HYD PRESSURE PUMP
NORMAL SYSTEM PRESSURE 1700 ± 100 PSI RELIEVES AT 2250 PSI VALVE RESEATS AT 1800 PSI
DIFFERENTIAL PRESSURE SWITCH CLOSES AT 250 PSI DIFFERENTIAL AND APU LIGHT ILLUMINATES UNTIL SWITCH OPENS AT 125 ± 25 PSI DIFFERENTIAL
STABILATOR AUXILIARY POWER UNIT
SEAL DRAIN (ALLOWABLE LEAKAGE 5 DROPS IN 3 MINUTES)
PUMP CASE DRAIN
AIR CASE BLEED VALVE (BUTTON TYPE)
RUDDER BACK UP SWITCHING VALVE
RUDDER DAMPER
RUDDER POWER CYLINDER
UTILITY RETURN
UTILITY PRESSURE
TO SECONDARY ACTUATORS
APU PRESSURE TRANSMITTER
UTILITY
UTILITY MIN PRESS SW
PILOT CONTROL
SOL VALVE
YAW MECHANICAL ISOLATION MECHANISM
PITCH MECHANICAL ISOLATION MECHANISM
DIRECTIONAL SECONDARY ACTUATOR
LONGITUDINAL SECONDARY ACTUATOR
LEGEND
POWER CONTROL 1 HYDRAULIC PRESSURE
POWER CONTROL 1 HYDRAULIC RETURN
POWER CONTROL 2 HYDRAULIC PRESSURE
POWER CONTROL 2 HYDRAULIC RETURN
PUMP SUPPLY
PUMP CASE DRAIN
UTILITY HYDRAULIC PRESSURE
UTILITY HYDRAULIC RETURN
AUXILIARY HYDRAULIC PRESSURE
AUXILIARY HYDRAULIC RETURN
VENT LINE
INTEGRAL FILTER
FILTER WITH BY PASS VALVE & CONTAMINATION INDICATOR
HYD IN LINE FILTER WITH BY-PASS VALVE
SYSTEM RELIEF VALVE
NON-STRUCTURAL PIVOT
STRUCTURAL PIVOT
EXTERNAL RECEPTACLE QUICK DISCONNECT AND DUST CAP
MANUAL BLEED VALVE
AIR CHARGE VALVE AND PRESSURE GAGE
QUICK DISCONNECT
CHECK VALVE
ONE-WAY RESTRICTOR.

FIGURE 12

SFCS FLIGHT CONTROL HYDRAULIC SYSTEM PHASE II A

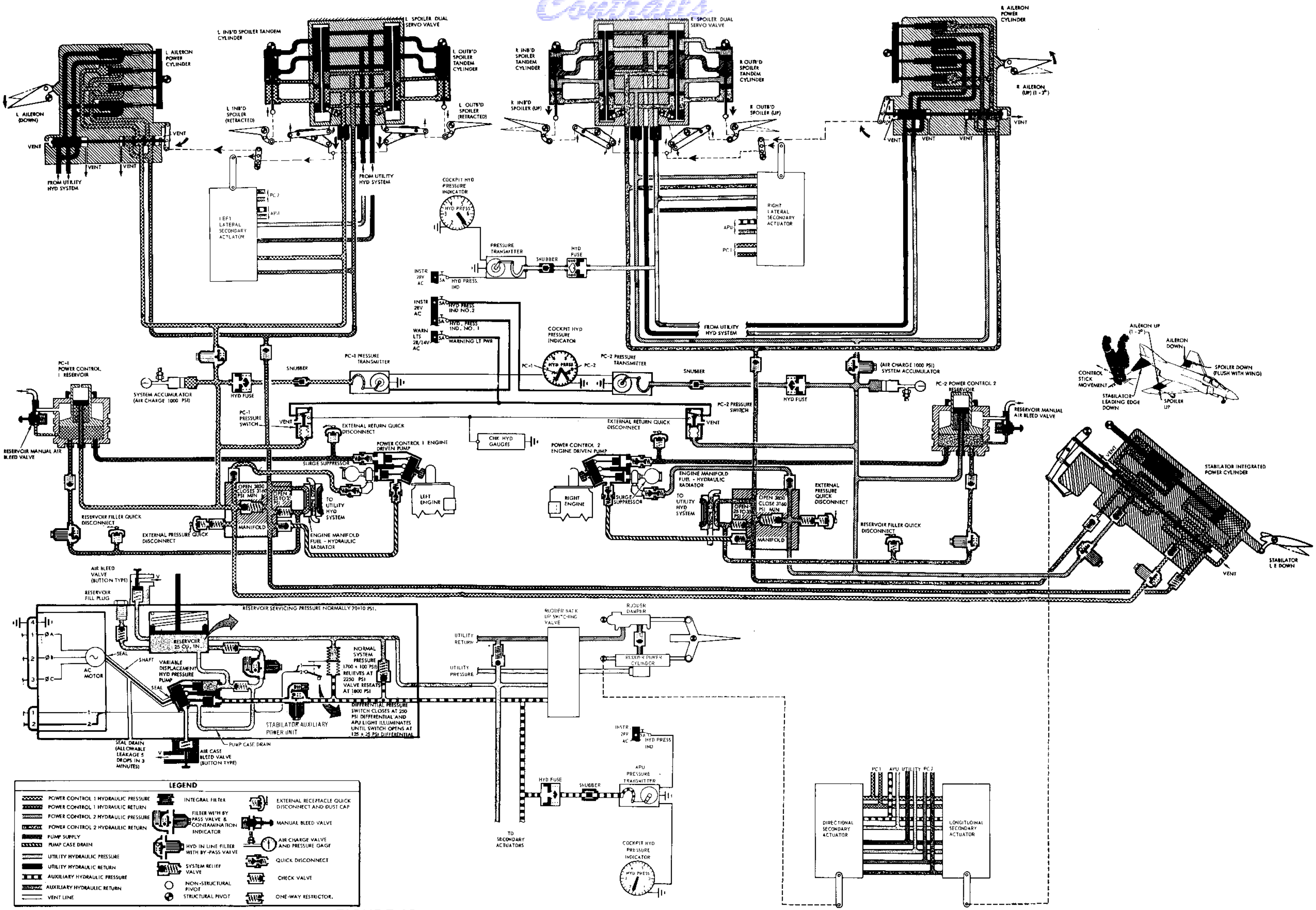

L AILERON POWER CYLINDER
L AILERON (DOWN)
VENT
VENT
VENT
VENT
FROM UTILITY HYD SYSTEM
L INB'D SPOILER TANDEM CYLINDER
L SPOILER DUAL SERVO VALVE
L OUTB'D SPOILER TANDEM CYLINDER
L INB'D SPOILER (RETRACTED)
L OUTB'D SPOILER (RETRACTED)
FROM UTILITY HYD SYSTEM
LEFT LATERAL SECONDARY ACTUATOR
PC 2
APU
COCKPIT HYD PRESSURE INDICATOR
HYD PRESS
PRESSURE TRANSMITTER
SNUBBER
HYD FUSE
INSTR 28V AC
5A
HYD PRESS. IND
INSTR 28V AC
5A
HYD PRESS IND NO.2
5A
HYD. PRESS IND. NO. 1
WARN LTS 28/14V AC
5A
WARNING LT PWR
R INB'D SPOILER TANDEM CYLINDER
R SPOILER DUAL SERVO VALVE
R OUTB'D SPOILER TANDEM CYLINDER
R INB'D SPOILER (UP)
R OUTB'D SPOILER (UP)
R AILERON POWER CYLINDER
R AILERON (UP) (1 - 3°)
VENT
VENT
VENT
VENT
RIGHT LATERAL SECONDARY ACTUATOR
APU
PC1
FROM UTILITY HYD SYSTEM
PC-1 POWER CONTROL 1 RESERVOIR
RESERVOIR MANUAL AIR BLEED VALVE
SYSTEM ACCUMULATOR (AIR CHARGE 1000 PSI)
HYD FUSE
SNUBBER
PC-1 PRESSURE TRANSMITTER
PC-1 PRESSURE SWITCH
VENT
EXTERNAL RETURN QUICK DISCONNECT
POWER CONTROL 1 ENGINE DRIVEN PUMP
SURGE SUPPRESSOR
OPEN 3850 CLOSES 3160 PSI MIN
OPEN 25 TO 35 PSI
MANIFOLD
TO UTILITY HYD SYSTEM
LEFT ENGINE
ENGINE MANIFOLD FUEL - HYDRAULIC RADIATOR
RESERVOIR FILLER QUICK DISCONNECT
EXTERNAL PRESSURE QUICK DISCONNECT
CHK HYD GAUGES
COCKPIT HYD PRESSURE INDICATOR
PC-1
HYD PRESS
PC-2
PC-2 PRESSURE TRANSMITTER
SNUBBER
HYD FUSE
(AIR CHARGE 1000 PSI) SYSTEM ACCUMULATOR
PC-2 POWER CONTROL 2 RESERVOIR
RESERVOIR MANUAL AIR BLEED VALVE
PC-2 PRESSURE SWITCH
VENT
EXTERNAL RETURN QUICK DISCONNECT
POWER CONTROL 2 ENGINE DRIVEN PUMP
RIGHT ENGINE
SURGE SUPPRESSOR
ENGINE MANIFOLD FUEL - HYDRAULIC RADIATOR
TO UTILITY HYD SYSTEM
OPEN 25 TO 35 PSI
OPEN 3850 CLOSE 3160 PSI MIN
MANIFOLD
EXTERNAL PRESSURE QUICK DISCONNECT
RESERVOIR FILLER QUICK DISCONNECT
AILERON UP (1 - 3°)
AILERON DOWN
SPOILER DOWN (FLUSH WITH WING)
CONTROL STICK MOVEMENT
STABILATOR LEADING EDGE DOWN
SPOILER UP
STABILATOR INTEGRATED POWER CYLINDER
VENT
VENT
STABILATOR L E DOWN
AIR BLEED VALVE (BUTTON TYPE)
V
RESERVOIR FILL PLUG
RESERVOIR SERVICING PRESSURE NORMALLY 70±10 PSI.
4
1 ØA
2 ØB
3 ØC
AC MOTOR
SEAL
SHAFT
RESERVOIR 25 CU. IN.
VARIABLE DISPLACEMENT HYD PRESSURE PUMP
SEAL
NORMAL SYSTEM PRESSURE 1700 ± 100 PSI) RELIEVES AT 2250 PSI VALVE RESEATS AT 1800 PSI
1
2
DIFFERENTIAL PRESSURE SWITCH CLOSES AT 250 PSI DIFFERENTIAL AND APU LIGHT ILLUMINATES UNTIL SWITCH OPENS AT 125 ± 25 PSI DIFFERENTIAL
STABILATOR AUXILIARY POWER UNIT
1
2
SEAL DRAIN (ALLOWABLE LEAKAGE 5 DROPS IN 3 MINUTES)
V
AIR CASE BLEED VALVE (BUTTON TYPE)
PUMP CASE DRAIN
UTILITY RETURN
UTILITY PRESSURE
RUDDER BACK UP SWITCHING VALVE
RUDDER DAMPER
RUDDER POWER CYLINDER
TO SECONDARY ACTUATORS
INSTR 28V AC
5A
HYD PRESS IND
APU PRESSURE TRANSMITTER
HYD FUSE
SNUBBER
COCKPIT HYD PRESSURE INDICATOR
HYD PRESS
PC1
APU
UTILITY
PC2
DIRECTIONAL SECONDARY ACTUATOR
LONGITUDINAL SECONDARY ACTUATOR
LEGEND
POWER CONTROL 1 HYDRAULIC PRESSURE
POWER CONTROL 1 HYDRAULIC RETURN
POWER CONTROL 2 HYDRAULIC PRESSURE
POWER CONTROL 2 HYDRAULIC RETURN
PUMP SUPPLY
PUMP CASE DRAIN
UTILITY HYDRAULIC PRESSURE
UTILITY HYDRAULIC RETURN
AUXILIARY HYDRAULIC PRESSURE
AUXILIARY HYDRAULIC RETURN
VENT LINE
INTEGRAL FILTER
FILTER WITH BY PASS VALVE & CONTAMINATION INDICATOR
HYD IN LINE FILTER WITH BY-PASS VALVE
SYSTEM RELIEF VALVE
NON-STRUCTURAL PIVOT
STRUCTURAL PIVOT
EXTERNAL RECEPTACLE QUICK DISCONNECT AND DUST CAP
MANUAL BLEED VALVE
AIR CHARGE VALVE AND PRESSURE GAGE
QUICK DISCONNECT
CHECK VALVE
ONE-WAY RESTRICTOR.

FIGURE 13
SFCS FLIGHT CONTROL HYDRAULIC SYSTEM PHASE II B

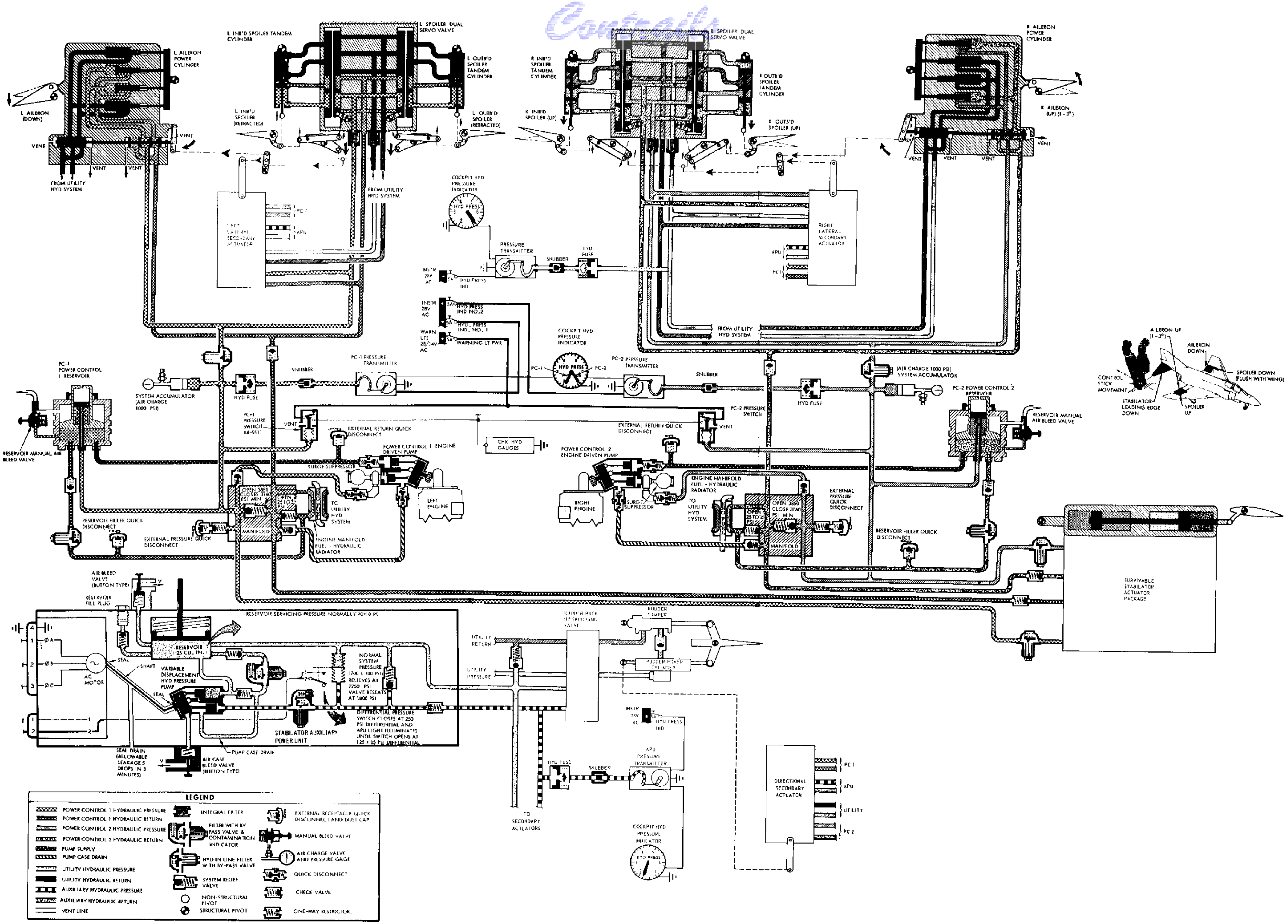

L AILERON POWER CYLINDER
L AILERON (DOWN)
VENT
VENT
VENT
VENT
FROM UTILITY HYD SYSTEM
L INB'D SPOILER TANDEM CYLINDER
L SPOILER DUAL SERVO VALVE
L OUTB'D SPOILER TANDEM CYLINDER
L INB'D SPOILER (RETRACTED)
L OUTB'D SPOILER (RETRACTED)
FROM UTILITY HYD SYSTEM
LEFT LATERAL SECONDARY ACTUATOR
PC 2
APU
COCKPIT HYD PRESSURE INDICATOR
HYD PRESS
PRESSURE TRANSMITTER
SNUBBER
HYD FUSE
INSTR 28V AC
5A
HYD PRESS IND
INSTR 28V AC
5A
HYD PRESS IND NO.2
5A
HYD. PRESS IND. NO. 1
WARN LTS 28/14V AC
5A
WARNING LT PWR
COCKPIT HYD PRESSURE INDICATOR
PC-1
HYD PRESS
PC-2
R INB'D SPOILER TANDEM CYLINDER
R SPOILER DUAL SERVO VALVE
R OUTB'D SPOILER TANDEM CYLINDER
R INB'D SPOILER (UP)
R OUTB'D SPOILER (UP)
R AILERON POWER CYLINDER
R AILERON (UP) (1 - 3°)
VENT
VENT
VENT
VENT
RIGHT LATERAL SECONDARY ACTUATOR
APU
PC1
FROM UTILITY HYD SYSTEM
AILERON UP (1 - 3°)
AILERON DOWN
SPOILER DOWN (FLUSH WITH WING)
CONTROL STICK MOVEMENT
STABILATOR LEADING EDGE DOWN
SPOILER UP
PC-1 POWER CONTROL, 1 RESERVOIR
RESERVOIR MANUAL AIR BLEED VALVE
SYSTEM ACCUMULATOR (AIR CHARGE 1000 PSI)
HYD FUSE
SNUBBER
PC-1 PRESSURE TRANSMITTER
PC-1 PRESSURE SWITCH 14-5511
VENT
EXTERNAL RETURN QUICK DISCONNECT
CHK HYD GAUGES
POWER CONTROL 1 ENGINE DRIVEN PUMP
SURGE SUPPRESSOR
OPEN 3850 CLOSES 3160 PSI MIN
OPEN 25 TO 35 PSI
TO UTILITY HYD SYSTEM
LEFT ENGINE
MANIFOLD
RESERVOIR FILLER QUICK DISCONNECT
EXTERNAL PRESSURE QUICK DISCONNECT
ENGINE MANIFOLD FUEL - HYDRAULIC RADIATOR
PC-2 PRESSURE TRANSMITTER
SNUBBER
HYD FUSE
(AIR CHARGE 1000 PSI) SYSTEM ACCUMULATOR
PC-2 POWER CONTROL 2 RESERVOIR
RESERVOIR MANUAL AIR BLEED VALVE
PC-2 PRESSURE SWITCH
VENT
EXTERNAL RETURN QUICK DISCONNECT
POWER CONTROL 2 ENGINE DRIVEN PUMP
RIGHT ENGINE
SURGE SUPPRESSOR
ENGINE MANIFOLD FUEL - HYDRAULIC RADIATOR
TO UTILITY HYD SYSTEM
OPEN 25 TO 35 PSI
OPEN 3850 CLOSE 3160 PSI MIN
MANIFOLD
EXTERNAL PRESSURE QUICK DISCONNECT
RESERVOIR FILLER QUICK DISCONNECT
SURVIVABLE STABILATOR ACTUATOR PACKAGE
AIR BLEED VALVE (BUTTON TYPE)
V
RESERVOIR FILL PLUG
RESERVOIR SERVICING PRESSURE NORMALLY 70±10 PSI.
4
1
ØA
2
ØB
3
ØC
AC MOTOR
SEAL
SHAFT
RESERVOIR 25 CU. IN.
VARIABLE DISPLACEMENT HYD PRESSURE PUMP
SEAL
NORMAL SYSTEM PRESSURE (1700 ± 100 PSI) RELIEVES AT 2250 PSI VALVE RESEATS AT 1800 PSI
DIFFERENTIAL PRESSURE SWITCH CLOSES AT 250 PSI DIFFERENTIAL AND APU LIGHT ILLUMINATES UNTIL SWITCH OPENS AT 125 ± 25 PSI DIFFERENTIAL
STABILATOR AUXILIARY POWER UNIT
1
2
SEAL DRAIN (ALLOWABLE LEAKAGE 5 DROPS IN 3 MINUTES)
PUMP CASE DRAIN
AIR CASE BLEED VALVE (BUTTON TYPE)
V
UTILITY RETURN
UTILITY PRESSURE
RUDDER BACK UP SWITCHING VALVE
RUDDER DAMPER
RUDDER POWER CYLINDER
INSTR 28V AC
5A
HYD PRESS IND
HYD FUSE
SNUBBER
APU PRESSURE TRANSMITTER
TO SECONDARY ACTUATORS
COCKPIT HYD PRESSURE INDICATOR
HYD PRESS
DIRECTIONAL SECONDARY ACTUATOR
PC 1
APU
UTILITY
PC 2
LEGEND
POWER CONTROL 1 HYDRAULIC PRESSURE
POWER CONTROL 1 HYDRAULIC RETURN
POWER CONTROL 2 HYDRAULIC PRESSURE
POWER CONTROL 2 HYDRAULIC RETURN
PUMP SUPPLY
PUMP CASE DRAIN
UTILITY HYDRAULIC PRESSURE
UTILITY HYDRAULIC RETURN
AUXILIARY HYDRAULIC PRESSURE
AUXILIARY HYDRAULIC RETURN
VENT LINE
INTEGRAL FILTER
FILTER WITH BY PASS VALVE & CONTAMINATION INDICATOR
HYD IN LINE FILTER WITH BY-PASS VALVE
SYSTEM RELIEF VALVE
NON-STRUCTURAL PIVOT
STRUCTURAL PIVOT
EXTERNAL RECEPTACLE QUICK DISCONNECT AND DUST CAP
MANUAL BLEED VALVE
AIR CHARGE VALVE AND PRESSURE GAGE
QUICK DISCONNECT
CHECK VALVE
ONE-WAY RESTRICTOR.

FIGURE 14
SFCS FLIGHT CONTROL HYDRAULIC SYSTEM PHASE II C

(a) The addition of the four quadruplex electrohydraulic secondary actuators, two in the lateral, one in the directional, and one in the longitudinal control systems

(b) The addition of a fourth hydraulic system

(c) The addition of the hydraulic controls and actuation for the MIMs in the longitudinal and directional control system

(d) The addition of a back-up hydraulic supply for the rudder surface actuator

(e) The auxilliary power unit, which has not previously been installed in the test airplane, will not be used to back up the stabilator surface actuator

(2) The Phase IIB configuration differs from the Phase IIA configuration by the removal of the hydraulic controls and actuation for the MIMs.

(3) The Phase IIC configuration differs from the Phase IIB configuration as follows:

(a) The removal of the production stabilator surface actuator

(b) The removal of the longitudinal secondary actuator

(c) The addition of the SSAP

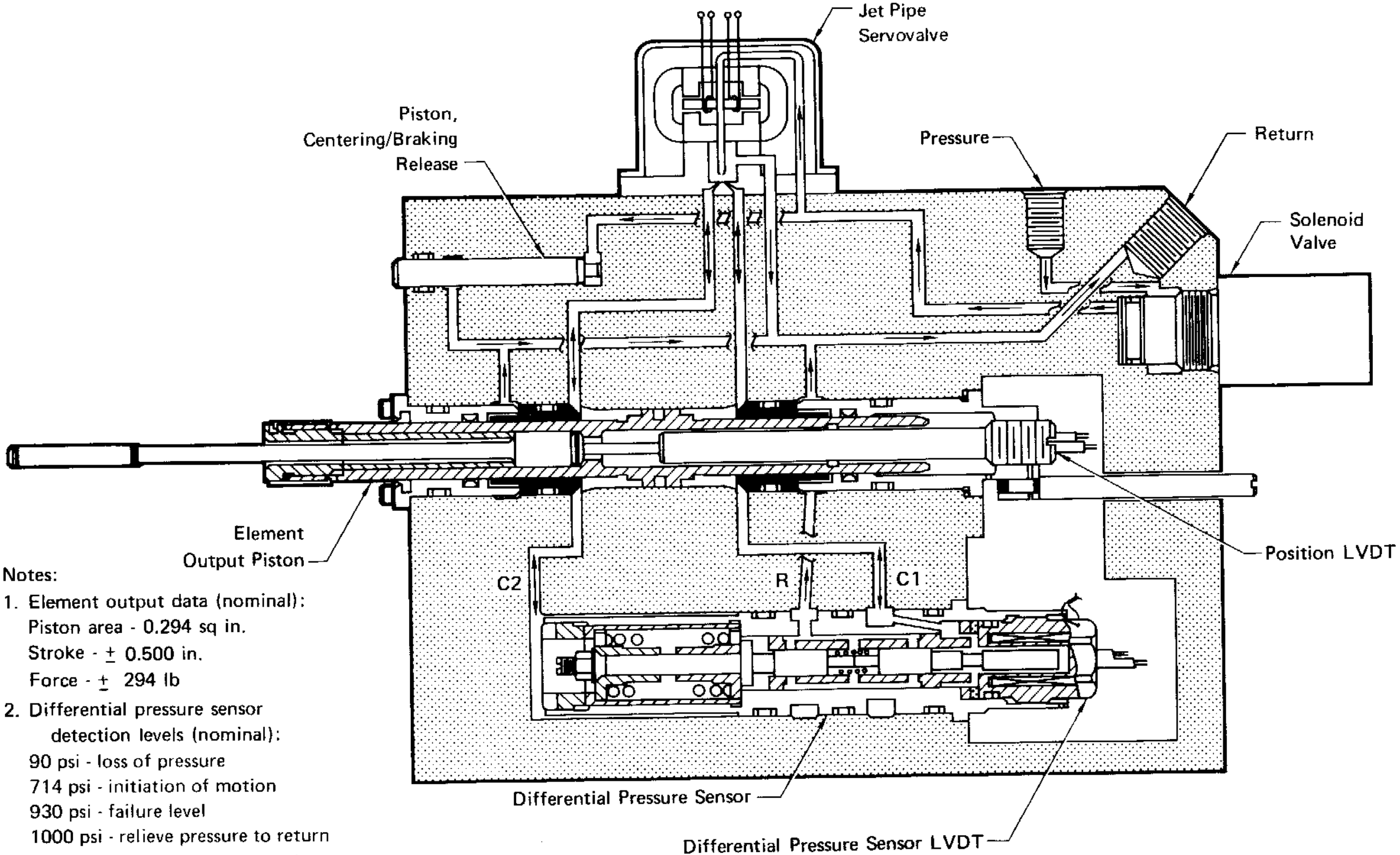
(d) The connection of PC-1 and PC-2 to the SSAP to provide emergency back-up hydraulic power

b. Secondary Actuators

Each Secondary Actuator contains four functionally identical elements which are mechanically tied together to a common output on which is summed the force output of the four elements. A more complete description of how the secondary actuator is mechanized is shown in AFFDL-TR-71-20, Section V. The hydraulic schematic of one element is shown on Figure 15. The pressure to each element is controlled by a solenoid operated shutoff valve. When the solenoid is energized, pressure is supplied to the servovalve and the centering or brake release mechanism.

The servovalve is a single stage jet pipe type which controls the actuator output force and velocity in response to an electric current signal.

A differential pressure (ΔP) sensor assembly is incorporated which senses differential pressure in the cylinder chambers of the element and indicates a failure when the pressure difference across the actuator piston exceeds 930 psi or the pressure in one cylinder chamber falls below approximately 90 psi. The ΔP sensor assembly also serves as a differential pressure relief valve to limit the maximum pressure difference between cylinder chambers C_1 and C_2 to 1000 psi, thus providing load limiting for the secondary actuator elements.



Jet Pipe Servovalve
Piston, Centering/Braking Release
Pressure
Return
Solenoid Valve
Position LVDT
Element Output Piston
C2
R
C1
Differential Pressure Sensor
Differential Pressure Sensor LVDT

Notes:

1. Element output data (nominal):
   Piston area - 0.294 sq in.
   Stroke - ± 0.500 in.
   Force - ± 294 lb
2. Differential pressure sensor detection levels (nominal):
   90 psi - loss of pressure
   714 psi - initiation of motion
   930 psi - failure level
   1000 psi - relieve pressure to return
3. The centering/braking release piston is shown in the locked/braked position. The centering/braking mechanism is spring driven and pressure released.
4. In case of element failure, pressure is shut off by the solenoid and the piston is by-passed through the jet pipe servo valve receiver.

FIGURE 15
HYDRAULIC SCHEMATIC, SINGLE ACTUATOR ELEMENT

(1) Flow Requirements

The equation for the flow requirement, Q, of a secondary actuator element in cubic inches per second (CIS), is $Q = K\sqrt{\Delta P}$, where ΔP is the pressure drop in psi between the supply and return of the secondary actuator element, and K is equal to 2.6×10^{-2}.

When $\Delta P = 3000$ psi $\quad Q = 1.42$ CIS (0.37 gpm)
When $\Delta P = 1270$ psi $\quad Q = 0.93$ CIS (0.24 gpm)
When $\Delta P = 1000$ psi $\quad Q = 0.83$ CIS (0.22 gpm)

(2) Pressure Requirements

Under secondary actuator stall conditions each secondary actuator requires a steady state ΔP greater than 1270 psi in order to recover 950 psi across the actuator piston. 950 psi is the minimum pressure required for failure indication. To meet slew rate requirements, the secondary actuator requires a minimum ΔP of 1000 psi.

c. Mechanical Isolation Mechanism (MIM)

The MIM is used in Phase IIA of the SFCS program to shift control of the surface actuators from mechanical to Fly-By-Wire mode and vice versa in the longitudinal and directional control systems. Further details are presented in AFFDL-TR-71-20, Section III. The MIM is removed for Phases IIB and IIC of the program. The hydraulic schematic of the MIM is shown in Figure 16.

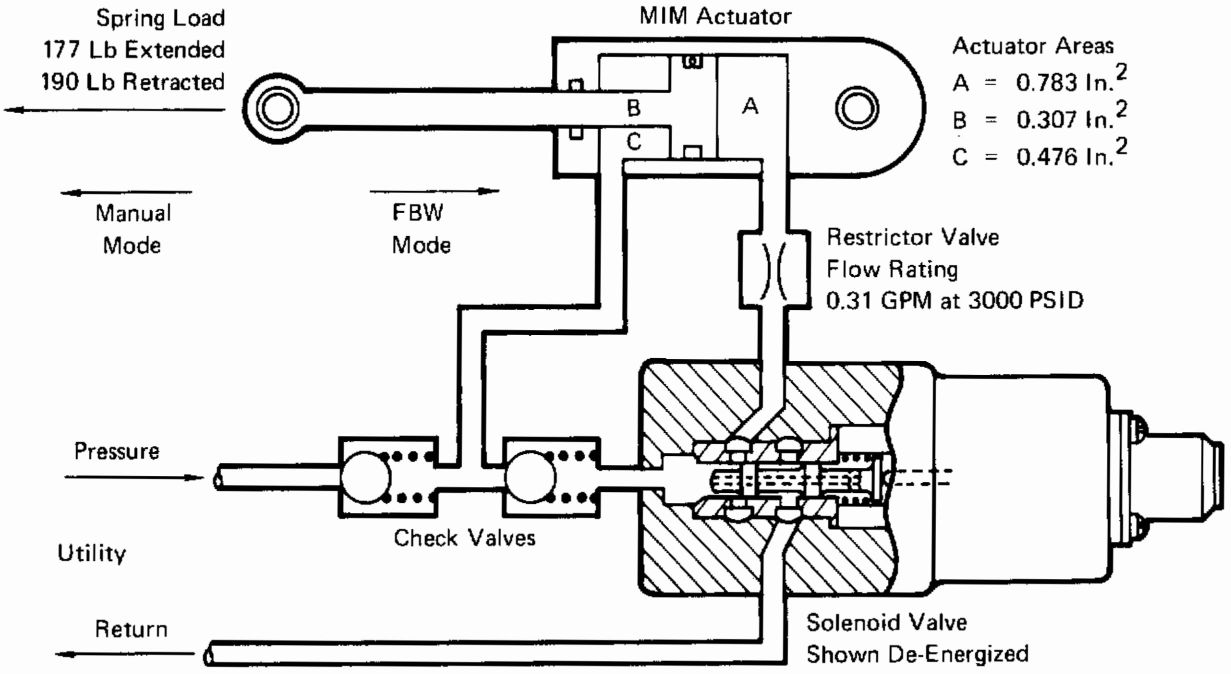


FIGURE 16
MIM HYDRAULIC SCHEMATIC

The utility system was chosen as the hydraulic supply for MIM actuation instead of PC-1, PC-2, or APU in order to obtain continuity of control for a greater period of time under engine out conditions. A single engine out condition results in the loss of the PC hydraulic system powered from that engine and therefore could result in an undesired shift of the MIM. The use of the APU hydraulic system was eliminated because the MIM would shift to mechanical back-up mode due to loss of AC electric power shortly after a dual engine flameout. Figure 17 illustrates typical engine RPM decay characteristics that may be expected after an engine flameout. The electrical generators are automatically disconnected from their associated bus at approximately 54% engine RPM. The three engine driven hydraulic supplies maintain usable pressure to a lower engine speed than that required to retain the electrical generators "on line".

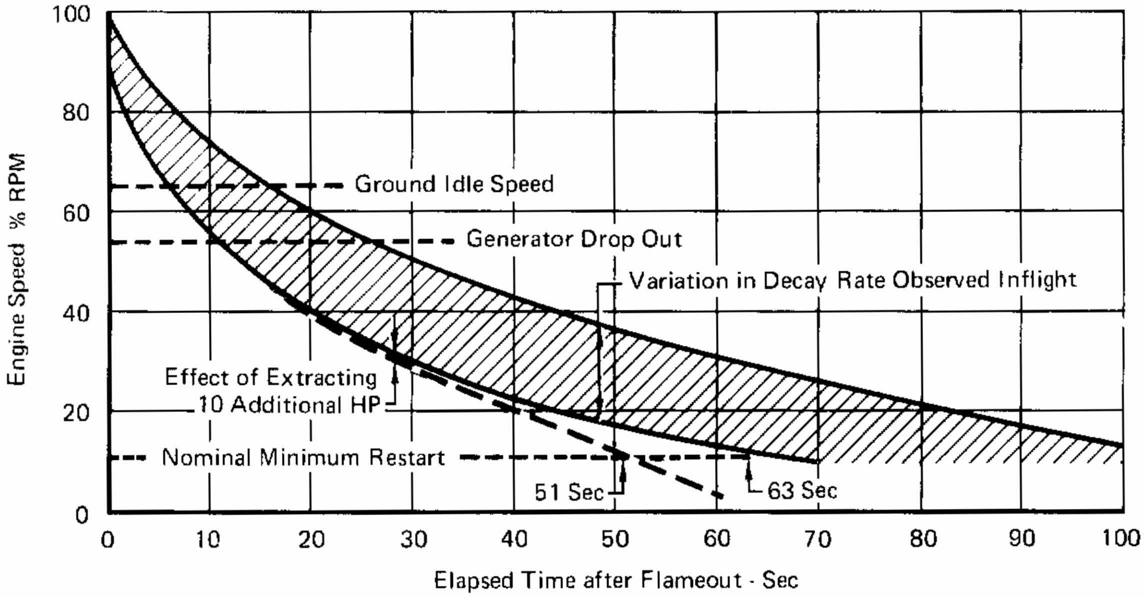


FIGURE 17

ENGINE RUN-DOWN CHARACTERISTICS
Altitudes = 5,000 Ft thru 55,000 Ft
Mach No. = 0.3 thru 1.2

Each MIM is mechanized so that an interruption of electrical power to the MIM solenoid valve will remove hydraulic pressure from one side of the actuator and allow the MIM to be shifted to the mechanical back-up position by a preloaded spring. Pilot action is then required to re-engage the Fly-By-Wire control mode. Figure 18 illustrates the electrical control circuitry planned for the MIM system.

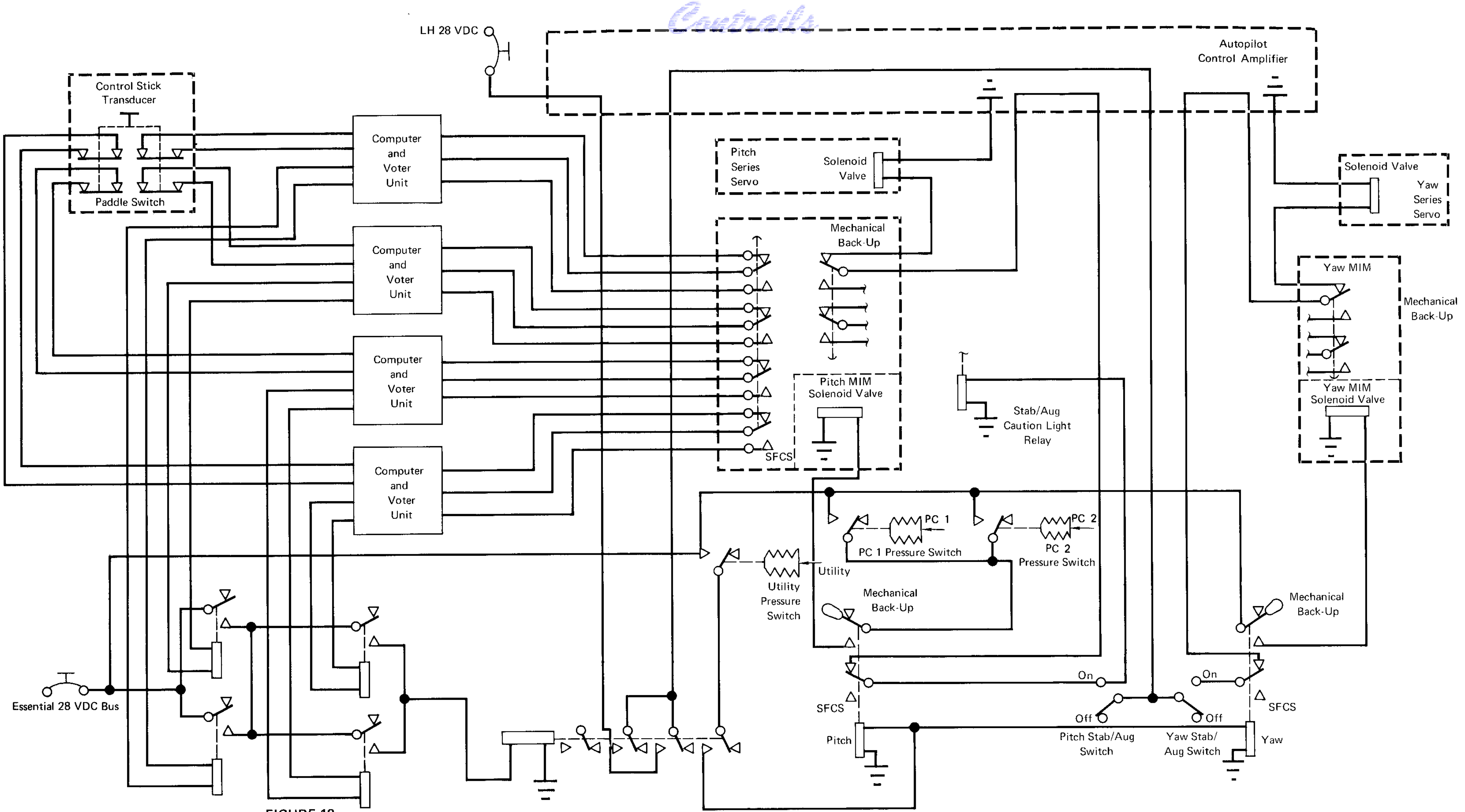


FIGURE 18
SFCS,STABILITY AUGMENTATION, AND MIM SWITCHING SCHEMATIC

Electrical power is supplied to each MIM solenoid circuit only when (1) utility hydraulic pressure is above 1000 psi, which will hold a pressure switch closed and (2) when the pilot operated, solenoid held SFCS/MECH Back-up switches are in the SFCS position. The solenoid-holding action of these switches may be interrupted by momentary operation of the control stick paddle switch. This action results in reversion of both MIM's to the mechanical back-up mode. In addition to the pilot operated and the utility pressure operated switches, the Pitch MIM also requires that the pressure in either PC-1 or PC-2 holds a PC pressure switch closed. Figure 18 also shows the circuitry which provides the Stability Augmentation function, which may be engaged when the pilot selects the Mechanical Back-up mode.

Hydraulic fluid flow required to actuate each MIM is approximately 0.5 cubic inches per second for 2.5 seconds when activating the system.

d. Rudder Back-up System

A rudder back-up supply is considered necessary to improve the reliability of the directional control system and give it fail operate performance in the unlikely event of complete loss of the lateral control system. Rudder deflection also is required to counter the effects of asymmetric thrust that would occur with the loss of a single engine. Therefore the fourth hydraulic system, which uses the same type of hydraulic fluid (MIL-H-5606) as the utility system was selected as the rudder back-up supply because it continues to operate after the loss of a single engine. The loss of a single engine causes the loss of a PC system; therefore, the use of a PC system for rudder back-up is not as acceptable.

The rudder back-up supply is pilot selected by a switch in the cockpit. Automatic switching was considered and discarded because of the difficulty of preflight testing and ground operation of the system.

The switching valve, see Figure 19, is pilot pressure operated with pilot pressure referenced to the fourth hydraulic system. A minimum pressure of 250 psi is required for valve actuation to select the back-up system.

(1) Flow Requirements

The flow requirement of the rudder surface actuator at maximum unloaded surface rate is approximately 3.8 CIS (0.99 gpm). The steady state quiescent leakage is 0.69 CIS (0.18 gpm) which is the combined leakage of the series servovalve and the main control valve.

(2) Pressure Requirements

The rudder surface actuator normally operates from a 3000 psi supply and will operate in normal power mode down to approximately 250 psi. At this pressure the mechanical input to the surface

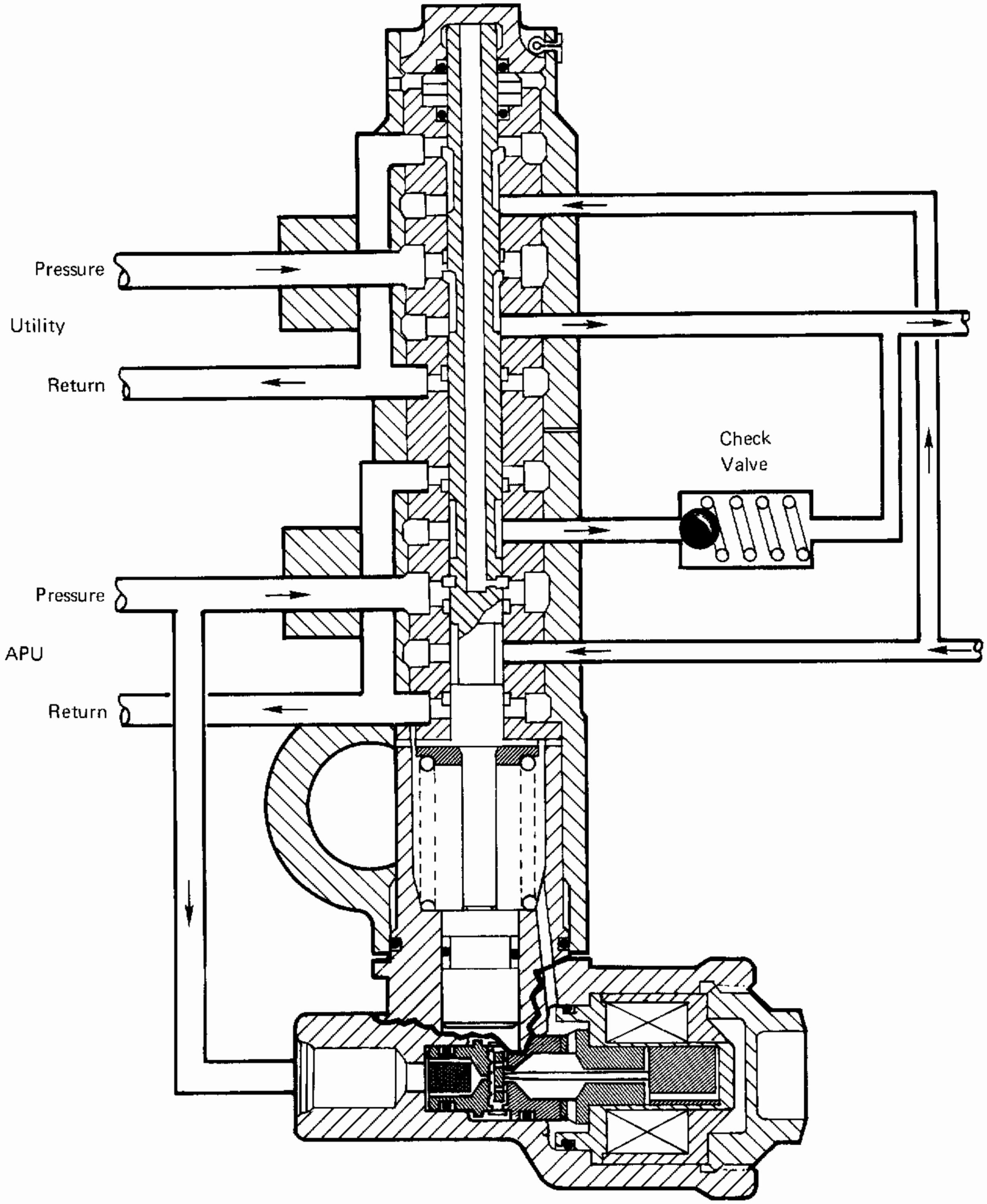


FIGURE 19
RUDDER BACK-UP SWITCHING VALVE
(Shown De-Energized with Utility System
Supplying the Rudder)

actuator is clamped to the mechanical input and is then able to drive the surface directly. It is intended to use the directional control system as an emergency back-up in the event of a total lateral system failure; however, this is predicated on the availability of a hydraulically powered rudder surface actuator. The Secondary Actuator force output with three elements operating is essentially the same as the manual force output in a conventional F-4. The output pressure of the fourth hydraulic system will provide approximately one half of the normal rudder hinge moment. However, at landing speeds the pressure available from the fourth hydraulic system is sufficient for full rudder deflection.

e. Survivable Stabilator Actuator Package (SSAP)

The SSAP replaces the production stabilator actuator and the longitudinal secondary actuator in Phase IIC of the SFCS program. Two integrated motor pump units normally supply the hydraulic power to the SSAP for control of stabilator surface position. The secondary actuator that is part of the SSAP is electromechanical and does not require hydraulic power. Each integrated motor pump system is backed up by one of the aircraft PC hydraulic systems. If a motor pump fails, a PC system is automatically switched in to replace the failed system.

When operating on the back-up system the hydraulic power requirements for the SSAP surface actuator are approximately the same as for the production stabilator actuator. When operating on the integral motor pump systems there is no hydraulic demand on the PC systems. Consequently no changes in PC system power capabilities are required for Phase IIC.

The maximum SSAP operating temperature is expected to be in the region of 350°F to 450°F, hence the fluid used in the SSAP must be capable of operating in this temperature environment. The fluid selected is MIL-H-83282 (MLO 68-5). For further information regarding the fluid selection see Section V of this report.

Fluid in the surface actuator LRU of the SSAP may be interchanged between the integrated hydraulic systems and the PC systems; therefore all must use the same fluid.

Furthermore, it must also be recognized that little information is available concerning long life seal materials for Type III hydraulic systems. For the SSAP the following philosophies and concepts have been established.

o Minimize "O" ring seals as far as possible.

o Use latest state of the art high temperature "O" rings where "O" rings must be used. For example a compound, Parker 747-75, similar to Viton but with better compression set characteristics, is utilized for all "O" rings.

- Use dual redundant techniques on dynamic seals.
- An air to fluid heat exchanger is employed to reduce the time at high temperature and attempt to limit maximum fluid temperature to something below 450°F.

3. ANALYSIS OF THE CENTRAL SYSTEMS

a. Hydraulic Supplies on Production F-4's

(1) PC-1 Hydraulic System

The PC-1 hydraulic supply is furnished by a constant pressure, variable delivery pump on the LH engine. The pump maximum flow versus pump rpm is as shown on Figure 20. The nominal pump displacement is 1.54 cubic inches per revolution giving a maximum flow of 75 CIS(19.45 gpm) at engine cruise RPM, which corresponds to a pump pad speed of 2950 RPM. The pressure versus flow relationship is shown on Figure 21. Within operating limits this relationship is independent of pump RPM.

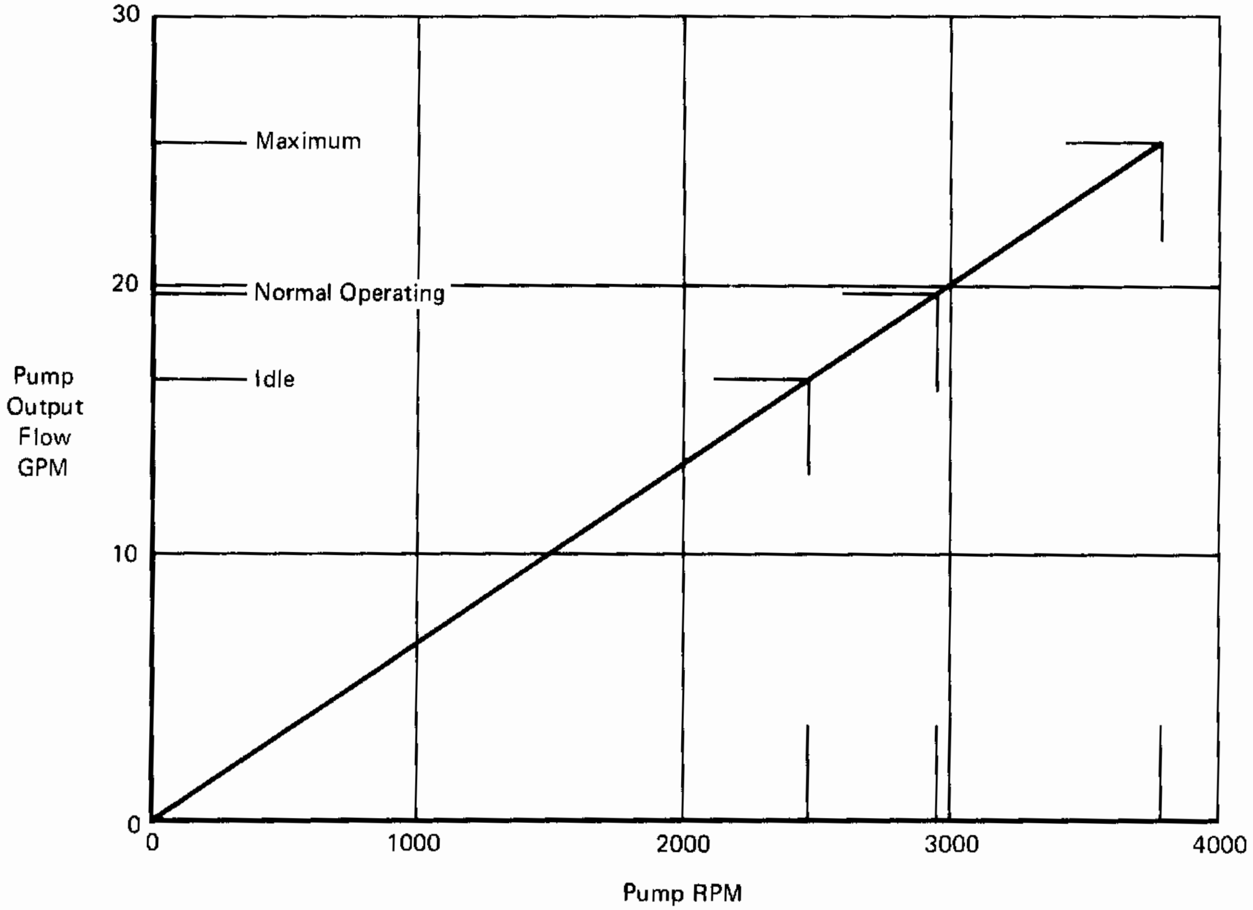


FIGURE 20
PC-1, PC-2 PUMP OUTPUTS

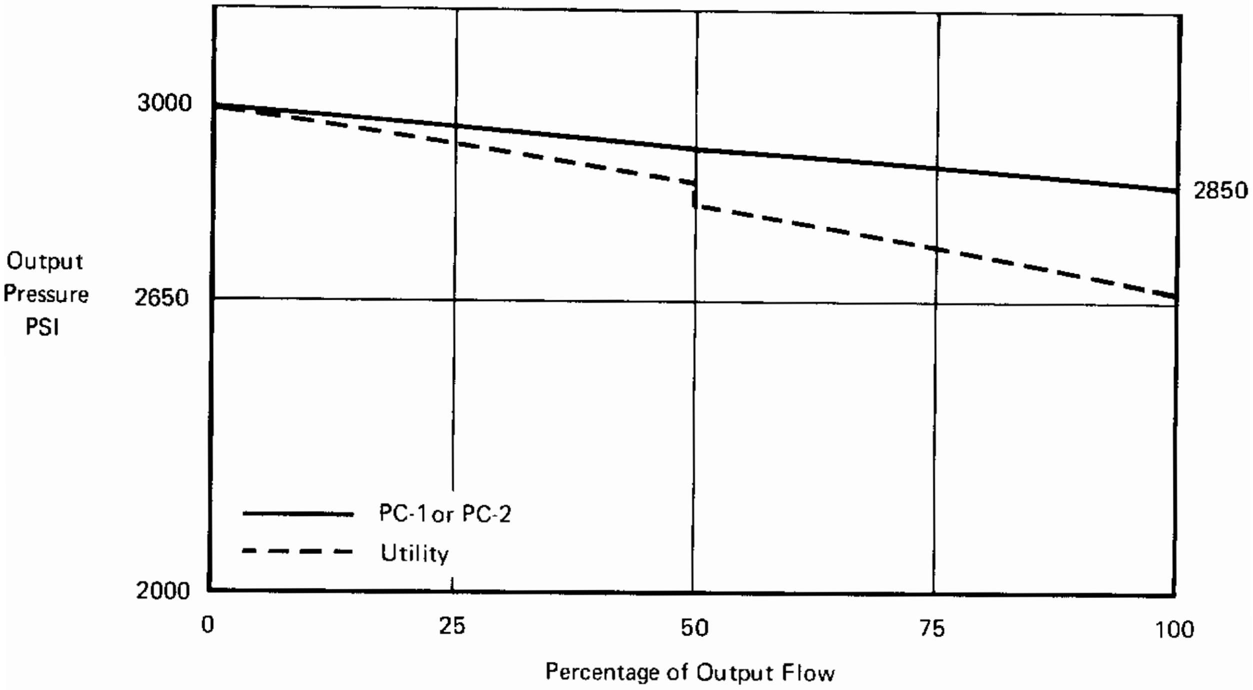


FIGURE 21
PC-1, PC-2 AND UTILITY PRESSURE/FLOW RELATIONSHIP

(2) PC-2 Hydraulic System

The PC-2 pump is driven by the RH engine and is identical to PC-1 with the same pressure-flow characteristics.

(3) Utility Hydraulic System

The utility supply is obtained from two pumps, one driven by each engine. The pump on the LH engine normally supplies the system flow requirements. Flow from the pump on the RH engine is not used until the supply pressure drops below 2,800 psi. The 2800 psi level is normally reached when the LH engine pump is operating at or near maximum flow. At this pressure level the RH pump capacity is added to the utility system.

The output flow capability of the utility system at any specific engine RPM is twice that of one PC system assuming that both engines are running at the same RPM. The output pressure vs. flow relationship is shown on Figure 21.

b. Effects of Additional Loads

PC-1, PC-2 and Utility will have the additional steady state loads of four secondary actuator elements which require a flow 5.7 CIS (1.48 gpm) from each system. This causes an increase of 2.6 HP in the steady state load on each system, but will have negligible effect on the maximum system flow demands listed in Table VIII. Most of the heat generated due to the additional loads will be taken

TABLE VIII
SURFACE ACTUATOR CHARACTERISTICS

Actuator		Maximum Flow CIS			Quiescent Flow CIS			Actuator Rate	Surface Rate	Actuator Area (Sq In.) (A1 is Rod End)			
		PC-1	PC-2	Utility	PC-1	PC-2	Utility	In./Sec	°/Sec	A1	A2	A3	A4
Stabilator	Extend	53.2	50.7		0.6	0.6		8.6	25	5.38	5.89	5.89	6.18
	Retract	50.7	46.3					8.6	25	PC-2	PC-2	PC-1	PC-1
SSAP	Extend	54.8	49.5		0.33	0.33		8.6	25	5.77	5.77	5.13	6.38
	Retract	44.2	49.5					8.6	25	PC-2	PC-2	PC-1	PC-1
Rudder				3.8			0.69	5.3	67.5	0.53	0.53		
				3.8						Util	Util		
L Aileron	Extend	49.6		49.6	0.5		0.5	6.32		2.44	7.86	2.44	7.86
	Retract	15.4		15.4				3.77		PC-1	PC-1	Util	Util
R Aileron	Extend		49.6	49.6		0.5	0.5	6.32		2.44	7.86	2.44	7.86
	Retract		15.4	15.4				3.77		PC-2	PC-2	Util	Util
L Inboard Spoiler	Extend	17.9		17.1	0.5		0.5	13.9	150	0.966	1.288	1.032	1.228
	Retract	12		12.8				12.4		PC-1	PC-1	Util	Util
L Outboard Spoiler	Extend	19.6		19.0	0.5		0.5	9.88	150	1.415	1.987	1.4	1.92
	Retract	12.1		12.0				8.6		PC-1	PC-1	Util	Util
R Inboard Spoiler	Extend		17.9	17.1		0.5	0.5	13.9	150	0.966	1.288	1.032	1.228
	Retract		12	12.8				12.4		PC-2	PC-2	Util	Util
R Outboard Spoiler	Extend		19.6	19.0		0.5	0.5	9.88	150	1.415	1.987	1.4	1.92
	Retract		12.1	12.0				8.6		PC-2	PC-2	Util	Util

out by the oil/fuel heat exchangers and will not cause any significant increase in fluid temperature. In the case of a flameout condition, these extra loads could be significant, since the PC and utility systems would be extracting more power from the windmilling engine than without these loads. The additional load demand from each system would be 5.7 CIS (1.48 gpm) requiring an additional 5.2 HP from the left engine and 2.6 HP from the right engine. The effect of an additional 10 HP load on an engine and the reduction in the time available for restart is shown on Figure 17 and indicates the time during which a restart must be accomplished is reduced from 63 seconds to 51 seconds, a 12 second reduction. For the purposes of this study, it can be assumed that the additional 5 HP load would have approximately half the effect of the 10 HP load and would reduce the restart time available from 63 to 57 seconds. From the above it can be concluded that the effects of the additional hydraulic loads imposed on the PC-1, PC-2, and Utility systems for Phases IIA and IIB of the SFCS program are negligible.

4. ANALYSIS OF THE FOURTH HYDRAULIC SYSTEM

A fourth hydraulic system is required to supply one element of each of the pitch, yaw, and lateral left and right secondary actuators. It will also supply back-up power for the rudder surface actuator. The power for the system will be obtained from an Auxiliary Power Unit (APU) driven by a 115/200 volt 3 phase 400 Hz motor. The APU is the same unit that is currently being used on the F-4E as a back-up hydraulic power source for the stabilator actuator. It is a Vickers APU, heavy-duty motor version, model MPEV3-011-15. The APU has completed actual qualification testing per Vickers Report No. 8-0643-268-310, which MCAIR is presently reviewing. It is important to note that the endurance section of the test consisted of 750 hours of running with a considerable portion at 275°F. The electrical power for the APU will normally be obtained from the LH bus. If an electrical power failure occurs in the LH bus then the RH bus is automatically switched in to provide electrical power.

a. Flow Requirements

The flow in each secondary actuator element is 0.97 CIS (0.25 gpm) at 1400 psi; therefore a total flow of 3.88 CIS (1.00 gpm) for all four secondary actuators is required. When the rudder is in the back-up mode, the steady state flow will be increased to 4.55 CIS (1.18 gpm) and the maximum flow demand will be 7.7 CIS (2.00 gpm).

b. Output Pressure

The output pressure versus flow characteristic of the APU is shown in Figure 22. With a flow demand of 3.88 CIS (1 gpm) due to the four secondary actuators, the nominal steady state output pressure will be 1400 to 1600 psi. The additional load of the rudder surface actuator will cause the supply pressure to drop 200 psi below nominal steady state levels during maximum demand and 50 psi below nominal steady state levels under quiescent conditions. The maximum flow

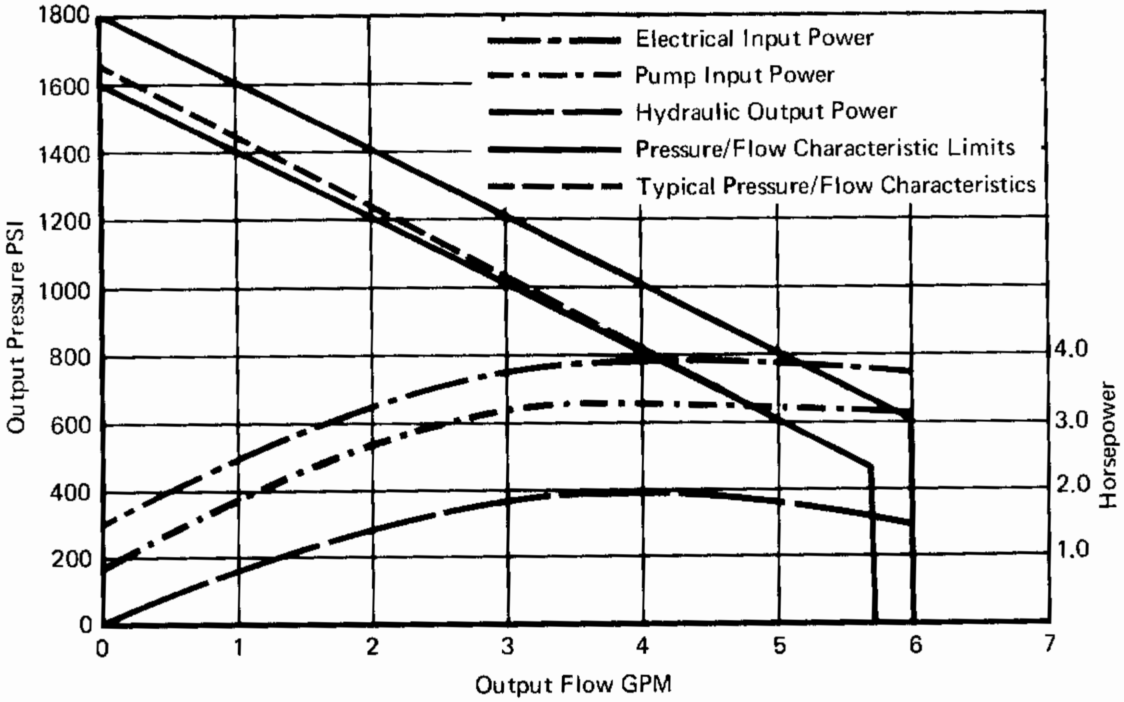
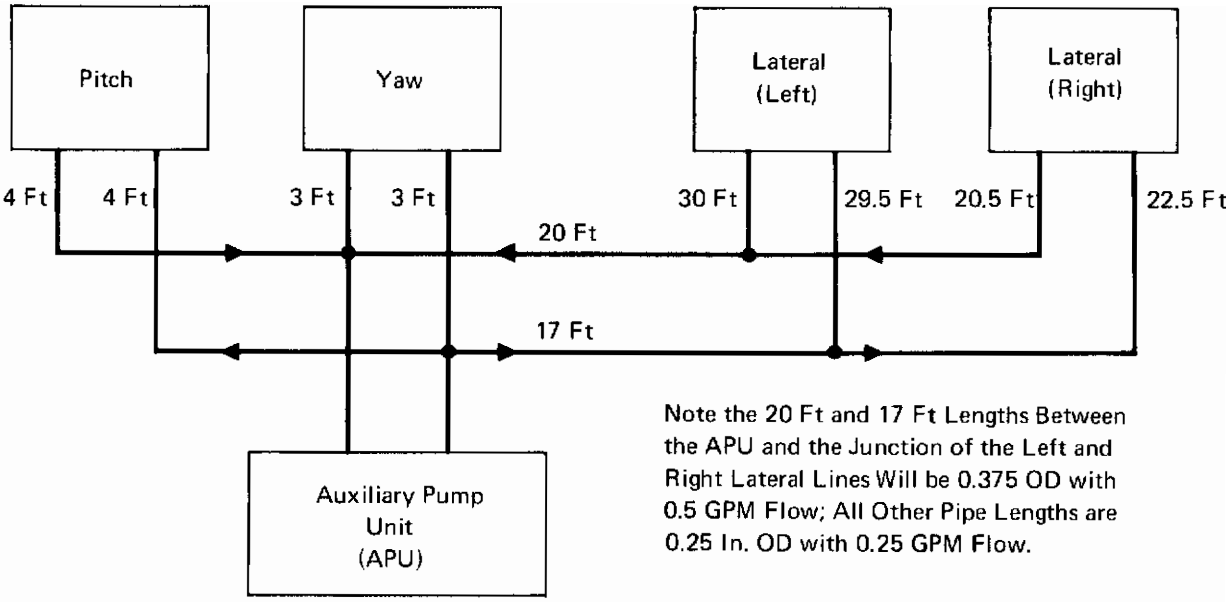


FIGURE 22
APU PRESSURE/FLOW CHARACTERISTICS

rate is based on an unloaded surface actuator and consequently is conservative. Lower load flow demands will produce a proportionally smaller drop in pressure. Transient pressure drops due to this load will not affect the secondary actuators because of their duration which is less than the one-half second delay required for a disconnect to occur.

c. Tubing Pressure Drop

The pressure drops due to the tubing lengths of the system have been calculated and are presented in Figure 23. The effects of the fitting and bends are incorporated by using an equivalent length of tubing and adding this to the straight line length. The viscosity, temperature and pressure relationship in the tubing is very complex and hence it is not practical to analyze the system with the true viscosity distribution. In general higher pressure in the supply line will cause an increased pressure drop due to the increase in viscosity with pressure. However, the fluid temperature will also be slightly higher thus compensating for the viscosity increase. In the return line the pressure will be about 50 psi and will have very little effect on the viscosity. The fluid will be losing heat due to convection and radiation from the tubing, thus given a gradual increase in viscosity in the direction of flow. Based on the available data it is assumed that the line pressure drops calculated on the basis of a constant temperature and zero pressure are sufficiently accurate to provide fluid at the differential pressures and flow required at the secondary actuators. Figure 23 shows the pressure available at the secondary actuators at the required flow rate and the data used for the calculations.



Note the 20 Ft and 17 Ft Lengths Between the APU and the Junction of the Left and Right Lateral Lines Will be 0.375 OD with 0.5 GPM Flow; All Other Pipe Lengths are 0.25 In. OD with 0.25 GPM Flow.

Fluid Temperature	$-20^{\circ}F$	$+20^{\circ}F$	$60^{\circ}F$	$100^{\circ}F$
ΔP At the Secondary Actuators with 1400 PSI ΔP at the APU				
Pitch	1353	1386	1394	1397
Yaw	1365	1389	1396	1398
Lateral Left	966	1268	1347	1370
Lateral Right	1063	1298	1359	1378
Line Pressure Drop PSI/Ft				
0.25 OD 0.02 Wall 0.97 CIS at 0 PSI	5.9	1.8	0.73	0.41
0.375 OD 0.028 Wall 1.93 CIS at 0 PSI	2.25	0.68	0.28	0.155

FIGURE 23

FOURTH HYDRAULIC SYSTEM LINE LOSSES FOR MIL-H-5606 FLUID

5. RELIABILITY ANALYSIS OF THE FOURTH HYDRAULIC SYSTEM

The fourth hydraulic system affects SFCS reliability primarily in three areas:

- It provides the required two-fail-operate capability for the SFCS secondary actuators in all three axes.
- In the directional system it enables fulfillment of the Military Specification requirement that aircraft employing powered or boost-control systems be provided with a suitable means for control following loss of primary power or boost.
- It reduces the probability of aircraft loss due to loss of control in the longitudinal and directional axes.

a. Two-Fail-Operate

Using majority voting at least three signals are required to identify the channel which has failed. If two-fail operate capability is required, then at least three signals must remain after the first failure. Consequently, four channels of signals are required before the first failure. Four SFCES signals are received by the secondary actuators, thus providing two-fail-operate capability to the input of the secondary actuators. With four active isolated sources of hydraulic power available, the two-fail-operate capability is extended to include the secondary actuator in all three axes.

b. Redundant Power Source

Military Specification MIL-F-9490 requires that aircraft employing powered or boosted control systems shall be provided with suitable means for control following complete loss of primary power or boost. In the production F-4, the longitudinal control system satisfies this requirement by using a dual tandem control surface actuator, powered by two independent hydraulic systems. The lateral system uses a 4 barrel aileron surface actuator and dual tandem spoiler actuators powered by two independent hydraulic systems. The directional control system is provided with a mechanical back-up should utility hydraulic power be lost. In the SFCS program the rudder mechanical back-up capability is removed with the removal of the directional MIM at the start of Phase IIB. The use of the fourth hydraulic system as a back-up for the utility hydraulic system at the rudder control surface actuator provides compliance with the MIL-F-9490 requirement.

c. Estimated Reliability

The effect of the fourth hydraulic system on estimated reliability, in terms of total loss of control, varies with the axis being considered. The effect on the longitudinal axis during Phase IIA would be minor because the secondary actuator is paralleled by the conventional mechanical system. In Phase IIB it becomes more important because removal of the mechanical system makes it possible for complete longitudinal failure in one hour due to loss of hydraulics from 7.91×10^{-9} to 7.64×10^{-9}. This is based on an estimated APU failure rate of 88 failures per million hours. Since the SSAP secondary actuator is not hydraulically operated, the fourth hydraulic system has no effect on longitudinal axis reliability in Phase IIC. In the lateral axis, both secondary actuators must fail before lateral control is lost. This probability of failure is so small that the effect of the fourth hydraulic system is negligible. A major reliability improvement takes place in the directional axis. The use of a second power source reduces the one hour probability of loss of rudder control due to the loss of the hydraulics system from 180×10^{-6} to 3.2×10^{-6}, an improvement of 60:1.

6. THERMAL ANALYSIS OF THE FOURTH HYDRAULIC SYSTEM

a. General

A thermal analysis of the fourth hydraulics system has been performed. An approach is established and identified herein whereby acceptable temperatures can be maintained in the fourth hydraulic system. The APU fluid and motor temperatures are acceptable for Phase IIA and IIB. Data from Phase IIA and IIB will be used to update the predicted temperature for Phase IIC during which the highest temperatures are predicted to occur. The APU motor winding temperature during Phase IIC may exceed the Supplier's recommended maximum by 25°F to 50°F This temperature excess is considered acceptable, however, because its occurrence is infrequent, short in duration, and does not significantly reduce winding life. Acceptable fluid temperatures for Phase IIC can be achieved by taking the following precautionary measures:

(1) SSAP ground operations with access doors closed must be monitored and limited to prevent excessive temperatures.

(2) Silicon grease will be installed between elements of the secondary actuators to assist in transferring heat from the fourth hydraulic system to the central hydraulic systems.

(3) The temperature instrumentation on the fourth hydraulic system must be monitored during ground and flight tests.

(4) Operation of the fourth hydraulic system with one or more of the central hydraulic systems or one or more of the fourth hydraulic system secondary actuator elements shut off or unpowered, might require limiting the time at some extreme flight conditions. The particulars of the limitations depend upon how many of the elements are inoperative, what the initial temperatures are, and the flight condition. However, operation in all configurations and all flight conditions is permissible even though the allowable duration may be reduced under these circumstances.

b. Hydraulic Fluid

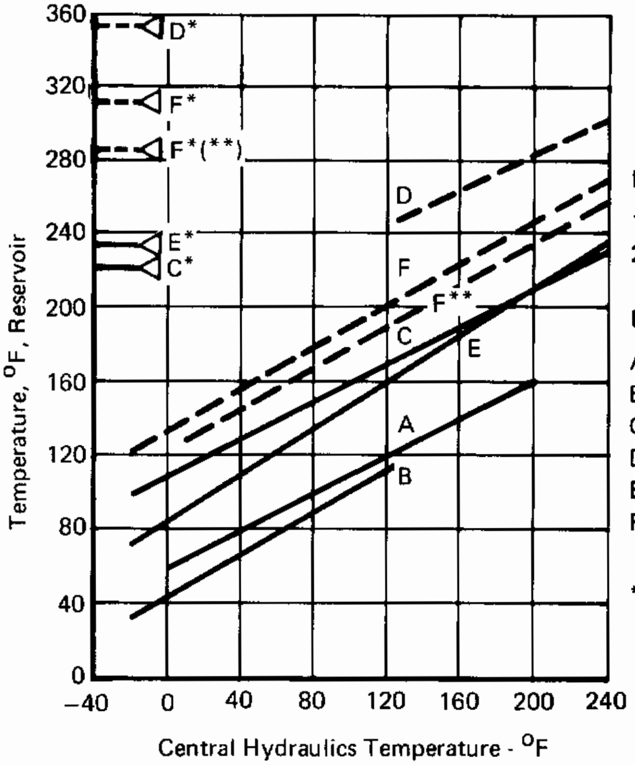
The nominal maximum allowable temperature of the fourth hydraulic system is 275°F, based on MIL-H-5440 Class II limits and the use of MIL-H-5606 fluid. The APU case drain temperature is expected to be about 50°F higher than the APU reservoir temperatures. Short duration temperatures in the case drain discharge above 350°F were experienced during Phase I quality assurance tests using the same basic Vickers pump, and without apparent pump damage. The case drain temperature of 325°F associated with a reservoir high temperature limit of approximately 275°F should therefore be acceptable. The operating minimum allowable fluid temperature is about +20°F, because the high fluid viscosity at low temperatures causes line pressure drops to increase, producing marginal inlet pressure at the secondary actuator. During normal operation, the two means presented below serve to hold the temperature of the fourth hydraulic system within allowable levels.

(1) The central hydraulics systems serve as the major heat sink for the fluid of the fourth hydraulic system. Their temperature determines to a substantial measure the temperature of the fluid in the fourth hydraulic system. The heat reaches this sink by being transferred by convection from,the fluid to the passage walls of the fourth hydraulic systems secondary actuator elements then by conduction throughout the secondary actuator assemblies, and finally by convection through the walls of the secondary actuator elements to the aircraft central hydraulic systems. Silicon grease could be applied between elements to attain the predicted conduction through the secondary actuators if the gap between them is large. To establish an analytical basis for predicting the worst case high extreme temperatures, the assumption is made that all three central hydraulic systems are disabled or deactivated at all the secondary actuators. In this condition, the central hydraulic systems are effectively isolated from use as a heat sink for the fourth hydraulics system. This condition is inconceivable in flight; however, it is considered to be a valid assumption for analytical purposes since it could occur on the ground. If the transient temperature consequences are acceptable, as predicted herein, the need for considering individually other less severe spatial arrangements of single and dual central systems failures is eliminated. The assumption adds a measure of conservatism to an analysis involving a great number of complex peripheral problems.

(2) Supplementing the heat transferred to the central hydraulic systems is the heat transferred from the secondary actuators and the other portions of the fourth hydraulic system to their local compartment air and adjacent structure.

Figure 24 presents the results of the thermal analysis in terms of steady state reservoir fluid temperature, central hydraulics fluid temperature, and several aircraft flight conditions.

Flight conditions were selected for analysis which variously produced low, high and normal temperatures. Of the conditions selected the following are presented because they are most indicative of fourth hydraulic system thermal capability.



Notes:

1. Steady state operation.
2. Three central hydraulics systems except where noted.

Legend:

A. Low temperature operation; sea level; 0°F day; IIA,B
B. Low temperature operation; 31,000 ft; cold day; IIA,B
C. Ground static; 125°F day; IIA,B
D. Ground static; 125°F day; IIC
E. Cruise; 45,000 ft; ANA hot day; IIA,B
F. Cruise; 45,000 ft; ANA hot day; IIC

*Without central hydraulic flow

**With 50% surplus ram airflow

FIGURE 24

FOURTH HYDRAULICS SYSTEM FLUID TEMPERATURES

(a) During ground operations, when the outside air temperature is 125°F, the central hydraulics systems could reach 175°F. Accordingly, during Phase IIA and IIB, the steady state APU reservoir temperature is expected to be approximately 225°F. For Phase IIC, the steady state APU reservoir temperature is expected to be approximately 272°F, assuming no stabilator activity. With the central hydraulics deactivated at the secondary actuators, the Phase IIC temperature is predicted to be 355°F.

(b) During cruise at Mach 0.94 and 45,000 feet in an ANA hot day atmosphere, assuming the central hydraulics fluid temperature is 85°F, the APU reservoir temperatures are estimated at 149°F for Phase IIA and IIB and 173°F for Phase IIC. With the central hydraulic systems deactivated at the secondary actuators, the steady state temperature for Phase IIC could be 312°F. If during Phase IIC a cruise flight condition, with the central hydraulics deactivated follows the high speed dash flight condition of the following paragraph, the cruise could continue for an estimated 1 hour before the reservoir temperature reaches 275°F.

(c) During a high speed dash, starting from the stabilized cruise condition of the previous paragraph in which all central hydraulic systems are active, the APU reservoir temperatures are not expected to exceed 180°F during Phase IIA and IIB and 200°F during Phase IIC. More severe than this high speed dash, but not expected to be a requirement for flight tests, is a rapid transition to a dash condition above 40,000 feet altitude after an extended period of operation on the ground during high temperature weather.

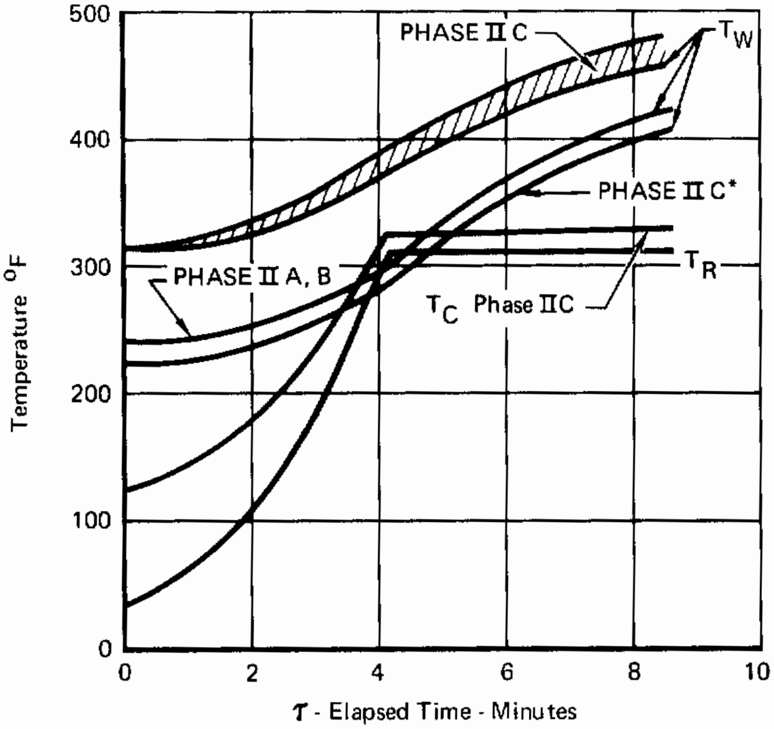
(d) The lowest in-flight reservoir fluid temperatures are expected to occur between 25,000 and 40,000 feet on a cold day. This expectation is based on the fact that in the general locale of where this program's flights are to originate, Edwards Air Force Base and St. Louis Lambert Field, standard cold day atmospheric temperatures do not extend to very low altitudes. Otherwise, the lowest inflight temperatures could occur during a minimum speed cold flight at sea level. A cursory analysis of fragmentary central hydraulic system temperature data previously recorded suggests that the system temperature in flight may drop to about -20°F to 0°F. The APU reservoir temperature therefore, will drop to about 40 to 50°F.

(e) The lowest reservoir fluid temperature during ground operations is expected to occur at start up. Although expected temperatures at ground level will not approach the standard cold day sea level extreme of -60°F they could be below the 20°F estimated minimum allowable for the fourth hydraulic system. If, at start up, IFM or BIT indicates inadequate pressure attributable to low ambient temperature, delaying the flight until normal warm-up of the fourth hydraulic system has occurred is a possible solution. When cold weather prevails, such a delay could be avoided by routinely turning on the fourth hydraulic system an adequate period of time prior to commencement of BIT.

The fact that the thermal time constant of the fourth hydraulic system is very large will not only help prevent the occurrence of an excessively high or low temperature, but also will help assure early detection of their impending occurrence by permitting trends to be observed in the flight test telemetry. The low temperature analysis is adequate to indicate that fourth hydraulic system fluid temperatures below 20°F are unlikely.

c. Electric Motor

The APU electric motor is cooled by a self-contained fan pulling aft fuselage air over the motor. The motor temperatures are therefore a function of aft fuselage air temperature, air density and motor loading. The motor loading during Phase IIA and IIB, with four electrohydraulic secondary actuators operating, is slightly more than 50% of full rated load, and during Phase IIC, with only three operating, it is slightly less than 50%. However, the aft fuselage compartment air is much hotter during Phase IIC because of the heat produced by the SSAP. The higher temperature more than offsets the reduced heat losses resulting from the reduced loading. Figure 25 presents an analytically developed time history of electric motor winding temperature during an acceleration to, and flight at, Mach 2.204 at 50,000 ft. from temperature stabilization at Mach 0.94 on an ICAO Standard Day. The last 4.4 minutes of the dash are at the peak Mach number with a ram air temperature of 310°F. During Phase IIA and IIB the winding temperature is estimated to increase to no higher than 420°F at the end of the dash. During Phase IIC the winding temperature could increase to approximately 455°F to 480°F at the end of the dash. By directly ventilating the electric motor with ram air to bypass the effect of SSAP compartment heating, the peak winding temperature during Phase IIC could be reduced to about 405°F. The motor used in the APU is the same or equivalent design as the one used on the Simplex package. The motor transient thermal characteristics are derived from Simplex motor temperature data recorded during Phase I of the SFCS Program as reported in Reference 2 . The maximum motor winding temperature recommended is 428°F, according to Simplex motor data obtained from Vickers during Phase I. Vickers stated that the winding life at this temperature is 10,000 hours and at 550°F is 1000 hours (no confidence level is stated). The Essex Wire Corporation, who provide the motor winding wire, has data which indicate that the average thermal life of the winding is 20,000 hours or more at 464°F. This temperature corresponds approximately to the maximum temperature expected in Phase IIC. Based on these data and both the infrequency and the short duration of the maximum speed dash condition, the predicted maximum winding temperature is considered acceptable.



Final T_W at $\tau = \infty$	Phase II A, B	518°F
	Phase IIC	527°F
	Phase IIC*	498°F

NOTES

1. T_W Winding Temperature
2. T_C Compartment Temperature
3. T_R Ram Air Temperature
4. * Indicates Temperature is Based on Ducting Ram Air to Motor Inlet to Achieve Same Motor Ambient Temperature as in Phase II A and II B
5. Acceleration from 0.94 to 2.204 M at 50,000 Ft Pressure Altitude, ICAO Standard Day
6. 50% Surplus Ram Airflow, Phase II C

FIGURE 25

FOURTH HYDRAULIC SYSTEM MOTOR TEMPERATURES DURING HIGH SPEED DASH

SECTION V

FLUID STUDY

1. GENERAL

The SSAP to be used in Phase IIC of the SFCS program will be powered by two motor-pump-reservoir systems integrated into the package. Each integrated hydraulic system will be backed up for emergency use by one of the existing F-4 central hydraulic systems. Studies made of the high power requirements of the SSAP in conjunction with its location in a hot portion of the airplane indicate that temperatures in the package could get as high as 450°F. This is discussed more fully in the SSAP degree of redundancy study in Section III. At this high temperature, close attention must be given to the characteristics of the hydraulic fluid to be used. Many of the fluid characteristics are so degraded that they become important parameters in designing the system. Consequently a study was undertaken to select a fluid which would meet with the SSAP requirements and still be compatible with the existing hydraulic systems of the production F-4 test airplane. Based on comparison of available data obtained on all the candidate fluids, the study narrowed down to two fluids, MIL-H-83282 (MLO 68-5) and MIL-H-27601A (MLO 60-294). MIL-H-83282 was subsequently selected. Although the advantages it had were small, they added up to a significant gain in performance at the temperature extremes. These advantages are as follows:

- o MIL-H-83282 has the lower viscosity at -40°F and consequently makes cold starts easier. MIL-H-83282 also has the higher viscosity at 450°F which will give lower leakage rates and better lubrication.
- o Comparison of bulk moduli is more difficult because of the overlapping characteristics. However, the isothermal secant bulk moduli at 330°F and 500 psi are equal and hence neither fluid has an advantage.
- o The combustion indices and flammability test results show that MIL-H-83282 is significantly less flammable.
- o The results of the four ball wear tests indicate that MIL-H-83282 causes less wear at the higher loads but the two fluids were essentially equal at low loads.
- o A series of high temperature pump tests was performed under AFML contract; however, the results were inconclusive. The test pump used was designed for operation at 275°F and included some copper components. Use of these components at 500°F is marginal at best. The conclusion drawn from these results is that the tests should be repeated using a pump designed for high temperature. In the absence of conclusive pump test results, the four ball wear tests have been assumed to be indicative of pump wear. This may be in error if the bearings loads are light, since the two fluids show similar performance at light loads.

The results of the fluid study are presented in the following paragraphs. Paragraph 2 introduces the fluid characteristics which are considered to be of prime importance. Paragraph 3 deals with the particular requirements of the system. Paragraph 4 gives the fluids chosen for comparison and the reasons other possible fluids have been omitted. Paragraph 5 compares the known properties of the chosen fluids with the system requirements. Paragraph 6 gives the recommendations for the use, control, and measurement of the fluid characteristics during component and flight testing of the SFCS.

2. FLUID CHARACTERISTICS

To some extent it is possible to categorize hydraulic fluid characteristics. The following listing sequence has no bearing on the relative importance of these characteristics. Reference 3 lists the SAE definitions and test methods for determining the following fluid characteristics.

a. Viscosity

Viscosity is a characteristic which describes the resistance of a fluid to the motion of its particles. It is one of the most important bases for measurement of the useability of a fluid.

(1) Maximum Viscosity

There appear to be two basic levels of maximum viscosity that are of importance in determining the useability of a fluid: maximum starting viscosity and maximum operating viscosity. A generally accepted level of maximum starting viscosity is 2500 centistokes. At this viscosity, full rate system operation cannot normally be achieved. The heat generated in the system rapidly increases the fluid temperature once the system has been started and thus decreases the viscosity. The maximum operating viscosity level is the viscosity at which full system operation can be expected. This viscosity level is in the order of 500 centistokes. Raising this level increases the system design problems, whereas a lowering of the level decreases them and results in a weight saving by reducing system pressure losses.

(2) Minimum Viscosity

Minimum viscosity is one of the fluid characteristics that establishes the upper economical thermal limit of a specific fluid. A level of 2.0 centistokes is generally accepted as a desired minimum for efficient pump operation. However, fluids with viscosities as low as 0.5 centistokes or less can be pumped, but with some types of pumps the efficiency will be reduced. It must also be noted that extremely low viscosity is often coupled with greatly reduced lubricity, with resultant effects on component life. Low viscosity results in increased internal and external leakage in slide, servo and similar valves.

(3) Pour Point

Normally this characteristic is of no importance to the hydraulic system or component designer. It does, however, indicate an absolute low thermal limit below which fluidity does not exist.

b. Low Temperature Stability

This is a characteristic not generally considered to be a design consideration. It describes the ability of a fluid to resist separation, decomposition and other forms of degradation during storage at extreme low temperature. Unless the results of low temperature reverse themselves when the fluid temperature is increased, the fluid is either unusable or will require special storage and handling.

c. Lubricity

There are two recognized tests for lubricity in a hydraulic fluid, the Shell Four Ball wear test and the Piston Pump wear test. The Shell Four Ball wear test is of considerable value as a preliminary indication of useability of a fluid. It should be run at increments of approximately 100°F from the lowest to the highest fluid working temperatures. The Piston Pump wear test is a much more reliable indication of useability of a fluid. Since this test requires a much longer running time than the Shell test, it is customary to run the test only at the maximum working temperature of the fluid. For specification purposes, it is desirable that the pumping lubricity test be run on a pump which is representative of the general type of pumps expected to be used with the fluid. Good indication of lubricity from wear characteristics in one pump will not mean similar indication in pumps of different manufacturers. Good lubricity is also necessary for slide, servo and similar valves and for its effect on packing life. However, in general, a fluid which has good indicated lubricity in a piston pump will also have the necessary lubricity for such valves.

An indication of poor pump lubricity does not necessarily indicate an unusable fluid, but instead indicates the need for further pump development or the use of a different type of pump.

d. Combustion Indices

The following basic indices of combustion should be stated as a function of pressure. They are of considerable importance in establishing a top usable temperature limit of a fluid and are an index of the temperature level at which special design precautions are necessary for its safe use.

(1) Flash Point

The Flash Point is the temperature to which a combustible fluid must be heated at one standard atmosphere to give off sufficient vapor to form momentarily a flammable mixture with air when a small flame is applied under specified conditions. The test method is referred to as "Cleveland Open Cup" and the result is expressed in increments of 5°F. This characteristic is of relatively small importance in hydraulic system design. Fluids can normally be used at temperatures greatly in excess of their flash point. The flash point is used as an index of volatility and vapor pressure. During system design, it can be used to consider the effect of a build-up of combustible fumes.

(2) Spontaneous Ignition Temperature

The spontaneous or autogenous ignition temperature is often used as a measure of the flammability characteristics of a fluid. It is one of the basic indices used in evaluating the fire resistance of a fluid and is the temperature at which ignition of a fluid takes place without a source of ignition being provided.

(3) Fire Point

Fire Point is the temperature to which a fluid must be heated at one standard atmosphere so that the released vapor will burn continuously when ignited under specified conditions. There is a closed cup (Pensky Marions) test used for determining flash and fire points, particularly for heavy fuel oils and other highly viscous materials. The American Society for Testing and Materials (ASTM) designation is D93. Due to the concentration of vapor evolved the value determined is somewhat lower than that obtained with the Cleveland Open Cup test.

e. Bulk Modulus

Bulk modulus is the volumetric modulus of elasticity. This is a measure of a fluid's resistance to volume reduction by pressure. It is consequently the reciprocal of compressibility. Bulk modulus is expressed in units of force per unit area, lb/in^2. Several types of bulk moduli measurement methods are referred to in published literature. Isothermal bulk modulus refers to modulus values taken under static isothermal conditions. It is used in the two forms given below.

- Isothermal secant bulk modulus refers to the volumetric change with pressure from atmospheric to the pressure of "interest".

- Isothermal tangent bulk modulus is the thermodynamically correct isothermal bulk modulus, and represents the true rate of change at the pressure of "interest".

Adiabatic or isentropic bulk modulus is the volumetric modulus of elasticity under conditions of constant entropy. It is greater than the isothermal tangent bulk modulus by the ratio of the specific heats (C_p/C_v) of the fluid. This value applies under conditions where pressure changes are rapid, with little opportunity for the temperature to come to equilibrium.

In discussing bulk modulus, it is necessary to distinguish between adiabatic and isothermal values. The consensus within the industry is that the adiabatic process giving "instantaneous" values approaches dynamic or transient conditions encountered by aircraft and missiles. Adiabatic modulus values have been determined for fluids at various pressures and over a wide temperature range. These values have been obtained by measuring the velocity of sound waves through the fluids and using the following expressions.

$$Bs = d\,c^2$$

where Bs = adiabatic tangent bulk modulus
d = density of the fluid
c = velocity of sound

Isothermal tangent bulk modulus values may be obtained from adiabatic data by use of the following relationship:

$$\frac{Bs}{Br} = \frac{Cp}{Cv}$$

where Br = isothermal tangent bulk modulus
Bs = adiabatic tangent bulk modulus
Cp = specific heat at constant pressure
Cv = specific heat at constant volume

f. Vapor Pressure

The vapor pressure of a fluid is dependent on the vapor pressure of the base fluid and the additives. The additives will frequently vaporize before the base fluid and hence the vapor pressure characteristic will tend to be non-linear. High fluid demand at the pump inlet at high operating temperatures can cause pump cavitation (vapor pockets in the hydraulic system) if the vapor pressure of the hydraulic fluid is high. Cavitation causes malfunctioning of hydraulic systems. Low vapor pressure results in less evaporation and thus means better retention of lubricating qualities at external seals such as piston rod seals. Low vapor pressure also reduces fire hazards due to flammable vapors.

g. Thermal Characteristics

To have an accurate appraisal of the thermal characteristics of a fluid, the following data should be included in the fluid description. They are important to the thermodynamic design of the system and its components.

(1) Thermal Stability

The thermal stability factor of a fluid should be presented as breakdown rate as a function of temperature. The nature of the products resulting from fluid breakdown should be stated in the descriptive data. These products must not be of a type which will result in excessive gumming of valves and other components, or tend to corrode or otherwise destroy the materials of which the components are manufactured. Where gaseous products are released in an irreversible process, means should be provided for their elimination from the system.

(2) Thermal Expansion

The coefficient of thermal expansion of a fluid should be given as a function of temperature.

(3) Specific Heat

The specific heat of a fluid should be given as a function of temperature.

(4) Thermal Conductivity

The thermal conductivity of a fluid should be given as a function of temperature.

h. Dissolved and Entrained Gases

Free or entrained air in a hydraulic system substantially reduces the effective bulk modulus of the system fluid. That is, an air-oil mixture appears to increase the compressibility of the fluid, making the system spongy. Test data indicate that dissolved air has no effect on bulk modulus. These facts appear paradoxical; however, if one visualizes a container filled to the brim with marbles (which represent the oil molecules), it is possible to pour in a fluid (representing the dissolved air) around them or to remove the fluid with no change in marble volume. The weight of the container changes, but not the volume occupied by the marbles.

(1) Dissolved Gas

Dissolved gas is gas which is in solution in a fluid; that is, stored in the cavities between the fluid molecules (intermolecular cavity). Dissolved gas behaves according to Henry's Law, which states that the amount of gas in solution is proportional to the volume of the intermolecular cavity and the absolute pressure existing in the cavity. Since the size of the cavity is dependent upon the fluid's molecular structure, dissolved gas content is a fluid property rather than a system property as is the case with entrained gas. If external gas pressure is greater than that within the cavities, gas will enter them, but only at the gas fluid interface. This gas will immediately go into solution, but will not permeate more than just a few of the adjacent fluid layers. Gas will leave the intermolecular cavities if there is a lower external pressure. For this condition, a gas fluid interface is not mandatory for release as it is for entry. Considerably more gas, however, will be liberated if the fluid is filmed. In summary, the mechanics of dissolved gas is essentially a film phenomenon, that is adsorption rather than absorption.

(2) Entrained Gas

Entrained gas is that which is suspended in a fluid in the form of bubbles and behaves according to Boyle's Law.

(a) Entrained gas is transient and therefore not a fluid property.

(b) The content of entrained gas is dependent upon the entrainment source; its release is a function of time and certain fluid properties such as specific gravity, viscosity and dissolved gas content.

(c) Sources of entrained gas in a system are:

- Residual gas resulting from inefficient bleeding techniques
- Inclusion or inward leakage of external gas through pump shaft seals, bladders, quick disconnects, etc.
- Release of dissolved gases due to pressure changes, e.g., at pump inlets, across orifices, altitude changes, etc.

(3) Adsorption Rate

Adsorption, rather than absorption, better describes the process by which bubbles under pressure in a hydraulic fluid are dissolved into the fluid. Adsorb means adhesion of extremely thin films of gases to surfaces with which the gases are in contact. Absorb means to soak up. The adsorption rate of bubbles (entrained gas) in a system is determined by their size. It is true that a fluid holds more dissolved gas as pressure increases (Henry's Law). However, the size of the bubble determines at what pressure it will dissolve. Bubbles, with diameters of 0.020 to 0.030 inch will dissolve at approximately 100 psi. Larger bubbles, however, will not dissolve until the pressure is proportionately greater. This behavior is true even when the amount of previously dissolved gas in the fluid is almost nil. We may say, therefore, that the rate of adsorption is a function of the pressure and an inverse function of the bubble diameter. Note that the bubble will reappear when the pressure is lowered.

(4) Origin of Entrained Gases

Dissolved gas will come out of solution when the fluid is exposed to a vacuum. A vacuum can occur in a hydraulic system across orifices, inside unsupercharged pumps, and inside double-acting actuators when driven by an external load at a rate greater than the rate at which fluid can fill the opposite end. Dissolved gas, once out of solution, can be partly readsorbed in a moving stream when the local static pressure again rises. When a stream containing bubbles is suddenly stopped, however, the bubbles will begin to migrate upwards and coalesce in the nearest high point. Repressurization may or may not drive the new, larger bubble back into solution. Even if all the air bubbles and pockets were removed from a hydraulic system and the system totally enclosed, after a short run-in period, bubbles would begin to reappear. The source of these bubbles is the working fluid itself because all fluids (except fully degassed ones) contain dissolved gas.

(5) Readsorbtion Rate

Fluid which has been forced to release its gas by a separator will quickly readsorb gas when left exposed. The degassed fluid acts like a sponge. This phenomenon can be beneficial in bleeding complex systems. The most remote areas of the system can be bled effectively by circulating degassed fluid through the system. Pockets of gas in nooks and crannies that would normally never get bled using conventional bleeding methods can be cleared easily of gas.

i. Hydrolytic Stability

A high degree of hydrolytic stability is a very desirable feature in a hydraulic fluid and should be stated in the fluid descriptive data as a function of water percentage. The nature of the products of hydrolytic breakdown should also be stated. In fluids which do not have the desired degree of hydrolytic stability, special precautions must be taken in the preparation, storage, handling and system design for the satisfactory use of that fluid.

j. Hydroscopic Tendency

This characteristic is to a certain extent coupled with the hydrolytic stability of a fluid and is an indication of the necessity of special precautions in the preparation, storage, handling and system design for satisfactory use of a fluid. It should be stated in the fluid descriptive data.

k. Foaming Tendency and Stability

The importance of this characteristic is entirely a function of hydraulic system design. Where a tank type reservoir is to be used, foaming is very definitely undesirable and any foam produced must break down rapidly. In the case of gas/oil separated systems, where piston type reservoirs are used, foaming is of lesser importance. For ground handling equipment, the foaming tendency of a fluid requires similar treatment to that in the air vehicle.

l. Compatibility with Other Hydraulic Fluids

In many instances it is desirable to increase hydraulic system performance by merely changing fluids. In these cases compatibility of the fluids involved becomes an important parameter.

m. Effect on Elastomers

Successful performance of high temperature systems requires that the fluid used be compatible with elastomeric gaskets and seals. Gaskets and seals must not only be stable over the extreme temperature range, but must also have the proper amount of swell to prevent leakage. A high temperature hydraulic fluid should act as a plasticizer for seals and gaskets to give low temperature flexibility because some elastomers do not contain a satisfactory plasticizer while others lose the plasticizer through leaching. The compatibility of fluids with commercially available seals and gaskets has been extensively investigated by laboratories and leading elastomer manufacturers.

n. Corrosion Stability

The existence of corrosion on the materials caused by the fluid in a hydraulic system is an important consideration to the system and component designer. The corrosion tests must be made at the maximum anticipated fluid temperatures. The materials to be tested may be varied to comprise those materials usable at the extreme temperature range of the fluid and should be clearly stated in the fluid descriptive data. The corrosion stability characteristic of a fluid must be evaluated to also determine the effects on storage and usage activities.

o. Oxidation Stability

The oxidation stability of a fluid should be stated not only at the upper working temperature of the fluid, but also at the storage temperature range. To a great extent, decreased oxidation stability can be compensated for by design, but the necessary precautions must be stated in the fluid descriptive data. It is desired that the fluid possess a reasonable degree of oxidation stability so that excessive precautions in system design and fluid storage will not be necessary.

p. Evaporation

A requirement that the product of evaporation be an oily residue is of particular importance in its effect on piston rods. A crystalline or glassy residue on piston rods will, in most cases, result in reduced packing life. It is also desirable that the residue be readily removable and preferably be soluble in the system fluid, so that extraneous materials will not be introduced into the system.

q. Toxicity

To be truly non-toxic, a fluid not only must not cause undesirable effects when ingested in small quantities, but its fumes must not result in lung or eye irritation. Contact with the skin must not result in irritation or allergic effects. A fluid having toxic effects will require special handling techniques.

r. Vehicle Compatibility

The fluid, when dripped or spilled on various portions of the vehicle, its wiring, its painting and its components should not cause damage or degradation. Leakage and spillage is unavoidable and special precautions should not be necessary to prevent damage resulting from this.

s. Particle Contamination

Cleanliness of fluid, as it is delivered to the user, is of great importance. The mere passing of the fluid, at some time during the course of its preparation, through a nominal 10 micron filter is insufficient to guarantee cleanliness. A finer degree of filtration must be used during the preparation for shipment, and precautions must be taken so that the shipping containers do not contaminate the fluid.

t. Viscosity Index Improvers and Shear Stability

The ability of the elements of a fluid formulation to withstand the shearing action of pumping and valve operation is of considerable importance in determining endurance life of a fluid. Shear breakdown is generally indicated by a lowering of viscosity and may also be accompanied by a loss of lubricity.

Polymers have been used widely to obtain viscosity-temperature characteristics not possible with the base stock alone. The use of these chemicals has made the viscosity-temperature characteristics of the fluid relatively independent of the base stock properties. Typical examples of polymer-containing fluids are multigrade crank-case oils and MIL-H-5606. Examples of commercially available polymers are polyacrylate esters, polymethacrylate esters, polybutene, polystyrene, and high molecular weight silicone polymers.

Polymer containing formulations, in general, display non-Newtonian viscosity properties. That is, the viscosity of these lubricants is a function of shear rate. The viscosity change with shear rate is a reversible change, provided only streamline flow is involved. The viscosity loss at a high shear rate is completely recovered at low shear rates under these conditions. Some advantages of non-Newtonian viscosity properties have been shown in fundamental studies of the hydrodynamic lubrication of journal bearings. It has been shown that a non-Newtonian fluid may show a 40 percent reduction in friction over a Newtonian fluid of the same low shear viscosity while both of these fluids exhibit the same film thickness in the bearing.

Polymeric materials usually introduce mechanical stability problems. These polymers tend to be permanently degraded to less viscous materials when subjected to turbulent flow under high shear, wire drawing, and attrition. This process results in a permanent loss in viscosity.

Both permanent and reversible viscosity-shear phenomena are a function of molecular type and molecular weight of the polymer. The degree of non-Newtonian properties that will be evident is determined by polymer chemistry and manufacture, and the mechanism to be lubricated.

u. Neutralization Number

The neutralization number is a measure of the acidity or basicity of a fluid. It is defined as milligrams of potassium hydroxide required to neutralize the acidity in one gram of fluid or the equivalent of the basicity expressed in a similar manner. A low neutralization number is not necessarily a reliable index of the corrosivity of a fluid, but a change in acidity or basicity is often used as a measure of deterioration of a fluid in use.

3. SYSTEM REQUIREMENTS

a. Present F-4 Hydraulic System

Figure 11 on page 55 shows the schematic of the present F-4, PC-1 and PC-2 hydraulic systems. The fluid used is MIL-H-5606 and the components have been tested using the fluid over the temperature range -65°F to 275°F.

The normal fluid working temperature range is from 60°F to 180°F. Small increases in viscosity over the temperature range will not cause significant effects on the system performance. However, a reduction in the fluid bulk modulus would be unacceptable. The fluid chosen must be compatible with MIL-H-5606 and the seals used in the present system components.

b. SFCS System

Figure 14 on page 61 shows the hydraulic schematic for Phase IIC of the SFCS program. The SSAP section of this schematic shows how PC-1 and PC-2 hydraulic systems are used as back-up supplies to the integrated hydraulic systems that are part of the SSAP.

The fluid used in the PC-1 and PC-2 hydraulic systems must be the same as that used in the integrated hydraulic systems.

(1) Working Temperature

Thermal studies show that the fluid temperature encountered in the integrated hydraulic systems may be as high as 450°F, with possible transients exceeding this value.

(2) Bulk Modulus

The fluid bulk modulus is lower at high temperatures. To achieve a usable package size, the adiabatic bulk modulus measured at 450°F and 1000 psi should not be less than 100,000 psi.

(3) Vapor Pressure

The vapor pressure of the fluid is considered to be important because of the high working temperature. For MIL-H-5606, the boost pressure increase required to prevent cavitation at 450°F is 10 psi greater than that required at 275°F. The fluid chosen should ideally have a vapor pressure of less than 1 psi absolute at 450°F, to prevent evaporation from the actuator piston rods and the build up of explosive vapors within the compartment during ground operation.

(4) Minimum Viscosity

The ideal minimum viscosity at the expected working temperature is 2.0 centistokes, but the available fluids cannot meet this requirement. Hence, a minimum viscosity of 0.5 centistokes must be accepted. Fluids which meet the 2.0 centistokes figure generally incorporate a viscosity index improver. When a viscosity index improver is used, the fluid viscosity will rapidly degenerate to the viscosity of the base stock, when subjected to shear, oxidation and thermal degradation.

(5) Shear Stability

Large variations in fluid viscosity due to shear instability would be unacceptable. Changes in viscosity would make correlation of performance data difficult. With only 150 cubic inches of oil in each integrated hydraulic system, the rate of change of viscosity due to shear could be high. The viscosity would change each time new oil is added to the system. Consequently, the fluid chosen should have high shear stability.

(6) Oxidation Stability

Systems using hydrocarbon fluids operating above 300°F are subject to severe oxidation problems if the fluid is exposed to air or contains dissolved air. This problem is normally overcome by exposing the fluid to an inert atmosphere; however, there remains the problem of air dissolved or entrained in the fluid during normal handling and filling operations. To avoid problems due to oxidation, the amount of dissolved and entrained air in the fluid may be controlled by de-aeration of the fluid prior to use.

4. CANDIDATE FLUIDS

Table IX lists all the fluids reviewed as possible candidates and the reason for elimination. Many of the fluids were eliminated from the study due to their incompatibility with a 450°F working temperature, MIL-H-5606 hydraulic fluid, and MS28775 Seals. Consequently, only two fluids, MIL-H-27601 and MIL-H-83282 (ML068-5), remained as candidates for use in the SFCS hydraulic systems. Two other fluids, MIL-H-5606B and Oronite M2V, are included for comparison. The MIL-H-5606B was chosen in preference to MIL-H-5606A because of its superior cleanliness and shear stability. The Oronite M2V was chosen because of its superior shear stability to fluids conforming to MIL-H-8446. This does not mean that the overall performance of Oronite M2V is considered to be superior to other fluids in its class, nor does it mean that MIL-H-27601 and ML068-5 are considered to be superior to other fluids capable of similar high temperature operation.

5. COMPARISON OF FLUID DATA

a. General Data

Figures 26 through 33 and Tables X through XXI show the characteristics of the two fluid candidates and the two baseline fluids. Where data from different sources are in conflict, the conflicting data are given, with no opinion expressed as to which data are considered more valid.

The bulk modulus data presented in Figure 27 were obtained using the test setup shown in Figure 34. The fluid shear and thermal stability test data presented in Tables XIII and XIV were obtained using the hydraulic test circuit shown in Figure 35. The high temperature test data presented in Tables XVI through XXI were obtained using the hydraulic pump circuit shown in Figure 36. Table XI compares some of the fluid characteristics of MIL-H-5606B, MIL-H-27601 and MIL-H-83282. Information included in the charts of the text concerning Oronite M2V was obtained from data sheets titled "Oronite High Temperature Fluid M2V", dated October 1966, by Chevron Chemical Company, Oronite Division.

(Text Continues on Page 120)

TABLE IX
CANDIDATE HYDRAULIC FLUIDS

Acceptance/Rejection Code

A	Acceptable
U	Unacceptable
C	Compatible
I	Incompatible
Blank	Unknown

Fluid	Fluid Type	Viscosity at -40^0F	Viscosity at 450^0F	Bulk Modulus at 450^0F	Shear Stability	450^0F Working Temp.	MIL-H-5606	MS-28775 Seals	
MIL-H-5559	Gycol-Water			U		I	I	C	
MIL-H-5606	Petroleum	A	A	U	U	I	C	C	
MIL-H-5606A	Petroleum	A	A	U	U	I	C	C	
MIL-H-5606B	Petroleum	A	A	U	A	I	C	C	
MIL-H-6083	Petroleum	A	A			I		C	
MIL-H-8446	Silicate Ester	A				C	I	I	
MIL-H-13862	Petroleum	A				I		C	
MIL-H-13866	Petroleum	U				I		C	
MIL-H-13919	Petroleum	U				I		C	
MIL-H-17111	Petroleum	A				I		C	
MIL-H-19557	Tri-aryl Phosphate			A		C	I	I	
MIL-H-22072	Glycol-Water	U	U	U		I	I	C	
MIL-H-27601	Super-Refined Mineral Oil	A	A	A	A	C	C	C	
MIL-H-460J4	Petroleum	A	U			I		C	
MIL-H-46001 Type I	Petroleum	U				I			
MIL-H-46001 Type II	Petroleum	U				I			
MIL-H-46001 Type III	Petroleum	U				I			
MIL-H-460C1 Type IV	Petroleum	U				C			
MIL-H-81091	Petroleum	A	U			I			
MIL-S-81087	Silicone	A	A			C	I		
Skydrol 500A	Phosphate Ester	A	A			I	I	I	
Skydrol 500B	Phosphate Ester	A	A			I	I	I	
Skydrol 7000	Phosphate Ester	A	A			I	I	I	
Skydrol HT	Porganate Phosphorus Compound	A	A				I	I	
Oronite Hyjet	Phosphate Ester	A	A	U			I	I	
Oronite Hyjet—W	Phosphate Ester	A	A	U			I	I	
MLO-8200	Silicate Ester	A	A	U			I	I	
AeroSafe 2300	Phosphate Ester	A	A	U			I	I	
AeroSafe 2300W	Phosphate Ester	A	A	U		C	I	I	
MLO-7277	Super-Refined Mineral Oil	U	A			C	C	C	
MLO-60-294 (MIL-H-27601)	Deep Dewaxed Mineral Oil	A	A	A	A	C	C	C	
Oronite 70	Silicate Ester	A	A		U	C	I	I	
Oronite M2V	Silicate Ester	A	A		A	C	I	I	
Monsanto OS-45 Type IV	Silicate Ester	A	A			C	I	I	

TABLE IX
CANDIDATE HYDRAULIC FLUIDS (Continued)

Acceptance/Rejection Code

A	Acceptable
U	Unacceptable
C	Compatible
I	Incompatible
Blank	Unknown

Fluid	Fluid Type	Viscosity at $-40^{\circ}F$	Viscosity at $450^{\circ}F$	Bulk Modulus at $450^{\circ}F$	Shear Stability	$450^{\circ}F$ Working Temp.	MIL-H-5606	MS-28775 Seals	
Monsanto MCS293	Polyphenyl Ether	U	A	U		C	I		
Monsanto MCS3101	Polyaryl	A		A		C	I		
G.E. SF-1147	Silicone	U	A	U		C	I		
G.E. SF-1148	Silicone	U	A	U		C	I		
Dow ET-378	Polyphenyl Ether (4P-3E)	U	A			C	I	I	
Monsanto CS-124	Polyphenyl Ether (5P-4E)	U	A			C	I	I	
NaK77	Liquid Metal Eutectic	U	A			C	I	I	
Esso 5251	Triester	U					I	I	
Dow-Corning XF-1-0294	Silicone	A	A			C	I		
Dow-Corning XF-1-0301	Silicone	A	A			C	I		
Brayco 713	Petroleum	A	U			I			
Brayco 718	Petroleum	U	A			C			
Brayco 745	Petroleum	U	A						
Brayco 762	Petroleum	A	U						
Brayco 775	Hydrocarbon	U		A					
Brayco 810	Perfluoro-Alkylpoly Ether	U	A						
Brayco 811	Perfluoro-Alkylpoly Ether	U	U						
Brayco 812	Perfluoro-Alkylpoly Ether	U	A						
Brayco 813	Perfluoro-Alkylpoly Ether	U	A						
DuPont Krytox 143AX	Perfluoro-Alkylpoly Ether	U	A						
DuPont Krytox 143AA	Perfluoro-Alkylpoly Ether	U	A						
DuPont Krytox 143AB	Perfluoro-Alkylpoly Ether	U	A	U					
DuPont Krytox 143AC	Perfluoro-Alkylpoly Ether	U	A	U					
DuPont Krytox 143AD	Perfluoro-Alkylpoly Ether	U	A						
Halocarbon 208-25H	Chlorofluoro-Carbon	A	U						
DuPont Freon E-1		A	U		A	I		C	
DuPont Freon E-2		A	U		A	I		C	
DuPont Freon E-3		A	U		A	I		C	
DuPont Freon E-4		A	U		A	I		C	
DuPont Freon E-5		A	U		A			C	
DuPont Freon E-6		U	U		A			C	
DuPont Freon E-7		U	U		A			C	
DuPont Freon E-8		U	U		A			C	
MLO 68-5 (MIL-H-83282)	Synthetic Hydrocarbon	A	A	A	A	C	C	C	

TABLE X
SUMMARY, FLUID CHARACTERISTICS

Fluid Characteristics	Units	MIL-H-5606	MIL-H-27601	MLO-68-5	Oronite M2V
Viscosity					
at –40°F	cs	488	3,936	2,978	700
at 100°F	cs	14.47	15.12	17.56	17.6
at 210°F	cs	5.26	3.27	3.82	5.45
at 450°F	cs	1.7	0.99	1.10	1.32
Pour Point	°F	–85	–70	–85	–110
Combustion Indicies					
Flash Point	°F	200	370	410	420
Fire Point	°F	255	425	495	500
Spontaneous Ignition	°F	475	740	685	760
Rubber Swell					
(Buna N Dumbells, - Hours at – °F)					
"L" Rubber					
168 Hours at 158°F	%	22.2	6.1	19.7	
72 Hours at 275°F	%			11.7	
Viton					
72 Hours at 400°F	%		0.65	1.0	
Shell 4 Ball Wear Test, Scar Diameter					
600 RPM at 167°F; 52, 100 Steel Balls					
1 Hour at 1 kg	mm		0.161		
1 Hour at 10 kg	mm		0.210		
1 Hour at 40 kg	mm		0.630	0.567	0.55
2 Hour at 40 kg	mm			0.579	
1200 RMP at 275°F 52,100 Steel Balls					
2 Hours at 10 kg	mm				0.78
2 Hours at 40 kg	mm				0.88
Thermal Stability					
6 Hours at 700°F					
Viscosity Change	%			–14.8	
Neutralization Change	mgKOH			+2.0	
Tested Per MIL-H-27601					
Viscosity Change	%		–15.25		
Neutralization Change	mgKOH		+0.02		

TABLE XI
FLUID CHARACTERISTIC COMPARISON

Fluid	MIL-H-5606(B)	MIL-H-27601 (MLO-60-294)	MIL-H-83282 (MLO-68-5)
Kinematic Viscosity (cst)			
at 500°F	–	0.64	0.72
at 400°F	–	0.99	1.10
at 300°F	2.36	1.64	1.89
at 210°F	5.26	3.27	3.82
at 130°F	10.22	9.02	10.43
at 100°F	14.47	15.12	17.56
at –40°F	488.0	3939.0	2978.0
Flash Point, °F	209	375	410
Fire Point, °F	230	420	495
S.I.T. °F	470	700	700
Corrosion and Oxidation Test		Pass	Excellent
48 Hours at 347°F, 5L, Air/Hour	–		
Pour Point, °F	–75	–75	–85
Specific Gravity	0.8681	0.8483	0.8433
Total Acid Number	0.03	0.07	0.02
"L" Rubber Swell			
168 Hours at 158°F, %	22.2	6.2	19.7
Viton "O" Rings			
72 Hours at 400°F			
Under N_2 Blanket			
Tensile Strength (PSI)	1535	1580	1580
Elongation (%)	275	225	235
Hardness (Shore A)	80	80	76
Volume Change (%)	–	+ 0.65	+ 1.0
Specific Heat			
at 100°F	0.475	0.480	0.499
at 300°F	0.573	0.580	0.598
at 500°F	0.690	0.680	0.697
Thermal Conducitivity			
BTU/Hr/Sq, Ft/°F/Ft			
at 100°F	0.078	0.078	0.097
at 300°F	0.074	0.071	0.076
at 500°F	0.069	0.063	0.072
Dielectric Constant			
at 79°F/1000 CPS	2.33	2.17	2.29
Electrical Conductivity			
PICOMHOS/Meter	9.6	0.3	1.2

TABLE XIII
FLAMMABILITY TESTS

Fluid	MIL-H-27601	MIL-H-5606(B)	MLO-68-5
High Temp. Spray Ignition (Ignition Source – Welding Torch)	Ignites and carries flame to hood in continuous stream. Burns vigorously in hood after flame is removed.	Ignites and carries flame to hood, burns vigorously in hood and in air after flame is removed.	Ignites and carries flame toward hood: consumed before reaching hood. Does not continue to burn after ignition source is removed.
Low Temp, Spray Ignition	Increases flame, carries to wall in continuous stream with audible roaring.	Increases flame, carries to wall in continuous stream with audible roaring.	Carries flame towards wall in continuous sheet of yellow flame; fluid consumed before reaching wall.
Wick Flammability (Windshield Wiper) Cycles to Continuous Burn	6	1	12
Navy 6-Wick Test			
Time to Leave Wick (Sec)	–	33	3600
Flame Speed	–	0.64 In./Sec	NIL
Gunfire Test			
50 Caliber Incendiary	6	6	10
No. of Shots	6 Fires out of 6 shots.	6 Fires out of 6 shots	3 Fires, but less intense than the MIL-H-5606(B) fires.

TABLE XIII

FLUID SHEAR AND THERMAL STABILITY TEST

100 Hours at 550°F Fluid Temperature
Effects on Fluids

Fluid	MLO 60-294	MLO 68-5
Flash Point, $^{\circ}$F	388	397
New	381	395
100 Hr	381	395
Fire Point, $^{\circ}$F		
New	435	483
100 Hr	412	477
Viscosity, Cst		
100°F		
New	14.08	18.09
100 Hr	13.79	17.85
210°F		
New	3.21	3.82
100 Hr	3.12	3.73
Neutralization No.		
New	0.06	*
100 Hr	0.06	*
Fluid Color		
New	Light Amber	Pale Yellow
100 Hr	Very Dark	Amber
IR Spectrum	None	None
Total Shear Cycles	9,700	16,500

*Too low to be calculated

TABLE XIV
FLUID SHEAR AND THERMAL STABILITY TEST

100 Hours at 550°F Fluid Temperature
Effects on Specimens

Specimen	Average Specimen Weight Change MG/CM^2		Specimen Appearance Change			
			Color		Texture	
	MLO 60-294	MLO 68-5	MLO 60-294	MLO 68-5	MLO 60-294	MLO 68-5
Aluminum, 2024-T4	–0.27	0	None	Slightly Darker	None	None
M-1 Tool Steel	–0.07	0	Slightly Darker	Much Darker	None	None
Chrome May Steel 4140	–0.06	0	Slightly Darker	Much Darker	None	None
302 Stainless Steel	–0.05	0	None	Slightly Darker	None	None
440 Stainless Steel	–0.075	0	Slightly Darker	Much Darker	None	None
Titanium, RCL30B	–0.055	0	Slightly Darker	Slightly Darker	None	None
Beryllium Copper QQ-C-530	–0.07	0	Much Darker	Slightly Darker	Pitted	None

TABLE XV
4 BALL WEAR TEST

400°F and 600 RPM for 1 Hour
Scar Diameter in Millimeters

Fluid	Ball Materials	Load in Kilograms				
		40	30	20	10	4
MIL-H-27601	M-10	0.99	0.99	0.76	0.22	0.18
MLO 68-5		0.46	0.38	0.27	0.21	0.19
MIL-H-27601	52,100	0.98	1.29	0.52	0.40	0.37
MLO 68-5		0.56	0.48	0.34	0.22	0.20

4 Ball Torque Transition Temperatures
Speed 600 RPM, Load 40 KG, Ball Material M-10 Tool Steel

Fluid	Transition Temperature, $^{\circ}$F
MLO 60-294	540
MLO 68-5	Above 650

TABLE XVI
HIGH TEMPERATURE HYDRAULIC PUMP TEST
50 Hours at 400°F Fluid Temperature
Test Profile

Fluid	MLO 60-294	MLO 68-5
Test Duration, Hrs	50	50
Maximum Fluid Temperature, °F	400*	401.7
Pumping Rate, GPM		
At Maximum Flow	7.80	8.06
At Minimum Flow	3.70	1.29
Pump Discharge Pressure, PSI		
At Maximum Flow	2900	2697
At Minimum Flow	3450/3140	3062
Inlet Filter Differential Pressure at Maximum Flow, PSI		
Initial	24	12
Final	32	25
Total Shear Cycles	9,580*	7,790

*Estimated

TABLE XVII
HIGH TEMPERATURE HYDRAULIC PUMP TEST
50 Hours at 400°F Fluid Temperature
Effects on Pumps

Parts	Average Part Weight Change (GMS)
Piston Assemblies	
MLO 60-294	–0.0096
MLO 68-5	–0.0339
Piston Sleeves	
MLO 60-294	–0.0005
MLO 68-5	+0.0004
Piston Collars	
MLO 60-294	–0.0002
MLO 68-5	–0.0004
	Longitudinal Movement Change (In.)
Piston/Slipper Ball Joint	
MLO 60-294	+0.0024
MLO 68-5	+0.0043

Pump Used: New York Air Brake Inline High Temperature Pump No. 69W03006-2.

TABLE XVIII
HIGH TEMPERATURE HYDRAULIC PUMP TEST

50 Hours at 400°F Fluid Temperature

Effects on Fluids

Fluid	MLO 68-5	MLO 60-294
Flash Point, °F		
New	397	382
50 Hours	395	387
Fire Point, °F		
New	483	420
50 Hours	478	426
Viscosity, Cst		
100°F		
New	18.09	13.88
50 Hours	17.69	14.04
210°		
New	3.82	3.37
50 Hours	3.76	3.13
Neutralization Number		
New	0.010	0.10
50 Hours	0.013	0.08

TABLE XIX
HIGH TEMPERATURE HYDRAULIC PUMP TEST
500°F Fluid Temperature
Test Profile

Fluid	MLO 60-294	MLO 68-5	
Test Duration	47	2.6	9.2
Maximum Fluid Temperature	500 ± 7	499.9	501.9
Pump Rate			
Maximum Flow	7.3 (6.3)	7.80	7.90
Minimum Flow	0.8	1;97	2.02
Pump Discharge Pressure			
Maximum Flow	2,700	2,700	2,538
Minimum Flow	3,100	3,032	2,812
Pump Filter Pressure			
At Maximum Flow			
Initial	10	12	12
Final	27	11	12
Total Shear Cycles	6,100	425	1,520

Pump Used: New York Air Brake Inline High Temperature Pump
No. 69W03006-2

TABLE XX
HIGH TEMPERATURE HYDRAULIC PUMP TEST
500° Fluid Temperature
Effects on Pumps

Part	Average Part Weight Change, GMS
Piston Assemblies	
MLO 60-294	−0.0089
MLO 68-5	−0.3710
Piston Sleeves	
MLO 60-294	+ 0.0006
MLO 68-5	+0.0015
Piston Collars	
MLO 60-294	−0.0009
MLO 68-5	−0.0015
	Longitudinal Movement Change, Inches
Piston/Slipper Ball Joint	
MLO 60-294	+0.0014
MLO 68-5	+0.0024

Pump Used: New York Air Brake Inline High Temperature Pump
No. 69W03006-2

TABLE XXI
HIGH TEMPERATURE HYDRAULIC PUMP TEST
500°F Fluid Temperature
Effects on Fluids

Fluid	MLO 60-294	MLO 68-5
Duration of Test, Hr	47	11.8
Flash Point, °F		
New	382	397
End of Test	385	405
Fire Point, °F		
New	420	483
End of Test	410	480
Viscosity, Cst		
100°F		
New	13.88	18.09
End of Test	13.40	17.73
210°F		
New	3.37	3.82
End of Test	3.05	3.75
Neutralization No.		
New	0.10	0.01
End of Test	0.16	0.02

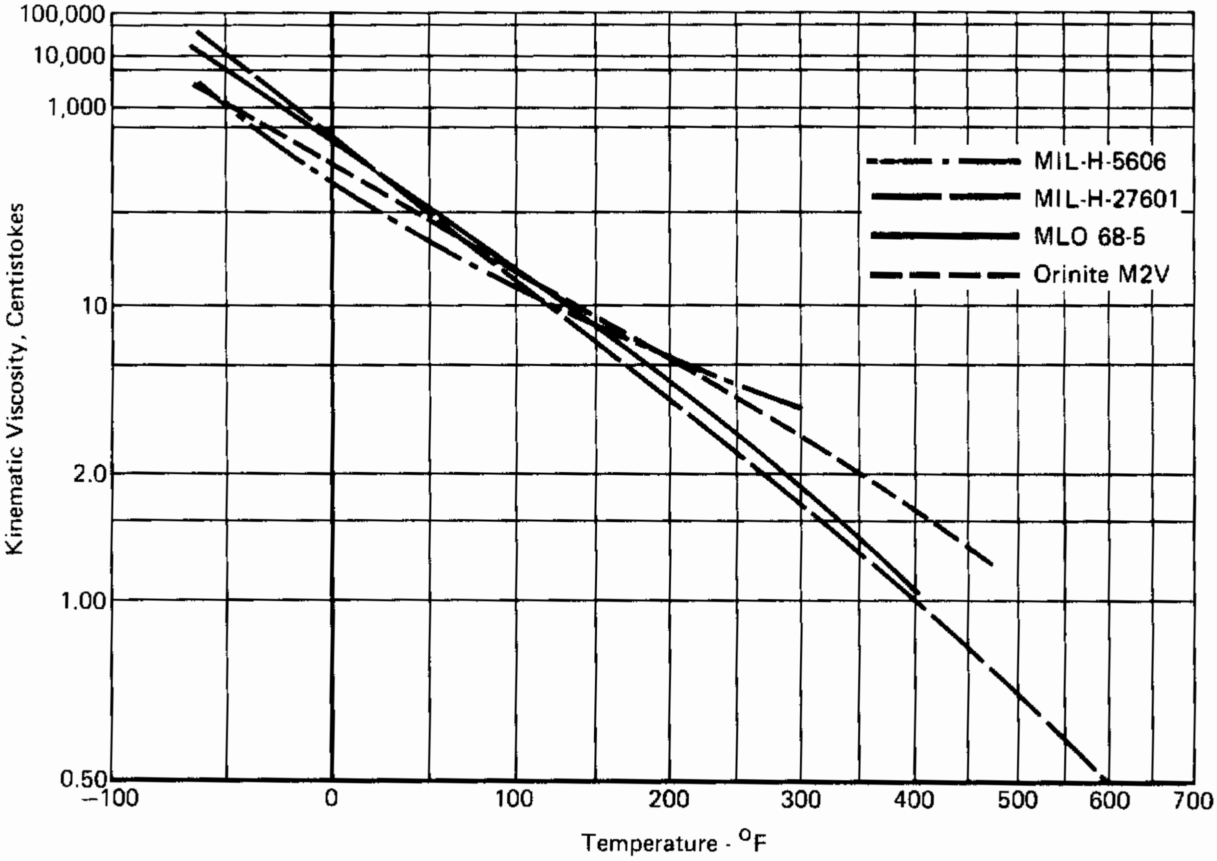
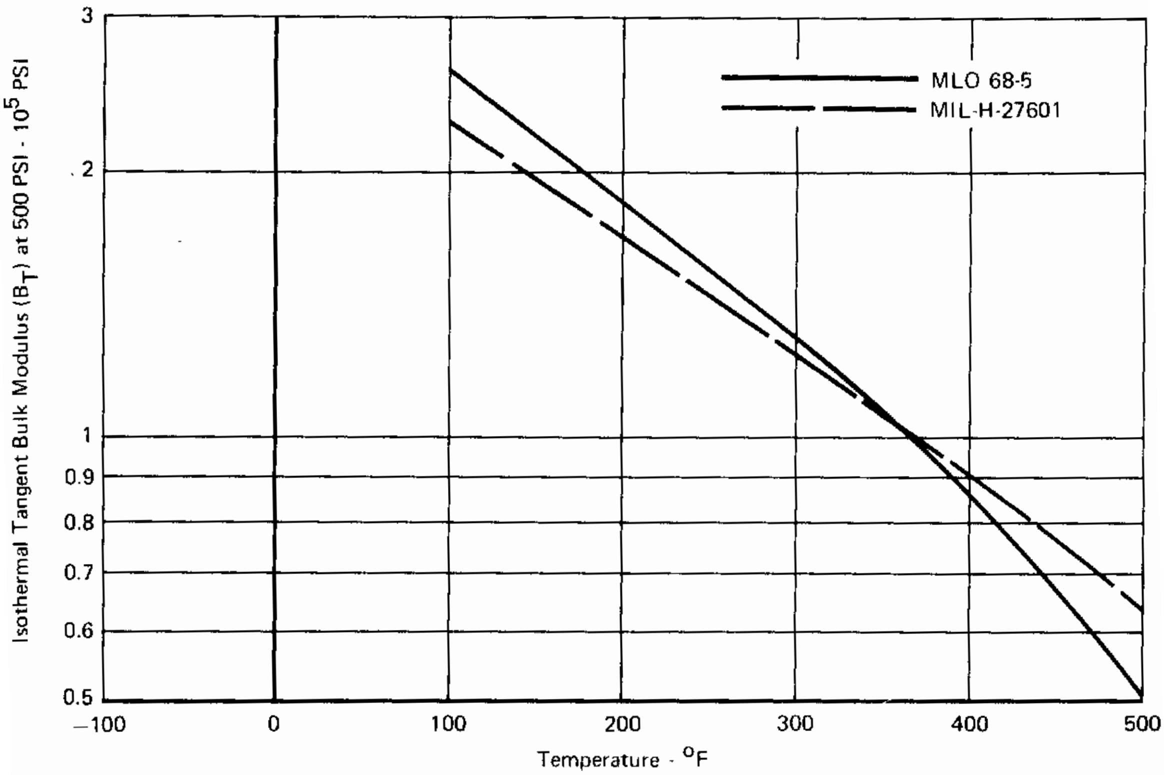


FIGURE 26
VISCOSITY



For values of isothermal tangent bulk modulus at higher pressures use
B_T at P_1 = B_T at 500 PSI + 12.5 x $(P_1 - 500)$

FIGURE 27
ISOTHERMAL TANGENT BULK MODULUS

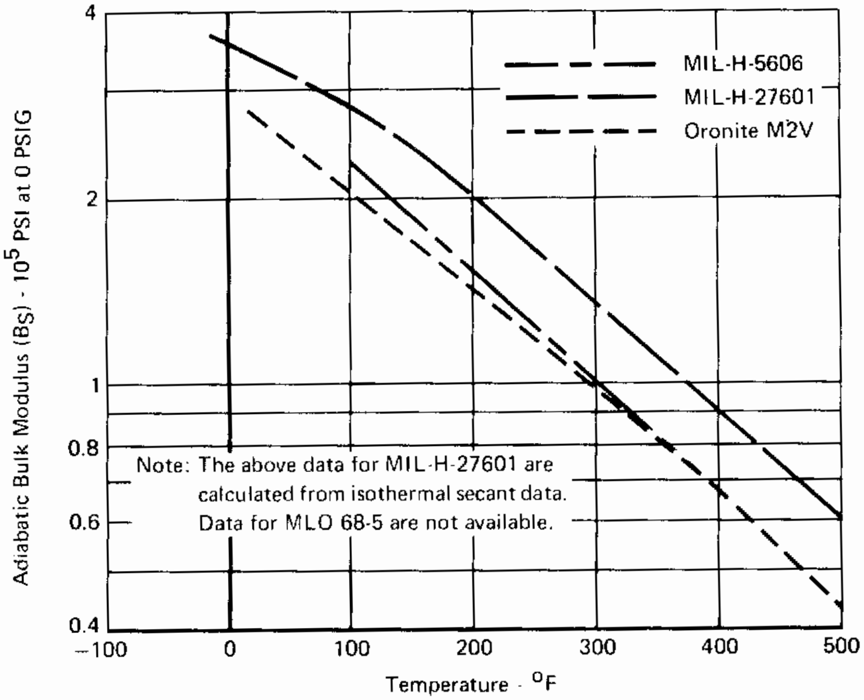


FIGURE 28
ADIABATIC BULK MODULUS

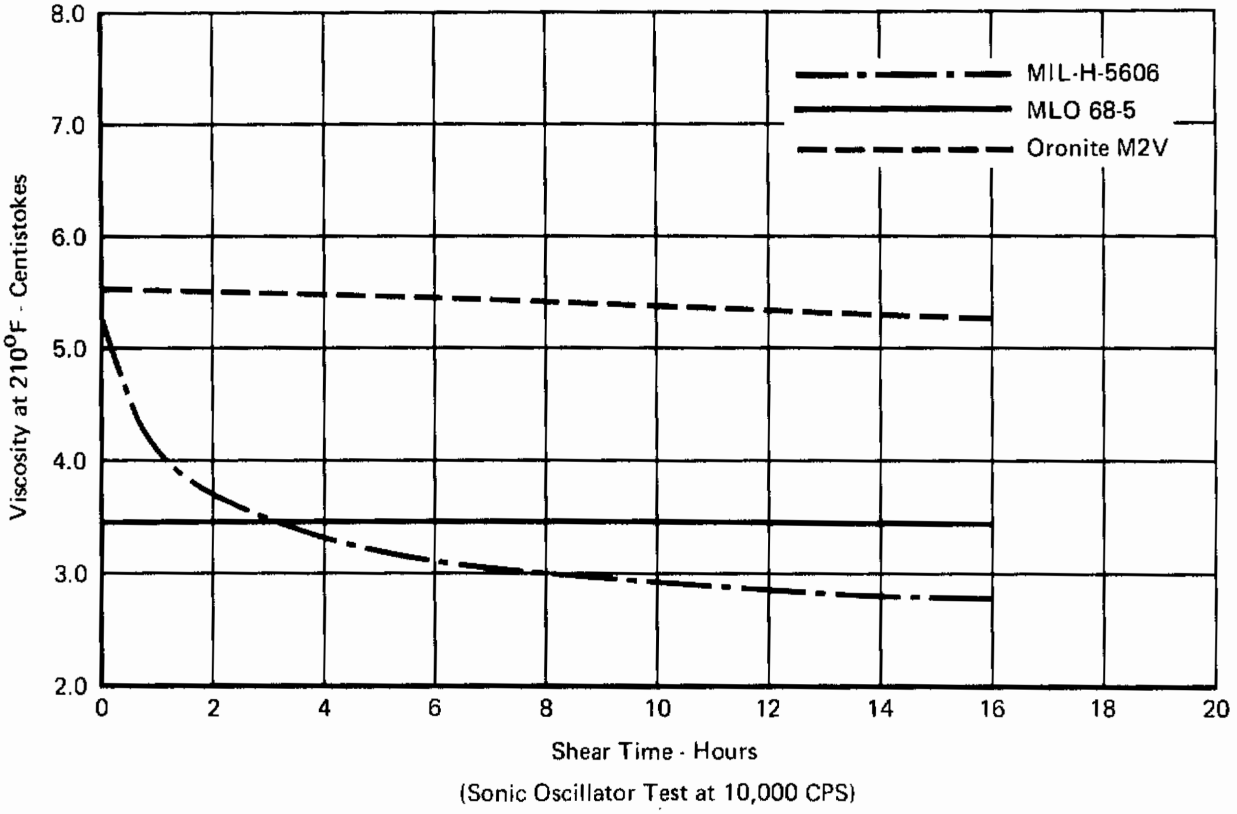


FIGURE 29
SHEAR STABILITY

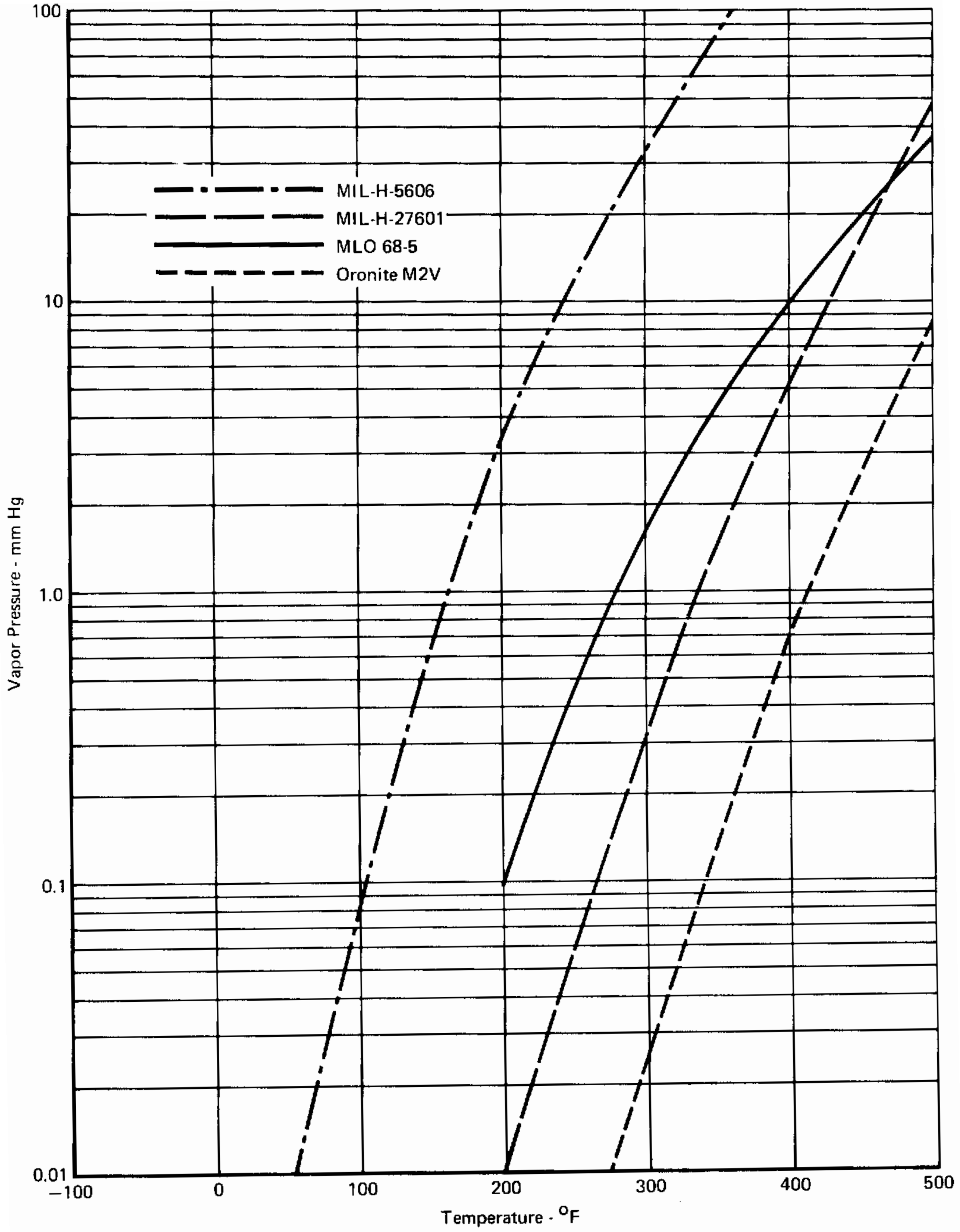


FIGURE 30
VAPOR PRESSURE

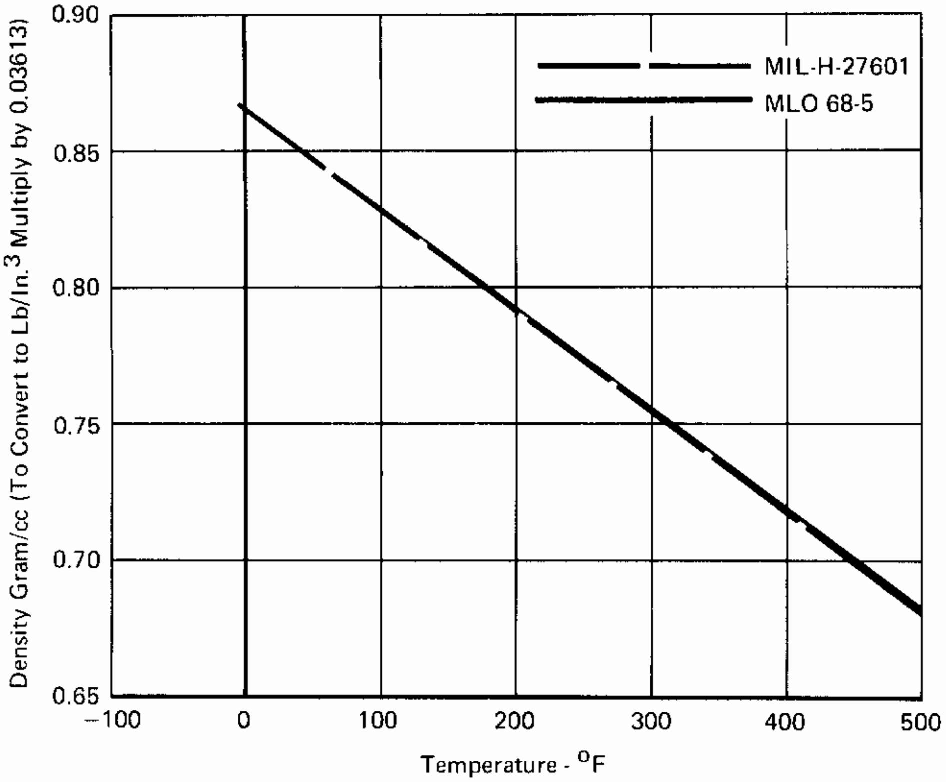


FIGURE 31
DENSITY

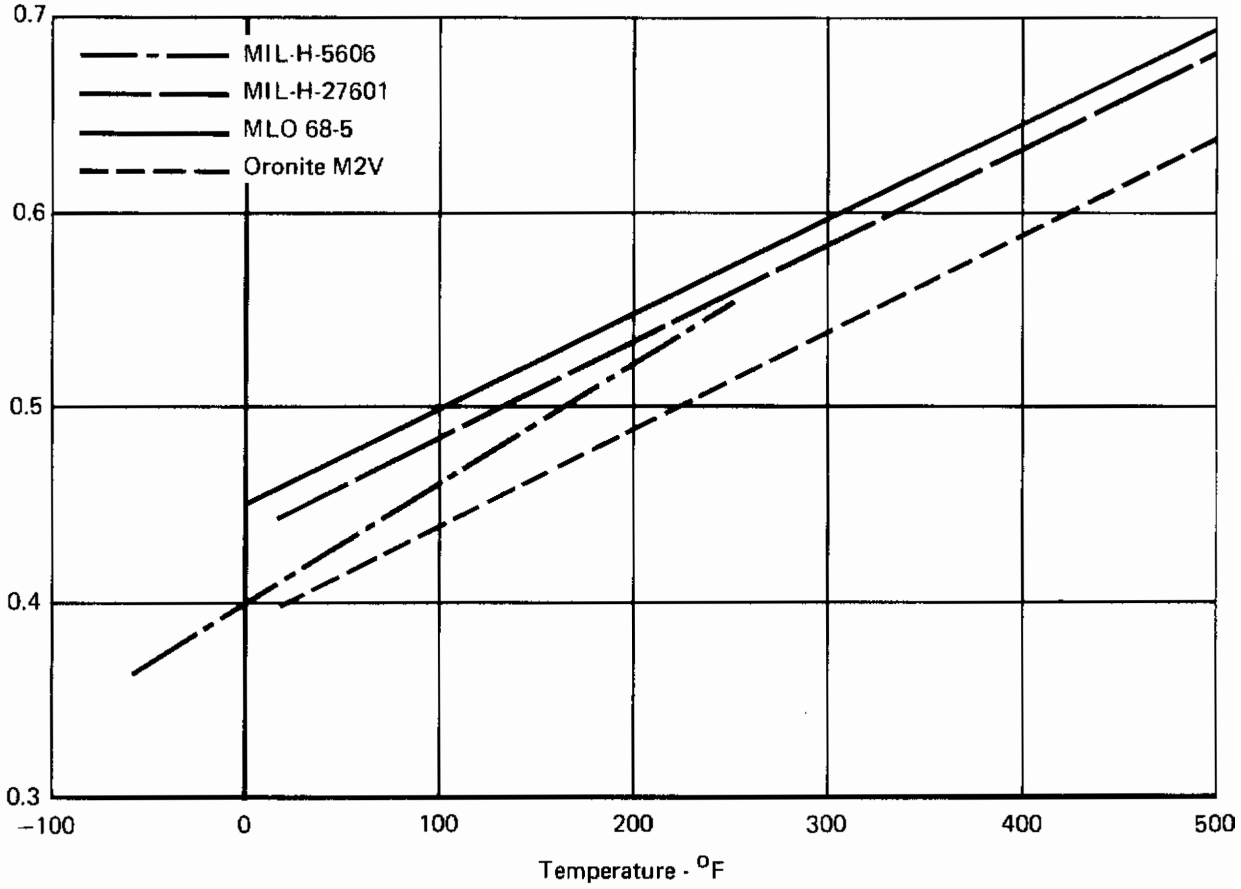


FIGURE 32
SPECIFIC HEAT

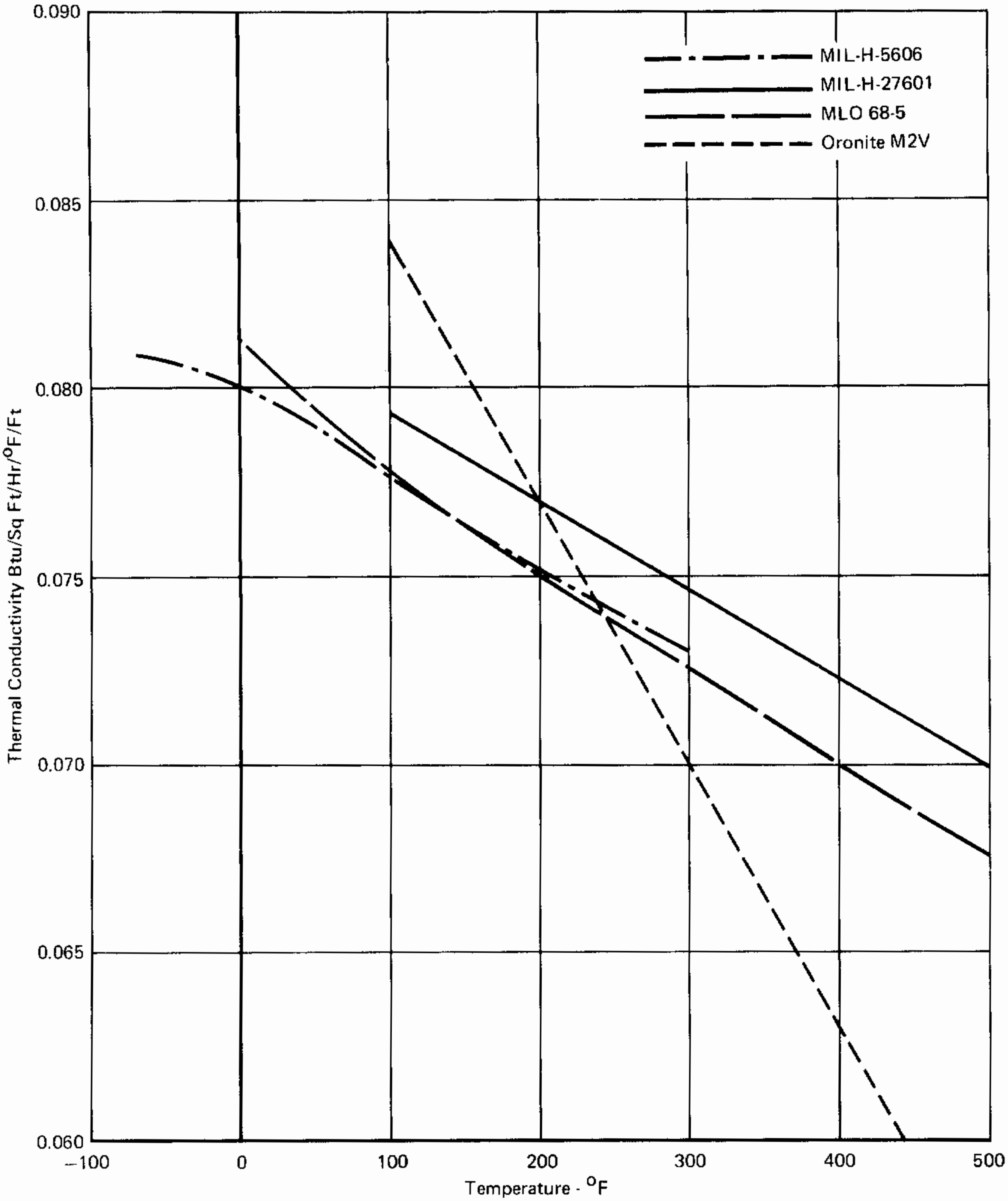


FIGURE 33
THERMAL CONDUCTIVITY

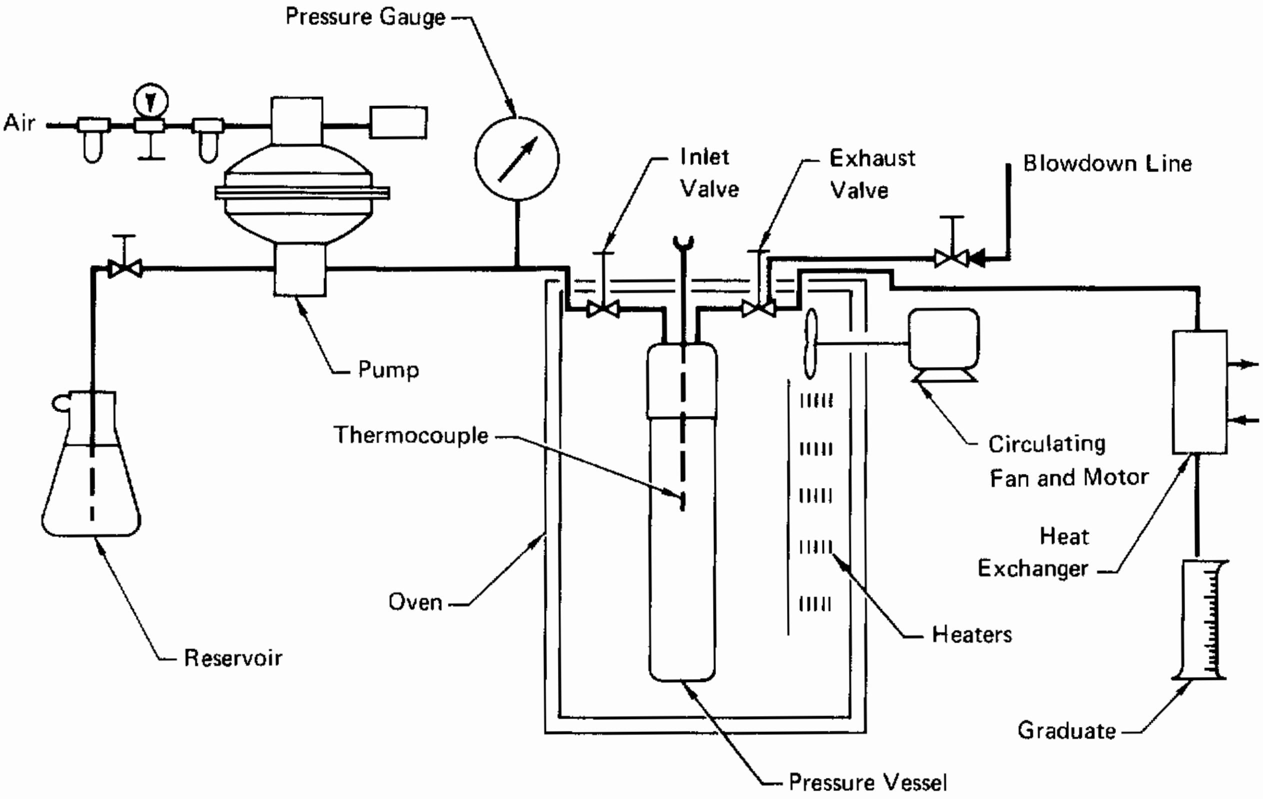


FIGURE 34
SCHEMATIC DIAGRAM OF BULK MODULUS APPARATUS

Note: This is not the same test apparatus required by the MIL-H-83282 specification.

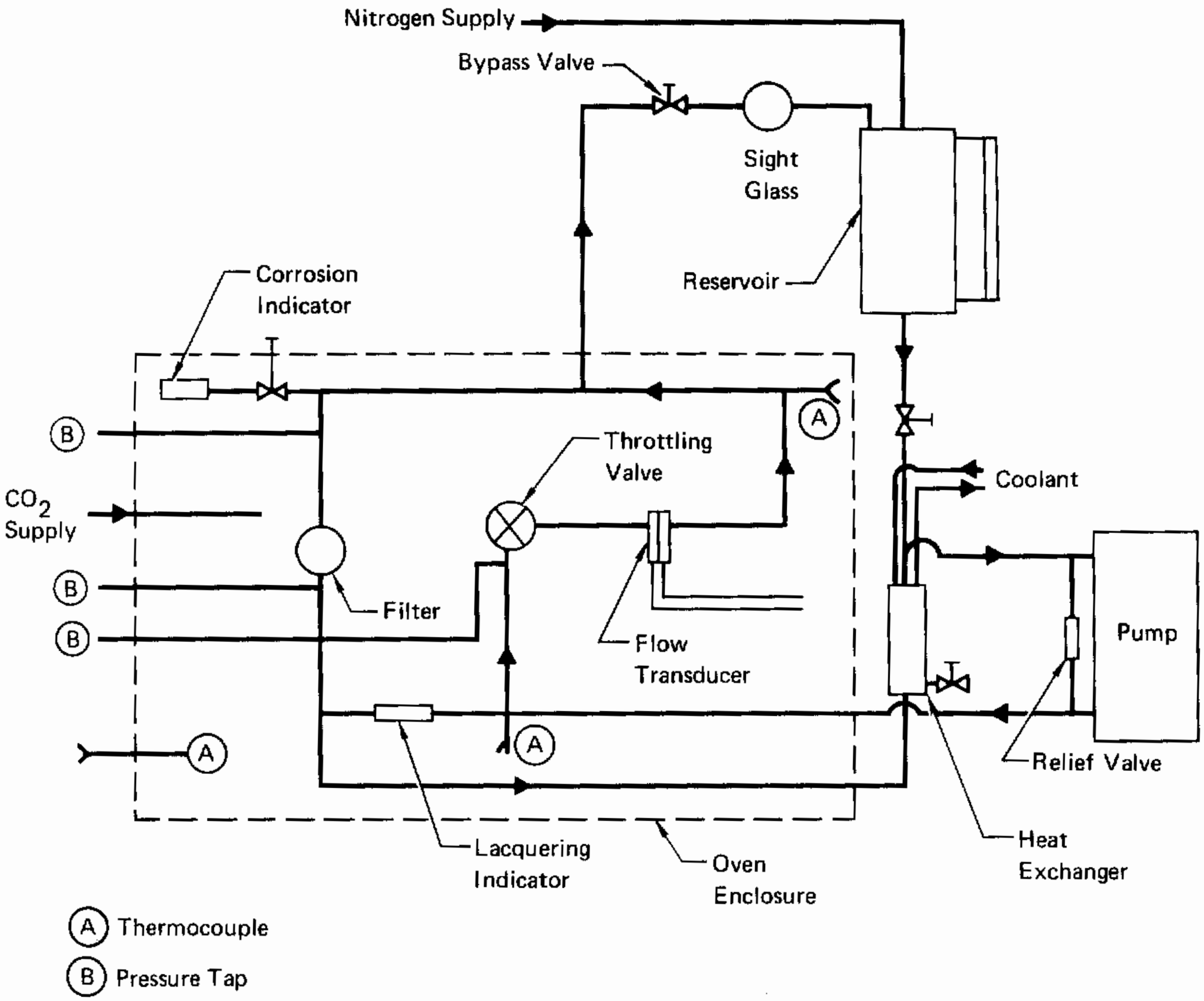


FIGURE 35
FLUID SHEAR AND THERMAL STABILITY TEST CIRCUIT

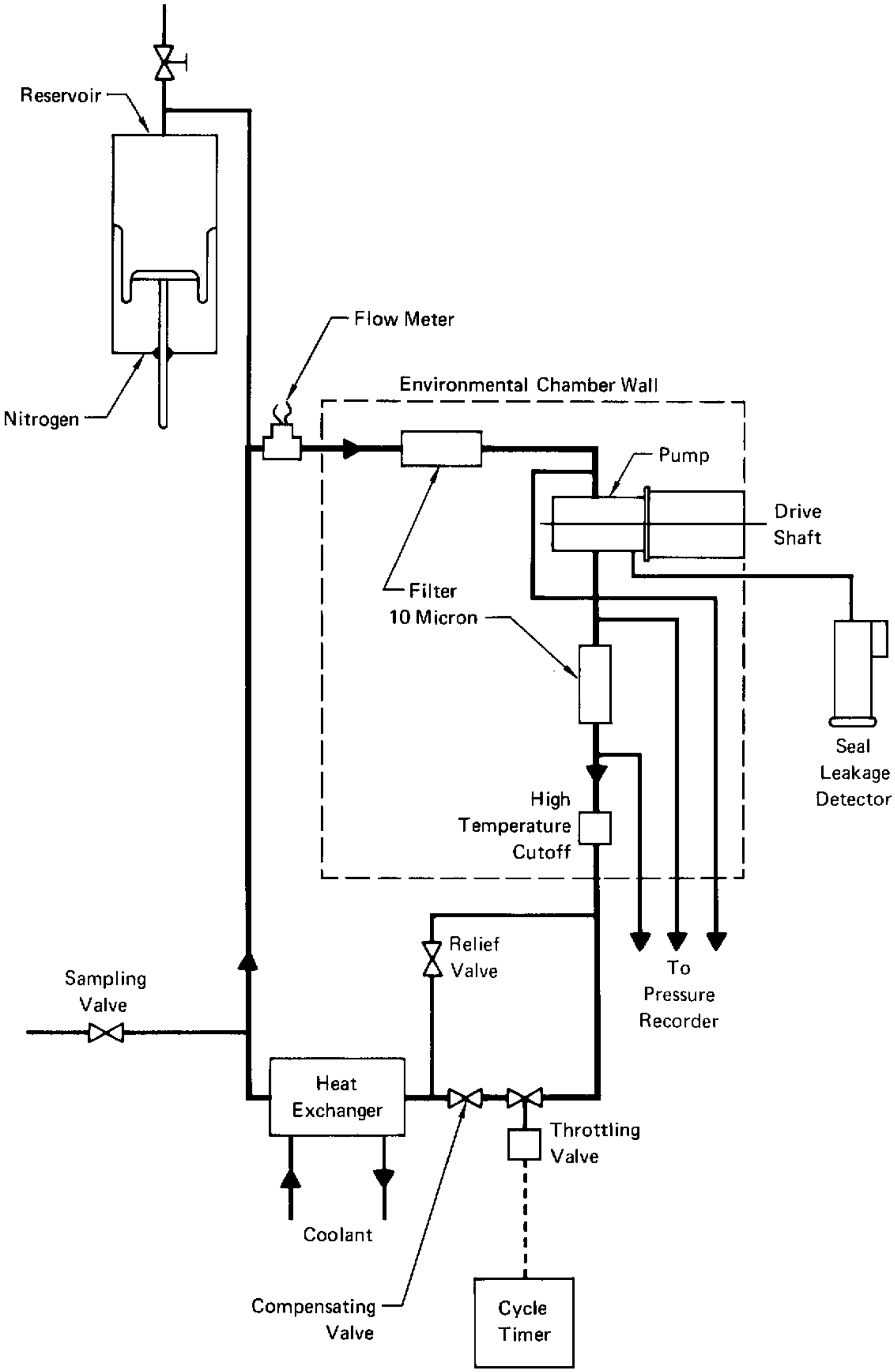


FIGURE 36
HIGH TEMPERATURE HYDRAULIC PUMP CIRCUIT

b. Thermal Data

To compare the thermal characteristics of the fluids, it is necessary to compare their relative performance in a heat exchanger at a given temperature. To establish this comparison the following information, taken from Reference (8) is used.

The Heating of Liquids in Streamline Flow Through Pipes

The equations which are used for streamline flow in pipes with viscous liquids like oils are more complicated than those for turbulent flow.

Experiments with viscous petroleum oils in tubes with inner diameters ranging from 0.39 to 1.57 in. and heated length ranging from 3 to 11.6 ft. were conducted as reported in Reference (9). As a result of this work the authors of Reference (9) arrived at the following formula:

$$\frac{hD}{k} = 1.86 \left(\frac{vDd}{\mu}\right)^{1/3} \left(\frac{\mu C_p}{k}\right)^{1/3} \left(\frac{D}{L}\right)^{1/3} \left(\frac{\mu}{\mu_s}\right)^{0.14}$$

In the experiments μ/μ_s ranged from 0.004 to 9.8.

C_p = specific heat
D = tube diameter
d = fluid density
h = heat transfer coefficient
K = general constant
k = thermal conductivity
L = line length
μ = viscosity
μ_s = viscosity at the tube surface temperature
v = average fluid velocity

All values are taken at the average fluid temperature except μ_s. All factors are dimensionless.

D = constant
L = constant
v = constant

The surface temperature is assumed to be 25°F below the average temperature.

$$\frac{h}{K} = (k^2\ d\ C_p)^{1/3}\ \left(\frac{\mu}{\mu_s}\right)^{0.14}$$

Using the data from Figures 26, 31, 32, and 33, the following comparative values of h/K were calculated.

AVERAGE TEMP.	MIL-H-5606	MIL-H-27601	ML068-5	ORONITE M2V
100°F	2.69	2.7	2.78	2.86
200°F	2.81	2.76	2.77	2.86
300°F	2.86	2.8	2.83	2.8
400°F	----	2.79	2.81	2.61

From the above results, it can be seen that any differences that may exist among the fluids will have very little effect on the heat transfer coefficient.

The numbers calculated for MIL-H-5606 do not take into account its non-Newtonian behavior at high shear rates. The effect of the non-Newtonian characteristics will be to modify the $(\mu/\mu_s)^{0.14}$ term. This term will also become velocity dependent. Due to this effect, it would be unwise to try to predict heat exchanger performance using the performance with MIL-H-5606 as a basis for comparison.

6. RECOMMENDATIONS

a. General

It is recommended that certain fluid characteristics be periodically measured during a flight test program. The bulk modulus is a critical part of the SSAP design and a significant reduction will affect the permissible flight envelope, the motor pump losses and package performance. A significant loss in bulk modulus can be caused by the presence of entrained gases within the package. Due to the interchange of fluid between the package and PC systems, elimination of entrained gases from both the PC systems and the package is important. Removal of entrained gases can be achieved using commercially available equipment. To give good correlation between data, it is necessary to control and/or measure the following fluid characteristics at regularly scheduled intervals.

- Dissolved Gas Content
- Contamination
- Acidity
- Viscosity
- Combustion Indices
- Color
- Vapor Pressure

It is recommended to initially sample and test every 50 hours, extending the time interval on some of the measurements whenever the rate of change of a characteristic is sufficiently low.

b. Justification

The reason for the measurements are as follows:

(1) Dissolved Gas

The dissolved gas content will change due to the contact between the fluid air film on the piston rod and gases produced as a by-product of fluid degradation. It is desirable to reduce this gas to a minimum to prevent a build up of entrained gas and to minimize fluid oxidation.

(2) Contamination

The contamination level will give some idea of wear taking place in the system. When small particles are generated they tend to be in a highly active state, causing fluid breakdown due to catalytic effect of the particles. Since filtration of all particles is impracticable, there will be a tendency for the smaller particles to build up gradually.

(3) Acidity

The acidity changes as a result of fluid degradation due to oxidation or thermal breakdown. The analysis of the acids should determine the source of degradation.

(4) Viscosity

Changes in viscosity can be expected due to fluid degradation. Extreme changes will necessitate corrective action.

(5) Combustion Indices

Some change in the combustion indices can be anticipated as the fluid characteristics change. From previous tests MIL-H-27601 degraded in this respect at high temperatures and there is certain evidence from tests carried out at MCAIR that ML068-5 behaves in a similar manner.

(6) Color

Color changes are normally indicative of a change in the fluid. Correlation between characteristic and color changes would be useful.

(7) Vapor Pressure

Pump performance at high temperatures is dependent on maintaining the vapor pressure characteristic; unfortunately degradation can change this characteristic. Correlation between vapor pressure and acidity may help to determine the useful life of the fluid based on the simpler acidity test.

(8) Filling, Refilling and Air Bleeding

Extreme care will be needed during the filling and air bleeding operations to make certain that the fluid is clean and contains a minimum of dissolved gas.

Before filling, the systems should be purged with nitrogen in order to exclude as much oxygen as possible. The fluid should be degassed under vacuum and then exposed to nitrogen only prior to filling. Using MIL-H-83282 fluid, the system should be filled and bled in the normal way except that much more care should be taken during the bleeding, which should be carried out at low pressure. If possible, a de-aerator should be used on the cart. Once the system is in normal operation, new degassed fluid should be added until sampling shows that the system fluid has a dissolved gas content less than that which would occur at normal atmospheric temperature and pressure, thus giving the fluid an affinity for any entrained gas that may be still in the system.

Additional degassed fluid should be added whenever the dissolved gas content rises above a specified level.

SECTION VI

ACTUATOR DYNAMIC ANALYSIS

1. GENERAL

Analyses were conducted to evaluate the dynamic characteristics of the Secondary Actuator and Survivable Stabilator Actuator Package (SSAP). This included an analysis of the stability and frequency response characteristics of both actuators and an evaluation of the capability of each actuator to comply with design criteria. The results of studies of secondary actuator monitoring techniques and electrohydraulic versus electromechanical characteristics are also reported herein.

The parameters used in the analyses were established theoretically with a few exceptions where experimental data were used. For failure transient and nuisance disconnect analyses, the availability of additional data is expected to make it possible to review the study results reported herein. In all instances, study results will be updated as actuator characteristics and the characteristics of equipment with which they interface become more firmly established and defined through hardware tests.

2. SECONDARY ACTUATOR

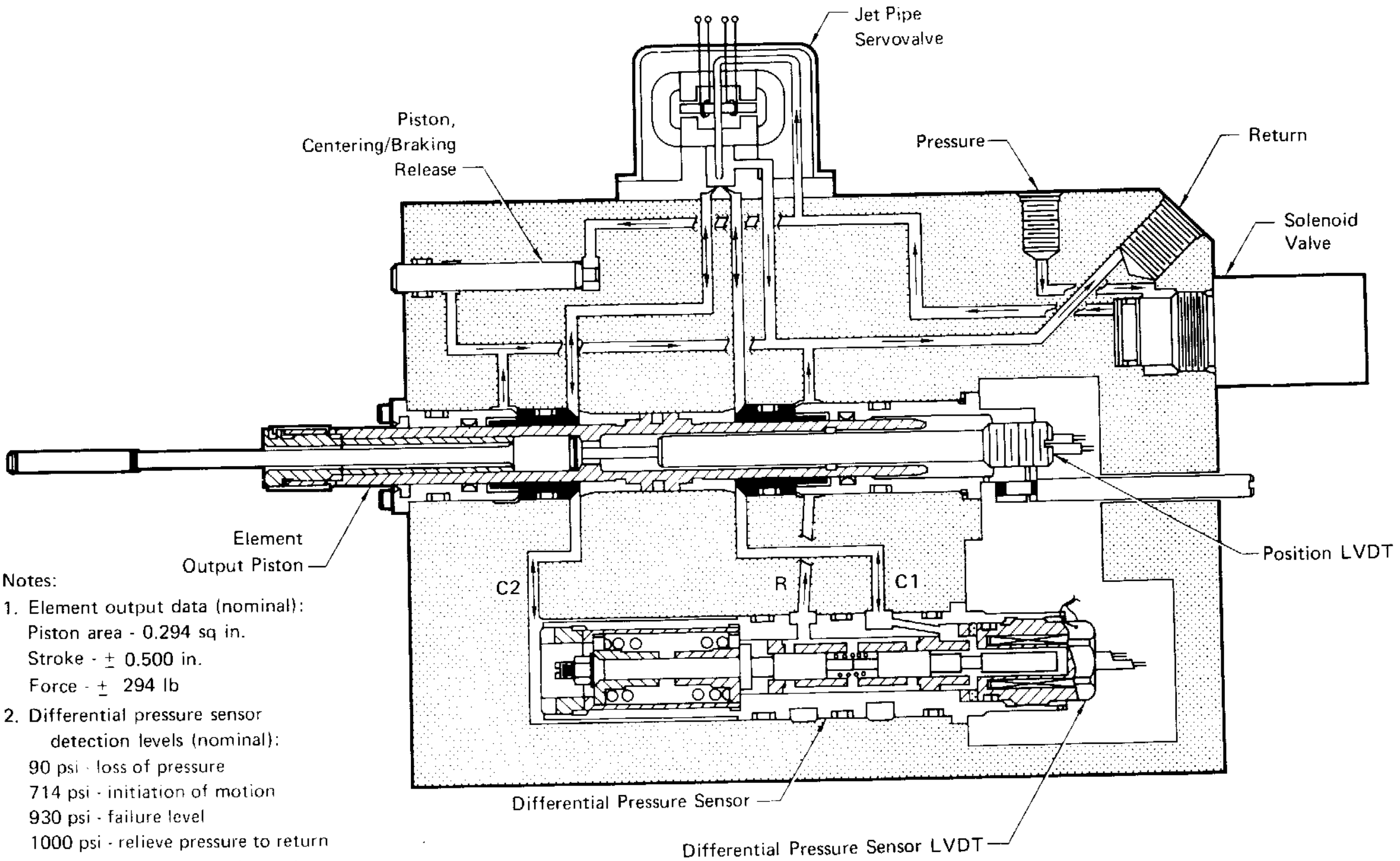
The secondary actuator is a quadruplex, force summing, electrohydraulic servomechanism. It is a self-contained unit consisting of four independent servocontrolled elements coupled to a common output, each element mechanized as shown in Figure 37.

In the dynamic analysis of the secondary actuator it is convenient to use two math models. The force summing model presented in Figure 38 is essential when investigating failure transients since the effects of force summing, dynamic loading, and multiloop operation can be easily included. The less complex performance model presented in Figure 39 suffices when investigating performance characteristics such as frequency response, where operational characteristics can be approximated by a single servo loop. The parameters used in the analysis are shown in Table XXII. The gains shown reflect the current status of the secondary actuator and were selected to satisfy the threshold, frequency response, failure transient, and nuisance disconnect requirements. They are subject to change as development progresses. The analysis and discussion which follows is based in large part on information from Reference 5.

a. Stability

The stability characteristics of the secondary actuator can be ascertained by constructing a transfer function from Figure 39. By eliminating filter dynamics and nonlinearities, a second order transfer function is obtained:

$$\frac{X_1}{V_i} = \frac{(K_a K_V)}{A_V \tau_V} \quad \frac{1}{S^2 + \frac{1}{\tau_V} S + \frac{K_a K_V H}{A_V \tau_V}} \qquad (1)$$


Jet Pipe Servovalve
Piston, Centering/Braking Release
Pressure
Return
Solenoid Valve
Element Output Piston
Position LVDT
C2
R
C1
Differential Pressure Sensor
Differential Pressure Sensor LVDT

Notes:

1. Element output data (nominal):
   Piston area - 0.294 sq in.
   Stroke - ± 0.500 in.
   Force - ± 294 lb
2. Differential pressure sensor detection levels (nominal):
   90 psi - loss of pressure
   714 psi - initiation of motion
   930 psi - failure level
   1000 psi - relieve pressure to return
3. The centering/braking release piston is shown in the locked/braked position. The centering/braking mechanism is spring driven and pressure released.
4. In case of element failure, pressure is shut off by the solenoid and the piston is by-passed through the jet pipe servo valve receiver.

FIGURE 37
HYDRAULIC SCHEMATIC, SINGLE ACTUATOR ELEMENT

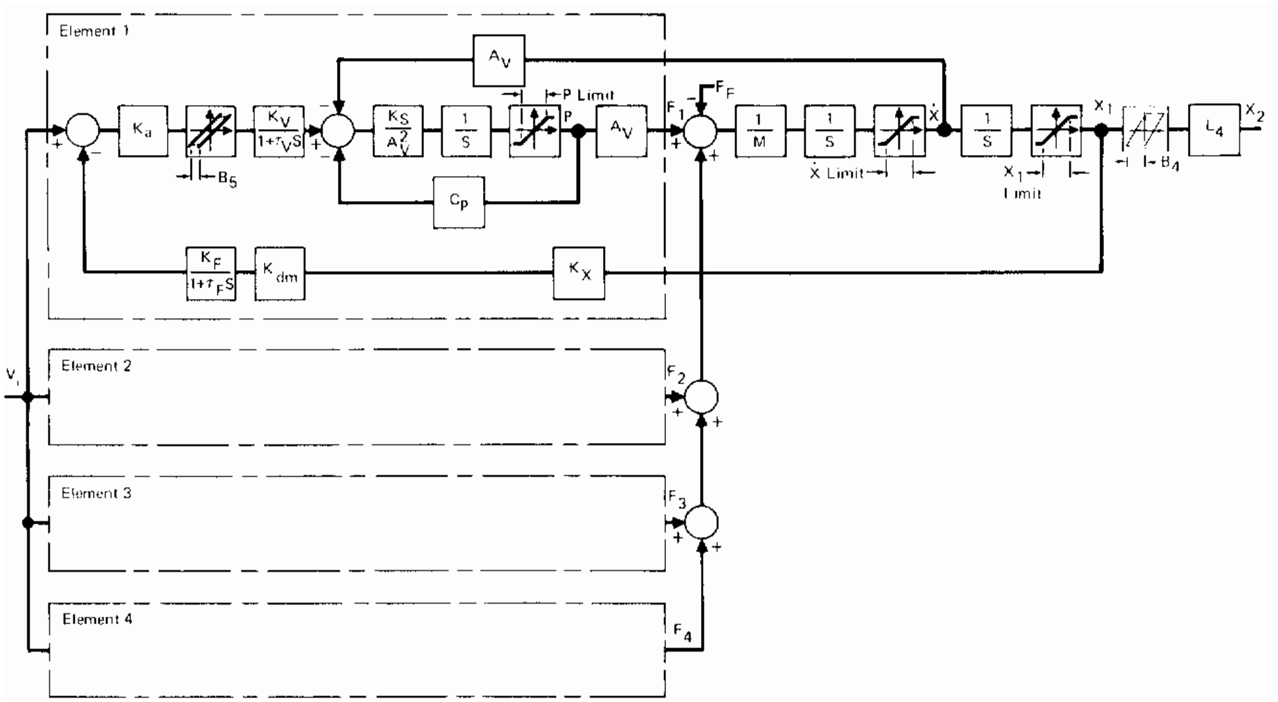


FIGURE 38
MATH MODEL - FORCE SUMMING

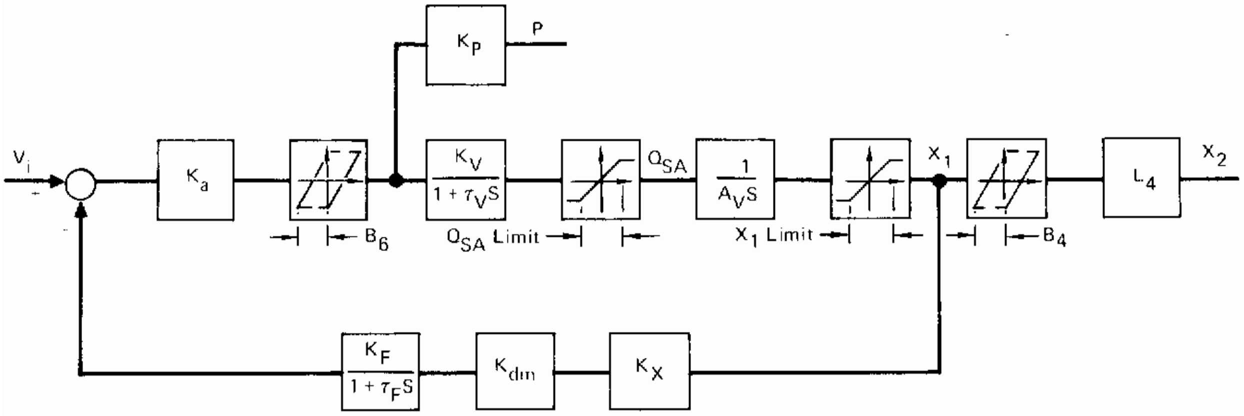


FIGURE 39
MATH MODEL - PERFORMANCE

TABLE XXII
SECONDARY ACTUATOR PARAMETERS

Symbol		Value	Units
A_V	Effective Secondary Actuator Piston Area	0.294	$In.^2$
B_4	Linkage Freeplay Secondary Actuator Elements to Summing Link	0.002	In.
B_5	T-Valve Hysteresis	0.0	Ma
B_6	Hysteresis Due to Friction Effects	0.047	Ma
C_p	Ratio of Flow Gain to Pressure Gain	0.000369	Cis/Psi
F_F	Coulomb Friction (for 4 Elements)	16.0	Lb
F_1	Force Output of Element 1 of Secondary Actuator	Definition	Lb
F_2	Force Output of Element 2 of Secondary Actuator	Definition	Lb
F_3	Force Output of Element 3 of Secondary Actuator	Definition	Lb
F_4	Force Output of Element 4 of Secondary Actuator	Definition	Lb
G_1	Preamplifier Gain	0.0142	Ma/Volt
G_f	Filter Gain	0.05	Ma/Volt
H	Feedback Gain, $K_X K_{dm} K_F$	5.19	Volts/Inch
K_a	Servo Amplifier Gain	57.6	Ma/V
K_{dm}	Demodulator Gain	1.25	VDC/VAC
K_F	Feedback Amplifier Gain	0.296	V/V
K_L	Open Loop Gain	122	Sec^{-1}
K_{p1}	Pressure Gain for 3000 psi Supply	325	Psi/Ma
K_{p2}	Pressure Gain for 1400 psi Supply	152	Psi/Ma
K_S	Structural Spring Constant	1.46×10^5	Lb/In.
K_V	Servo Valve Gain	0.12	Cis/Ma
K_X	LVDT Scale Factor	14.0	Volts/In.
L	Total Secondary Actuator Piston Stroke	1.0	Inch
L4	Linkage Ratio - Summing Link to Elements of Secondary Actuator	1.37	In./In.
M	Effective Mass of Linkage	0.456	$Lb\text{-}Sec^2/In.$
N_1	Number of Operating Elements on 3000 psi	Definition	
N_2	Number of operating Elements on 3000 psi Opposing a Failed Element or an Element with a Large Tolerance Buildup	Definition	
$P, \Delta P$	Pressure Differential Across a Secondary Actuator Piston	Definition	Psi
P_M	Maximum Differential Pressure across a Secondary Actuator Piston	1000	Psi
P_R	Return Pressure	Definition	Psi
P_S	Supply Pressure	Definition	Psi
P_T	Tripout Pressure of Differential Pressure Sensor	930 Psi	Psi
Q	Flow	Definition	Cis
Q_{SA}	Flow Rate for Secondary Actuator Element	Definition	Cis
V_e	Error Signal into Servo Amplifier	Definition	Volts
V_i	Input Command Signal	Definition	Volts
V_S	Signal in One Element Due to Tolerance Buildup	Definition	Volts
X_1	Displacement - Secondary Actuator Elements	Definition	Inch
X_2	Displacement - Secondary Actuator Summing Link	Definition	Inch
ξ	Damping Ratio	Definition	
τ_F	Feedback Amplifier Time Constant	0.001	Sec
τ_V	Servo Valve Time Constant	0.00177	Sec

Using the parameters from Table XXII the damping ratio (ξ) is found to be 1.07, which indicates a very stable configuration. This result confirms past experience, which has shown that small electrohydraulic servo actuators with relatively insignificant dynamic loading are singularly free of closed loop stability problems. Moreover, it also shows that the present open loop gain ($K_L = 122$) can be increased appreciably, without causing stability problems, should a need be indicated as development progresses.

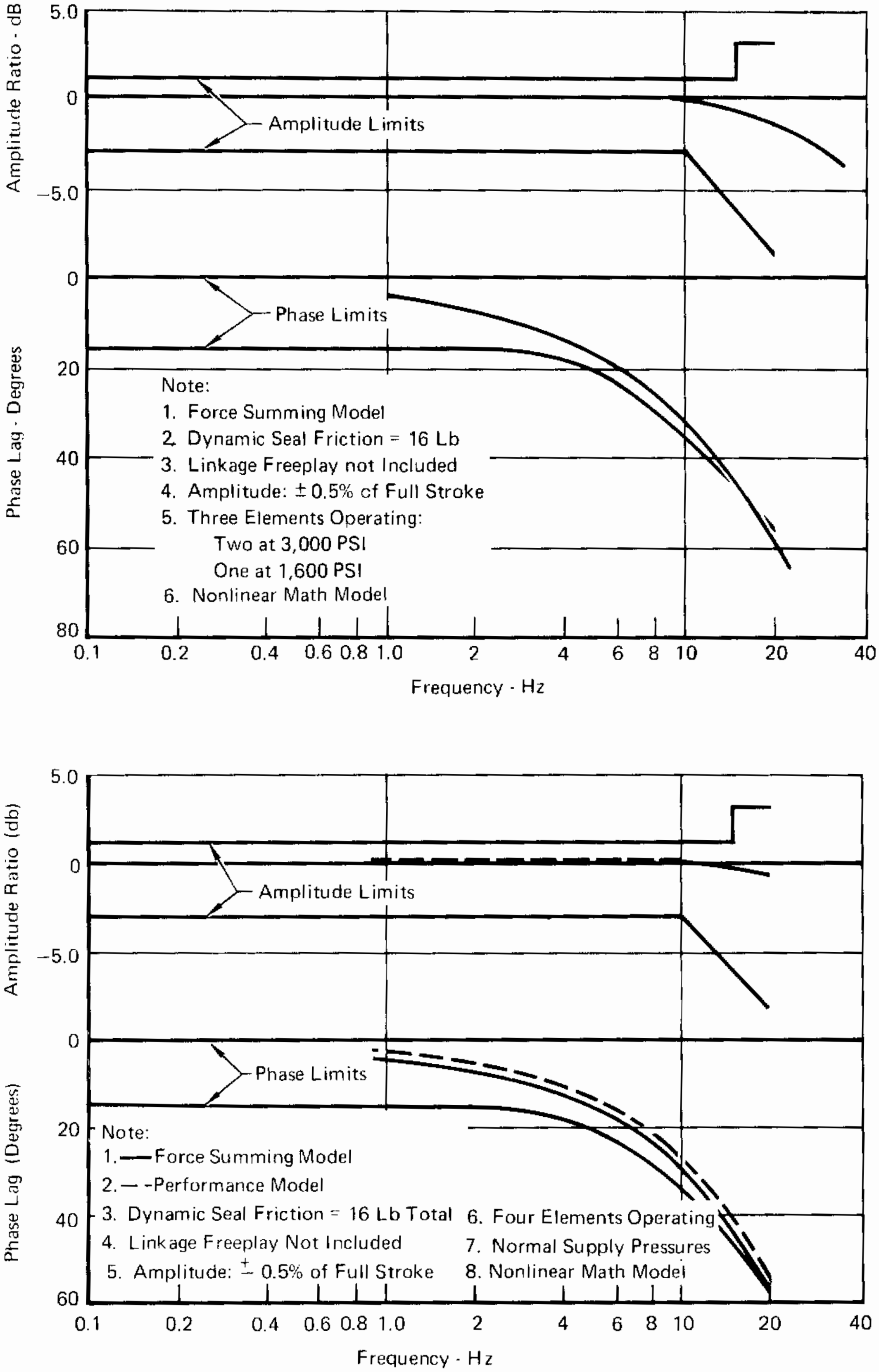
b. Frequency Response

The frequency response characteristics represent the overall performance capability of the secondary actuator with nominal parameters. On a linear basis, the frequency response of the secondary actuator can be approximated by a first order lag with an open loop gain (K_L) of 122. However, nonlinearities in the secondary actuator result in significant deviation from the linear approximation and their effect was evaluated. Nonlinearities included in the investigation of the frequency response include:

- dynamic seal friction (coulomb)
- flow limits
- electrohydraulic valve hysteresis
- force limits

The results of the frequency response evaluation are shown in Figure 40 and Figure 41. Figure 40 shows the small signal frequency response capabilities for four and three elements operating, respectively. With four elements operating, the phase lag and amplitude ratio are within envelope limits; while with three elements operating, the phase lag exceeds the envelope at higher frequencies. For small signal response, the dynamic seal friction in concert with the relatively low pressure gain of the single stage electrohydraulic valve contributes significantly to the phase lag. The hysteresis of the electrohydraulic valve was found to be only a small percentage of the hysteresis caused by friction. Force and flow limits were not encountered. The effect of linkage freeplay was not included since it is outside the secondary actuator loop and, therefore, can properly be lumped and considered with the freeplay of control system linkage between the secondary and surface actuators.

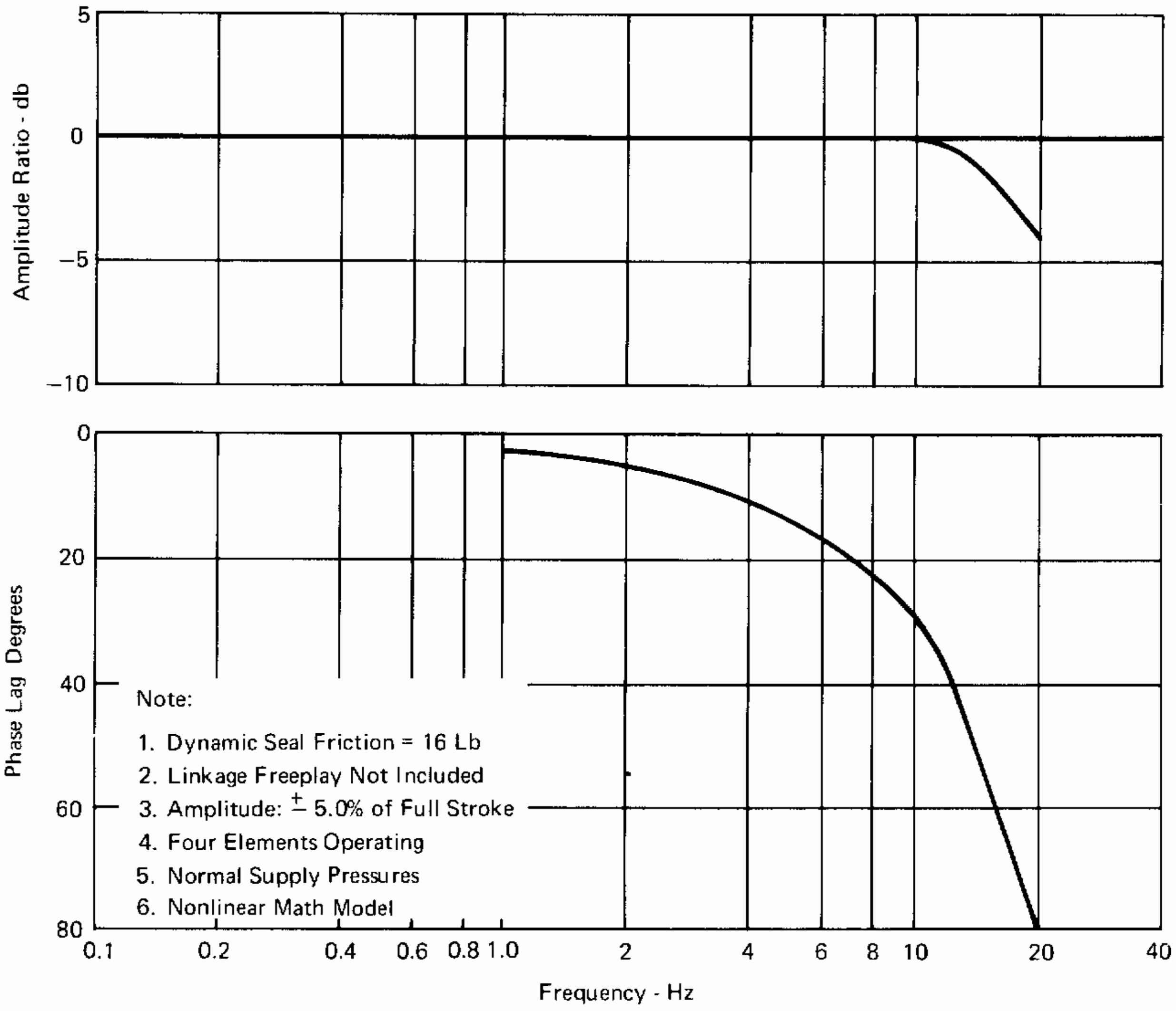
The large signal frequency response shown in Figure 41 indicates a greater phase lag at the higher frequencies than would be predicted using a linearized model. The additional phase shift is due to flow limiting (velocity saturation) in the electrohydraulic servo valve. Force limiting (acceleration saturation) was not encountered due to the large signal commands and is not ordinarily a problem with small electrohydraulic servo actuators. The effects of structural spring rates and dynamic loading on frequency response were also investigated and found to be negligible.



Note:
1. Force Summing Model
2. Dynamic Seal Friction = 16 Lb
3. Linkage Freeplay not Included
4. Amplitude: ± 0.5% of Full Stroke
5. Three Elements Operating:
Two at 3,000 PSI
One at 1,600 PSI
6. Nonlinear Math Model

Note:
1. —— Force Summing Model
2. – – Performance Model
3. Dynamic Seal Friction = 16 Lb Total
4. Linkage Freeplay Not Included
5. Amplitude: ± 0.5% of Full Stroke
6. Four Elements Operating
7. Normal Supply Pressures
8. Nonlinear Math Model

FIGURE 40
SMALL SIGNAL FREQUENCY RESPONSE



Note:

1. Dynamic Seal Friction = 16 Lb
2. Linkage Freeplay Not Included
3. Amplitude: ± 5.0% of Full Stroke
4. Four Elements Operating
5. Normal Supply Pressures
6. Nonlinear Math Model

FIGURE 41
LARGE SIGNAL FREQUENCY RESPONSE

In summary, the frequency response characteristics appear to be essentially consistent with the system requirements, provided the nominal characteristics of Table XXII are realized. However, if higher than nominal friction levels are encountered, which result in low amplitude frequency response degradation, the open loop gain can be increased in order to obtain relief. An increase in open loop gain is permissible only to the extent that nuisance disconnect characteristics permit.

c. Threshold

Threshold is defined here as the maximum peak to peak command signal for which no output displacement occurs. It can be expressed in terms of a percentage of total secondary actuator stroke. The threshold in percent of total secondary actuator stroke can be determined from:

$$\% X_1 = \frac{200 F_F}{H K_a A_V (N_1 K_{P_1} + K_{P_2}) L} \qquad (2)$$

The symbols are defined in Table XXII. Equation (2) represents the threshold which is the result of coulomb friction only. Estimates of the effect of electrohydraulic valve hysteresis were found to be insignificant relative to that of coulomb friction and are not considered here. It was further assumed that leakage effects on threshold are negligible. Using the nominal parameters from Table XXII, the threshold as a function of the number of elements operating is:

N_1	No. of Elements	% X_1	% Allowable
3	4	0.032	0.04
2	3	0.045	0.04
3	3	0.037	0.04

The permissible threshold is a maximum of 0.04 percent of full stroke with three or more elements operating. As indicated, the threshold exceeds the permissible value with three elements (including the 1600 psi channel) operating and this is reflected in the small amplitude frequency response results of Figure 40. It should be noted that N_1 represents the number of operating elements which are supplied by 3000 psi hydraulic systems. From equation (2), it can be seen that threshold increases with increases in friction (F_F) and decreases with increases in servo amplifier gain (K_a). Thus, an increase in servo amplifier gain can be used to compensate for an increase in friction provided other operating characteristics permit.

d. Failure Transients

A failure transient will result whenever an active failure occurs in an element. When such a failure occurs, the actuator output is displaced until a force balance is achieved. The force balance is maintained until the failed element is switched off line. On third failure, a brake is applied to hold position when the last two elements are switched off line.

Under dynamic conditions, failure transients are influenced by control linkage and surface actuator dynamic characteristics. The effect of these characteristics on failure transients are currently under investigation. However, when dynamic characteristics are ignored, the failure transients can be approximated by the following equation:

$$\% \, X_1 = \frac{100 \; P_M}{L \; H \; K_a \; (N_2 K_{P_1} + K_{P_2})} \tag{3}$$

Equation (3) expresses the output displacement resulting from an active failure in terms of the percentage of total stroke. The limits on output displacement have been specified at 0.56, 0.95, and 2.2 percent of full stroke for first, second, and third failures, respectively. It should be noted that the N_2 term in equation (3) represents the number of elements on 3000 psi supply which oppose the failed element. The tripout pressure (P_M) may be 1000 psi for an active failure, which is assumed here, or 930 psi for a slowover failure.

The failure transient calculations are summarized in Table XXIII. Worst case failures are those in which a failure occurs in an element supplied by a 3000 psi supply. Also listed in Table XXIII are the number of elements in operation, the type of failure, and the maximum signal difference between one element and other elements to cause tripout. The maximum signal difference required to cause tripout can be determined from:

$$\% \, V_S = \frac{100 P_T}{K_a \; K_{P_1} \; V_{i_{MAX}}} \left[\frac{1}{1 - \dfrac{1}{N_2 + 1 + K_{P_2}/K_{P_1}}} \right] \tag{4}$$

The difference in signal between elements is expressed in terms of the percent of maximum signal. Equation (4) is for a failure which occurs in an element supplied by 3000 psi. A similar expression can be developed for a failure in the element supplied by 1600 psi.

TABLE XXIII
FAILURE TRANSIENT SUMMARY

Failure Transient (Percent of Full Stroke)	Maximum Element Difference to Cause Failure (Percent of Maximum Signal)	Number of Elements		Type of Failure
		P_s = 1400 Psi K_{P_2} = 152 Psi/ma	P_s = 3000 Psi K_{P_2} = 325 Psi/ma	
0.35	4.8	1	3	1st Low Pressure
0.42 Worst Case	2.74	1	3	1st High Pressure
0.52	5.1	1	2	2nd Low Pressure
0.71 Worst Case	3.28	1	2	2nd High Pressure
1.03	6.1	1	1	3rd Low Pressure
2.2 Worst Case	6.1	1	1	3rd High Pressure
1.03	3.8	0	2	3rd High Pressure

The signal developed by differential pressure sensors across each element is zero for differential pressures less than 700 psi and is linear from 700 to 1000 psi. The threshold signal level for tripout is nominally set to correspond to 930 psi. As a result of the deadband of ±700 psi, the failure for first or second failures can be generated entirely within the failed element. On third failure, both of the remaining elements may contribute to the failure signal. On a slowover failure, for example, the failure signal threshold would be reached when the differential pressure in each of the two remaining elements reaches approximately 815 psi.

Secondary actuator design parameters are such that a failed element cannot overpower a good element on third failure. The maximum differential pressure which can be developed across an element is 1000 ± 20 psi. Thus the maximum pressure difference between elements will be 40 psi which is equivalent to approximately 12 pounds. Therefore, the incorporation of the force limiting valve in the design will maintain a "force fight" on third failure which produces an unbalanced force below the friction level to be expected from the two already disengaged elements.

e. Nuisance Disconnect

A nuisance disconnect or tripout is one which occurs as a result of tolerance buildups between the four secondary actuator elements. The tolerance buildup between elements may be the result of differences in command signals, excitation to LVDT's, LVDT tracking, valve nulls, LVDT nulls, etc., i.e., any differences which result in a differential pressure buildup between elements.

The potential for nuisance disconnect may be analyzed in a number of different ways in that the tolerances may be handled additively, root sum squared, or dealt with statistically utilizing different distributions. It is MCAIR's view that the statistical approach with normal distribution represents the most realistic approach since the study concerns randomly distributed variables. The results of such an analysis are summarized from Reference 5. In addition, the sensitivity of the secondary actuator to nuisance disconnect is evaluated for a special set of conditions.

An examination of the following equation provides some insight into the sensitivity of the secondary actuator to nuisance disconnects:

$$\Delta P = 1 - \left[\frac{1}{N_2 + 1 + K_{P_2}/K_{P_1}} \right] V_S K_a K_{P_1} \qquad (5)$$

This equation applies only for a special set of conditions, namely for a tolerance buildup in only one element on 3000 psi supply. The term V_S represents the net tolerance buildup in the element and ΔP represents the resulting differential pressure in that element. Note that N_2 represents the number of operating elements on 3000 psi supplies which are opposing the element with the tolerance buildup.

With four elements operating and a tolerance buildup equivalent to one percent of maximum signal in an element on 3000 psi, the resulting differential pressure in that element is approximately 340 psi. After one 3000 psi element failure, the differential pressure buildup is approximately 280 psi for one percent tolerance, and after two 3000 psi element failures the differential pressure buildup is approximately 150 psi for a one percent tolerance. For a tripout pressure of 930 psi, a tolerance of 2.7, 3.3, and 6.2 percent of maximum signal would be required to produce a nuisance disconnect for four, three and two elements operating, respectively.

Thus, the sensitivity to tolerance buildup is greatest with four elements operating. Also evident in examining equation (5) is the effect of servo amplifier gain (K_a) on the buildup of differential pressure. For a given tolerance (V_S), an increase in servo amplifier gain results in a corresponding increase in differential pressure. Therefore, while an increase in open loop gain improves threshold and failure transient characteristics, it aggravates the tendency for nuisance disconnect.

Nuisance disconnect can also occur when the pressure in the extend chamber of the secondary actuator falls below approximately 100 psi. When the pressure falls below this value, a spring in the differential pressure sensor forces the error LVDT hardover and provides a failure signal. This type of nuisance disconnect is most likely to occur in the element on 1600 psi where, at no load and no signal conditions, the pressure on each side of the piston is approximately 500 psi. With a pressure gain in this element of 152 psi/ma, the pressure in the extend chamber will decrease (or increase) approximately 75 psi for each ma of error signal due to tolerance buildup. For tripout to occur, it would be necessary to have an error signal due to tolerance buildup in excess of 5 ma. Since 5 ma represents an extremely large error signal to result from tolerance buildup under normal operating conditions, the probability of a nuisance disconnect due to a reduced pressure in the extend chamber is considered to be relatively slight.

f. Tolerance Analysis

The statistical tolerance analysis discussed here is based primarily on material contained in Reference 5. Steady state conditions with no output force on the load and a maximum input signal in conjunction with maximum deflection of the output are assumed. It should be noted that not all of the tolerances in the associated electronics were available when this analysis was conducted. The analysis will therefore be repeated when all tolerances have been defined.

A block diagram of all four channels is shown in Figure 42 which is based on the block diagram of Figure 43. For steady state conditions, the sum of the forces applied to the load is zero.

$$\sum_{i=1}^{4} F_i = 0 \tag{6}$$

$$\sum_{i=1}^{4} (E_i + M_i/K_i - L_{ni}H_i\ K_iV_iA_{vi} - X_1 \sum_{i=1}^{4} H_iK_iV_iA_{vi} = 0 \tag{7}$$

$$X_1 = \frac{\sum_{i=1}^{4} (E_i + M_i/K_i - L_{ni}H_i)\ K_iV_iA_{vi}}{\sum_{i=1}^{4} H_iK_iV_iA_{vi}} \tag{8}$$

$K_i = (G_iK_{a_i})$; $H_i = (G_{f_i}/G_i)\ K_{dmi}K_{x_i}$; L_{ni} = LVDT Null

$V_i = (K_{p_i})$; M_i = Valve Null ; E_i = Input

Using the above equations, the differential pressure in each element may be found from the following equation:

$$P_i = (E_i + M_i/K_i - L_{ni}H_i - X_iH_i)\ K_iV_i \tag{9}$$

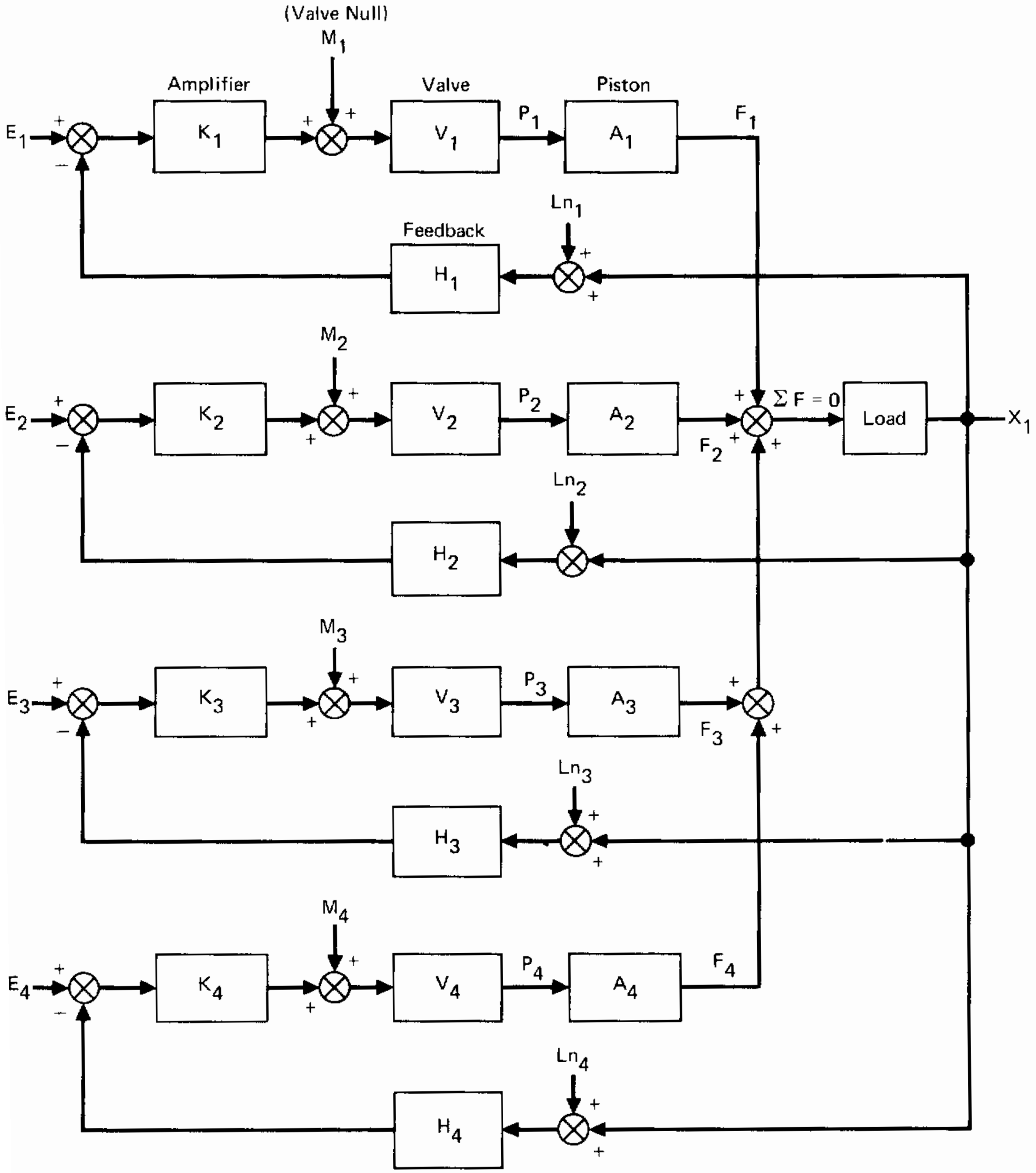
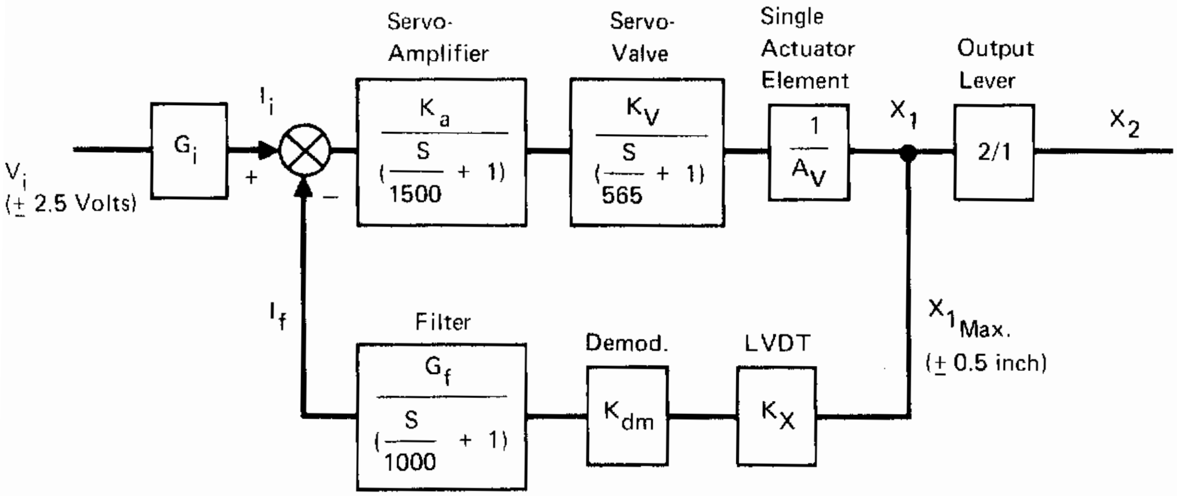


FIGURE 42
FORCE SUM BLOCK DIAGRAM, STATIC CONDITIONS



Nominal Gains

G_i = 0.05 ma/volt = $1/R_i$ = 1/20 K Ω
$\dot{K}_a$ = 1194 ma/ma
K_V = 0.12 (in.^{3}/sec)/ma
A_V = 0.294 in.^{2}
K_X = 14.0 vac/in.
K_{dm} = 1.26 vdc/vac
G_f = 0.0142 ma/volt = $1/r_f$ = 1/70.5 K Ω

Open Loop Gain

$$K_L = \frac{I_f}{I_i} = \frac{K_a K_V K_X K_{dm} G_f}{A_V S} = \frac{122}{S} \quad \text{(At Low Frequency)}$$

FIGURE 43
SECONDARY ACTUATOR SERVOLOOP BLOCK DIAGRAM

The pressure in each element is a function of the gains and nulls in all elements.

The values for gain and null tolerances are shown in Table XXIV. Normal distribution was assumed for the tolerances. For gains, the specified gain was assumed to be the mean and the specified tolerance was assumed to be the 3 sigma value. For nulls, the mean was assumed to be zero, except for the element on 1600 psi, for which the specified null was assumed to be the 3 sigma value. Table XXIV includes the mean, the maximum or 3 sigma tolerance, and the standard deviation for each variable.

TABLE XXIV
BASELINE CHARACTERISTICS

Variable	Mean	Sigma	Tolerance (3x Sigma)
1. Amplifier Gain (K_i) i = 1, 2, 3, 4	18.8 ma/V	0.0627	± 1.0%
2. Valve Pressure Gain (V_i) i = 1, 2, 3	325 PSI/ma	10.8	± 10.0%
3. Feedback Gain (H_i) i = 1, 2, 3, 4	16 Volts/Inch	0.0583	± 1.1%
4. Piston Area (A_i) i = 1, 2, 3, 4	0.294 $(\text{Inch})^2$	0.00157	± 1.6%
5. Input Signal (E_i) i = 1, 2, 3, 4	8 Volts	0.0133	± 0.5%
6. Valve Null (M_i) i = 1, 2, 3	0 ma	0.217	± 8.15%
7. LVDT Null (L_{ni}) i = 1, 2, 3, 4	0 Inch	0.000357	± 0.21%
8. V_4, Valve Pressure Gain	152 PSI/ma	5.06	± 10.0%
9. M_4, Valve Null	0.16 ma	0.184	± 6.9%

The results of the analysis show that the 3 sigma differential pressure for any of the four elements does not exceed 520 psi. The results also show the most significant tolerances to be those of the input signal, LVDT excitation, and LVDT tracking. Valve nulls, LVDT nulls, and preamplifier tolerances also contribute significantly to the tolerance buildup.

g. Output Velocity and Centering Time

The secondary actuator is required to travel through full stroke in 0.6 seconds with 3 elements and a 1000 psi pressure differential. Following the loss of each element, the slew rate is allowed to decrease by 25 percent of the speed with four active elements.

For the secondary actuator, the slew rate of the piston must be 1.67 inches/sec. Each secondary actuator servo valve will have a nominal no load flow of 0.96 CIS when supplied by 3000 psi. The no load slew rate would be 0.555/0.294 = 1.89 inches/sec. with 1000 psi supply pressure disregarding the effects of bypassing through the control ports of a failed actuator element. To slew at a rate of 1.89 in/sec requires a back flow of 0.144 gpm through the C_1 or C_2 receiver orifice in the single stage jet pipe servo valve (See Figure 37). Typical back flow through receiver control orifice C_1 with C_2 and return open and no jet pipe pressure is:

Flow	Diff Press
0.20 gpm	30 psi
0.30 gpm	80 psi

These data show that less than 10 psi per active element of 3 operating would have to be imposed on the operating valve load pressure vs. flow curves which would have only a small effect in reducing the no load flow. The equivalent load pressure due to friction would be distributed over 3 elements which amounts to about 18 psi per active element for the proposed actuator. The friction load pressure is greater than the orifice load pressure. Assuming a straight line approximation to the load pressure vs. flow curve between 0.21 gpm and 1000 psi maximum (no flow), the loaded flow would decrease from 0.21 gpm to 0.20 gpm with a 30 psi load pressure. The actuator would then have a slew rate of 0.52/0.294 = 1.77 inches/sec which is greater than the required 1.67 inches/sec. This analysis together with data taken on the Supplier's prototype quadruplex actuator indicates that bypass valves are not necessary to meet the Procurement Specification slew rate requirements.

The following data on centering time were measured using the Supplier's prototype quadruplex force summing actuator.

Spring Force - Friction Force =	Net Force	(psi) / (element)	Centering Slew Rate
24.5 pounds - 12.5 pounds	12 pounds	30	1.56 in/sec
16.5 pounds - 12.5 pounds	4 pounds	10	0.81 in/sec

The required centering time of 2 seconds from 1 inch displacement to neutral is equivalent to a slew rate of 0.5 in/sec. The data presented above shows that the centering time can be achieved using a centering force that is at least 4.0 pounds greater than the friction force. Since the secondary actuator will use a pumped back spring centering device, the required force necessary to center within 2.0 seconds can be easily obtained.

h. Compatibility With The 1600 PSI System

Assuming a leakage flow in each actuator of 85.0 in^3/min = 1.42 in^3/sec

at 3000 psi, the leakage flow at 1420 psi is given by

$$Q_{1420} = \left(\frac{1420}{3000}\right)^{1/2} (1.42) = 0.98 \text{ in}^3/\text{sec and } 4(Q_{1420}) = 3.92 \text{ in}^3/\text{sec}$$

The total flow from the fourth hydraulic system at $P_S = 1420$ is approximately 3.9 in^3/sec.

Using the value of 1.96 in^3/sec for flow in the 25-foot and 23-foot line lengths and 0.98 in^3/sec in all other line lengths shown in Section III, the approximate differential pressures across each actuator element are listed below.

Pitch	$P_S - P_R = 1420-4 = 1416$ psi
Yaw	$P_S - P_R = 1420-3 = 1417$ psi
Left Lateral	$P_S - P_R = 1420-65 = 1355$ psi
Right Lateral	$P_S - P_R = 1420-60 = 1360$ psi

The threshold computations assumed a pressure gain of 152 psi/ma using an average of 1400 psi. With a pressure recovery of 0.75 $(P_S - P_R)$, the actual differential pressure available across the piston is listed below.

Pitch	= 1060 psi
Yaw	= 1060 psi
Left Lateral	= 1015 psi
Right Lateral	= 1020 psi

The force limit valve in the actuator is set at (1000 ± 20) psi which is compatible with the above recovery pressures. The failure detect level will be set to trip at a level equivalent to approximately (930 ± 30) psi.

i. IFM (In-Flight-Monitor)

While the design and fabrication of the in-flight monitor is the responsiblity of the SFCES Supplier, it is essential that the IFM function be compatible with Secondary Actuator operating characteristics. That function is to detect and to isolate(shutoff)elements in which failures have occurred.

Figure 44 presents a block diagram of the Secondary Actuator in-flight monitor circuit. The diagram shows that the signal from the differential pressure (ΔP) LVDT is demodulated and compared in six cross channel comparators whose outputs are connected to collection logic for failure detection and shutdown of a failed secondary actuator element. The diagram also shows a carrier sensor circuit which will

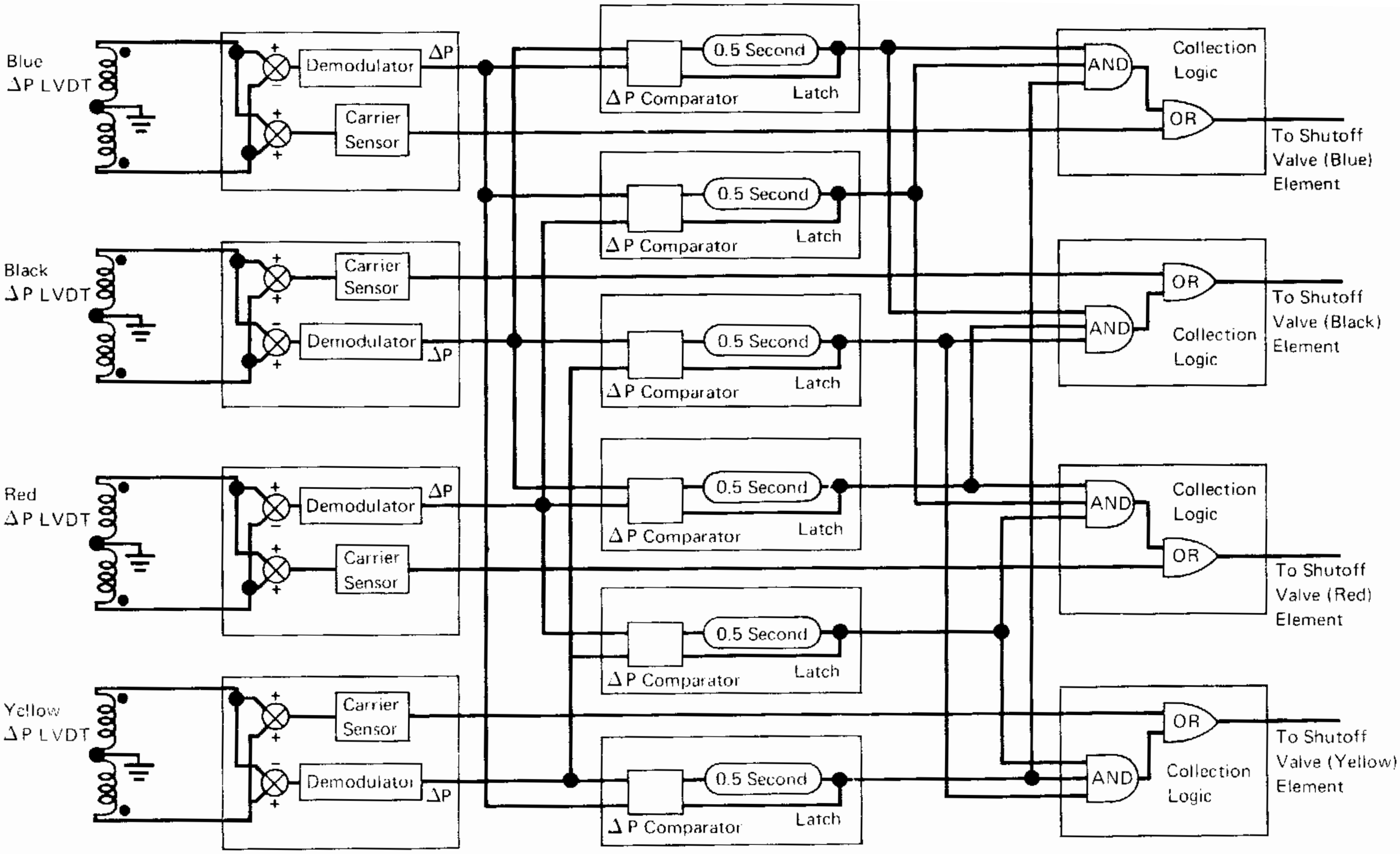

Blue
ΔP LVDT
Black
ΔP LVDT
Red
ΔP LVDT
Yellow
ΔP LVDT
Demodulator
Carrier Sensor
ΔP
0.5 Second
Latch
ΔP Comparator
AND
OR
Collection Logic
To Shutoff Valve (Blue) Element
To Shutoff Valve (Black) Element
To Shutoff Valve (Red) Element
To Shutoff Valve (Yellow) Element

FIGURE 44
IN-FLIGHT MONITOR BLOCK DIAGRAM

indicate a failure if the LVDT output is open, shorted or the power fails. The monitor time delay for the comparators is 0.5 seconds.

An active failure, i.e., a failure which results in a hardover signal, is easily isolated by the circuit of Figure 44. An active failure in the blue element, for example, produces three signals to the AND gate in the blue element collection logic approximately 0.5 seconds later, and results in a shutoff signal to the blue element shutoff valve. A single passive failure, on the other hand, will not be detected under normal operating conditions because the failed element does not generate enough differential pressure to produce a failure signal. Detection and fault isolation will occur under stall conditions, however, since all three operating elements will generate failure signals. When two passive failures have occurred, switchout of either passively failed element can not be accomplished even under stall conditions because it is no longer possible to produce three signals to an AND gate.

The operation of the in-flight monitor circuitry is considered to be satisfactory since the presence of one or two passive failures does not degrade the secondary actuator performance appreciably. In addition, the exercise of BIT before each flight will detect the existence of passive failures at that time as discussed in AFFDL-TR-71-20, Section III. The probability of multiple passive failures should, therefore, be reduced to an acceptable level.

j. Pressure Variations

The Secondary Actuator is designed to function with rather wide pressure variations in the supply systems. Since the pressure sensors observe cylinder differential pressures, the pressure variations should have only a small affect upon the actuator when the output is static. Dynamically the time delay in the shutdown circuitry prevents any nuisance failure signals should a transient pressure differential be developed that is of sufficient magnitude to trigger the failure detection mechanism.

The use of low pressure gain servo valves reduces the sensitivity to the effect of supply pressure variations. Tests have been performed on the demonstrator hardware which verify that supply pressure variations do not jeopardize performance. Each element of the secondary actuator is equipped with a pressure sensor. This pressure sensor has three functions:

- It will transmit an error signal (LVDT voltage) in the absence of pressure or too low a pressure.
- It will signal when the differential pressure in the actuator cylinders reaches a predetermined limit.
- The force limiting feature will limit the cylinder pressures when the pressure differential reaches a predetermined limit.

The two phenomena affected by pressure variations are cylinder pressures and flow (actuator output velocity). Since the pressure sensor reacts only to cylinder differential pressure, a base pressure difference between elements does not cause a large cross element pressure fight. A small pressure fight may occur due to the change in servo valve null bias with pressure. However, the tolerance study shows that this effect is negligible compared to feedback or input signal tolerances.

The pressure variations specified are: supply, 1000 psi to 135 percent of normal (4050 psi); and return, 0 to 500 psi for the 3000 psi systems and 0 to 150 psi for the 1600 psi system.

Supply pressures affect the actuator output velocity. The output velocity required is full stroke slew in less than 0.60 seconds with a 1000 psi pressure differential across the actuator with either four or three elements operating. The velocity is relatively unaffected by the number of elements in operation since this is a force summing type of actuator whereby the velocity of a single element is approximately the same as it is with four elements.

The potential effect on actuator velocity performance would be caused by the pressure extremes causing an internal pressure level of an element that would result in a force fight. In order for a failure to be indicated, the differential pressure in a cylinder in an offending element must reach the failure detect level. Most of the performance is with virtually no load force. The failure detection circuits in the electronics has a built-in-time delay of 0.5 second. This time delay prevents nuisance failure indications should maximum actuator velocities cause pressure fights sufficient to reach the detection level.

k. Voltage Variations

The circuits contained in the path from the position feedback LVDT through the servo-amplifier to the servo valve include two operational amplifiers that use power from a ±12 VDC power supply. The possible variation in the output of an operational amplifier per volt change in excitation is 0.15 millivolt/volt. Thus on a worst case addition, the output of two operational amplifiers might vary 0.30 millivolt/volt. Since the power supply may vary within ±25% from low temperature to high temperature, the +12 VDC or -12 VDC power supply should not exceed ±3.0 volts above or below the nominal value. Therefore, the output of the two amplifiers should not vary more than ±0.9 millivolt from nominal over the temperature range due to excitation variations.

The other major effect on the output of a secondary actuator is due to changes in the 26 VAC excitation voltage to the feedback LVDT. The change in excitation voltage does not have any significant effect at the neutral position of the actuator and LVDT. However, any change in excitation affects the gradient of the LVDT which directly affects the position gain of the servo actuator. The gradient selected for the secondary actuator is 14 volts/inch at 26 VAC excitation. If the excitation voltage is allowed to vary, worst case ± 0.5 percent maximum over temperature and the actuator is positioned at some point other than neutral, the actuator output position would vary a maximum of ±0.25 percent of total stroke. If the point was at full extend stroke of one inch, the position change would be ±0.0025 inch due to excitation variations over the temperature range of -54°C to + 71°C. This maximum position variation at one inch assumes that all four independent power supplies increase over temperature to the maximum allowed value of ±0.5 percent. However, this will in all probability not occur and two of the supplies might be at +0.2 percent and two at -0.4 percent so that small pressure fights will occur between elements of the actuator at full stroke output. These small pressure differences would tend to reduce the position variation to less than 0.0025 inch in voltage output position. The effects of voltage variations on the system in terms of positionability are negligible.

3. SURVIVABLE STABILATOR ACTUATOR PACKAGE (SSAP)

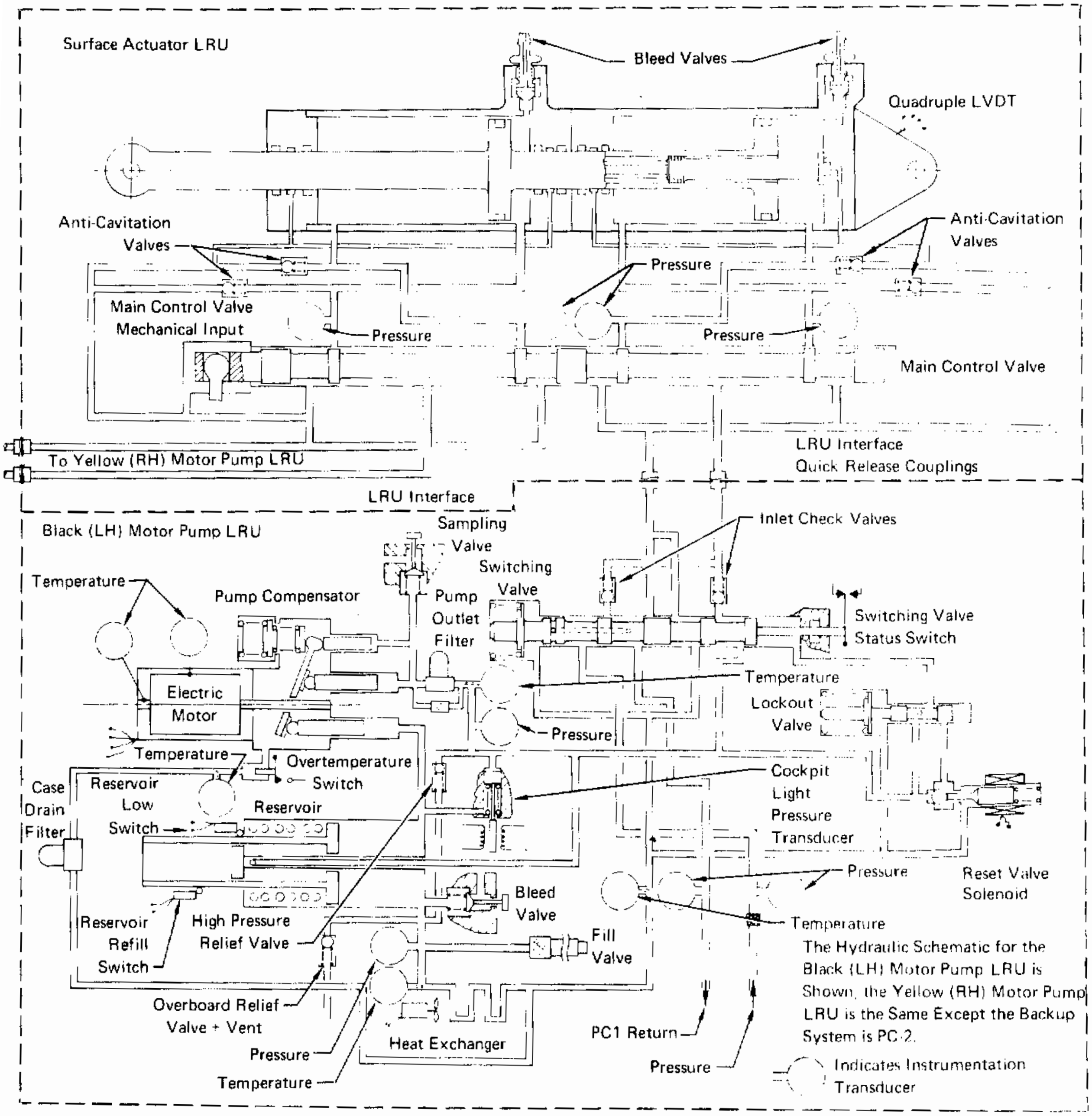
The SSAP is an integrated actuator package utilizing both power-by-wire and fly-by-wire concepts which is designed to replace the secondary actuator and F-4 stabilator actuator combination for Phase IIC of the SFCS program. The SSAP consists of a quadruplex velocity summing electromechanical secondary actuator and a surface actuator with dual tandem pistons which are powered by two integral motor pump units. The SSAP also contains two switching valves which allow the surface actuator to be powered by aircraft hydraulic supplies in the event that one or both of the integral motor pumps fail. A hydraulic schematic of the SSAP is shown in Figure 45.

The integral hydraulic pumps used on the SSAP are the soft cutoff type pumps. The pressure-flow characteristic of this type of pump is such that the pressure supplied is dependent upon the flow being demanded. This pump characteristic causes the SSAP response to be dependent upon the amplitude of the input command signal, and results in a nonlinear response for the SSAP.

In the dynamic analysis of the SSAP it was convient to use both linear and nonlinear math models. The linear math model was used primarily for stability, failure transient, and disconnect analysis and the nonlinear model was used primarily for frequency response and threshold analysis. The math models are developed in the following paragraphs. The parameters used in the analysis are shown in Table XXV. They reflect the current status of the SSAP, but they are subject to change as development progresses.

a. Stability

A linear analysis was conducted to determine the stability of the SSAP when installed in an F-4 aircraft and driving the stabilator surface. The stability analysis was conducted in three parts: (a) the surface actuator alone, (b) the secondary actuator alone, and (c) the total SSAP. The stability of the individual actuators was determined from the roots of their transfer functions. The stability of the total SSAP was determined via the root locus method.



The Hydraulic Schematic for the Black (LH) Motor Pump LRU is Shown, the Yellow (RH) Motor Pump LRU is the Same Except the Backup System is PC-2.

FIGURE 45
SSAP HYDRAULIC SCHEMATIC

TABLE XXV
SSAP PARAMETERS

		Units	Normal Operation	System "B" Operating on One Integral Motor Pump (See Notes 1 and 2)	System "B" Operating On One Aircraft Hydraulic Supply (See Notes 1 and 3)
A	Surface Actuator Piston Area	$In.^2$	11.54	5.76	5.76
ABS	Absolute Value	–	–	–	–
B_1	Backlash: Secondary Actuator Gears, Differentials and Ballscrew	In.	0.005	0.005	0.005
B_2	Linkage Freeplay: Surface Actuator Linkage	In.	0.0046	0.0046	0.0046
C_m	Coefficient of Damping - Surface Mass	Lb-Sec/In.	30.0	30.0	30.0
K_3	Ratio: Actuator Velocity to Master Control Valve Stroke	Sec^{-1}	102	102	140
K_4	Ratio: Master Control Valve Conductance to Piston Area Squared	In./Lb-Sec	1.45×10^{-5}	5.78×10^{-5}	5.78×10^{-5}
K_a	Secondary Actuator Amplifier Gain	% Pulse Width/Volt	2000	2000	2000
K_G	Secondary Actuator Gearing Gain	In./Rad	0.00105	0.00105	0.00105
K_M	Secondary Actuator Servo Motor Gain	Rad/Sec/% Pulse Width	396.0	396.0	396.0
K_T	Secondary Actuator Tachometer Gain	Volts/Rad/Sec	3.39×10^{-4}	3.39×10^{-4}	3.39×10^{-4}
K_{ea}	Secondary Actuator Error Signal Amplifier	Volts/Volt	3.86	3.86	3.86
K_{f1}	Secondary Actuator Feedback Gain	Volts/In.	2.59	2.59	2.59
K_{f2}	Main Ram Feedback Gain	Volts/In.	1.17	1.17	1.17
K_{pa}	Pre-amplifier Gain	Volts/Volt	0.346	0.346	0.346
K_B	Spring Rate: Backup Structure, End Fitting and Actuator Cylinder (Axial)	Lb/In.	1.10×10^6	1.24×10^6	1.24×10^6
K_D	Spring Rate: Hydraulic Fluid Between Inlet and Outlet Orifices, Hoop Tension of Actuator Barrel, and Piston Rod	Lb/In.	0.255×10^6	0.1345×10^6	0.189×10^6
K_o	Spring Rate: Piston Rod End and Surface Horn	Lb/In.	0.285×10^6	0.285×10^6	0.285×10^6
K_{Σ}	Effective Spring Rate: Complete System	Lb/In.	0.120×10^6	0.085×10^6	0.104×10^6
K_{mv}	Master Control Valve Constant	$In.^3/Sec(Lb)^{1/2}$	See Table XXVI	See Table XXVI	See Table XXVI
L_o	Linkage Ratio: Effective Mass to Stabilator	Deg/In.	2.924	2.924	2.924
L_2	Linkage Ratio: Summing Link to Control Valve	In./In.	1.0	1.0	1.0
L_3	Linkage Ratio: Piston to Summing Link	In./In.	0.1385	0.1385	0.1385
L_4	Linkage Ratio: Secondary Actuator to Summing Link	In./In.	0.667	0.667	0.667

TABLE XXV (CONTINUED)
SSAP PARAMETERS

		Units	Normal Operation	System "B" Operating on One Integral Motor Pump (See Notes 1 and 2)	System "B" Operating on One Aircraft Hydraulic Supply (See Notes 1 and 3)
M_m	Effective Mass of Stabilator: Surface Inertia Reflected to Actuator Piston	$Lb\text{-}Sec^2/In.$	3.56	3.56	3.56
P_L	Pressure Differential: Surface Actuator Piston	PSI	Definition	Definition	Definition
P_s	Pressure Supplied by Pump	PSI	Definition	Definition	Definition
Q	Flow Through Master Control Valve	CIS	Definition	Definition	Definition
S	Laplace Operator	—	—	—	—
SQRT	Square Root	—	—	—	—
V_c Limit	Max. Pulse Width into Servo Motor	% Pulse Width	±45.0	±45.0	±45.0
V_F	Feedback Voltage from Surface Actuator LVDT	Volts/In.	Definition	Definition	Definition
V_i	Input Command Signal: SSAP	Volts	Definition	Definition	Definition
V_{is}	Input Command Signal: Secondary Actuator	Volts	Definition	Definition	Definition
X_c	Displacement: Surface Actuator Cylinder	In.	Definition	Definition	Definition
X_F	Displacement: Surface Actuator Feedback Linkage	In.	Definition	Definition	Definition
X_m	Displacement: Effective Load Mass	In.	Definition	Definition	Definition
X_o	Displacement: Secondary Actuator Output	In.	Definition	Definition	Definition
X_p	Displacement: Surface Actuator Piston	In.	Definition	Definition	Definition
X_v Limit	Max. Master Control Valve Stroke	In.	±0.10	±0.10	±0.10
$\dot{\theta}$ Limit	Max. Servo Motor Speed	Rad/Sec	±1200	±1200	±1200
τ_a	Time Constant: Secondary Actuator Amplifier	Sec	0.00067	0.00067	0.00067
τ_M	Time Constant: Servo Motor	Sec	0.035	0.035	0.035
τ_{pa}	Time Constant: Pre-amplifier	Sec	0.0010	0.0010	0.0010
τ_{f1}	Time Constant: Secondary Actuator Feedback	Sec	0.0010	0.0010	0.0010
τ_{f2}	Time Constant: Main Ram Feedback	Sec	0.0050	0.0050	0.0050

Note 1 - See Figure 68 for a definition of System "B".

Note 2 - System "A" inoperative, two source failures have occurred.

Note 3 - System "A" inoperative, three source failures have occurred.

The parameters used in the SSAP stability analysis are shown in Table XXV. Parameters such as actuator structural spring rates and master control valve characteristics were furnished by the SSAP Supplier. The aircraft structural spring rates and stabilator inertia were determined by MCAIR analysis. The surface damping coefficient reflects an inherent surface damping. This coefficient was based on experience with the F-4 stabilator actuator. An electrical feedback is included in the actuator portion of the package for use in the longitudinal flight control system.

(1) Surface Actuator

The surface actuator is a closed loop servo system which makes use of mechanical linkage for position feedback to the master control valve. The schematic of the SSAP installation in Figure 46 shows the relationship between the surface actuator, control linkage, installation geometry, structural spring rates and stabilator surface inertia. The surface actuator portion of the SSAP can be represented by the linear math model shown in Figure 47. The masses of the actuator piston and cylinder are small when compared to the inertia of the stabilator surface reflected to the actuator piston as an effective mass. These smaller masses have very little effect on the overall system dynamics and therefore, were not included in the math model. If it is assumed that the integral pumps produce a constant 1600 psi, this math model can be used to represent the surface actuator when operating on integral motor pumps or aircraft hydraulic supplies. This assumption is valid

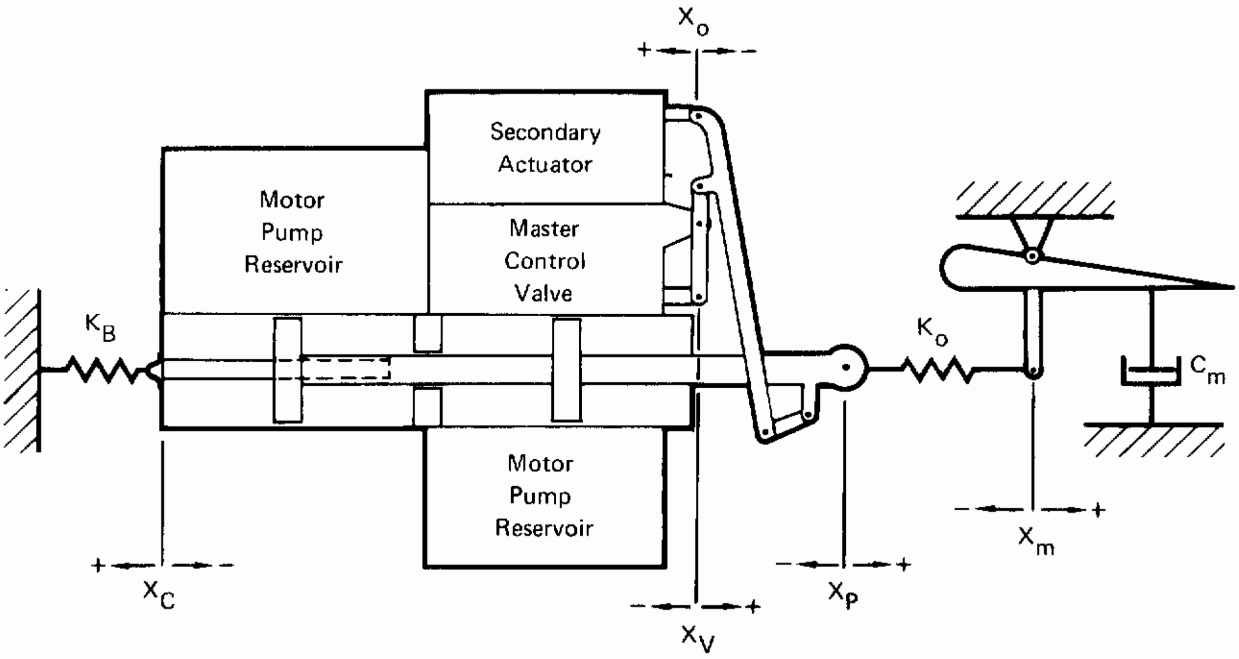


FIGURE 46
SCHEMATIC - SSAP

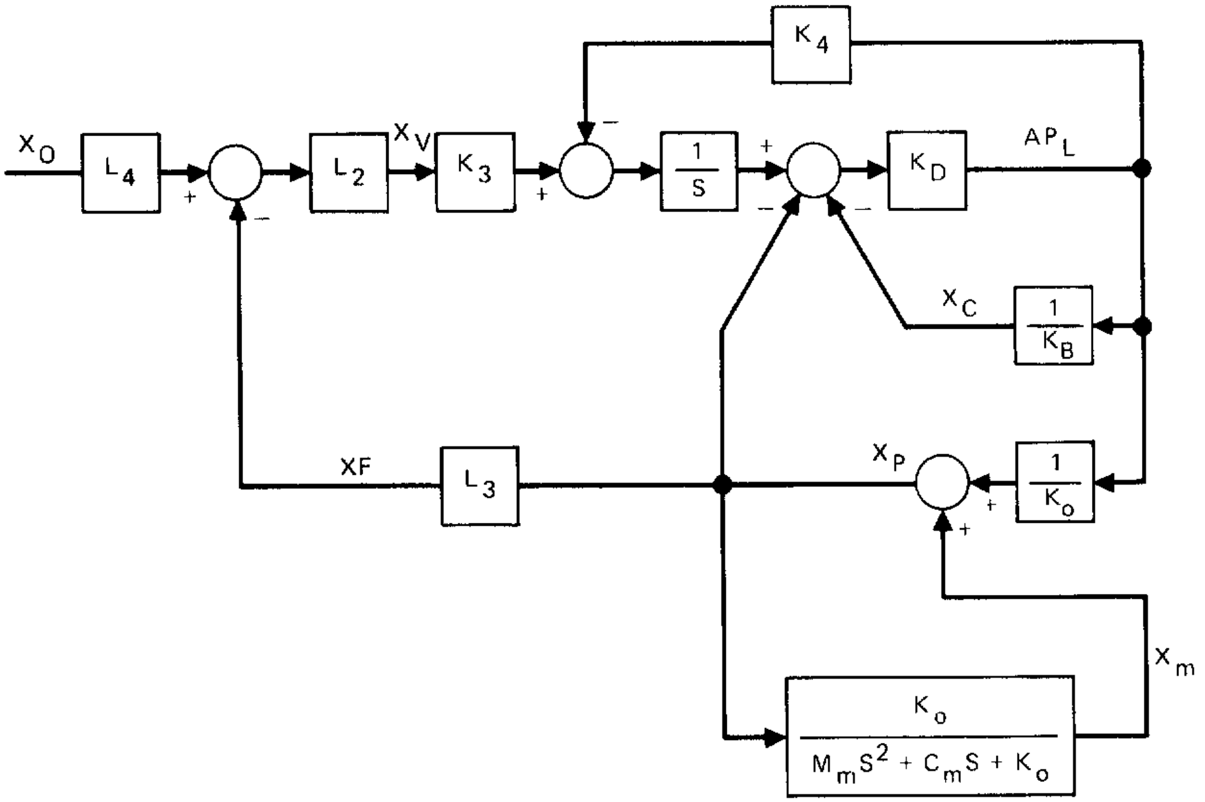


FIGURE 47
LINEAR MATH MODEL - SSAP SURFACE ACTUATOR

for stability analysis. A closed loop transfer function for the surface actuator math model can be represented by:

$$\frac{X_p}{X_o} = \frac{K_3 K_\Sigma L_2 L_4 \left(M_m S^2 + C_m S + K_o\right)}{(K_o M_m)S^3 + [(K_o K_\Sigma K_4 + K_3 K_\Sigma L_2 L_3)M_m + K_o C_m]S^2 + [(K_o K_\Sigma K_4 + K_3 K_\Sigma L_2 L_3)C_m + K_o K_\Sigma]S + K_o K_\Sigma K_3 L_2 L_3} \tag{10}$$

Two modes of operation of the surface actuator were considered: 1) one system operating from an integral motor pump unit, and 2) one system operating from an aircraft hydraulic supply, with the second system inoperative in both cases. These modes of operation are analogous to operation of the actuator after two and three failures, respectively. From a stability standpoint the worst case for the surface actuator is when only one hydraulic system is operating. This is due to the fact that the spring rate of the hydraulic fluid is reduced by 50% when only one hydraulic system is operating. For this analysis it was assumed that the system that is shut off does not contribute to the total spring rate of the hydraulic fluid and actuator parts (K_D) and does not contribute to the overall system damping.

Substitution of parameter values from Table XXV into surface actuator closed loop transfer function, equation (10), results in the following relationships:

Operating from one integral motor pump:

$$\frac{X_p}{X_o} = \frac{(2.06 \times 10^7)S^2 + (1.74 \times 10^8)S + 1.65 \times 10^{12}}{(1.015 \times 10^6)S^3 + (1.782 \times 10^7)S^2 + (2.43 \times 10^{10})S + 3.43 \times 10^{11}} \tag{11}$$

Operating from one aircraft hydraulic supply:

$$\frac{X_p}{X_o} = \frac{(3.46 \times 10^7)S^2 + (2.91 \times 10^8)S + (2.77 \times 10^{12})}{(1.015 \times 10^6)S^3 + (2.18 \times 10^7)S^2 + (2.97 \times 10^{10})S + 5.75 \times 10^{11}} \tag{12}$$

The roots of these transfer functions are:

Operating from one integral motor pump:

Numerator	Denominator
-4.22 + j 283.0	-14.1 + j 0.0
-4.22 - j 283.0	-1.73 + j 155.0
	-1.73 - j 155.0

Operating from one aircraft hydraulic supply

Numerator	Denominator
-4.22 + j 283.0	-19.35 + j 0.0
-4.22 - j 283.0	-1.08 + j 171.0
	-1.08 - j 171.0

From the roots of the closed loop transfer function for the surface actuator it can be seen that the actuator is more stable when operating on one integral hydraulic supply than when operating on one aircraft hydraulic supply. These roots also show that in either of the two modes of operation the system is very lightly damped. Due to the low damping further analysis was performed to determine the relative stability of the surface actuator. This analysis was performed only for the least stable of the two modes of operation, the case when the actuator was operating from one aircraft hydraulic supply.

The open loop transfer function for the surface actuator can be represented by:

$$\frac{X_F}{X_o} = \frac{L_3 K_3 K_\Sigma L_2 L_4 (M_m S^2 + C_m S + K_o)}{S\{(K_o M_m)S^2 + [K_o K_\Sigma K_4) M_m + K_o C_m]S + [(K_o K_\Sigma K_4) C_m + K_o K_\Sigma]\}} \tag{13}$$

For the case when the actuator is operating on one aircraft hydraulic supply, substitution of parameter values from Table XXV into equation (13), results in the following relationship:

$$\frac{X_F}{X_o} = \frac{4.719\ [S^2 + (8.43)S + 8.01 \times 10^4]}{S[S^2 + (14.43)S + 2.92 \times 10^4]} \tag{14}$$

A frequency response of the open loop transfer function is shown in Figure 48. This frequency response indicates that for the worst case, the SSAP surface actuator has a gain margin of approximately 5 dB.

(2) Secondary Actuator

The SSAP secondary actuator is a quadruplex, velocity summing electromechanical servomechanism. It uses LVDT's for position feedback and tachometer's for rate feedback. A schematic of the secondary actuator is shown in Figure 49. For the stability analysis it was assumed that all gears and linkages are infinitely rigid and the inertia of the gears, differential, and the ball-screw was reflected to the servo motor output. The secondary actuator and its associated electronics can be represented by the linear math model shown in Figure 50. Assuming all elements are identical, a closed loop transfer function for this math model can be represented by:

$$\frac{X_o}{V_{is}} = \frac{K_{ea} K_a K_M K_G N_a (\tau_{fl} S + 1)}{(\tau_a \tau_M \tau_{fl}) S^4 + (\tau_a \tau_{fl} + \tau_M \tau_{fl} + \tau_a \tau_M) S^3 + [(1 + K_a K_M K_T)\tau_{fl} + \tau_a + \tau_M] S^2 + (1 + K_a K_M K_T) S + K_{ea} K_a K_M K_G N_a K_{fl}} \tag{15}$$

Substitution of parameter values from Table XXV into equation (15) and factoring the resulting relationship yields the following roots.

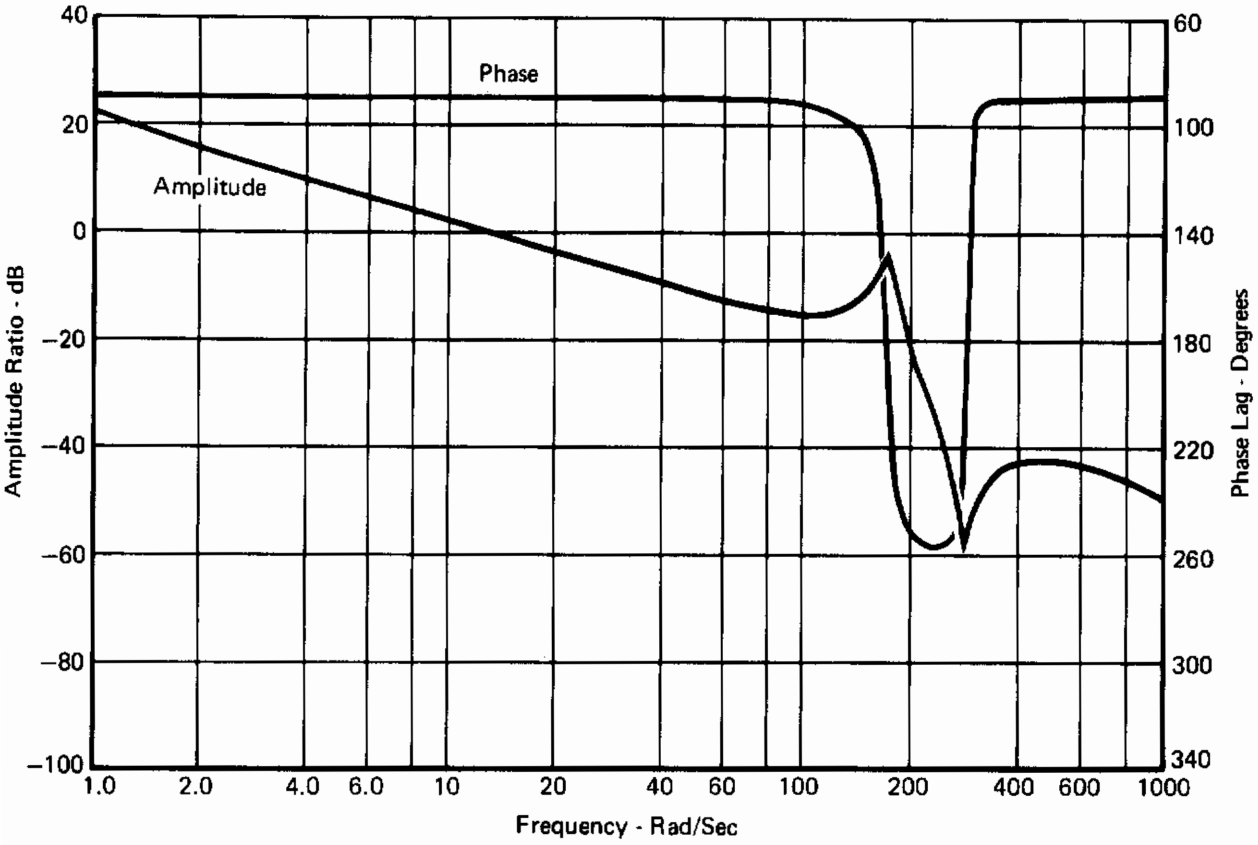


FIGURE 48
OPEN LOOP FREQUENCY RESPONSE
SSAP SURFACE ACTUATOR

Four elements operating:

Numerator	Denominator
-1000 + j 0.0	-147.4 + j 0.0 -846.2 + j 0.0 -767.1 + j 3291. -767.1 - j 3291.

Three elements operating:

Numerator	Denominator
-1000 + j 0.0	-104.7 + j 0.0 -890.5 + j 0.0 -766. + j 3296. -766. - j 3296.

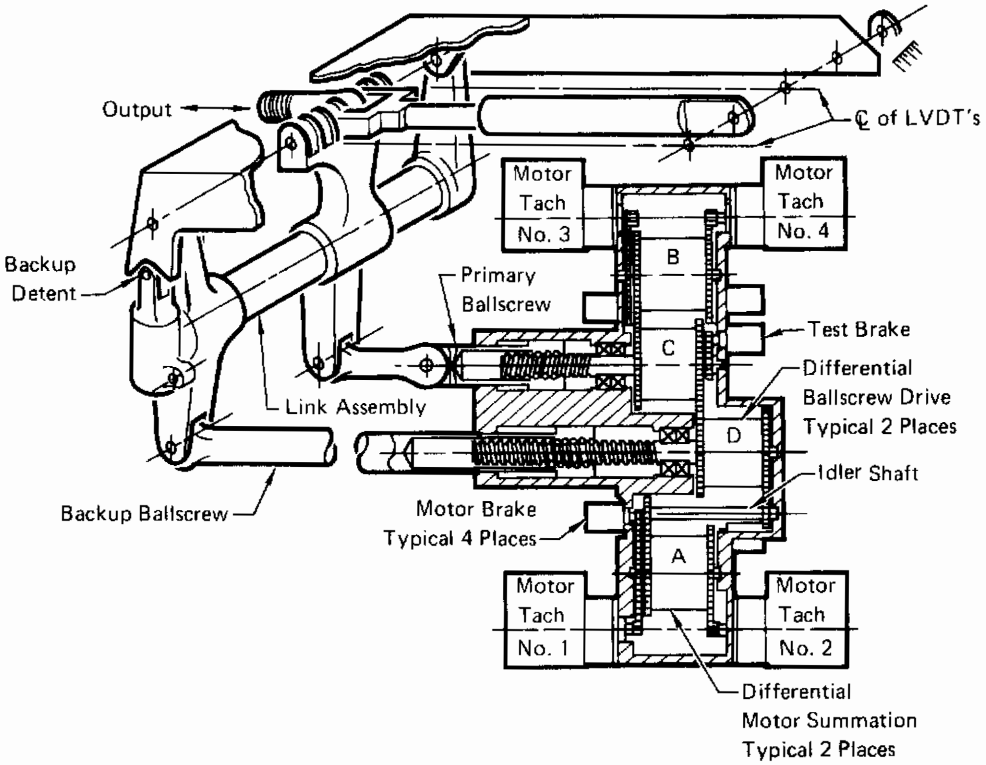


FIGURE 49
ELECTROMECHANICAL SECONDARY ACTUATOR SCHEMATIC

Two elements operating:

Numerator	Denominator
-1.000 + j 0.0	-66.7 + j 0.0
	-930.1 + j 0.0
	-765.6 + j 3301.
	-765.6 - j 3301.

The roots of the closed loop transfer function indicate that no stability problems are anticipated in the secondary actuator with tachometer feedback.

(3) Total SSAP

The total SSAP consists of the series combination of the secondary actuator and the surface actuator with an electrical feedback loop closed around the two actuators. The total SSAP can be represented by the linear math model shown in Figure 51. Using

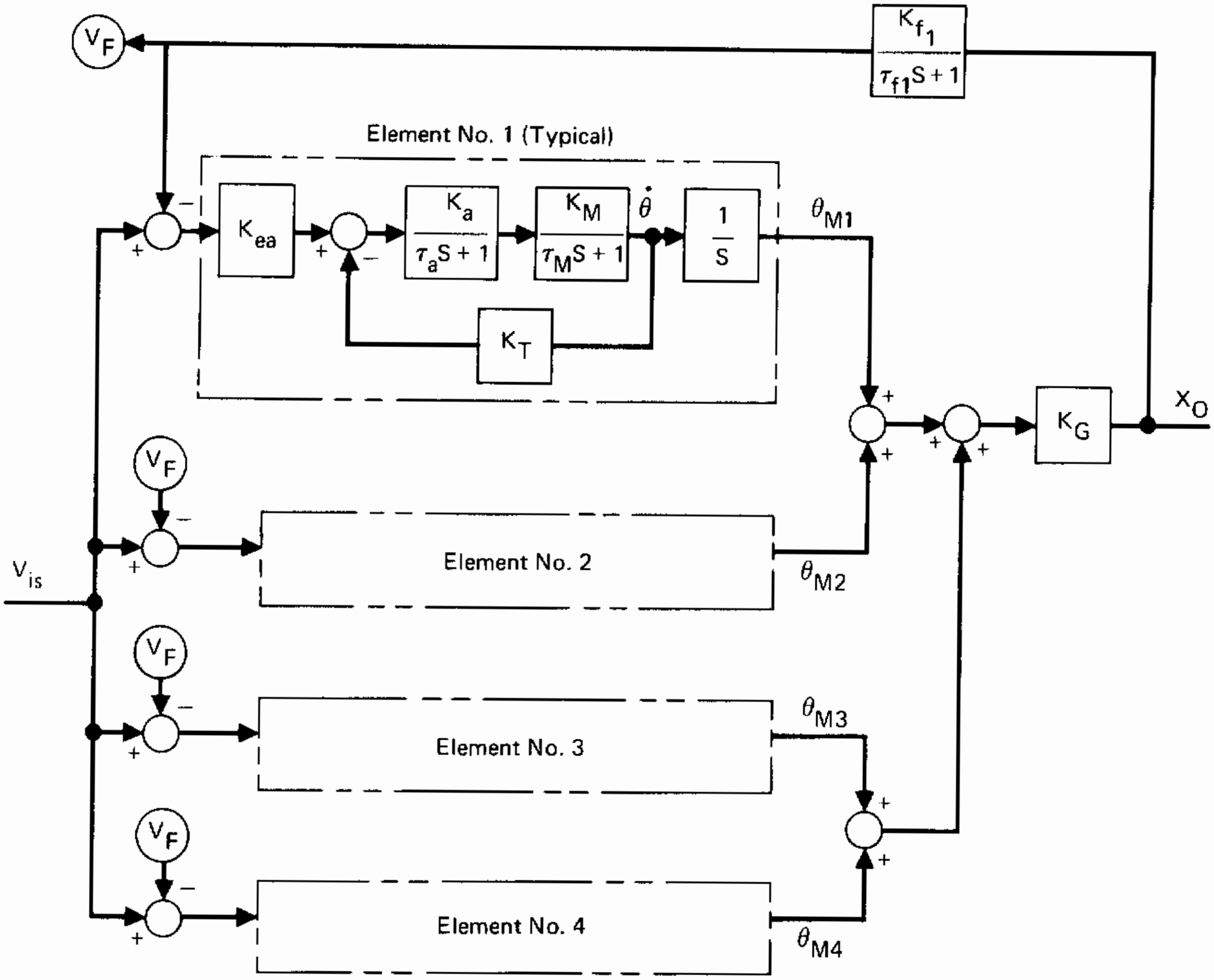


FIGURE 50
LINEAR MATH MODEL - SSAP SECONDARY ACTUATOR

this math model, an open loop root locus stability analysis was performed to determine the gain for the electrical feedback loop. For this analysis the worst case for the surface actuator stability, operating from one aircraft hydraulic supply, was chosen.

This analysis indicated the need for some type of compensation to stabilize the SSAP. The compensation chosen was a simple first order lag in the electrical feedback loop.

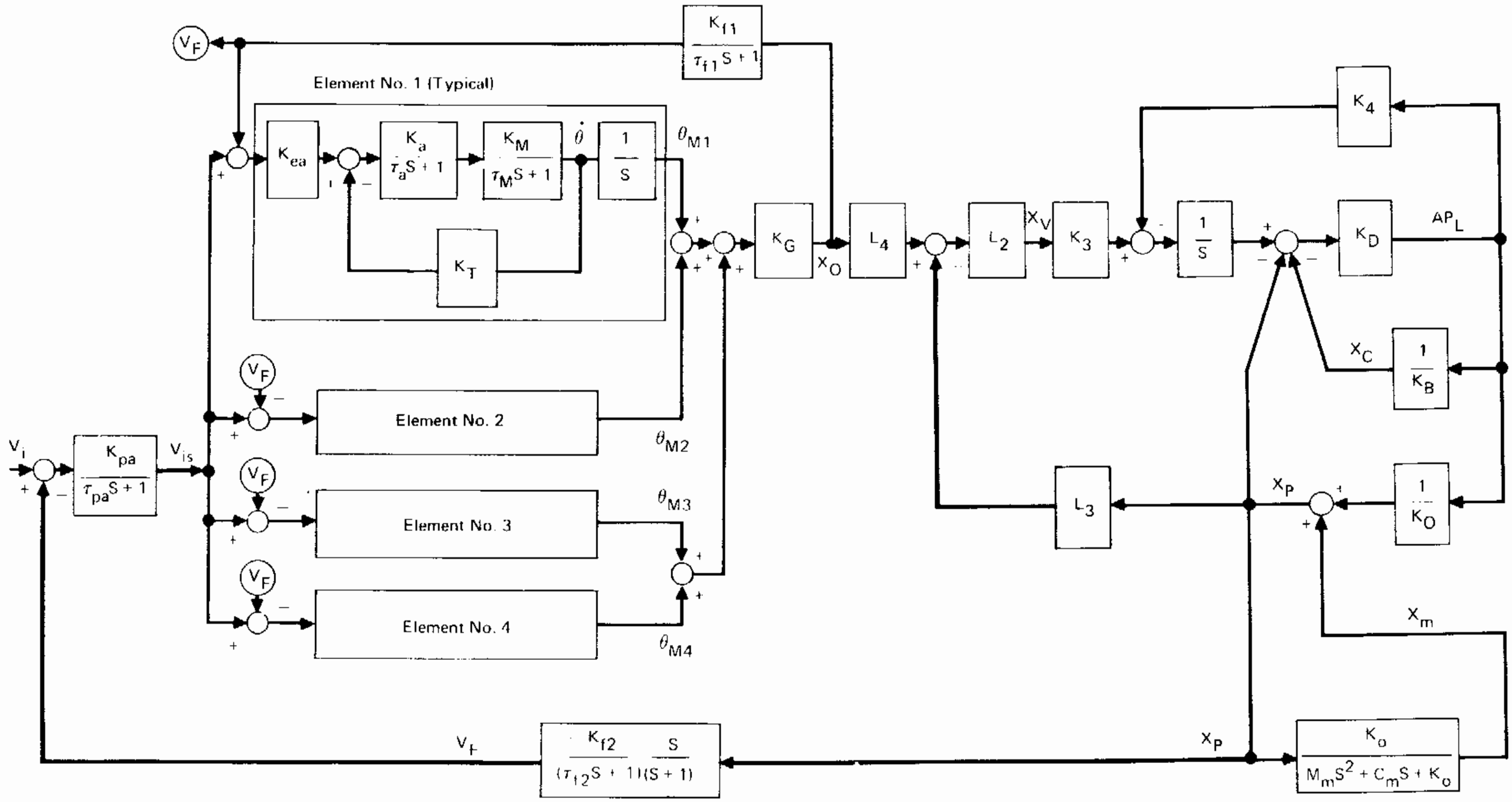


FIGURE 51
LINEAR MATH MODEL -SSAP

The open loop transfer function for the total SSAP can be presented by:

$$\frac{V_F}{V_i} = \left[\frac{K_{pa}}{\tau_{pa} S + 1}\right]\left[\begin{array}{c}\text{SECONDARY ACTUATOR} \\ \text{CLOSED LOOP TRANSFER} \\ \text{FUNCTION, EQUATION} \\ \text{(15)}\end{array}\right]\left[\begin{array}{l}\text{SURFACE ACTUATOR} \\ \text{CLOSED LOOP TRANS-} \\ \text{FER FUNCTION,} \\ \text{EQUATION (10)}\end{array}\right]\left[\frac{K_{f_2}}{(\tau_{f_2} S + 1)}\frac{S}{(S+1)}\right]$$

Substitution of parameter values from Table XXV into the open loop transfer function and factoring results in the following relationships:

For 4 elements of the secondary actuator operating:

$$\frac{V_F}{V_i} = \frac{K_{f2}(1.30 \times 10^{15})(S + 1000.)(S + 4.21 \pm j283.0)(S + 0.0)}{(S+1000.)(S+846.)(S+767 \pm j3291.)(S+200.)(S+147.)(S+19.3)(S+1.08 \pm j171)(S+1)} \tag{16}$$

For 3 elements of the secondary actuator operating:

$$\frac{V_F}{V_i} = \frac{K_{f2}(9.72 \times 10^{14})(S + 1000.)(S + 4.21 \pm j282.0)(S + 0.0)}{(S+1000.)(S+890.)(S+766. \pm j3296)(S+200.)(S+105.)(S+19.3)(S+1.08 \pm j171.)(S+1)} \tag{17}$$

For 2 elements of the secondary actuator operating:

$$\frac{V_F}{V_i} = \frac{K_{f2}(6.48 \times 10^{14})(S + 1000.)(S + 4.21 \pm j283.0)(S + 0.0)}{(S+1000.)(S+930.)(S+766. \pm j3300.)(S+200.)(S+67.)(S+19.3)(S+1.08 \pm j171.)(S+1)} \tag{18}$$

Root loci for these open loop transfer functions are shown in Figures 52 through 54. From these root locus plots it can be seen that the SSAP is stable for an electrical feedback gain of 6.15 volts/inch or less. A feedback gain of 1.17 volts/inch was chosen for normal operation. This will result in a gain margin of approximately 14.4 dB.

The results of this analysis indicate that no stability problems should be anticipated with the SSAP per se. However, the effective bulk modulus of the hydraulic fluid being used has not been well defined when the fluid is used in an actuator at high temperature. Also, many of the parameters used were based upon preliminary estimates. Therefore, it is recommended that the SSAP be subjected to careful and extensive stability testing, and more analysis when better information is available.

b. SSAP Response Characteristics

An analysis was conducted to determine the response characteristics of the SSAP. In the SFCS, the servo controlled actuators have full authority over the aircraft control surfaces. Therefore, the math models used in determining the overall system response must reflect the response characteristics of the proposed hardware over the full range of actuator authority. Also the SSAP makes use of soft cutoff

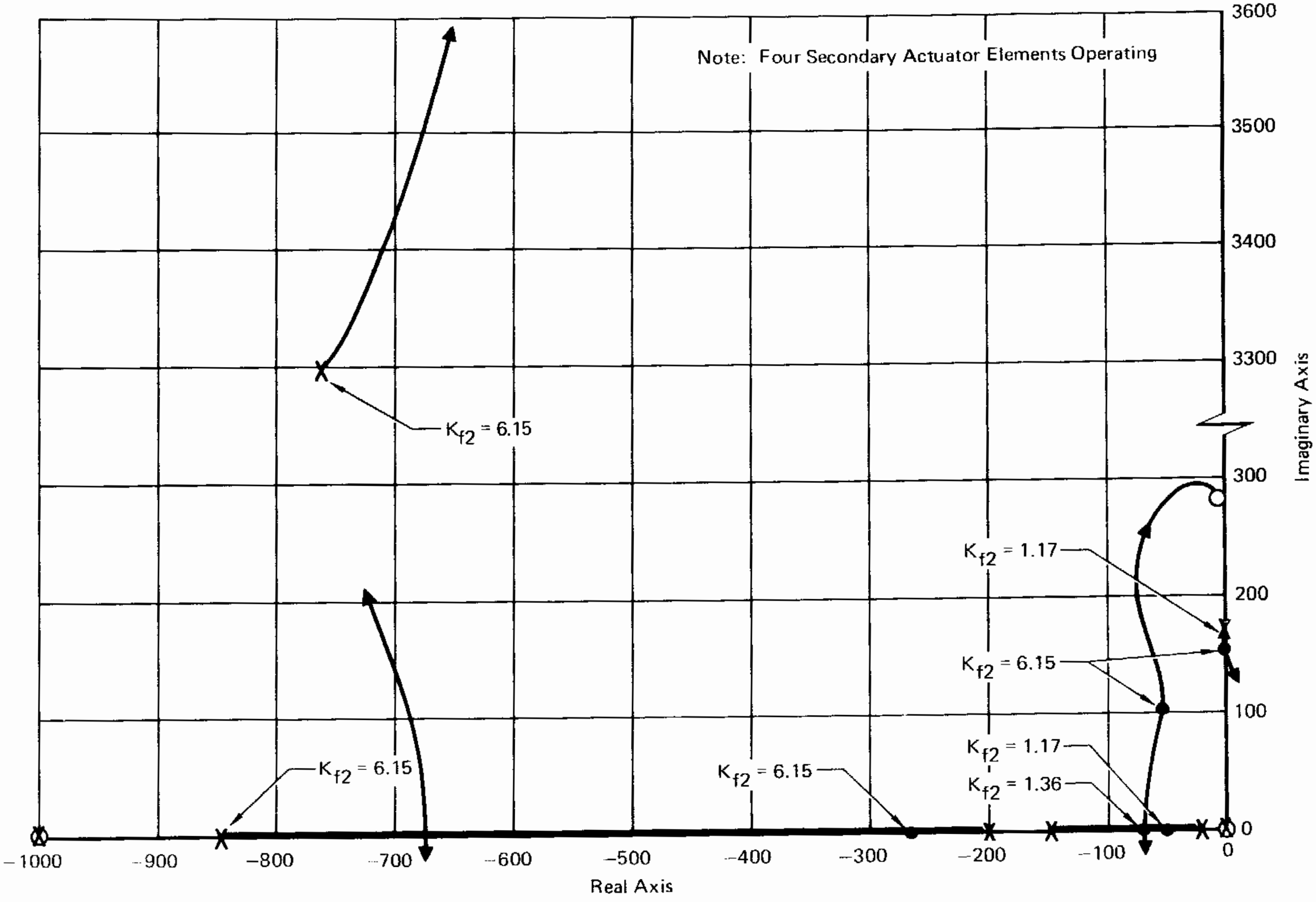


FIGURE 52
OPEN LOOP ROOT LOCUS - SSAP

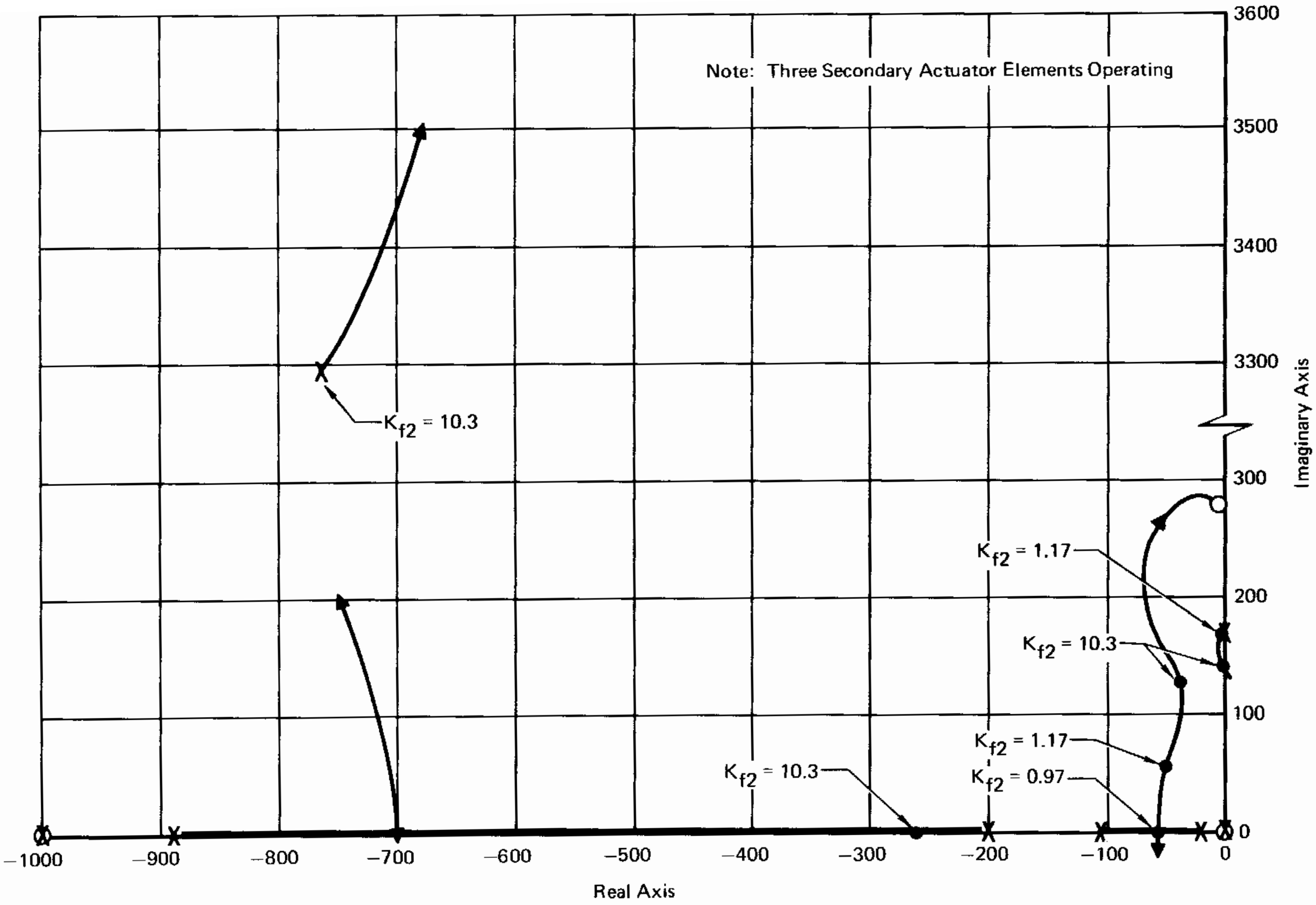


FIGURE 53
OPEN LOOP ROOT LOCUS - SSAP

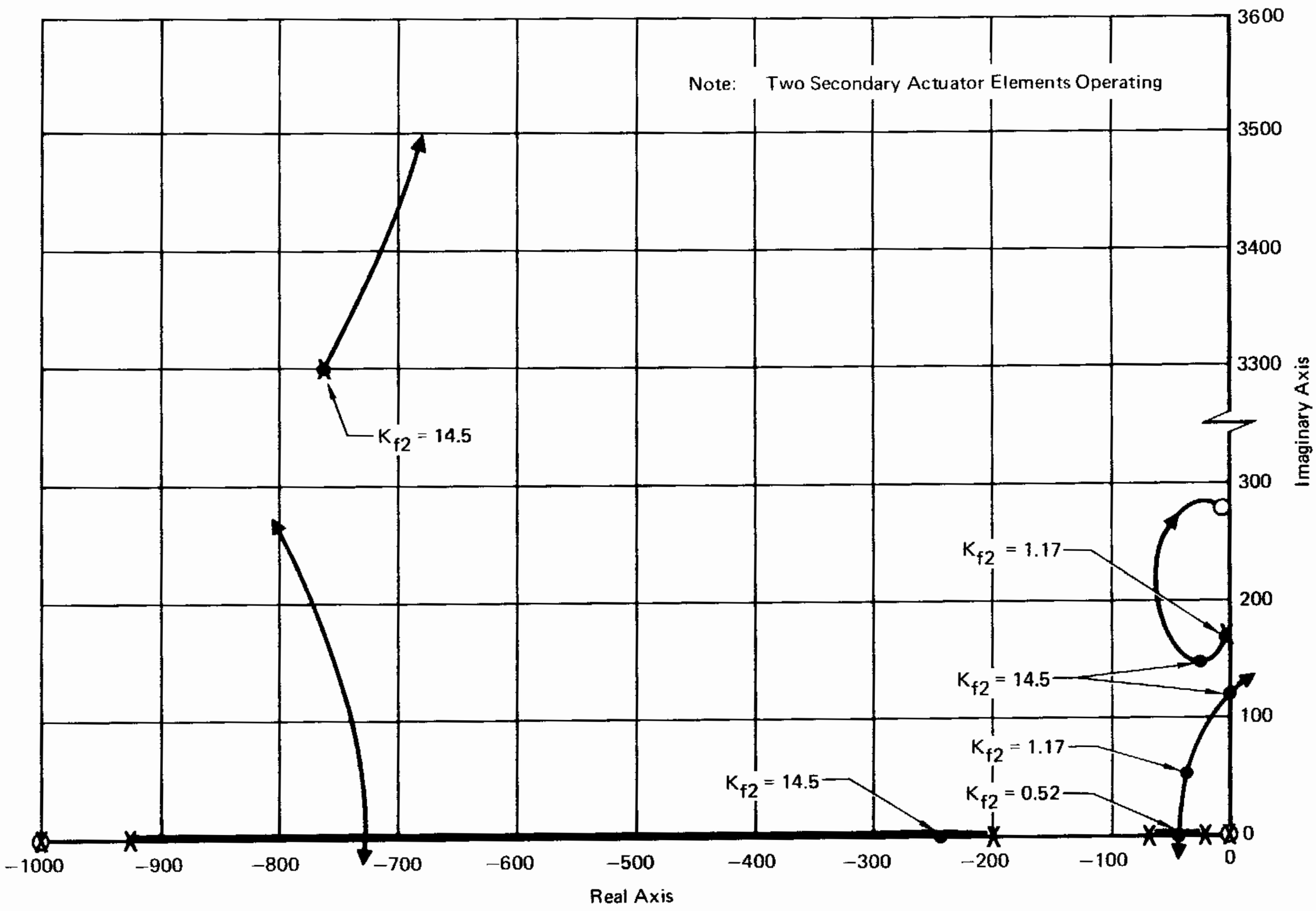


FIGURE 54
OPEN LOOP ROOT LOCUS - SSAP

hydraulic pumps. The pressure-flow characteristic of this type of pump is such that the pressure supplied by the pump is dependent upon the flow being demanded. This pump characteristic causes the SSAP response to be dependent upon the amplitude of the input command signal, and causes the SSAP to have a nonlinear response.

For the reasons noted above, linear analysis techniques do not provide an adequate description of the SSAP response characteristics. Therefore, nonlinear techniques were used to determine the response characteristics of the SSAP. Nonlinear analysis is also required if the dynamic representation of the SSAP is to be refined in the future to demonstrate correlation between the math model and characteristics of the physical hardware.

A nonlinear math model was developed for the SSAP. For the sake of clarity this model is presented in two parts; surface actuator and secondary actuator. These two parts are then combined to produce the nonlinear math model for the SSAP.

(1) Surface Actuator

A nonlinear math model was developed for the surface actuator. It was based on the linear math model shown in Figure 47. In the linear math model the master control valve representation was developed from:

$$Q = \frac{\partial Q}{\partial X_v} X_v + \frac{\partial Q}{\partial P_L} P_L \tag{19}$$

This relationship is valid for small perturbations about an operating point when the supply pressure is constant. Because the surface actuator is supplied by soft cutoff hydraulic pumps it is obvious that the linear relationship for the master control valve is not valid over the full range of response for the SSAP. However, the master control valve can also be represented by:

$$Q = K_{mv} X_v \sqrt{P_S - P_L} \tag{20}$$

In this relationship P_S represents the pressure supplied by the soft cutoff pump. The nonlinear representation for the master control valve and the pump characteristic are included in the nonlinear math model for the surface actuator shown in Figure 55 and 69 are also shown in Table XXVI. The only remaining major nonlinearity in the surface actuator is the physical limit on the master control valve stroke. This limit is also shown in the nonlinear math model. The significance of the limit is more fully explained in the description of the total SSAP model. Figures 56 and 57 present frequency response plots generated by the surface actuator nonlinear math model. The difference in performance for small inputs is due to the low gain of the master control valve in the area around null. The low gain around null is required for stability. It should also be noted that the

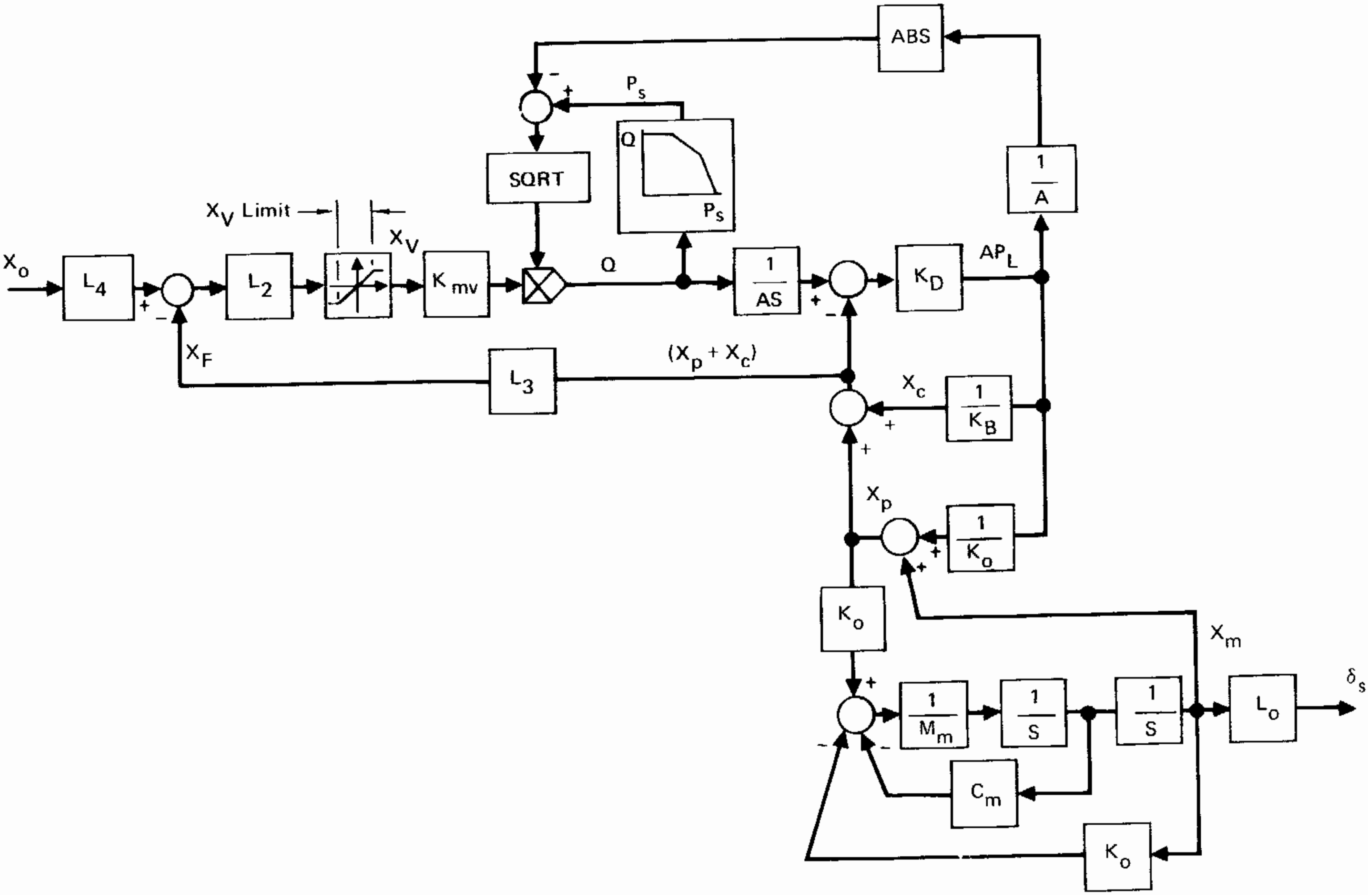


FIGURE 55
NONLINEAR MATH MODEL - SSAP SURFACE ACTUATOR

TABLE XXVI
MASTER CONTROL VALVE CONSTANT - SSAP SURFACE ACTUATOR

X_v	K_{mv} (One System Operating)	K_{mv} (Both Systems Operating)
±0.005	7.2	14.4
±0.01	10.8	21.6
±0.02	14.4	28.8
±0.03	16.2	32.4
±0.04	17.1	34.2
±0.05	17.7	35.4
±0.06	18.3	36.6
±0.07	18.6	37.2
±0.08	18.8	37.6
±0.09	19.0	38.0
±0.10	19.2	38.4

surface actuator response shows no signs of velocity or acceleration limiting in the large signal response.

(2) Secondary Actuator

A nonlinear math model, shown in Figure 59, was developed for the secondary actuator. It was based on the linear math model shown in Figure 50. The major nonlinearities considered in this model are limitations of the physical hardware. These limits were added to the linear math model as explained below.

The electronics associated with the secondary actuator contain a limiting circuit which limits the maximum width into the servo motor. This limit, shown in Figure 70, was placed in the math model at the output of the servo amplifier. The speed vs. pulse width characteristic for a prototype servo motor contains a deadspace and a steep rise, such that full motor speed is obtained with a pulse width of less than 25% of maximum.

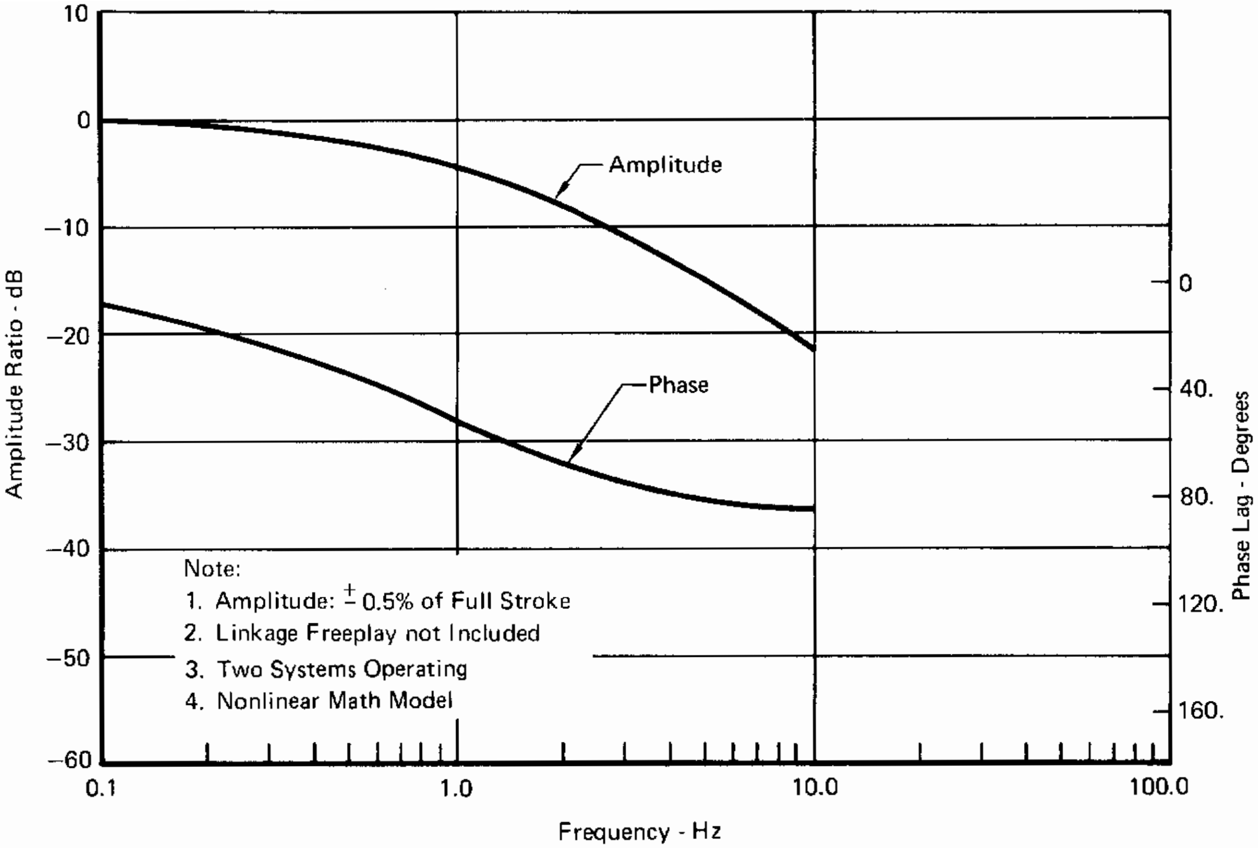


FIGURE 56
SMALL SIGNAL FREQUENCY RESPONSE - SSAP SURFACE ACTUATOR

The speed vs. pulse width characteristic of the prototype servo motor shown in Figure 58 was placed in the math model as a nonlinear function. The time constant for the servo motor is a function of the combined motor, tachometer and gear train inertias and the torque developed by the servo motor. This time constant was adjusted such that data generated using the math model would match empirical data from the prototype servo motor.

When the secondary actuator is stalled, the speed of the servo motors goes to zero. To perform this function a limit was included in the math model to limit the servo motor speed. This limit is a function of secondary actuator output position limit. When the output is against its positive position limit the positive motor speed limit is set to zero. The same relationship is true for the negative position limits.

The differentials, gears and the ballscrew in the secondary actuator have some inherent backlash. All of the backlash in the secondary actuator was reflected to the output and mechanized as friction controlled backlash. This backlash is inside the feedback loop and therefore, does not effect the actuator positionability. The backlash will, however, result in an output

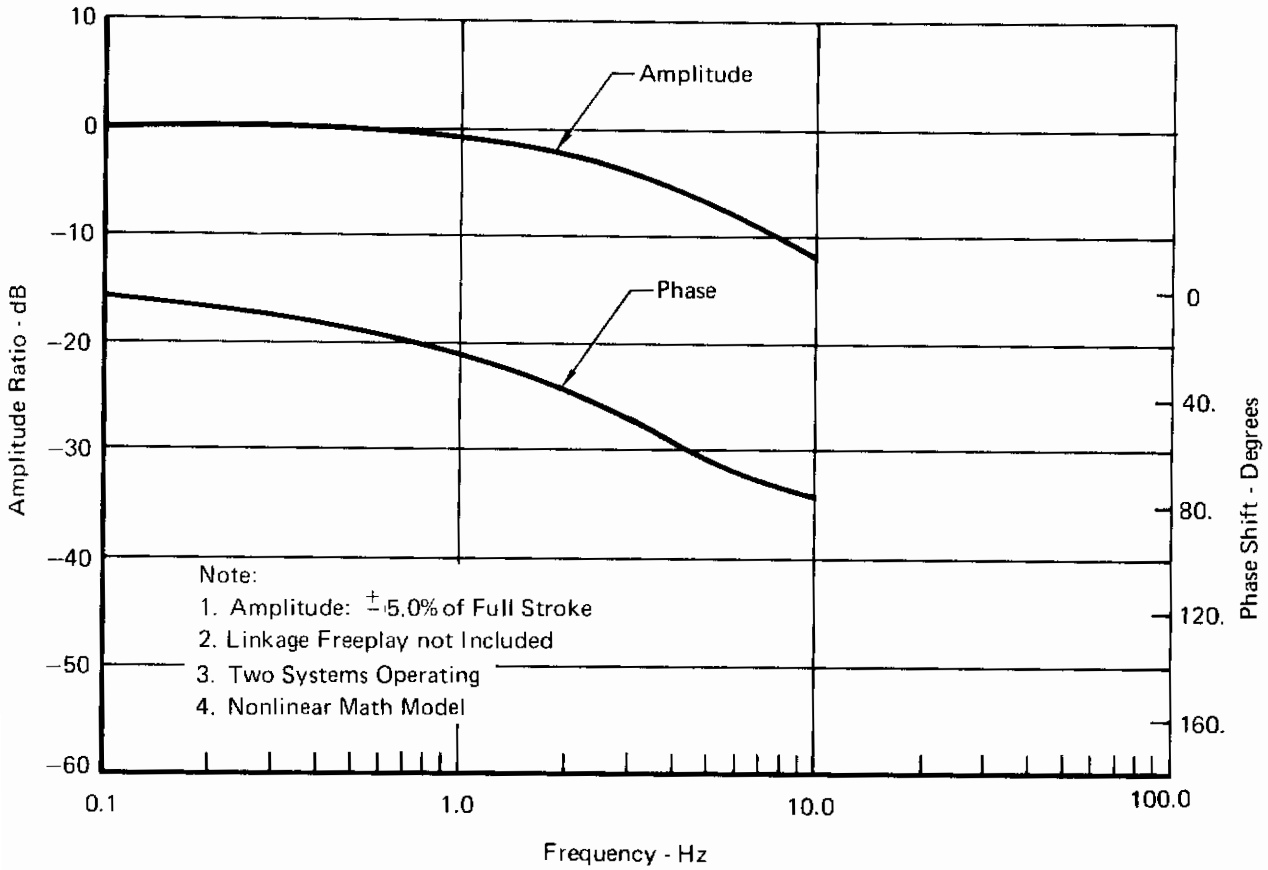


FIGURE 57
LARGE SIGNAL FREQUENCY RESPONSE - SSAP SURFACE ACTUATOR

with a nonlinear wave form. The fundamental component of this nonlinear wave will exhibit a considerable amount of phase shift for a small input signal.

The nonlinear math model for the secondary actuator is shown in Figure 59. Frequency response plots generated using this math model are shown in Figure 60 and Figure 61. The small signal frequency response in Figure 61 shows that backlash causes the actuator to exceed the small signal phase limits above 5.5 hz. The backlash used in this response was based upon the maximum allowable backlash for each component, and thus constitutes a worst case. This problem will be investigated later in the program to establish more realistic backlash estimates and to determine the effect on the overall system.

The large signal frequency responses presented in Figure 60 show that for certain frequencies the actuator exhibits excessive phase shift and amplitude attenuation. The basis of this problem is that the servo motors do not develop sufficient torque to accelerate the combined motor, tachometer, and gear train inertias at a rate sufficient to follow the input command signal. The acceleration saturation is verified by the fact that

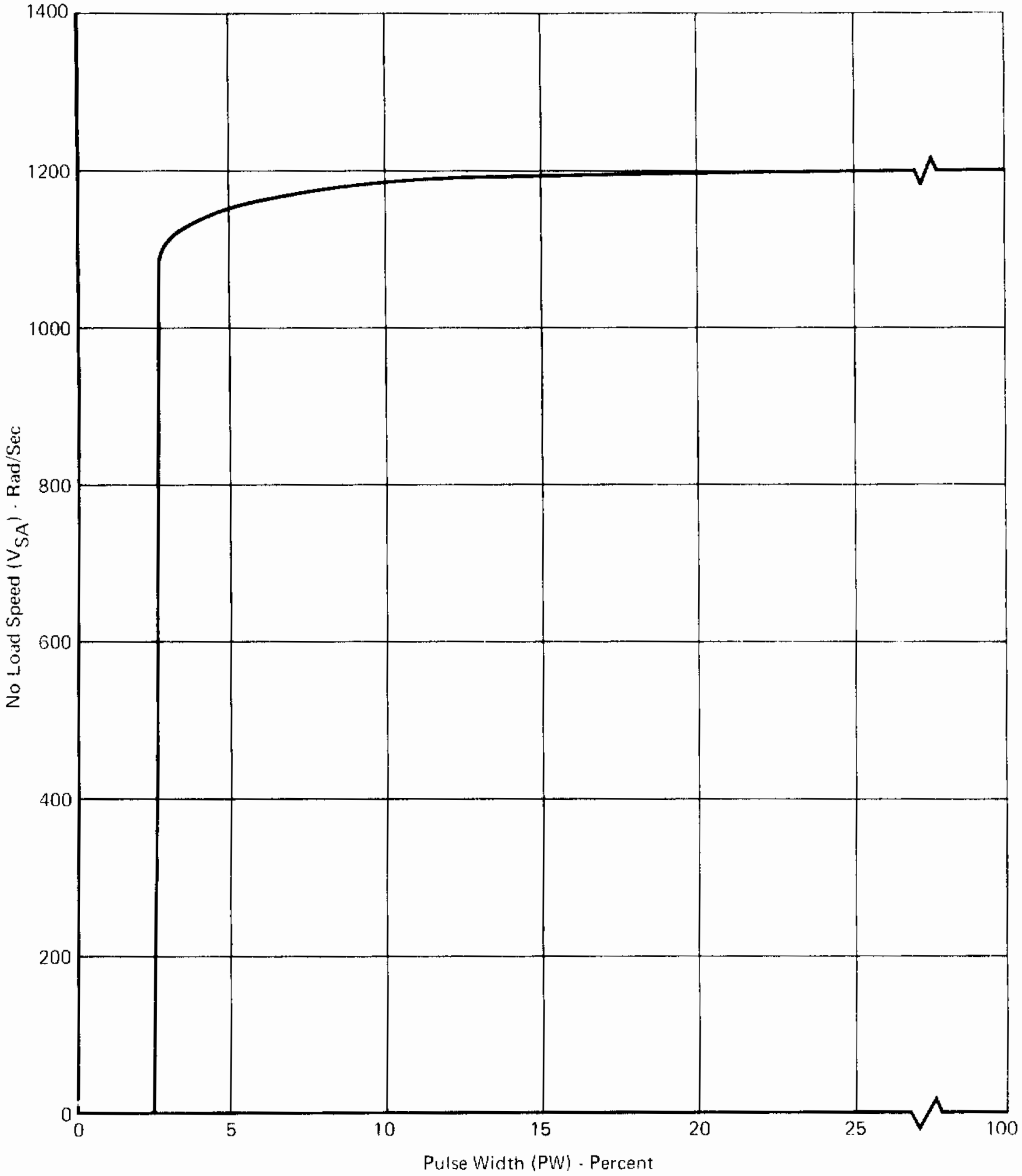
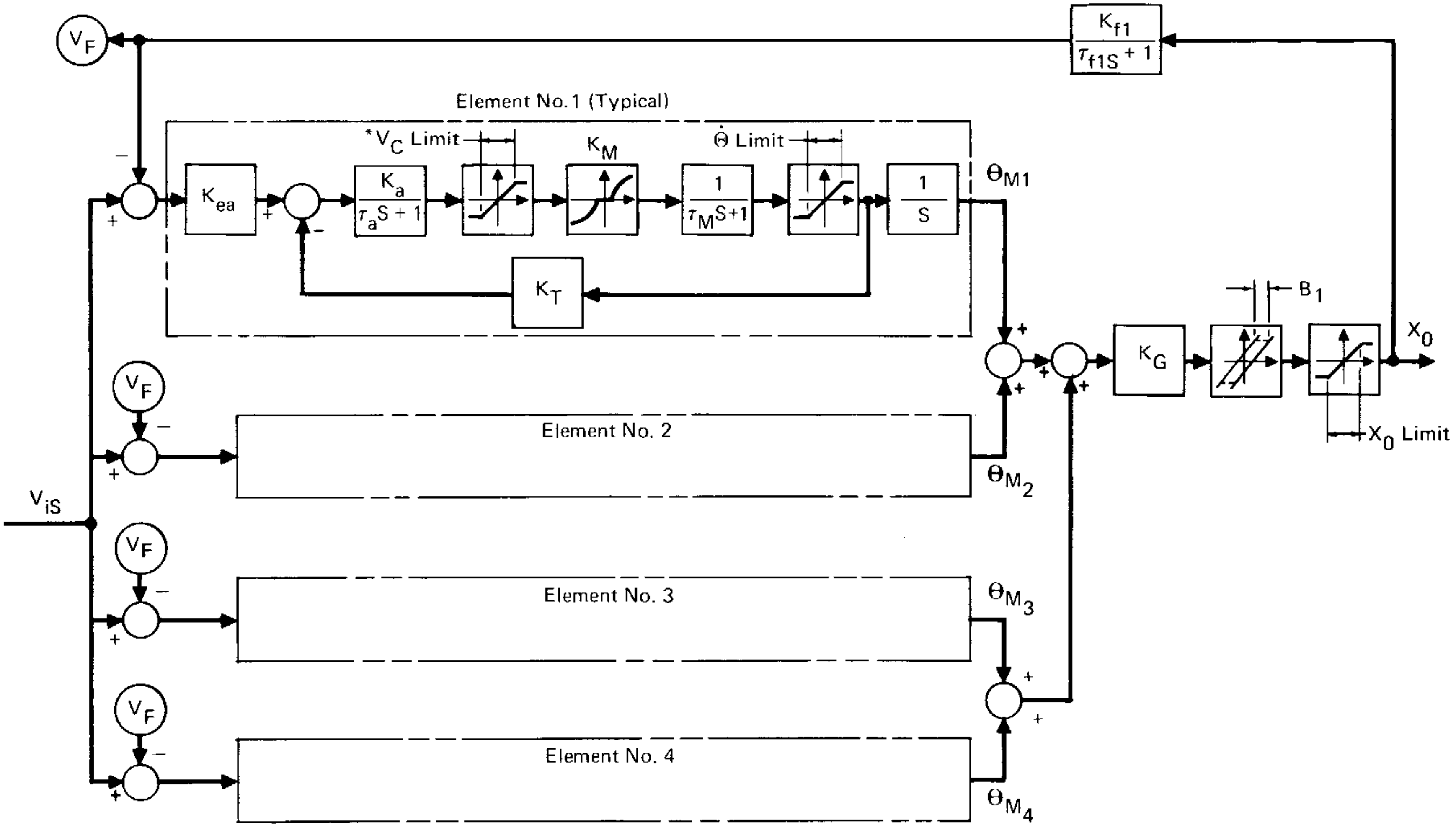


FIGURE 58

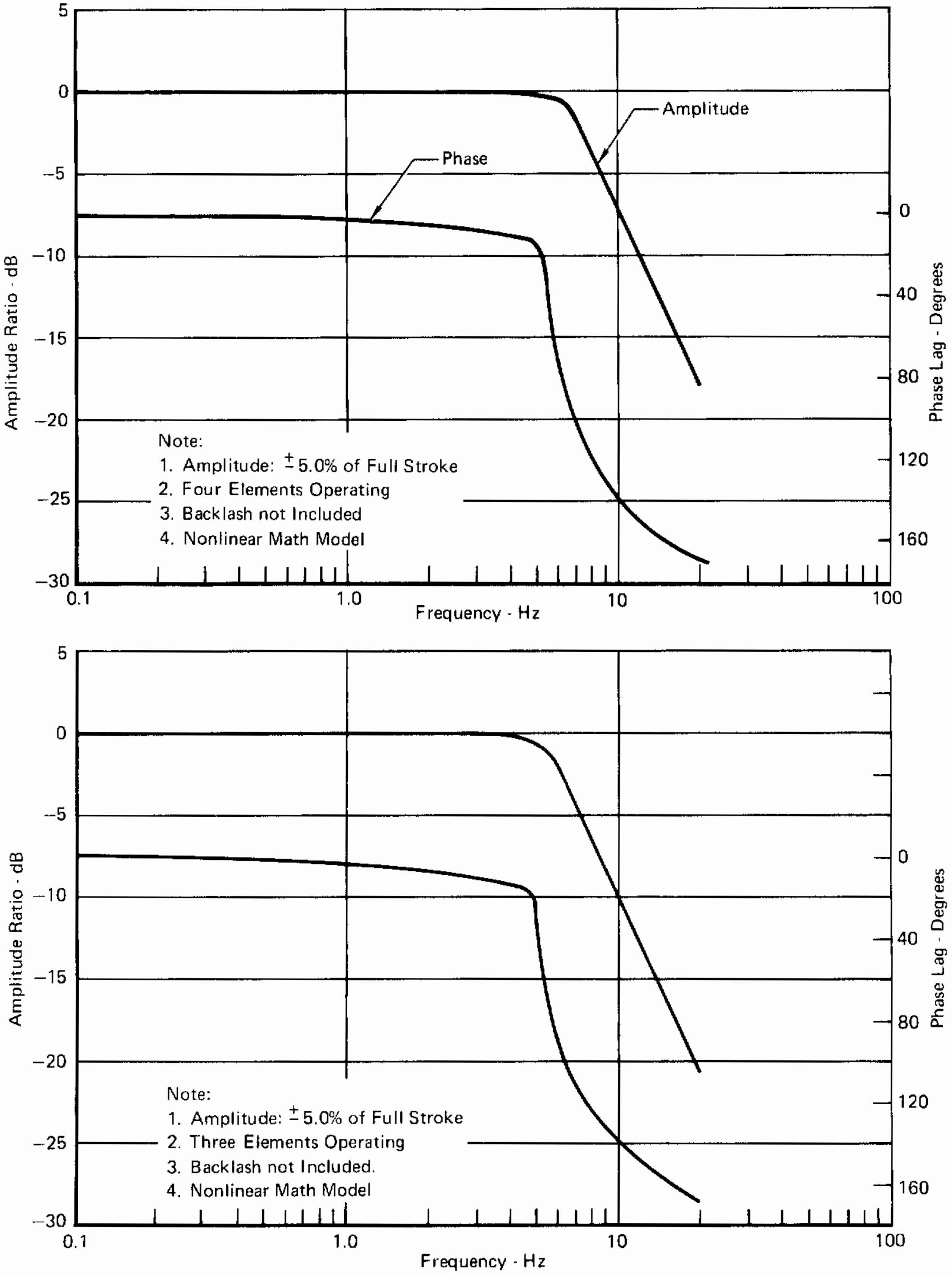
NO LOAD SPEED vs PULSE WIDTH

SSAP Secondary Actuator Servo Motors



*The effect of the pulse width limit is simulated by means of the V_C limit

FIGURE 59
NONLINEAR MATH MODEL - SSAP SECONDARY ACTUATOR



Note:
1. Amplitude: ± 5.0% of Full Stroke
2. Four Elements Operating
3. Backlash not Included
4. Nonlinear Math Model

Note:
1. Amplitude: ± 5.0% of Full Stroke
2. Three Elements Operating
3. Backlash not Included.
4. Nonlinear Math Model

FIGURE 60
LARGE SIGNAL FREQUENCY RESPONSE - SSAP SECONDARY ACTUATOR

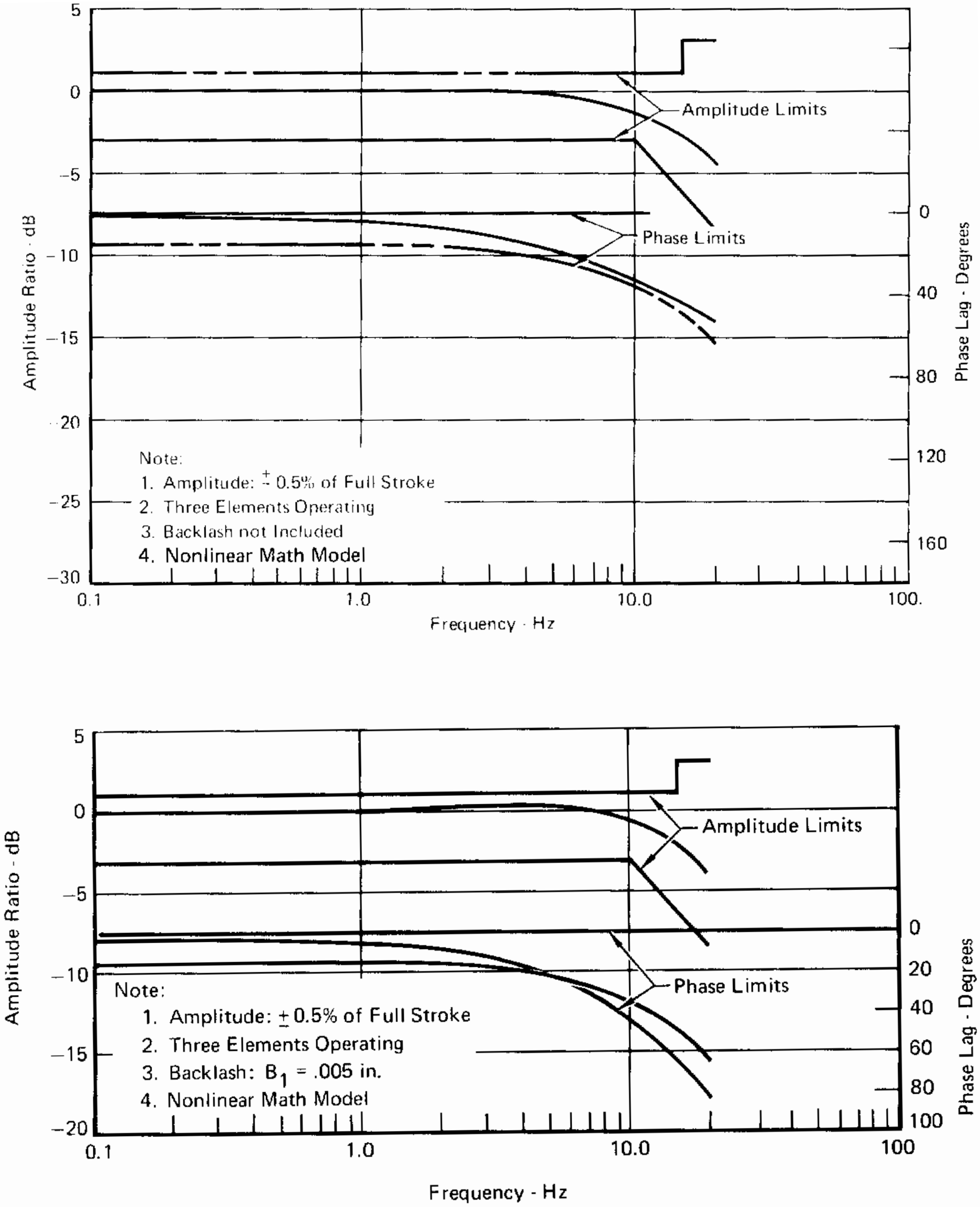


FIGURE 61
SMALL SIGNAL FREQUENCY RESPONSE - SSAP SECONDARY ACTUATOR

after the point of saturation, the amplitude ratio changes by approximately 12 dB per octave. As explained below, the excessive phase shift caused by acceleration saturation in the secondary actuator will cause the SSAP to be unable to meet the large signal frequency response requirements. If velocity saturation were to occur it would be at a higher frequency; it therefore will probably not be encountered.

(3) The Total SSAP

The total SSAP consists of the series combination of the secondary actuator and the surface actuator with an electrical feedback loop closed around both actuators. The nonlinear math model for the SSAP contains all the nonlinearities shown in the surface and secondary actuator models. In addition, it contains linkage freeplay and logic to adjust the maximum rate of the secondary actuator.

The mechanical linkage between the secondary actuator and the surface actuator and the surface actuator feedback linkage contains some inherent linkage freeplay at each bearing joint. All of this freeplay was reflected to the master control valve and mechanized as friction controlled backlash.

Logic was added to the SSAP math model to adjust the secondary actuator rate as a function of master control valve position. When the master control valve is against its position limit, the secondary actuator can not move the master control valve in the same direction as the surface actuator, at a rate greater than the surface actuator rate. To perform this limiting, logic was added to adjust the speed limit on the secondary actuator servo motors such that when the master control valve is against its position limit the secondary actuator rate is reduced to match the rate of the surface actuator. The surface actuator does not require a position limit due to the fact that the secondary actuator will hit its position limit before the surface actuator bottoms out.

The nonlinear math model for the SSAP is shown in Figure 62. The frequency response requirements for the SSAP are specified without electrical feedback from the surface actuator and with three elements of the secondary actuator operating. Frequency response plots were generated using the SSAP nonlinear math model to determine if the SSAP will meet the frequency response requirements.

Figure 63 and Figure 64 show the SSAP small signal frequency response plots. The SSAP response falls slightly outside the amplitude and phase limits even without considering the effects of backlash and linkage freeplay. Backlash and linkage freeplay cause a further degradation in the SSAP response. The impact of this degradation in response on the performance of the longitudinal control system is currently under investigation. Figure 65 shows the SSAP large signal frequency response plots. For frequencies above 5.4 Hz the SSAP falls outside the phase limits.

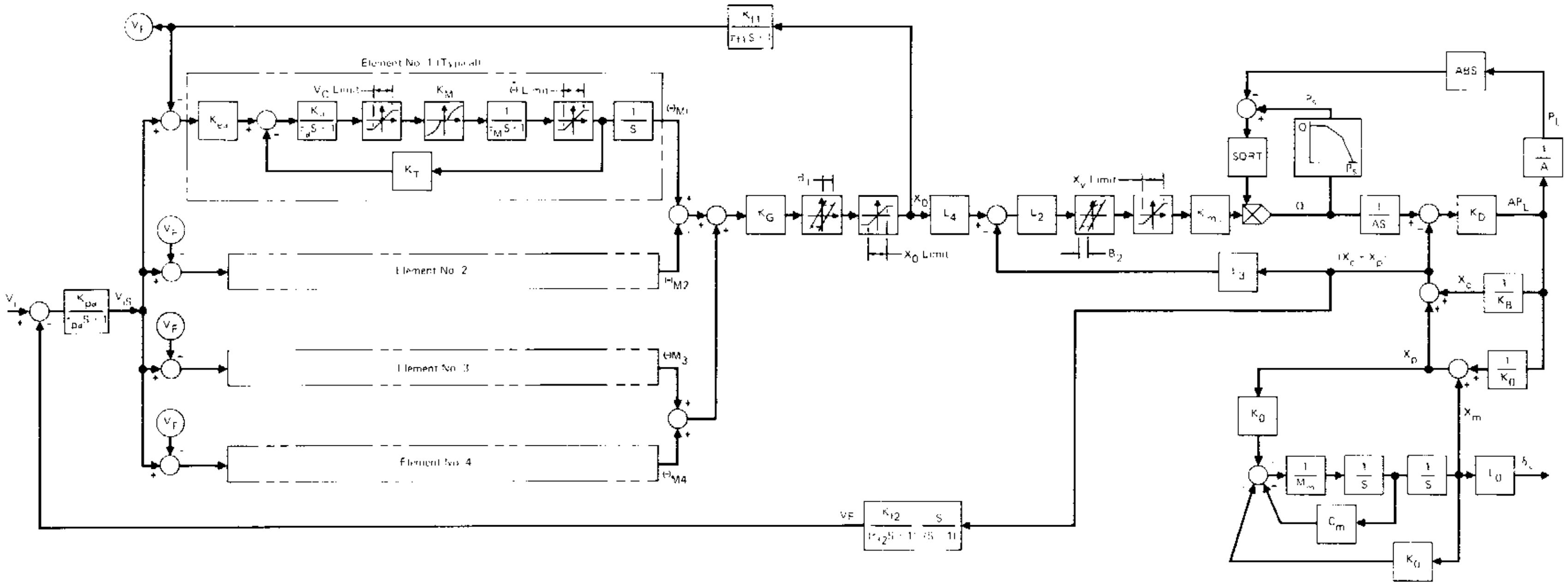


FIGURE 62
NONLINEAR MATH MODEL - SSAP

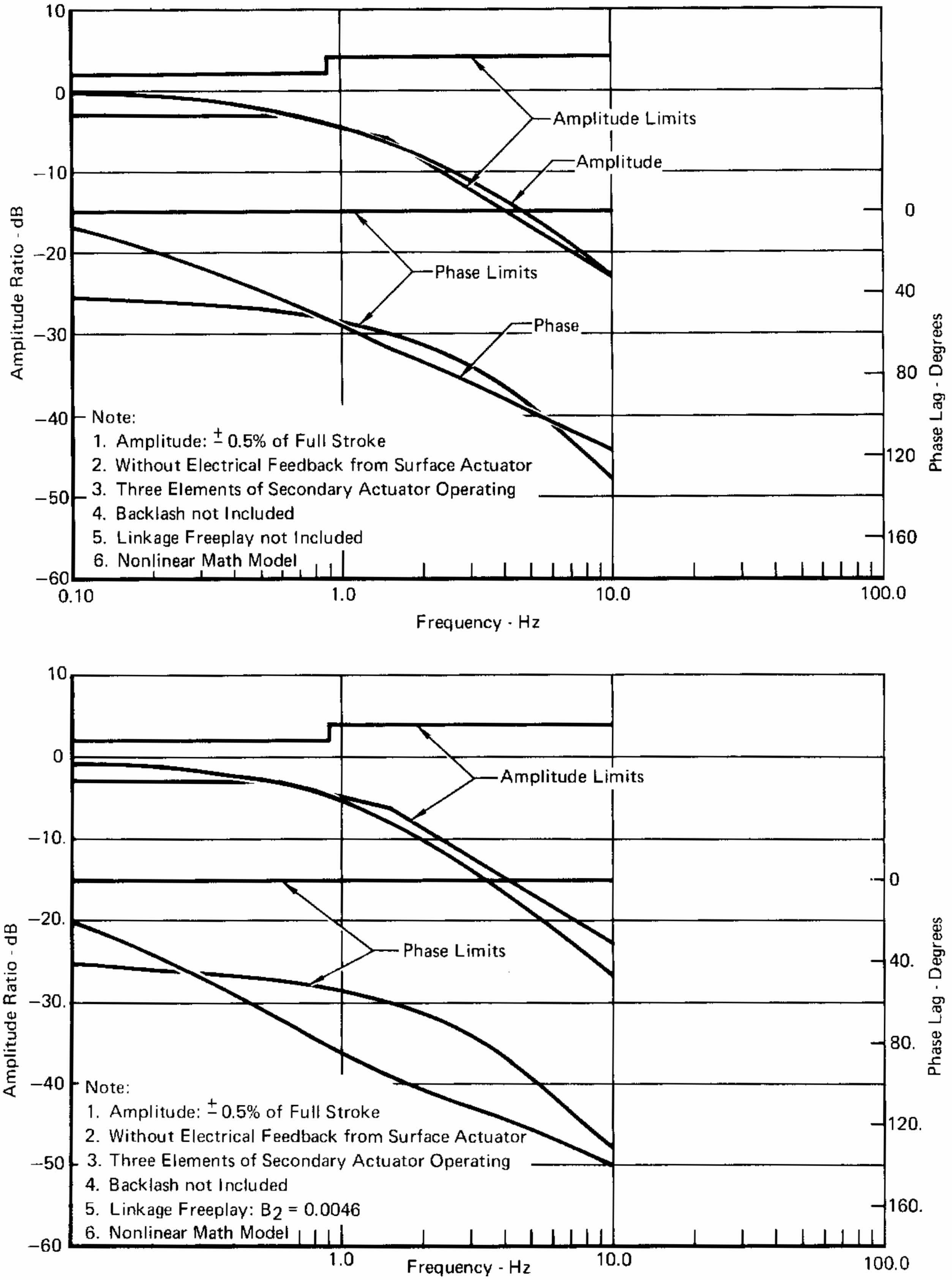
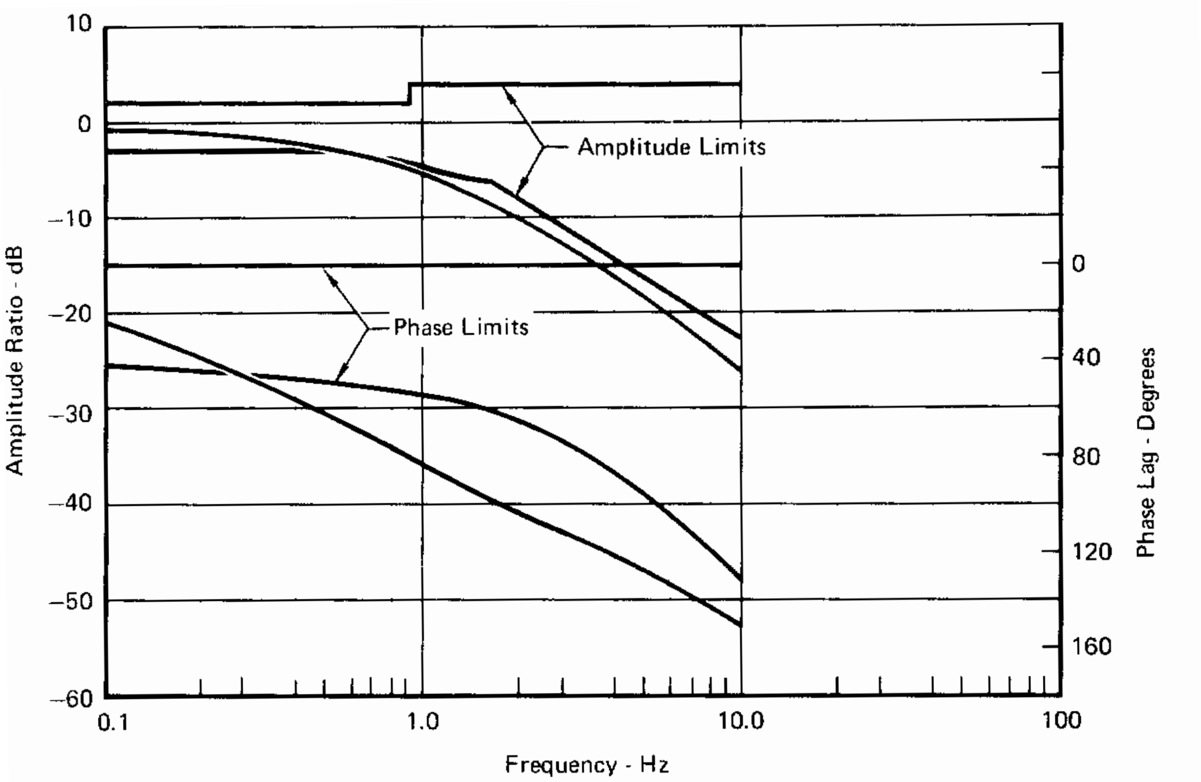


FIGURE 63
SMALL SIGNAL FREQUENCY RESPONSE - SSAP



Notes:

1. Amplitude: ± 0.5% of Full Stroke
2. Without Electrical Feedback From Surface Actuator
3. Three Elements of Secondary Actuator Operating
4. Backlash: B_1 = 0.005
5. Linkage Freeplay: B_2 = 0.0046
6. Nonlinear Math Model

FIGURE 64
SMALL SIGNAL FREQUENCY RESPONSE - SSAP

This is a direct result of acceleration saturation in the secondary actuator. The effect of this saturation on the stability of the longitudinal control system is discussed in Supplement 2 of this report. Figures 66 and 67 show the large and small frequency response plots for the SSAP with electrical feedback from the surface actuator.

This analysis was based on data from a prototype servomotor and preliminary estimates on many of the other parameters. Therefore, it is expected that this analysis will be updated when data is available on the production hardware.

c. SSAP Threshold

The threshold for the SSAP is specified as 0.06% of the full stroke of the surface actuator. This threshold corresponds to 0.0058 inches. The threshold for the SSAP is dependent upon the threshold of the secondary actuator and the linkage freeplay between the secondary actuator and the surface actuator.

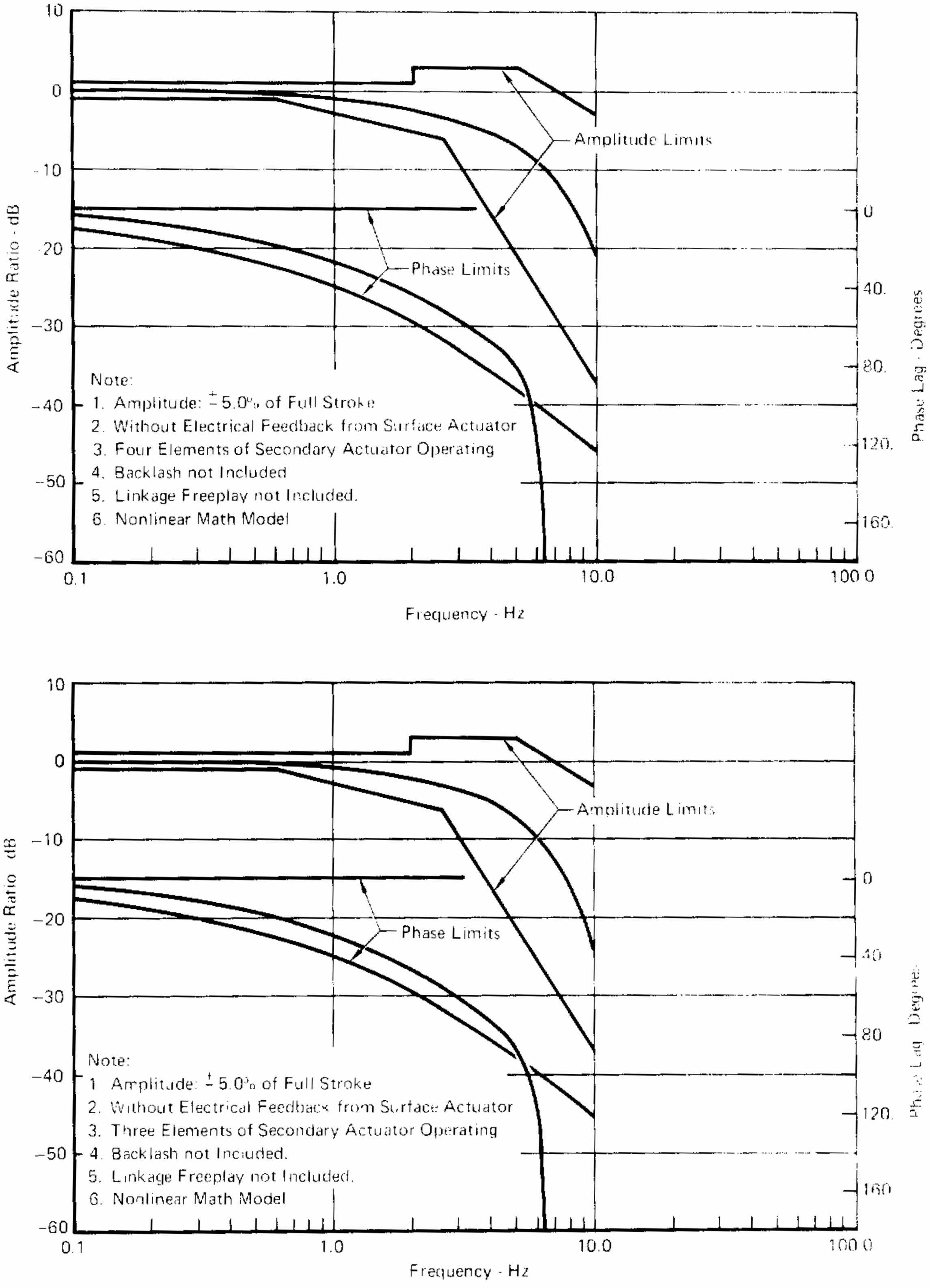
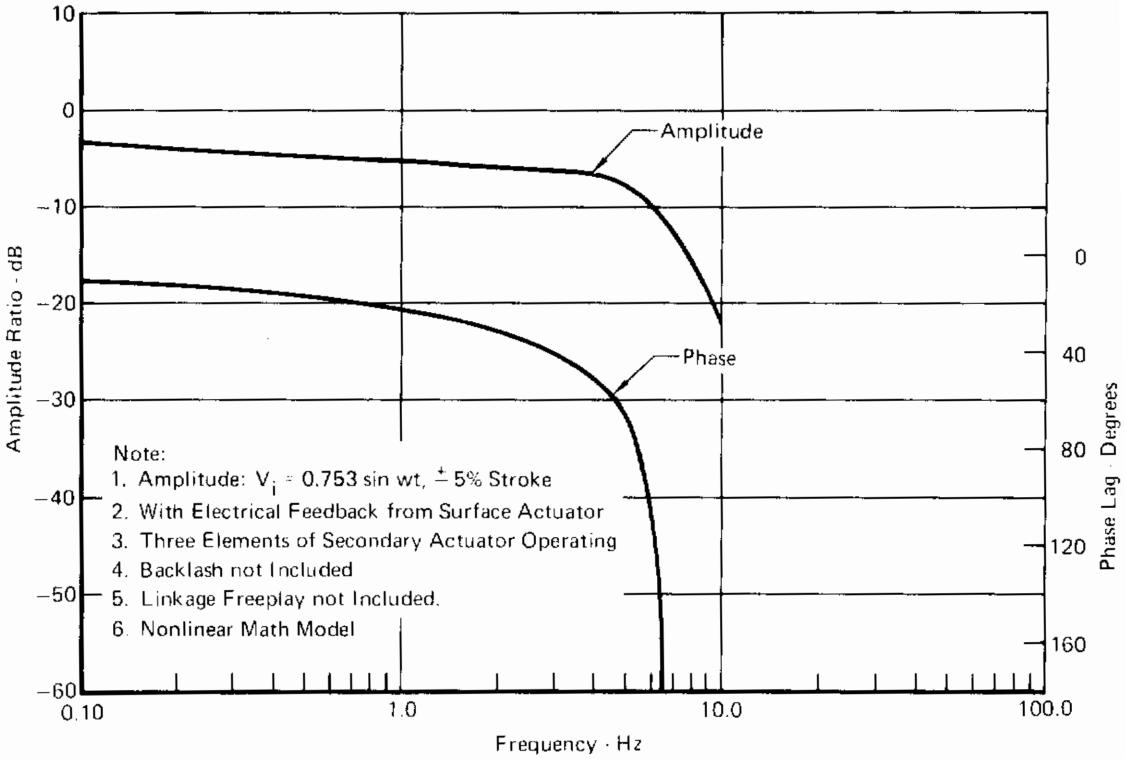
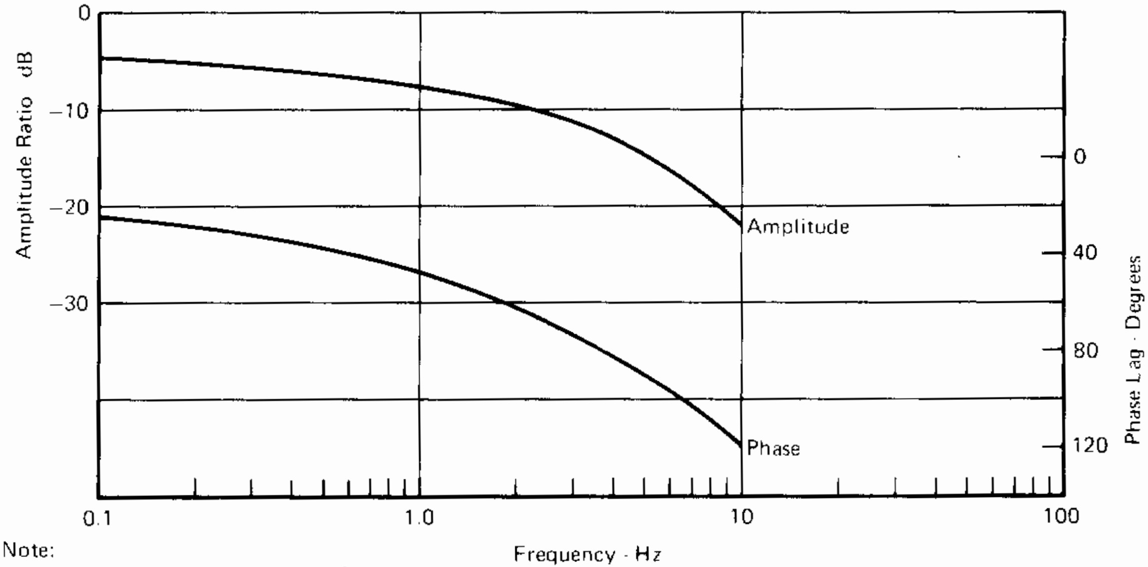


FIGURE 65
LARGE SIGNAL FREQUENCY RESPONSE - SSAP



Note:
1. Amplitude: V_i = 0.753 sin wt, ± 5% Stroke
2. With Electrical Feedback from Surface Actuator
3. Three Elements of Secondary Actuator Operating
4. Backlash not Included
5. Linkage Freeplay not Included.
6. Nonlinear Math Model

FIGURE 66
LARGE SIGNAL FREQUENCY RESPONSE - SSAP



Note:
1. Amplitude: V_i - 0.0753 sin wt, ± 0.5% Stroke
2. With electrical feedback from surface actuator
3. Three elements of secondary actuator operating
4. Backlash not included
5. Linkage freeplay not included
6. Nonlinear Math Model

FIGURE 67
SMALL SIGNAL FREQUENCY RESPONSE - SSAP

The threshold for the secondary actuator is due primarily to the threshold of the servo motors. From the speed vs. pulse width curve for a prototype servo motor shown in Figure 58, it can be seen that the motor has a threshold of 5.0% pulse width. The servo motor threshold can be reflected to the surface actuator output by:

$$X_p \text{ (sec.act.)} = \frac{L_4 \text{ (motor threshold)}}{L_3 \, K_{f_1} \, \text{Kea Ka}} \qquad (21)$$

Using the parameters from Table XXV, the threshold due to the secondary actuator is 0.0012 inches.

The threshold due to freeplay in the linkage between the secondary actuator and the surface actuator can be determined by reflecting the freeplay at each linkage bearing joint to the surface actuator output. Assuming that the inherent freeplay in each bearing joint is 0.001 inches, the threshold due to linkage freeplay is 0.033 inches at the surface actuator output. This results in a total threshold of 0.36% of full stroke or approximately six times greater than the specified value. Due to the fact that the primary source of threshold is linkage freeplay, precision bearings and bolts are being used to reduce the threshold to values consistent with overall SFCS requirements.

d. Failure Transients

A failure transient will result whenever an active failure occurs in a secondary actuator element. When such a failure occurs, that actuator output is displaced until a velocity balance is achieved or the brake is applied and the failed element is switched off line. On third failure the brakes are applied to hold position when the last two elements are switched off line.

Under dynamic conditions, failure transients are influenced by control linkage and surface actuator dynamic characteristics in addition to monitor delay time. Differences in maximum servo motor speeds may also be a factor. The effect of these characteristics on failure transients will be investigated during the SSAP simulation scheduled for later in the program. However, when these characteristics are ignored, the failure transients for the secondary actuator may be estimated by means of a linear analysis. The output displacement of the secondary actuator is specified not to exceed 0.56, 0.95, and 2.2 percent of full stroke for first, second, and third failures, respectively.

A linear analysis was performed to determine the maximum failure transients for the secondary actuator. This analysis was based on the linear math model shown in Figure 50. Assuming a hardover into one element and assuming a steady state condition is reached before shutoff, the resulting failure transient can be approximated by the following equation:

$$X_o = \frac{(1 + K_T K_a K_M) \, [\dot{\theta}_{Mi}]}{K_{f1} \, K_{ea} \, K_a \, K_M \, (N_a - 1)} \qquad (22)$$

It should be noted that $\dot{\theta}_{Mi}$ is the maximum speed of servomotor in the failed element and N_a is the number of active elements prior to the hardover failure.

The maximum failure transients are shown in the table below:

	1st Failure	2nd Failure	3rd Failure
X_o (Percent of full stroke)	0.68	1.01	2.04

These failure transients are based on allowing the actuator to reach a steady state condition before the failed element is shut off. Although the 1st and 2nd failure transients do not meet the requirement they can be reduced by setting the trip out time such that the failed element is shut off before the actuator reaches a steady state condition. The trip out time is more critical on the third failure than for the first or second failure. This is due to the fact that if the maximum speeds of the servo motors are not identical, the output will drift until the elements are shut off. For example, if the maximum speed of the servo motor in the failed element is 10% greater than the other motor, the output will drift at a rate of approximately 6% of full stroke per second. The SSAP failure transients will be fully investigated during the SSAP simulation.

e. Nuisance Disconnect

Nuisance disconnect in the SSAP may occur in the motor pumps or in the secondary actuator. In the SSAP, hydraulic fluid is normally supplied to the surface actuator by two motor pumps at pressures greater than 500 psi. Should motor pump output pressure fall below 500 psi, the hydraulic switching valve will switch the surface actuator from an integral to an aircraft hydraulic system. Once switchout has taken place, the surface actuator will remain on the aircraft hydraulic supply until the hydraulic switching valve is reset manually. The pilot can accomplish reset in flight by pushing a reset button on the SSAP hydraulic system status panel.

In the electromechanical secondary actuator, nuisance disconnects will occur if the tolerance buildup between elements results in an angular rate difference in excess of the tripout threshold between one element and all other elements not previously failed. For the special case where the tolerance buildup is assumed to occur in one element only, the difference in angular velocity between the element with a toleranc buildup and the elements with zero tolerance buildup can be found from the following equation.

$$\Delta\dot{\theta} = \frac{K_{ea} K_a K_M V_s}{1 + K_a K_m K_t} \simeq \frac{K_{ea} V_s}{K_T} \qquad (23)$$

For a tolerance buildup (V_S) equivalent to one percent of maximum signal, the angular velocity between elements is 295 rad/sec. Since the tripout threshold is 785 rad/sec (7500 rpm), a tolerance buildup of approximately 2.7 percent of maximum signal will result in a nuisance disconnect. Equation (23) is independent of the number of elements which are operating so that the sensitivity of the secondary actuator to nuisance disconnect does not change.

An abbreviated statistical analysis, similar to that discussed in Paragraph 2.e, was conducted to provide a statistical estimate of the probability of nuisance disconnect. This analysis included only those tolerances found significant in the analysis described in paragraph 2.e, namely, input signal, LVDT excitation, and LVDT tracking. The three sigma value of differential angular velocity was found to be less than 400 rad/sec, which is well below the tripout threshold of 785 rad/sec. A review of this analysis will be conducted once tolerances in the servomotor drive electronics are established and operating characteristics under stall conditions have been evaluated.

f. Output Velocity

Both the secondary and surface actuators must meet maximum slew rate requirements under no-load conditions. In addition, the surface actuator must achieve a specified output velocity when retracting against an 8600 pound load which is the worst case condition.

(1) Secondary Actuator No-Load Velocity

The velocity of the secondary actuator along its two inch stroke is approximately:

$$\dot{X} = N_a K_G \dot{\theta}_{MAX} \tag{24}$$

where N_a is the number of elements operating. The no-load velocity with four elements operating is approximately 5.0 inches per second. With the loss of each element a 25 percent reduction in velocity occurs. Since the requirement is to travel full stroke in 0.9 seconds, the requirement is met with two or more elements operating.

(2) Surface Actuator No-Load Velocity

The surface actuator may operate on either integral and/or aircraft hydraulic supplies. With either hydraulic supply system, the maximum velocity under no-load conditions is determined by supply system characteristics, master control valve characteristics, and internal line losses. With an aircraft hydraulic supply, the no-load velocity is also influenced by external line losses. The master control valve characteristics for the SSAP differ from the F-4 production actuator valve characteristics in that the orifice areas are larger near maximum travel. This results in a higher output velocity for the SSAP surface actuator whether operating on integral or aircraft hydraulics. The change in valve characteristics was made to increase the output velocity under load and an increase in no-load velocity is a natural consequence.

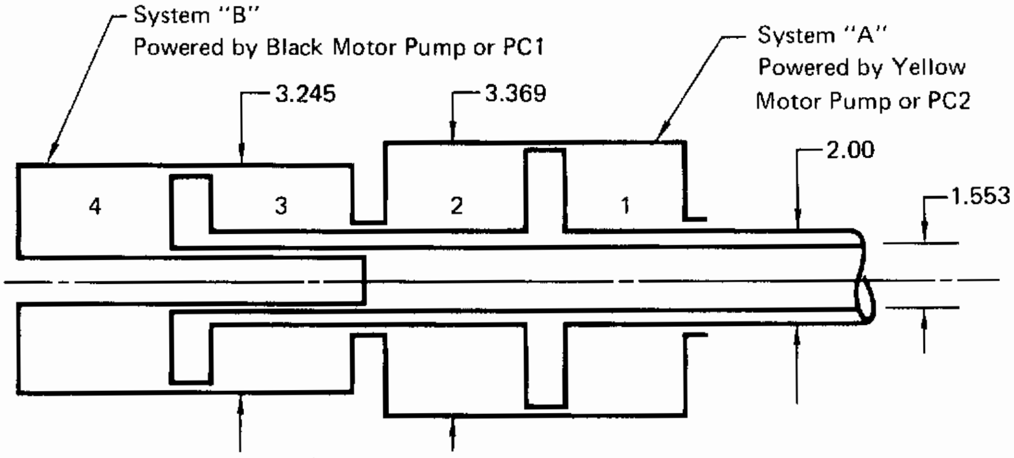
(3) Velocity Under Load

The purpose of this study was to determine the adequacy of the pump flow pressure characteristics to provide an SSAP output velocity of 6.9 in/sec while operating against a retract load of 8600 pounds. This requirement applies with both pumps operating and provides an adequate stabilator recovery rate when the aircraft encounters a low static stability flight condition. An evaluation of the adequacy of the pump characteristics involves the matching of pump, master control valve, and surface actuator characteristics as discussed in the following paragraphs.

(a) Neutral Pressures

With 1000 psi applied to the actuator, the output velocity is required to be approximately 10.7 inches per second. Using this as a design condition and assuming an ideal neutral pressure of 500 psi, the no-load pressure in each cylinder, based on the piston areas shown in Figure 68 will be:

$$P_1 = 500 \text{ psi}$$
$$P_2 = 500 \text{ psi}$$
$$P_3 = (A_3 + A_4)/2\ (500/A_3) \simeq 560 \text{ psi}$$
$$P_4 = P_3(A_3/A_4) \simeq 450 \text{ psi}$$



$A_1 = (3.369^2 - 2.00^2)\ \pi/4 = 5.77 \text{ in.}^2$ $A_3 = (3.245^2 - 2.00^2)\ \pi/4 = 5.13 \text{ in.}^2$

$A_2 = (3.369^2 - 2.00^2)\ \pi/4 = 5.77 \text{ in.}^2$ $A_4 = (3.245^2 - 1.553^2)\ \pi/4 = 6.38 \text{ in.}^2$

FIGURE 68
ACTUATOR AREAS

(b) Valve Pressure Drop

From above, the valve pressure drop at 10.7 in/sec will be:

	Extend	Retract
ΔP_1	500 psi	500 psi
ΔP_2	500 psi	500 psi
ΔP_3	560 psi	440 psi
ΔP_4	550 psi	450 psi

and the pressure drop at 6.9 in/sec will be:

$$\Delta P = \Delta Po \ (V/Vo)^2 = \Delta Po \ (6.9/10.7)^2 = .416 \ \Delta Po$$

	Extend	Retract
ΔP_1	208 psi	208 psi
ΔP_2	208 psi	208 psi
ΔP_3	233 psi	183 psi
ΔP_4	229 psi	187 psi

(c) Pump Pressure Available

The flow rates required are determined from

$$Q = \dot{X} A_p$$

At 6.9 in/sec the flow rates are:

$$Q_1 = 6.9(5.77) = 39.9 \text{ in}^3\text{/sec or } 10.35 \text{ gpm}$$

$$Q_2 = 6.9(5.77) = 39.9 \text{ in}^3\text{/sec or } 10.35 \text{ gpm}$$

$$Q_3 = 6.9(5.13) = 35.4 \text{ in}^3\text{/sec or } 9.2 \text{ gpm}$$

$$Q_4 = 6.9(6.38) = 44.0 \text{ in}^3\text{/sec or } 11.4 \text{ gpm}$$

From Figure 69, the minimum pump output pressures will be:

	Extend	Retract
Syst A Pressure	1170 psi	1170 psi
Syst B Pressure	1050 psi	1300 psi

(d) Output Force

The force capability, neglecting friction, will be:

$$F_{ext} = (P_{Aext} - \Delta P_{2ext}) \ A_2 - (\Delta P_{1ext}) A_1$$

$$+ (P_{Bext} - \Delta P_{4ext}) \ A_4 - \Delta P_{3ext} A_3$$

$$= (1170 - 208) \ 5.77 - (208) \ 5.77$$

$$+ 1050 - 229) \ 6.38 - (233) \ 5.13$$

$$= 8393 \text{ lbs.}$$

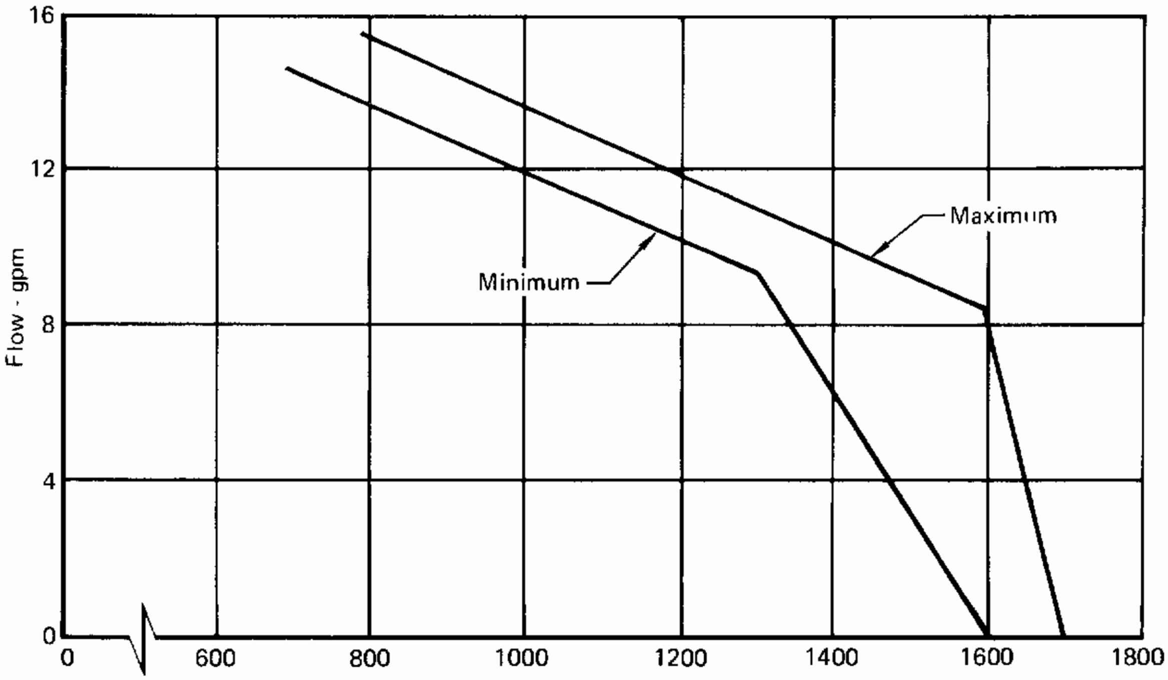


FIGURE 69
SSAP PUMP PRESSURE - FLOW CHARACTERISTICS

$$F_{ret} = (P_{Aret} - \Delta P_{1ret})\, A_1 - (\Delta P_{4ret})\, A_2$$

$$+ (P_{Bret} - \Delta P_{3ret})\, A_3 - (\Delta P_{4ret})\, A_4$$

$$= (1170 - 208)\ 5.77 - (208)\ 5.77$$

$$+ (1300 - 183)\ 5.13 - (187)\ 6.38$$

$$= 8887 \text{ lbs.}$$

With a 6.9 inch per second rate, the forces available are 8887 pounds retract and 8393 pounds extend which exceeds the requirements in both directions. Since the critical direction is retract, (stabilator trailing edge down) the presently specified pumps, with the characteristics shown in Figure 69 , will provide acceptable performance.

g. In-Flight Monitoring (IFM)

In-flight monitoring for the SSAP includes monitoring for the two motor pumps in addition to the secondary actuator. Switching valve position, low reservoir fluid level, pump over-temperature, and low pump output pressure are monitored on each motor pump. Should the output pressure of an integral pump fall below a specified pressure the switching valve will automatically switch the faulty motor pump off and switch in an aircraft hydraulic supply. The in-flight monitor provides the pilot with information on integral to aircraft hydraulic system switchover, on empty reservoir, a high motor pump temperature, and degraded motor pump operation, respectively. All of these provide malfunction indication to the pilot. Of these only the hydraulic system switch over signal is the result of a pilot initiated action; the others are automatically activated.

For the SSAP secondary actuator, the in-flight monitoring circuitry is functionally equivalent to that of Figure 44 and discussed in Paragraph 2.i. The signals to six comparators are obtained from dual tachometer windings connected to each of four servo motors. A center tapped tachometer output winding and a special coil which couples excitation voltage to the output at zero rpm allow the use of the carrier sensor monitoring scheme. The carrier sensor signal will provide a failure signal if the tachometer windings are open, shorted, or a power failure occurs. The monitor time delay has been tentatively set at 100 milliseconds.

h. Voltage Variations

A study of the effects of voltage variations was conducted to determine the effects on the performance of the SSAP. The results of the study are presented in the following section.

The supply bus voltage variations are expected to be 27.0 to 28.3 VDC when the battery is being charged from a properly functioning transformer-rectifier, and 20.0 to 24.5 VDC when the battery is not being charged.

The effects of supply voltage variations on the performance of the secondary actuator are as follows:

(1) Brakes

In order to prevent overheating in the brake solenoids, a regulated voltage of +19 volts, derived from the battery voltage, will be used for brake power. With a regulated supply, brake performance and heating will be unaffected by battery voltage variations.

(2) Maximum Actuator Speed

The maximum speed is dependent almost entirely upon the frequency of the 400 Hz generated by the SFCES and is virtually independent of the voltage over the supply range of 20 to 28.3 VDC. Below a stall torque load the tachometer feedback maintains a nearly constant motor speed relative to supply voltage.

(3) Maximum Power and Torque

In order to reduce power dissipation in the electronics and in the motor, pulse width modulation of both the fixed and control phases of the motor is employed. The maximum pulse width is limited as a function of supply voltage as shown in Figure 70. By limiting the maximum pulse width, the maximum power from the battery and the maximum dissipation in the electronics are not appreciably affected by supply voltage variations. The maximum torque is also limited so the velocities of channels supplied from separate batteries will not differ appreciably under stall conditions. This prevents nuisance disengagement resulting from differences in stall torque with different battery voltages.

(4) Static and Dynamic Performance

If it is assumed that there are no tolerances on pulse width limiting, the performance of the Secondary Actuator is almost unaffected by supply voltage variations during stall conditions because the maximum pulse width is limited as described above. If the supply voltages of two channels are +20.0 volts and +28.3 volts, respectively, there is little affect on the Secondary Actuator stall torque. However, pulse width limiting does not compensate for differences in channel performance when pulse widths are below the maximums. The probability of nuisance disengagement is slightly increased due to the combination of steady state and dynamic difference velocities described below.

(a) Steady-State Difference Velocities

The steady-state difference velocities of any two channels increase at high torque loads. For example, if the two channels have, respectively, error offsets of +0.1 and -0.1 volt (referred to a ±10 input), the difference velocity at no load is 592 rad/sec-based on speed-torque data of a prototype motor. If the load is increased to 7.7 in-oz, with zero output displacement velocity, the velocity difference increases to 652 rad/sec, an increase of 60 rad/sec.

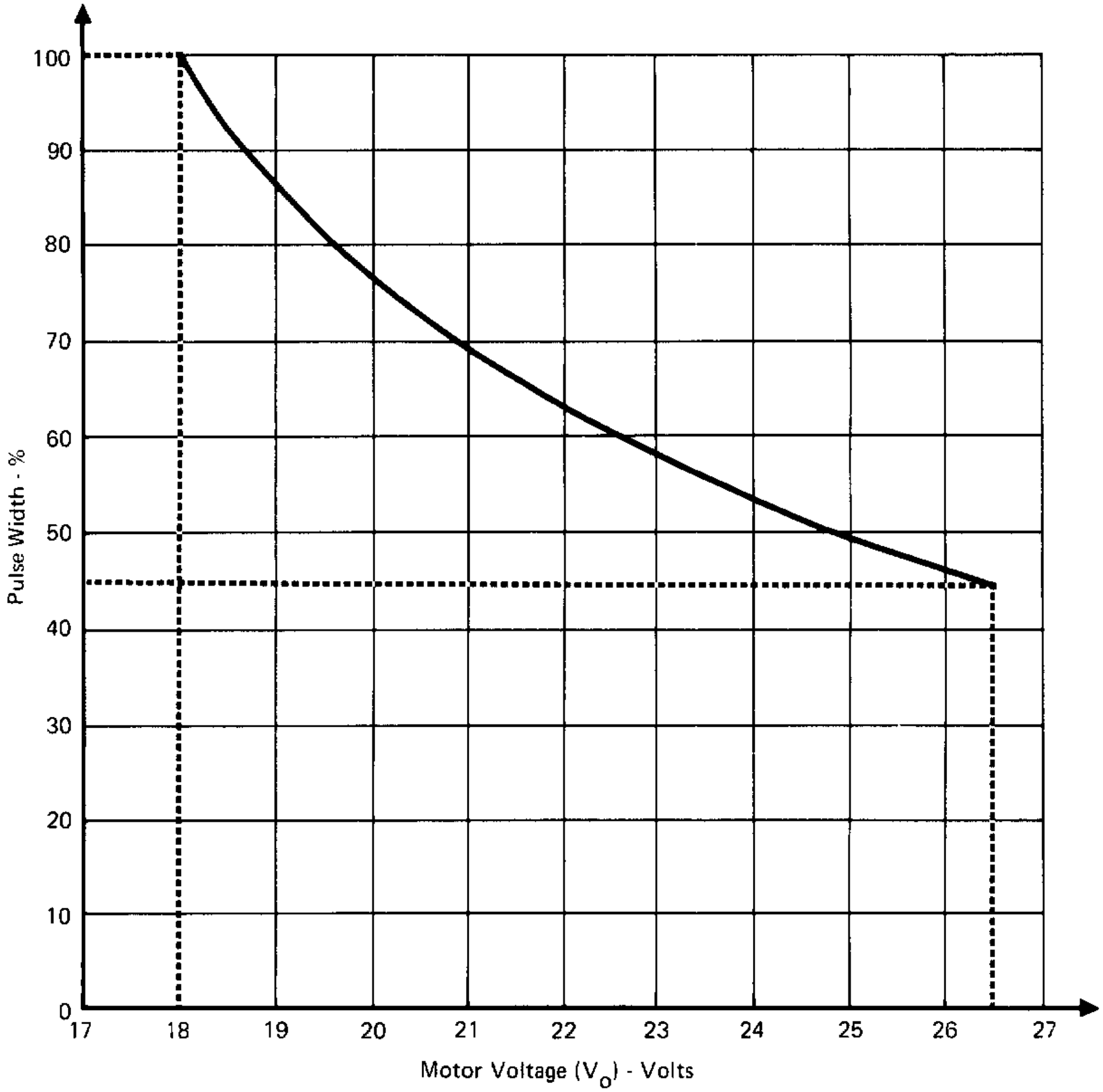
(b) Dynamic Velocity Differences

The dynamic performances of any two channels differ because greater torque can be generated in the high-voltage channel for a given error signal. The time constants of the tachometer feedback loops of the two channels differ by more than a two-to-one ratio:

τ_{20v} = 285 rad/sec. $\tau_{28.3v}$ = 909 rad/sec.

When equal sinusoidal signals are applied simultaneously to both channels, the differences in phase shift result in sinusoidal velocity differences. For example, an 8 Hz signal of sufficient amplitude applied to both channels can produce the following difference velocity:

$$\Delta\dot{\theta} = 84 \sin 16\pi t \text{ rad/sec}$$



Notes:
1. V_o levels assume approximately 2 VDC drop in switch elements from Vbatt levels.
2. Limit is based on nominal motor input power limit of 150 watts total with absolute limiting of 160 watts into motor.

FIGURE 70
PULSE WIDTH LIMITING AS A FUNCTION OF SUPPLY VOLTAGE

This difference is superimposed on the no-load velocity difference and the probability of nuisance disengagement is slightly increased.

The above examples of steady-state and dynamic difference velocities show that the probability of nuisance disengagement will be increased if only the maximum pulse widths are limited as a function of supply voltage. The increases in difference velocity of 60 rad/sec and $84 \sin 16\Pi t$ rad/sec are added to the no-load difference velocities. If the no-load difference velocity is in the order of 592 rad/sec, such increases are significant with the 785 rad/sec threshold.

i. Stabilator Actuator Feedback

The purpose of this study was to evaluate the merits of mechanical and electrical feedback for stabilator actuator operation. Study results indicated that mechanical feedback should be used around the surface actuator and electrical feedback should be used to close a loop around both secondary and surface actuators as shown in Figure 71. The considerations which resulted in a decision to employ both mechanical and electrical feedbacks of surface position are as follows:

- After a third failure, the secondary actuator is held in its last position. If electrical feedback only were used around the surface

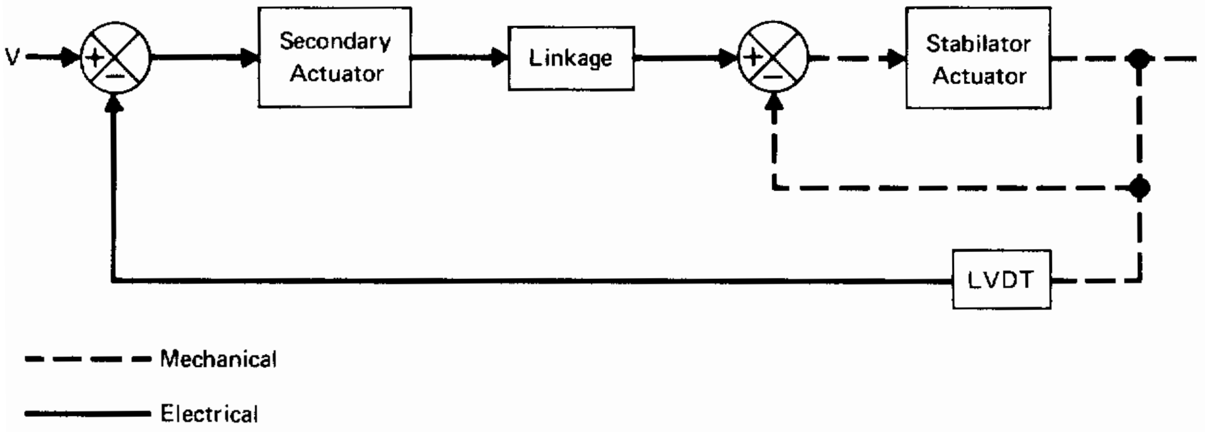


FIGURE 71
STABILATOR ACTUATOR FEEDBACK

actuator with hydraulic power still applied, the surface actuator would drift and the effect of this drift could be catastrophic. Mechanical feedback, on the other hand, would hold the surface actuator in place.

- The integral control law, as discussed in AFFDL-TR-71-20, Supplement 2, Section III, can be mechanized by utilizing an electronic high-pass network in the secondary actuator only if a mechanical feedback of surface position is provided.

- Using both electrical and mechanical feedback of surface position in Phase IIA and IIC will minimize changes in the electronic set during Phase II, since the existing moving body actuator which is retained in Phases IIA and B automatically provides mechanical surface position feedback.

Electrical feedback from the surface actuator can also be utilized on the secondary actuator - F-4 stabilator actuator combination and the SSAP in order to improve the response of the longitudinal control system. This electrical feedback will also tend to minimize the effect of nonlinearities within the actuators. The DC gain of the actuator loops will not be affected due to the fact that the electrical feedback from the surface actuator contains a high pass filter as shown in Figure 51.

The linear transfer function for the SSAP is very similar to the transfer function for the secondary actuator - F-4 stabilator actuator combination. Therefore, the general effect of the electrical feedback for either of these actuator systems can be demonstrated by referring to the SSAP actuator dynamic analysis in Paragraph 3.b.3. From the SSAP stability analysis, the root locus plots shown in Figures 53 through 54 show that the electrical feedback loop increases the SSAP bandpass from approximately 19 radians to 50 radians when operating on aircraft hydraulics. The benefit of the electrical feedback loop is further demonstrated in the SSAP response analysis. The large signal frequency response plots in Figures 65 and 66 show that at 3 Hz the addition of the electrical feedback loop decreases the phase lag by 18 degrees and increases the amplitude attenuation by 2.4 dB when operating on integral hydraulics. The direct effect of adding the electrical feedback is to increase the overall gain and phase margins for the longitudinal control system by reducing the phase contribution of the actuator loop.

4. SYSTEM STUDIES

The results of actuator monitoring and electrohydraulic versus electromechanical secondary actuator studies are presented below. Information on the actuator monitoring study is based primarily on Reference 7. This study provides the basis for the selection of an actuator monitoring concept which will be used to detect and isolate element failures in the quadruplex secondary actuators. The electrohydraulic versus electromechanical study results present the advantages and disadvantages of each concept relative to the SFCS program. This study was based on References 5, 6, and 7.

a. Actuator Monitoring

Actuator monitoring techniques have been investigated for both the electromechanical and electrohydraulic SFCS secondary actuators. Three candidate monitoring configurations were studied, each one compatible with SFCS redundancy requirements. The selected monitoring configuration consists of eight cross-element comparators packaged two in each computer and voter unit. Six of the eight comparators are selected by the aircraft wiring harness to perform the actual monitoring function. Thus the computer voter units are identical in all channels and therefore interchangeable. This particular configuration was chosen on the basis of self-test complexity, interface requirements and relative circuit complexity. Monitoring circuitry can be used in Phases IIA, IIB, and IIC of the SFCS program with minor modifications since the selected monitoring scheme is identical in format for either the electromechanical or electrohydraulic secondary actuator mechanizations.

(1) Secondary Actuator Configurations

(a) Electrohydraulic Secondary Actuator

A block diagram of the Electrohydraulic (E/H) secondary actuator is presented in Figure 72. As evidenced by this diagram, the E/H secondary actuator is a quadruplex, force summed unit which uses differential pressure sensors to derive monitoring intelligence. The relatively high position loop gain combined with a common feedback point transforms the quadruplex servo into a limited averaging system, i.e., the contribution of each servo element to the position average is limited by pressure saturation. Assuming no load, which closely approximates normal operating conditions, the differential pressure is a direct measure of servo element deviation from the average position output. During normal operation, tolerances must permit operation substantially below pressure saturation levels for differential pressure monitoring purposes. Stringent failure transient requirements, which produce proportionately high position loop gain, thus place heavy accuracy constraints on circuit and component designs.

The differential pressure sensor will detect any failure within the servo loop capable of producing a significant error signal at the valve. Error signals of this type result in a large pressure buildup in the failed element and a pressure change in the remaining elements sufficient to balance out the failed element. This discrepancy may be recognized and isolated by several differential pressure monitoring schemes as demonstrated later in this section.

Another failure mode, which is not readily detectable with a simple differential pressure monitor, may be postulated in the E/H secondary actuator. By nature, failure modes which result in zero pressure or zero indicated pressure present a detection problem. Failures which fall into this

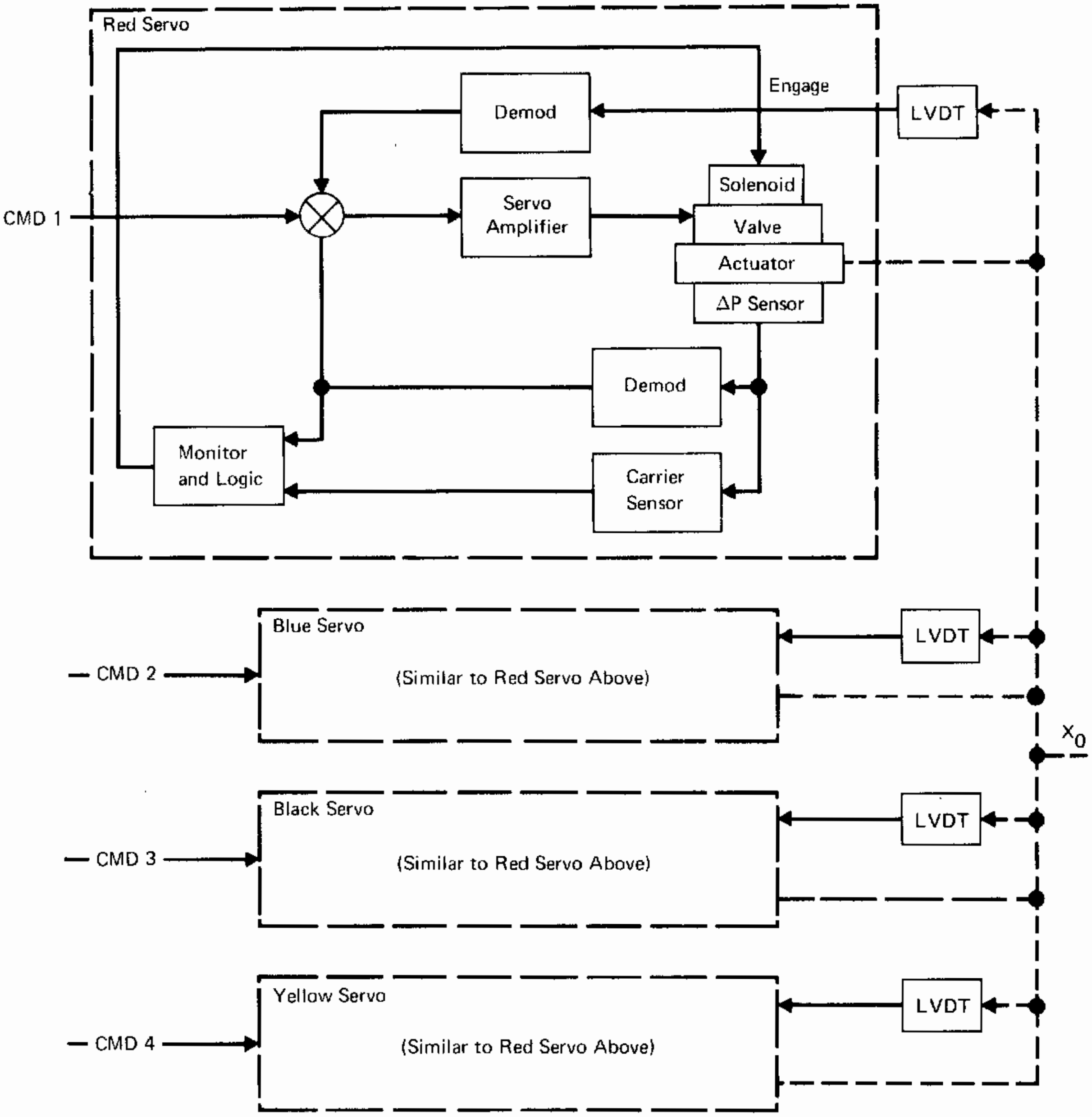


FIGURE 72
ELECTROHYDRAULIC SECONDARY ACTUATOR BLOCK DIAGRAM

category are zero servoamplifier failures, failed to center valves, failed to zero hydraulic supplies and failed to zero differential pressure. A preload in the differential pressure sensor detects hydraulic supply failures by displacing the differential pressure LVDT to a hardover condition when pressure is removed. Passive failure modes within the differential pressure sensors are detected by using a center tapped LVDT output winding. A carrier sensor sums the two LVDT winding voltages together directly (not differentially) and checks for a sum greater than a minimum level. Loss of LVDT excitation, open LVDT windings and shorted LVDT windings are thus detected. More elaborate schemes could be postulated to detect centered valves or failed to zero servoamplifiers; however, the complexity increase is not warranted based upon the low failure probability and the use of BIT to detect these types of failures prior to flight. Passive failures of this type cannot cause serious problems until two additional active failures have occurred, a situation which has a probability of occurrence comparable with the probability of system failure for the quadruplex system. This probability is negligible.

(b) Electromechanical Secondary Actuator

The electromechanical (E/M) secondary actuator, shown in Figure 73 is similar to the E/H unit from a monitoring standpoint even though the principle used in deriving a position output is entirely different.

Three differentials sum the four rotational outputs of the electromechanical actuator channels. A common position feedback point downstream of the differential allows limited averaging of the input commands, i.e., the servo output represents an average of the four input commands during normal operation. In an abnormal condition, however, each servo element is limited in its contribution to the averaged output by velocity saturation. Tachometers connected to each motor output provide a measure of servo performance much like pressure sensing in the E/H servo, but are not normally zero signals. Because of this trait, monitoring techniques most logically applied involve some form of comparison monitoring.

The basic fault detection capability of the tachometer is essentially the same as that previously discussed with respect to the differential pressure sensor. Although loss of tachometer output may be argued as detectable, "in-line" monitoring of the tachometer is used to prevent undesirable effects such as servo loop oscillation. A center tapped output winding and a special coil which couples excitation voltage to the output at zero rpm allow the use of the carrier sensor monitoring scheme.

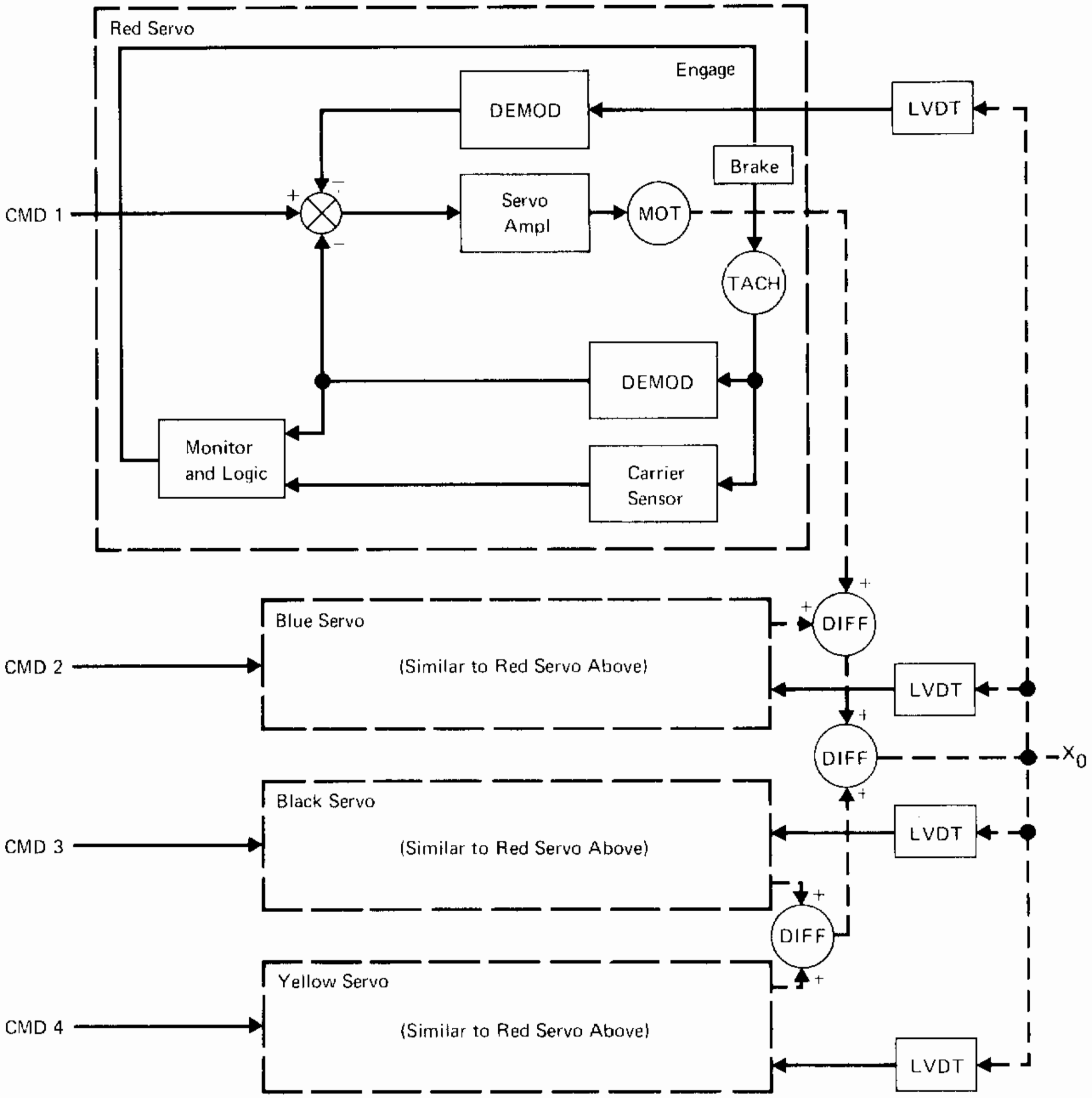


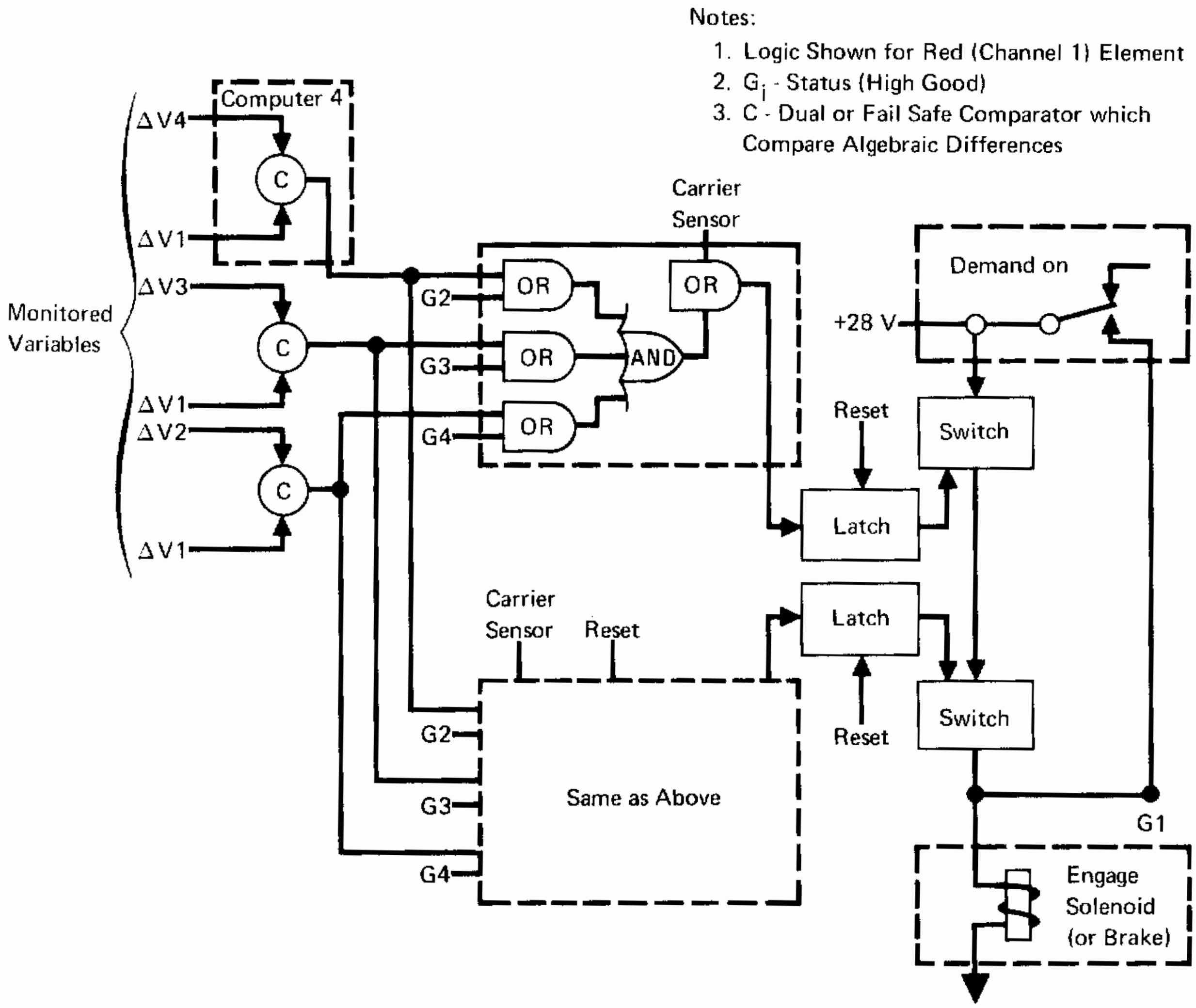
FIGURE 73
ELECTROMECHANICAL SECONDARY ACTUATOR BLOCK DIAGRAM

(2) Secondary Actuator Monitoring and Logic Study

Three candidate monitoring and logic schemes can be postulated. These are compatible with the basic E/M and E/H secondary actuator configurations. The cross-element monitoring candidate shown in Figure 74 has been chosen for the SFCES. It uses direct comparison to ascertain servo status. The monitoring variable in each element is compared to the corresponding variable in the remaining three servoelements through the use of a six comparator network. Since six comparators cannot be distributed among the four Computer and Voter Units (CVUs) equally, eight comparators are used to satisfy CVU interchangeability requirements. Six of these eight comparators are selected for use at CVU installation by a nonsymmetrical airplane wiring harness. The logic circuitry of Figure 74 defines a failure when all three of the comparator limits associated with a particular servo element are exceeded for a time duration determined by comparator delay time. Each of these comparator outputs is enabled by the corresponding status signal to eliminate the need for latching comparators, to provide cross-element intelligence of power supply status, and to prohibit single element operation in the event of three failures. Servo status is also dependent on a carrier sensor valid discrete and contains a latching element resettable from the master control and display panel. This latching element cannot be reset by power interruptions and, like the remainder of the logic, is dual to ensure fail-safe failure detection and correction performance.

A demand-on switch, located on the master control and display panel, has the capability of engaging the servo irrespective of engage logic conditions. The servo status signal is derived from the engage solenoid (or brake) drive and is transmitted to external LRU's via a format which defines opens or low impedance to ground as a failure and B+ as good. This particular monitoring and logic candidate is applicable to both secondary actuator types, permitting commonality of hardware in the two program phases.

Another form of "monitoring variable" comparison is achieved through the use of voting mechanisms as shown in Figure 75. In each element, a voter selects one of the monitored variables or its output in accordance with a selection rule (such as the least magnitude of the two middle values). By comparing a monitoring variable against the voted value, an index of servo performance is obtained. Minimum logic is required for this configuration as a result of direct association between comparator indication and servo element status. As in the previous case, servo element status signals are included in the logic implementation to prevent single element operation and the servo status is verified by a carrier sensor discrete. The output circuitry is identical to that previously described and the logic string is again fail-safe.

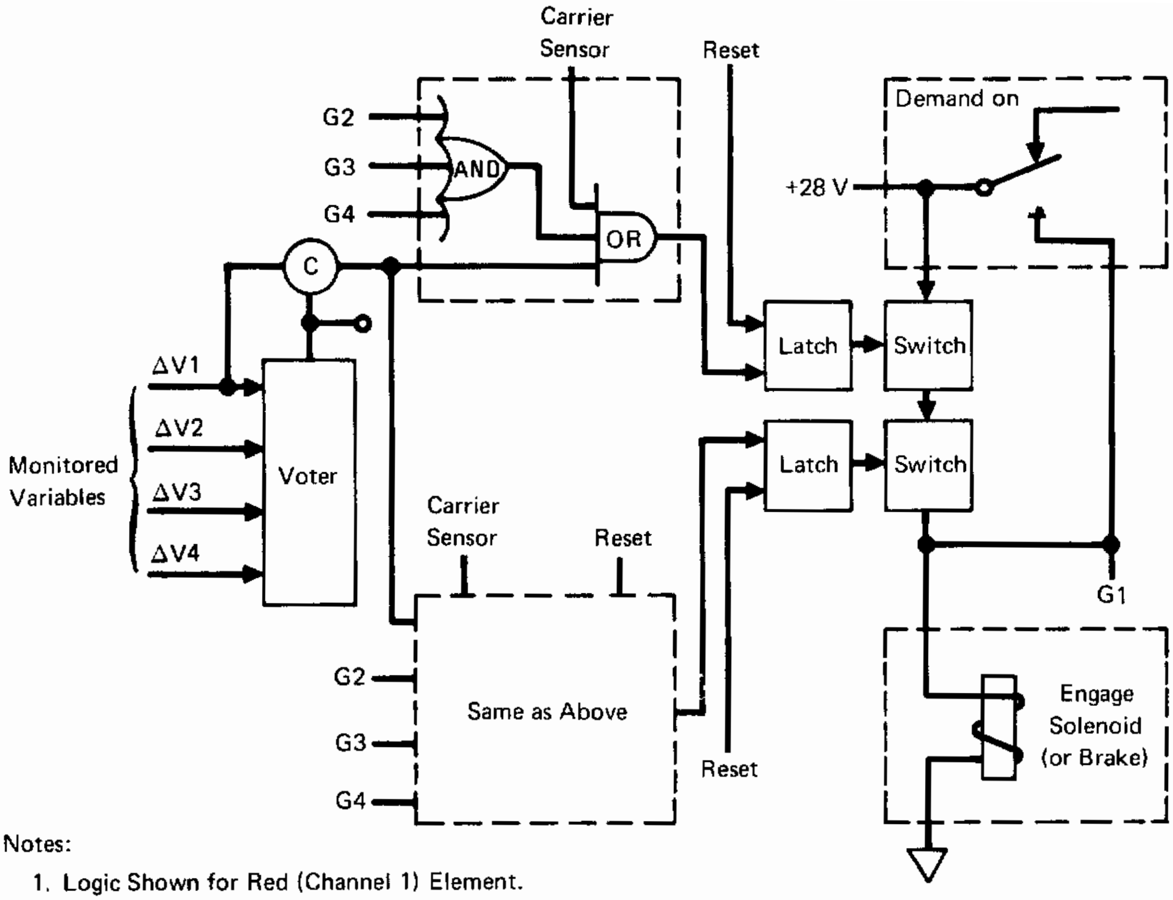


Notes:

1. Logic Shown for Red (Channel 1) Element
2. G_i - Status (High Good)
3. C - Dual or Fail Safe Comparator which Compare Algebraic Differences

FIGURE 74
CROSS ELEMENT SERVO MONITORING CANDIDATE

An in-line threshold logic monitoring candidate is illustrated in Figure 76. Ideally, in-line configurations minimize interelement data exchange and in doing so enhance element isolation. In its purest form, this particular candidate is more suitable for the E/H secondary actuator because the E/H monitoring variable (ΔP) is a nominally zero signal, the presence of which signifies an abnormal condition. The detection threshold must be set at a level greater than normal tolerance levels to eliminate nuisance disconnects. Additional logic is required, however, to allow actuator bottoming or chip shearing in the main control valve. This logic (Figure 76) permits excessive differential pressure if at least one other element is in agreement within a time period defined by the latch delay. Threshold discretes are qualified with servo status to eliminate crossfeed of a high differential pressure threshold discrete after failure. This is necessitated by the current E/H actuator design which outputs a



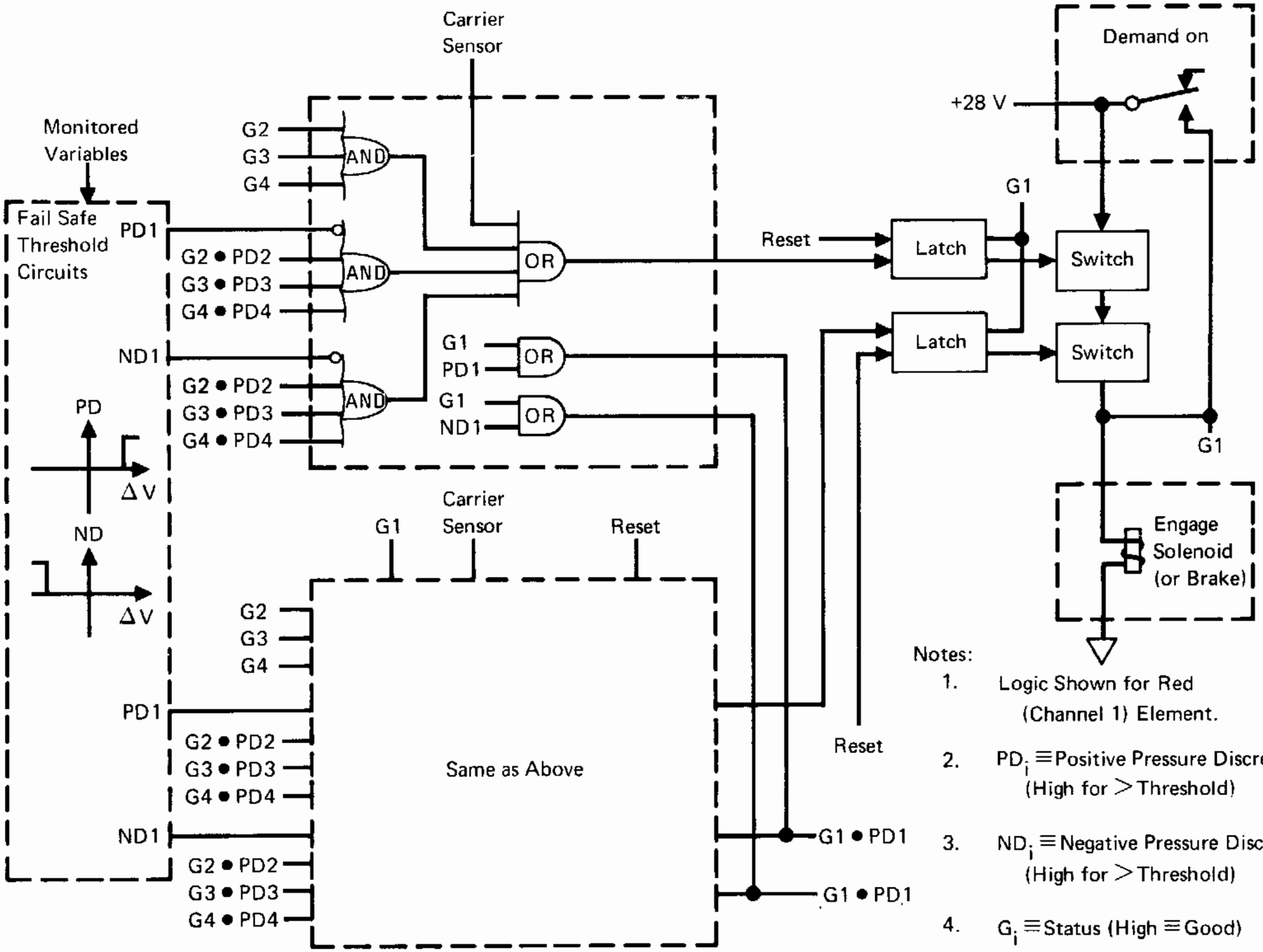
Notes:

1. Logic Shown for Red (Channel 1) Element.
2. G_i = Status (High = Good).
3. C = Dual or Fail Safe Comparator which Compares Algebraic Differences

FIGURE 75
CROSS VOTER SERVO MONITORING CANDIDATE

high discrete if pressure drops (solenoid is disengaged). The remainder of the logic is similar to that previously presented. At this point, it is evident that element isolation has not been maintained in the practical case although it may be argued that the interface is limited to discrete levels. The in-line monitoring concept described may also be applied to the E/M servo since the same logic which must permit actuator bottoming will accept a normally non-zero monitoring variable.

The three secondary actuator monitoring and logic candidates described above can be compared on the basis of BIT complexity, interface requirements, and relative circuit complexity. Table XXVII presents a summary of the results of this comparison. The cross-element comparison candidate was selected for use in the SFCS program on the basis of minimum test and circuit complexity.



Notes:

1. Logic Shown for Red (Channel 1) Element.
2. $PD_i \equiv$ Positive Pressure Discrete (High for $>$ Threshold)
3. $ND_i \equiv$ Negative Pressure Discrete (High for $>$ Threshold)
4. $G_i \equiv$ Status (High $\equiv$ Good)

FIGURE 76
THRESHOLD-LOGIC SERVO MONITORING CANDIDATE

TABLE XXVII
SERVO MONITORING CANDIDATE TRADEOFF SUMMARY

Tradeoff Parameter Candidate	Bit Complexity	Interface Requirements	Relative Complexity (Reliability)
Cross-Element Comparison	1	1.3	1.0
Cross-Voter Comparison	3	1	1.5
Threshold and Logic	3	1.7	2.0

b. Electrohydraulic Versus Electromechanical Actuators

The purpose of this study was to compare electrohydraulic and electromechanical secondary actuators. A secondary actuator is a device that converts four electrical command signals into a mechanical signal for positioning the primary actuator input. The force and stiffness requirements of this actuator are sufficiently low so that either an electrohydraulic or an electromechanical actuator can be used.

The actuators considered in this study were the electrohydraulic force summing actuator and the electromechanical velocity summing actuator. The electrohydraulic force summing system utilizes four individual servoactuators that are mechanically tied together to a single output ram. The cylinder pressures of each individual actuator are monitored and used for failure detection. The electromechanical velocity summing system utilizes four servomotors. The outputs of these motors are summed through three differentials, and a ballscrew is used to convert the output from rotary motion to linear motion. Each servomotor has its own brake which is used in the event of a failure. The tachometer on each of the servomotors is used in the feedback loop and for failure detection.

When comparing the two types of actuators there are two major areas of consideration; the advantages and disadvantages of each type of actuator from a general design standpoint and the effect of each type of actuator on the design of the electronic set.

The advantages and disadvantages of the two types of actuator are listed below:

Electrohydraulic, Force Summing

Advantages

- High power density
- Slew rate is independent of number of elements operating
- High attainable speed of response
- Hydraulic components proven in primary flight control systems
- Not subject to acceleration saturation

Disadvantages

- Requires four independent hydraulic supplies
- Requires special design consideration to compensate for hydraulic pressure variations
- A leakproof system is hard to obtain

Electromechanical, Velocity Summing

Advantages

- Provides quadruple redundancy without hydraulic supplies
- Eliminates some vulnerable hydraulic lines thus improving survivability
- Actuator is not effected by hydraulic pressure variations caused by the primary actuator
- Has constant output force

Disadvantages

- Motors, gearbox, and ballscrew require high temperature lubrication
- Slew rate is reduced with each failure
- High altitude aggravates motor cooling problem
- Requires complex electronic drive circuitry

The impact of an electromechanical versus electrohydraulic actuator configuration is significant in terms of circuit complexity and packaging. Figures 77 and 78 provide a relative comparison of circuitry involved for the two types of actuators; the electromechanical servomotor amplifier is more complex than the valve driver amplifier. The fundamental reason for the complexity contrast is the significantly greater power requirements imposed on the electromechanical servo amplifer.

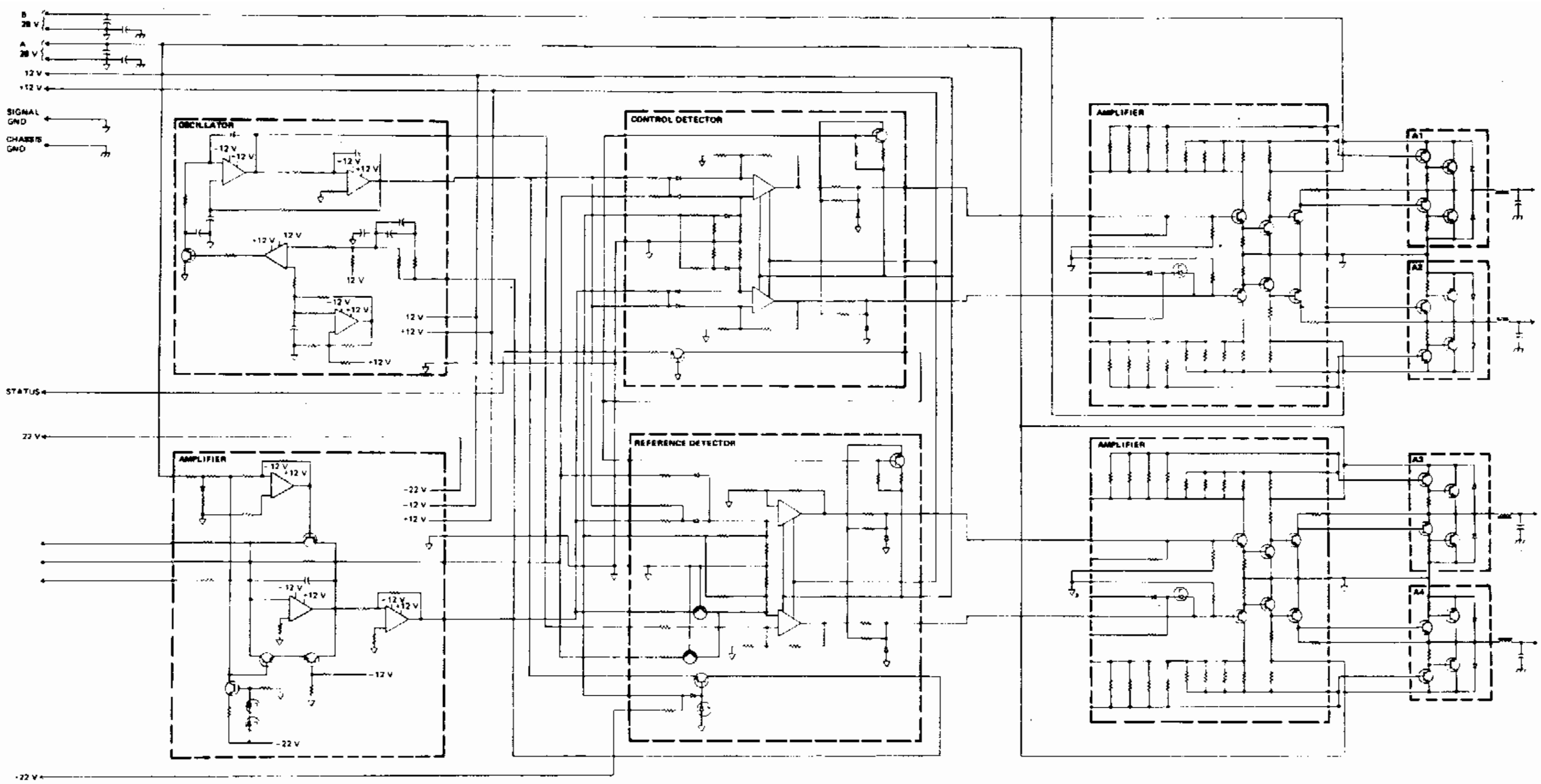


FIGURE 77
ELECTROMECHANICAL ACTUATOR CIRCUITRY

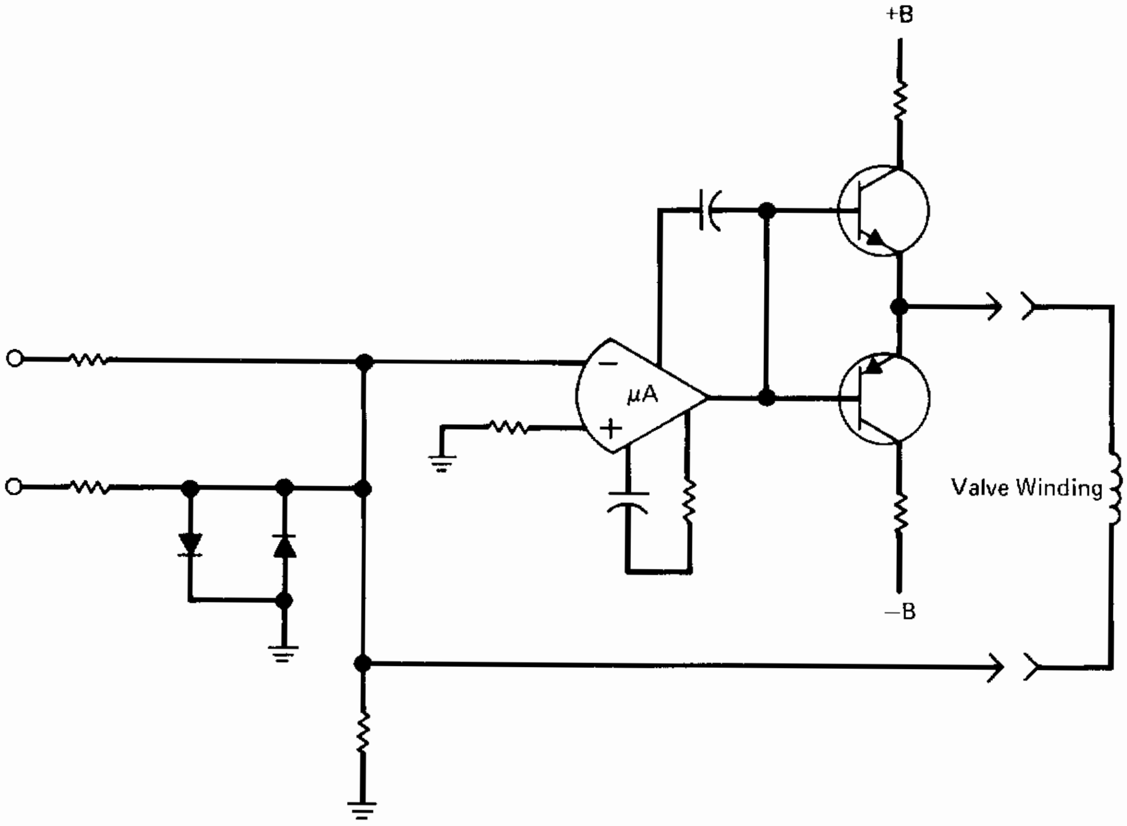


FIGURE 78
ELECTROHYDRAULIC ACTUATOR CIRCUITRY

Table XXVIII provides a quantitative summary of the significant electronic tradeoff factors. In all instances, the electromechanical interface represents a significant penalty. If all the servo loop electronics contained in each computer were scaled up by the relative weight, volume, and power figures of Table XXVIII, the volum of the computer and voter unit would increase by at least 100 cubic inches.

TABLE XXVIII
ACTUATOR ELECTRONICS TRADEOFF SUMMARY

Configuration	Relative Volume	Relative Weight	Relative Cost	Relative Power Dissipation at Full Load
Electromechanical Actuator Electronics	10.0	9.0	4.0	65.0
Electrohydraulic Actuator Electronics	1.0	1.0	1.0	1.0

Considering the comparison outlined above, both actuator concepts prove to be feasible for the secondary actuator application, and both types of actuators will be used in the SFCS program. The SFCES is being designed with the additional quadruplex electromechanical actuator interface electronics in the pitch axis only to accommodate the SSAP which will replace the hydraulic actuator in Phase IIC. Common electronics are used wherever possible with the main difference being the power stage required for the SSAP.

A more complete comparison between the electrohydraulic and the electromechanical actuators will be made at the end of the SFCS program. At that point the study will be able to draw on the extensive ground and flight test data that will be generated during the program.

5. CONCLUSIONS

Conclusions drawn on the basis of study results are necessarily contingent on the present status of actuator design and the current definition of parameters. While no major problems have been uncovered, it is evident that additional study and investigation will be required in specific areas. Specific conclusions are:

- Secondary actuator nuisance disconnect characteristics are acceptable. However, a review of the analysis will be necessary when tolerances in the associated electronics are more fully defined. The status of SSAP nuisance disconnect characteristics is comparable.
- A cross-element comparison monitoring technique has been selected for the SFCS in-flight monitoring.
- Both electrohydraulic and electromechanical quadruplex concepts are considered acceptable for the SFCS program and both will be used, the former on the secondary actuator and the latter in the SSAP.
- The secondary actuator, SSAP surface actuator, and SSAP secondary actuator will be stable.
- The electrohydraulic secondary actuator frequency response is adequate to permit proper SFCS operation.
- The SSAP frequency response results indicate that the phase lag will be in excess of required limits.
- Secondary actuator failure transients are acceptable. However, the effect of control linkage and surface actuator dynamic characteristics on failure transients should be investigated when the necessary information is available.
- SSAP failure transients may be excessive on a third failure unless maximum rpm differences between servo motors can be reduced.

APPENDIX I

SAMPLE SURVIVABILITY CALCULATIONS

1. GENERAL

The purpose of this appendix is to describe the ground rules and show the basis of the survivability calculations performed in support of the Degree of Redundancy Trade Study for the SSAP.

2. CONFIGURATION SCHEMATICS

The components considered for the various configurations are shown in Table XXIX. Included in the table are the abbreviation and code letter used to represent each component in the study. The configuration schematics are shown in Figures 79-83 for each of the nineteen configurations considered.

3. STUDY BASIS

For the SSAP trade study, survivability was evaluated from the standpoint of each configuration's ability to survive, without loss of longitudinal control, a specific number of projectile hits, e.g., one, two, and three hits. The probability of hit was not considered; therefore, survivability is conditioned upon the various configurations receiving the indicated specific number of hits. The study assumes that the components comprising the various configurations must absorb the hits. A hit on any component is assumed to destroy, or kill, that component. Multiple hits on a single component were considered, but multiple hits on a single component were considered not required to kill any component. A second or third hit on any component is simply overkill. The possibility of a hit destroying more than one component was not considered. Components downstream from a hit component are disabled in accordance with the redundancy paths shown in Figures 79-83. A break in any stabilator actuator chamber will cause the loss of all upstream hydraulic pressure sources associated with that chamber.

The component hit combinations causing loss of the longitudinal control system are shown in Tables XXX-XLVI. These are shown for one, two, and three hits for all the configurations considered. For all configurations, a single hit on the stabilator actuator attachment would cause aircraft kill. On the two hit matrix, the killing combinations are indicated by the letter "Y". The letter "Y" equals a "yes kill" and the letter "N" equals a "no kill". For three hits, the killing combinations are tabulated using the code letters for the components involved.

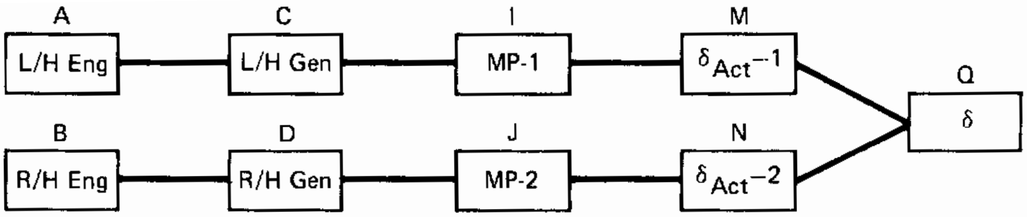
A hit survivability summary is shown in Table XLVII. This shows for each configuration the total number of possible hit combinations, both killing and non-killing, and the number of killing combinations causing loss of the longitudinal control system. These are shown for one, two, and three projectile hits. The total hit combinations are determined by standard combinatorial mathematics and are a function of the number of components in the configuration. The number of killing combinations are those evaluated and enumerated in Tables XXX-XLVI.

The evaluation of each configuration is made by determining the probability of kill given a hit ($P_{k/h}$) and its associated survivability ($1 - P_{k/h}$). The $P_{k/h}$ for each configuration is the ratio of killing combinations to total hit combinations. This has been done for each configuration, for one, two and three projectile hits, with the results shown in Table XLVII. As can be seen, survivability for one or two hits does not provide for sufficient discrimination between the majority of the configurations. However, with three hits, discrimination between configurations is now sufficient so that an ordering of configuration acceptability is possible.

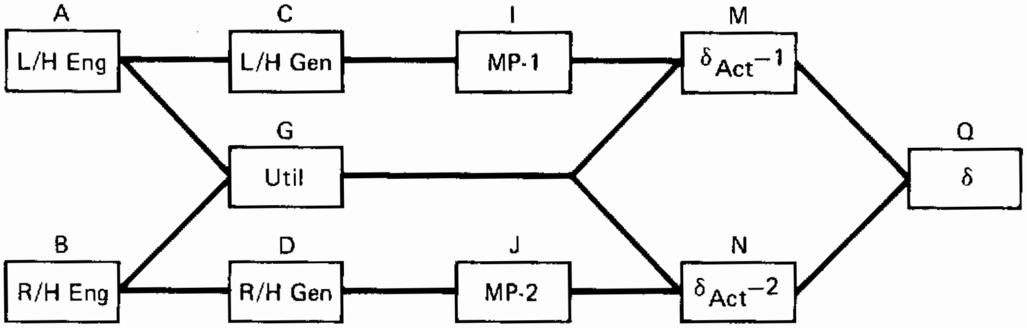
Shown in Table XLVIII is the ordered ranking of the nineteen configurations considered. The ranking is based on the configuration's ability to survive, given three hits on the system. The ratings assigned to the configurations are shown, and are the ones used in the Comparison Matrix of SSAP Design Approaches, Table II.

TABLE XXIX
COMPONENTS CONSIDERED FOR SURVIVABILITY STUDY

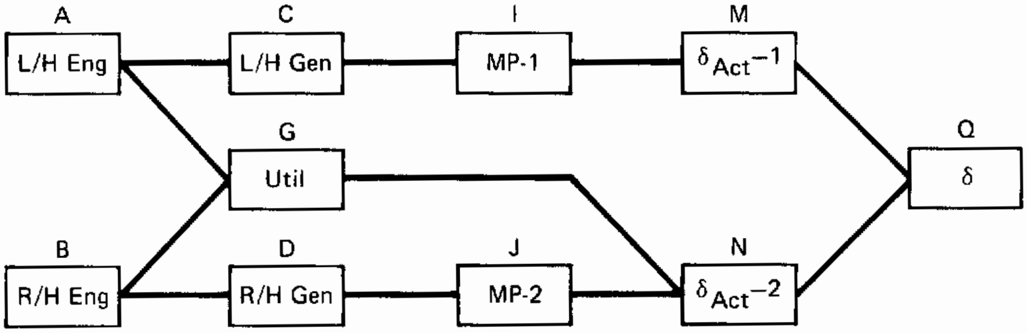
Component	Abbreviation	Code
Left Hand Engine	L/H ENG	A
Right Hand Engine	R/H ENG	B
Left Hand Generator	L/H GEN	C
Right Hand Generator	R/H GEN	D
PC-1 Hydraulics	PC-1	E
PC-2 Hydraulics	PC-2	F
Utility Hydraulics	Util	G
Hydraulic Driven Electric Generator	HDEG	H
Motor Pump No. 1	MP-1	I
Motor Pump No. 2	MP-2	J
Motor Pump No. 3	MP-3	K
Motor Pump No. 4	MP-4	L
Actuator Chamber No. 1	$\delta_{ACT}-1$	M
Actuator Chamber No. 2	$\delta_{ACT}-2$	N
Actuator Chamber No. 3	$\delta_{ACT}-3$	O
Actuator Chamber No. 4	$\delta_{ACT}-4$	P
Actuator Attachment	δ	Q



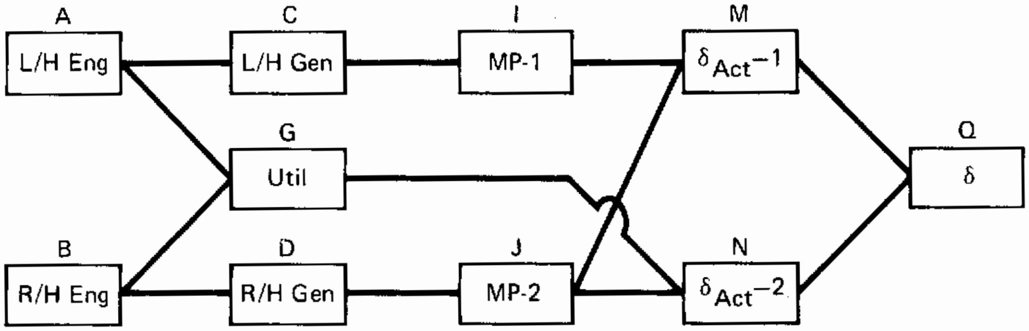
Configuration 1



Configuration 2A

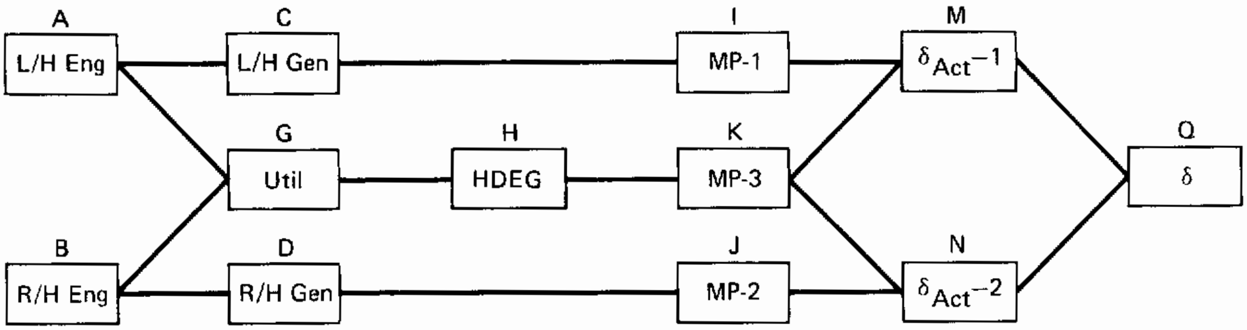


Configurations 2B and 2C

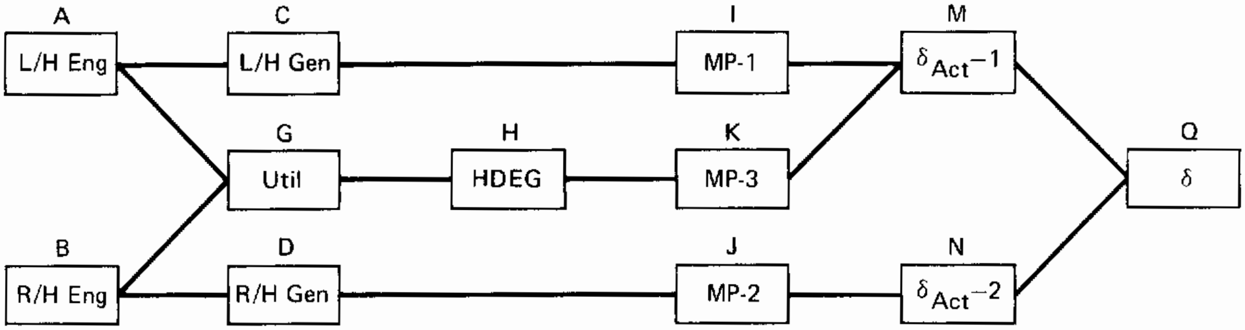


Configuration 2D

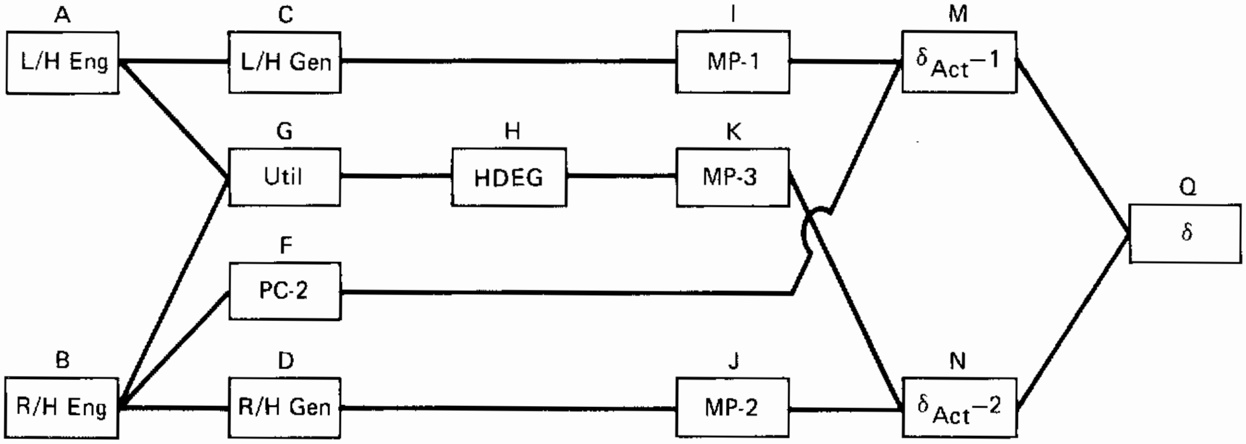
FIGURE 79
CONFIGURATION SCHEMATICS FOR SURVIVABILITY STUDY



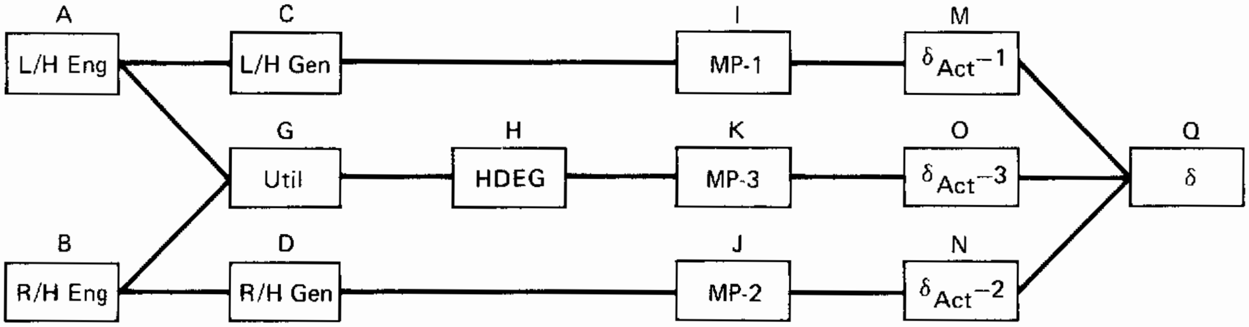
Configuration 3A



Configuration 3B

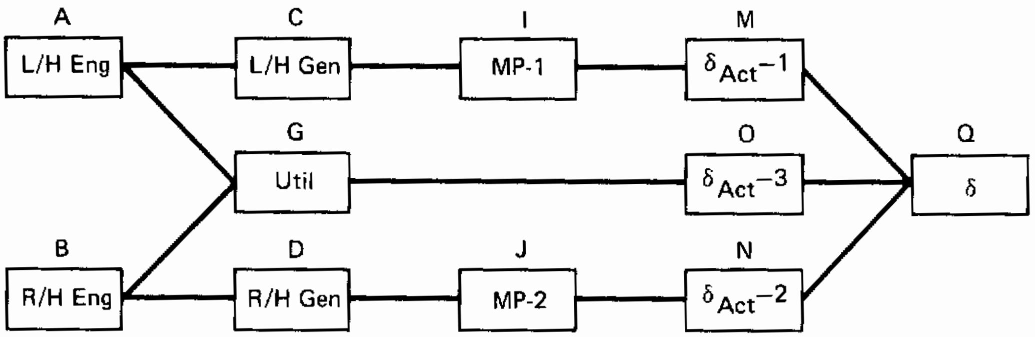


Configurations 4 and 11A

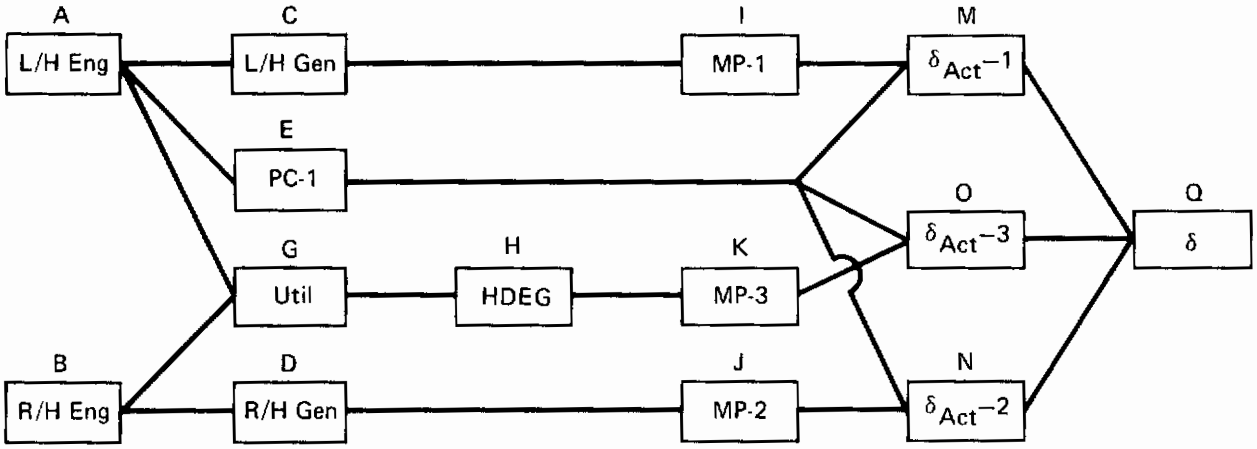


Configuration 5

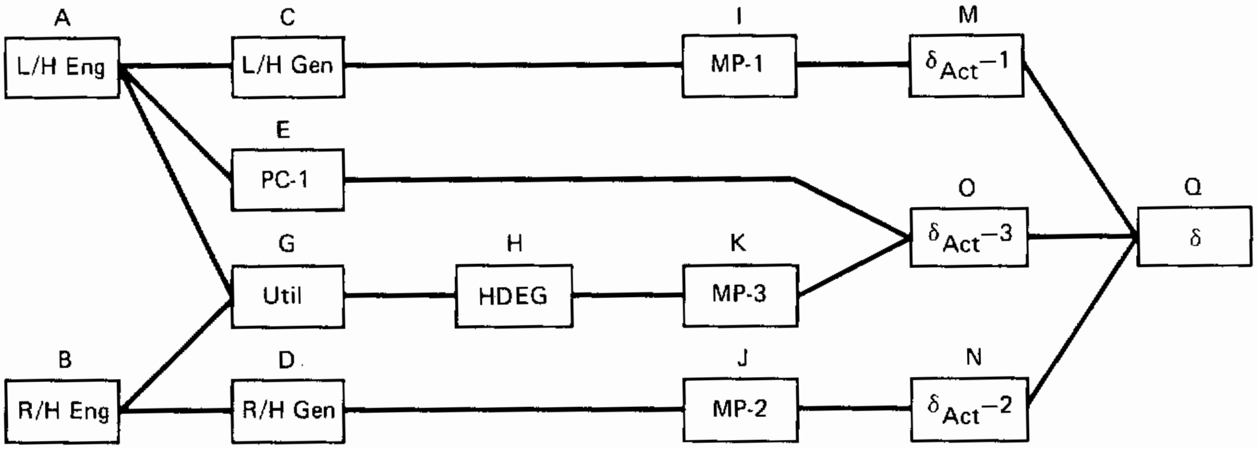
FIGURE 80
CONFIGURATION SCHEMATICS FOR SURVIVABILITY STUDY



Configuration 6

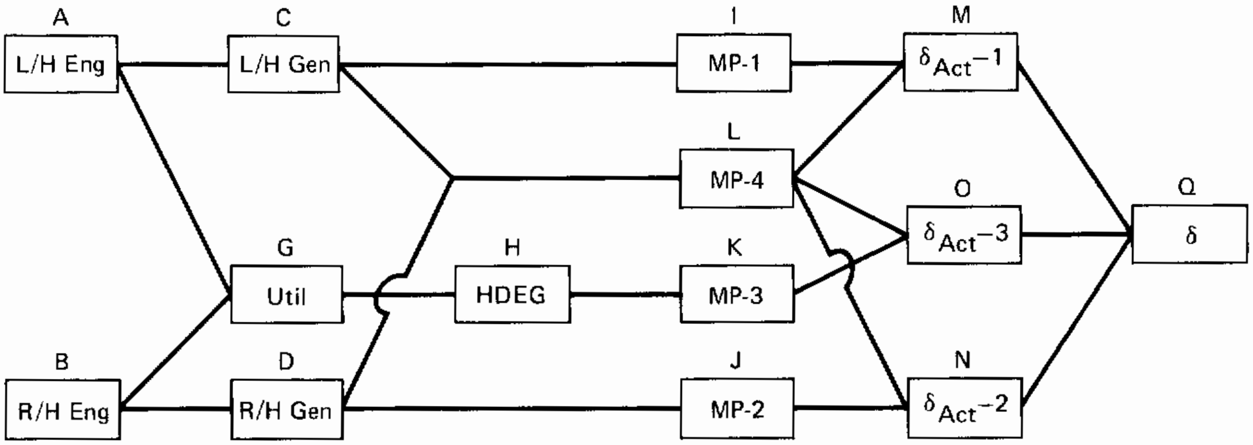


Configuration 7A

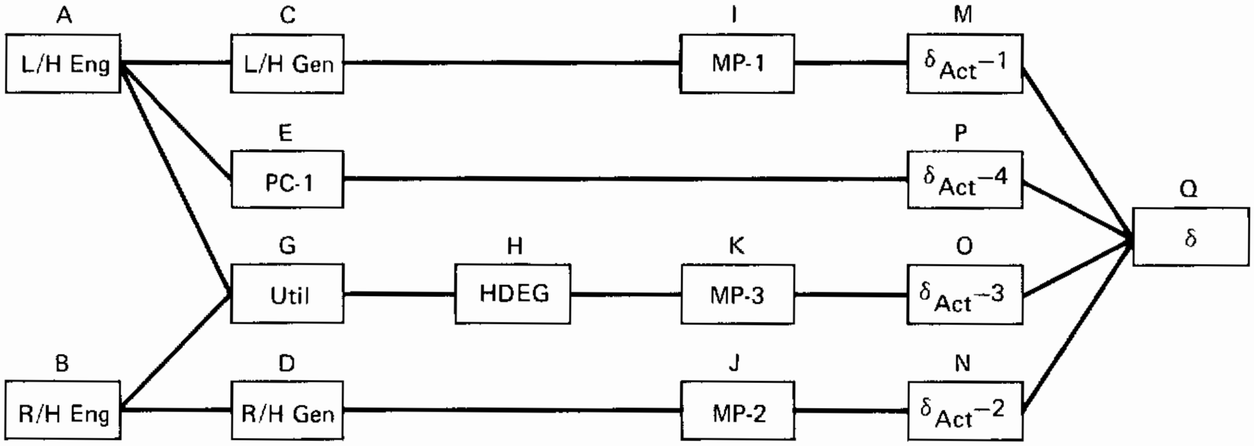


Configuration 7B

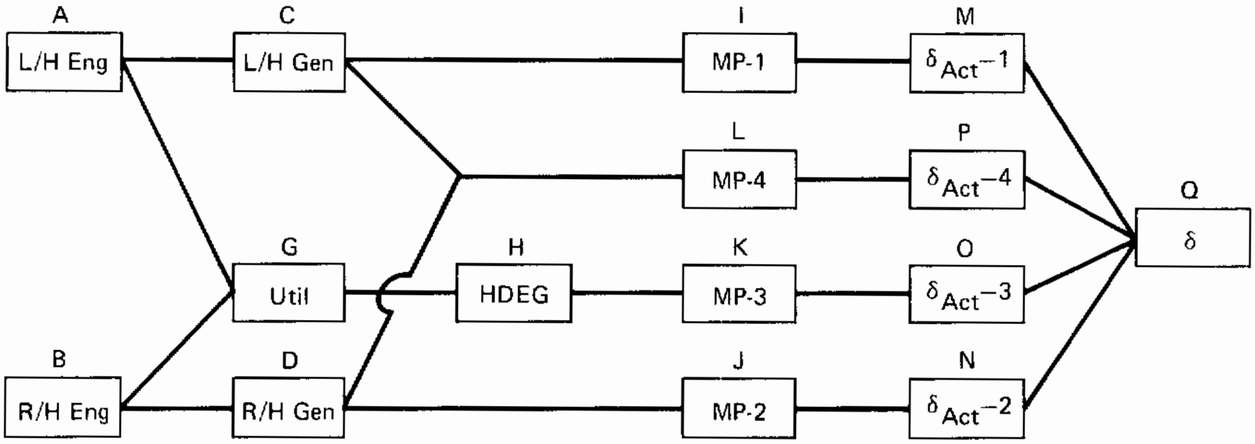
FIGURE 81
CONFIGURATION SCHEMATICS FOR SURVIVABILITY STUDY



Configuration 8

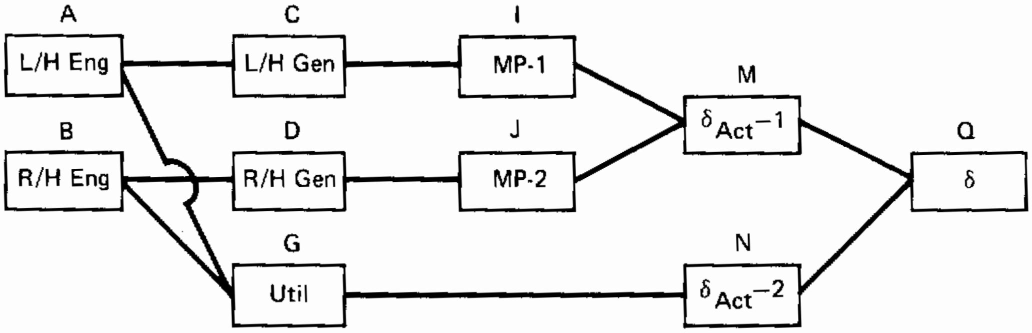


Configuration 9

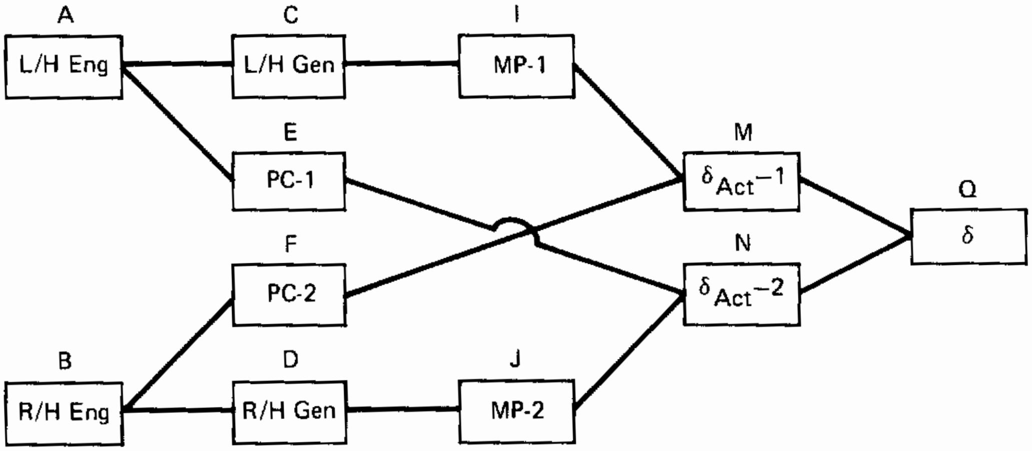


Configuration 10

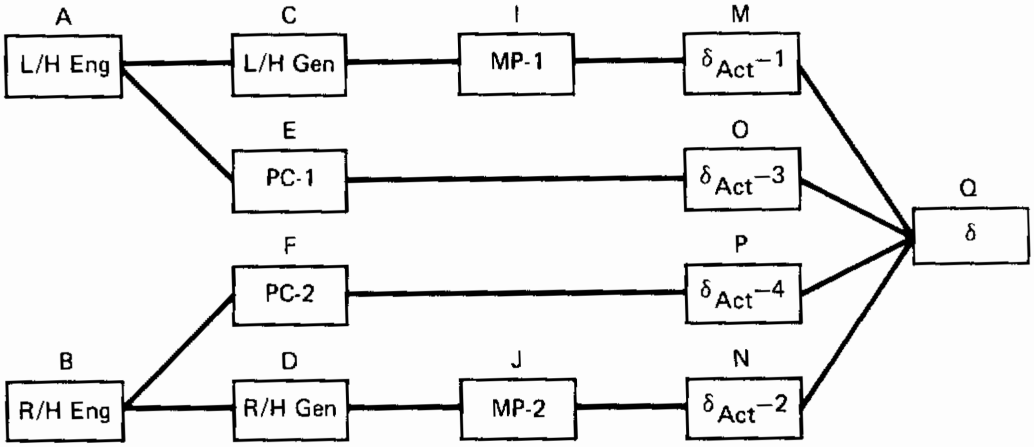
FIGURE 82
CONFIGURATION SCHEMATICS FOR SURVIVABILITY STUDY



Configuration 11B



Configuration 12



Configuration 13

FIGURE 83
CONFIGURATION SCHEMATICS FOR SURVIVABILITY STUDY

TABLE XXX

[illegible] ING HIT COMBINATIONS [illegible]

[illegible]

[illegible]	[illegible]									
[illegible]	[illegible]	N								
[illegible]	[illegible] NO	N	Y							
[illegible]	[illegible] N	N	Y	N						
[illegible]	[illegible] GEN	Y	N	Y	N					
[illegible]	MF 1	N	[illegible]	N	[illegible]	N				
[illegible]	MF 2	Y	N	Y	[illegible]	[illegible]	[illegible]			
M	PART 1	N	Y	N	[illegible]	N	[illegible]	[illegible]		
N	[illegible] 2	N	N	Y	N	Y	[illegible]	[illegible]		
[illegible]	[illegible]	Y	Y	Y	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
[illegible]	ADN	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
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[illegible]	CAO	[illegible]	[illegible]	[illegible]	[illegible]		
[illegible]	DHM	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
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[illegible]	[illegible]	[illegible]	[illegible]	[illegible]			

TABLE XXXI
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 2A

1 HIT	
1 KILLING COMBINATION	Q Only

2 HITS

18 KILLING COMBINATIONS

	CODE	A	B	C	D	G	I	J	M	N	Q
		L/H ENG	R/H ENG	L/H GEN	R/H GEN	UTIL	MP-1	MP-2	δACT−1	δACT−2	δ
A	L/H ENG	N									
B	R/H ENG	Y	N								
C	L/H GEN	N	N	N							
D	R/H GEN	N	N	N	N						
G	UTIL	N	N	N	N	N					
I	MP−1	N	N	N	N	N	N				
J	MP−2	N	N	N	N	N	N	N			
M	δACT−1	N	Y	N	Y	N	N	Y	N		
N	δACT−2	Y	N	Y	N	N	Y	N	Y	N	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

121 KILLING COMBINATIONS

AAB	AAN	AAQ	ABB	ABC	ABD	ABG	ABI	ABJ	ABM
ABN	ABQ	ACN	ACQ	ADG	ADM	ADN	ADQ	AGJ	AGN
AGQ	AIN	AIQ	AJM	AJN	AJQ	AMN	AMQ	ANN	ANQ
AQQ	BBM	BBQ	BCG	BCM	BCN	BCQ	BDM	BDQ	BGI
BGM	BGQ	BIM	BIN	BIQ	BJM	BJQ	BMM	BMN	BMQ
BNQ	BQQ	CCN	CCQ	CDG	CDM	CDN	CDQ	CGJ	CGN
CGQ	CIN	CIQ	CJM	CJN	CJQ	CMN	CMQ	CNN	CNQ
CQQ	DDM	DDQ	DGI	DGM	DGQ	DIM	DIN	DIQ	DJM
DJQ	DMM	DMN	DMQ	DNQ	DQQ	GGQ	GIJ	GIN	GIQ
GJM	GJQ	GMN	GMQ	GNQ	GQQ	IIN	IIQ	IJM	IJN
IJQ	IMN	IMQ	INN	INQ	IQQ	JJM	JJQ	JMM	JMN
JMQ	JNQ	JQQ	MMN	MMQ	MNN	MNQ	MQQ	NNQ	NQQ
QQQ									

TABLE XXXII
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configurations 2B and 2C

1 HIT 1 KILLING COMBINATION	Q Only

2 HITS

15 KILLING COMBINATIONS

	CODE	A L/H ENG	B R/H ENG	C L/H GEN	D R/H GEN	G UTIL	I MP-1	J MP-2	M δ_{ACT-1}	N δ_{ACT-2}	Q δ
A	L/H ENG	N									
B	R/H ENG	Y	N								
C	L/H GEN	N	N	N							
D	R/H GEN	N	N	N	N						
G	UTIL	N	N	N	N	N					
I	MP-1	N	N	N	N	N	N				
J	MP-2	N	N	N	N	N	N	N			
M	δ_{ACT-1}	N	N	N	N	N	N	N	N		
N	δ_{ACT-2}	Y	N	Y	N	N	Y	N	Y	N	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

103 KILLING COMBINATIONS

AAB	AAN	AAQ	ABB	ABC	ABD	ABG	ABI	ABJ	ABM
ABN	ABQ	ACN	ACQ	ADG	ADN	ADQ	AGJ	AGN	AGQ
AIN	AIQ	AJN	AJQ	AMN	AMQ	ANN	ANQ	AQQ	BBQ
BCG	BCN	BCQ	BDQ	BGI	BGM	BGQ	BIN	BIQ	BJQ
BMN	BMQ	BNQ	BQQ	CCN	CCQ	CDG	CDN	CDQ	CGJ
CGN	CGQ	CIN	CIQ	CJN	CJQ	CMN	CMQ	CNN	CNQ
CQQ	DDQ	DGI	DGM	DGQ	DIN	DIQ	DJQ	DMN	DMQ
DNQ	DQQ	GGQ	GIN	GIQ	GJM	GJQ	GMN	GMQ	GNQ
GQQ	IIN	IIQ	IJN	IJQ	IMN	IMQ	INN	INQ	IQQ
JJQ	JMN	JMQ	JNQ	JQQ	MMN	MMQ	MNN	MNQ	MQQ
NNQ	NQQ	QQQ							

TABLE XXXIII
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 2D

<table>
<tr><td>1 HIT
1 KILLING COMBINATION</td><td>Q Only</td></tr>
<tr><td>2HITS
16 KILLING COMBINATIONS</td><td>
<table>
<tr><th colspan="2">CODE</th><th>A</th><th>B</th><th>C</th><th>D</th><th>G</th><th>I</th><th>J</th><th>M</th><th>N</th><th>Q</th></tr>
<tr><th></th><th></th><th>L/H ENG</th><th>R/H ENG</th><th>L/H GEN</th><th>R/H GEN</th><th>UTIL</th><th>MP-1</th><th>MP-2</th><th>$\delta_{ACT}-1$</th><th>$\delta_{ACT}-2$</th><th>δ</th></tr>
<tr><td>A</td><td>L/H ENG</td><td>N</td><td></td><td></td><td></td><td></td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>B</td><td>R/H ENG</td><td>Y</td><td>N</td><td></td><td></td><td></td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>C</td><td>L/H GEN</td><td>N</td><td>N</td><td>N</td><td></td><td></td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>D</td><td>R/H GEN</td><td>N</td><td>N</td><td>N</td><td>N</td><td></td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>G</td><td>UTIL</td><td>N</td><td>N</td><td>N</td><td>N</td><td>N</td><td></td><td></td><td></td><td></td><td></td></tr>
<tr><td>I</td><td>MP-1</td><td>N</td><td>N</td><td>N</td><td>N</td><td>N</td><td>N</td><td></td><td></td><td></td><td></td></tr>
<tr><td>J</td><td>MP-2</td><td>N</td><td>N</td><td>N</td><td>N</td><td>N</td><td>N</td><td>N</td><td></td><td></td><td></td></tr>
<tr><td>M</td><td>$\delta_{ACT}-1$</td><td>N</td><td>N</td><td>N</td><td>N</td><td>Y</td><td>N</td><td>N</td><td>N</td><td></td><td></td></tr>
<tr><td>N</td><td>$\delta_{ACT}-2$</td><td>Y</td><td>N</td><td>Y</td><td>N</td><td>N</td><td>Y</td><td>N</td><td>Y</td><td>N</td><td></td></tr>
<tr><td>Q</td><td>δ</td><td>Y</td><td>Y</td><td>Y</td><td>Y</td><td>Y</td><td>Y</td><td>Y</td><td>Y</td><td>Y</td><td>Y.</td></tr>
</table>
</td></tr>
<tr><td>3 HITS
109 KILLING COMBINATIONS</td><td>
<table>
<tr><td>AAB</td><td>AAN</td><td>AAQ</td><td>ABB</td><td>ABC</td><td>ABD</td><td>ABG</td><td>ABI</td><td>ABJ</td><td>ABM</td></tr>
<tr><td>ABN</td><td>ABQ</td><td>ACN</td><td>ACQ</td><td>ADG</td><td>ADN</td><td>ADQ</td><td>AGJ</td><td>AGM</td><td>AGN</td></tr>
<tr><td>AGQ</td><td>AIN</td><td>AIQ</td><td>AJN</td><td>AJQ</td><td>AMN</td><td>AMQ</td><td>ANN</td><td>ANQ</td><td>AQQ</td></tr>
<tr><td>BBQ</td><td>BCG</td><td>BCN</td><td>BCQ</td><td>BDQ</td><td>BGI</td><td>BGM</td><td>BGQ</td><td>BIN</td><td>BIQ</td></tr>
<tr><td>BJQ</td><td>BMN</td><td>BMQ</td><td>BNQ</td><td>BQQ</td><td>CCN</td><td>CCQ</td><td>CDG</td><td>CDN</td><td>CDQ</td></tr>
<tr><td>CGJ</td><td>CGM</td><td>CGN</td><td>CGQ</td><td>CIN</td><td>CIQ</td><td>CJN</td><td>CJQ</td><td>CMN</td><td>CMQ</td></tr>
<tr><td>CNN</td><td>CNQ</td><td>COQ</td><td>DDQ</td><td>DGI</td><td>DGM</td><td>DGQ</td><td>DIN</td><td>DIQ</td><td>DJQ</td></tr>
<tr><td>DMN</td><td>DMQ</td><td>DNQ</td><td>DQQ</td><td>GGM</td><td>GGQ</td><td>GIJ</td><td>GIM</td><td>GIN</td><td>GIQ</td></tr>
<tr><td>GJM</td><td>GJQ</td><td>GMM</td><td>GMN</td><td>GMQ</td><td>GNQ</td><td>GQQ</td><td>IIN</td><td>IIQ</td><td>IJN</td></tr>
<tr><td>IJQ</td><td>IMN</td><td>IMQ</td><td>INN</td><td>INQ</td><td>IQQ</td><td>JJQ</td><td>JMN</td><td>JMQ</td><td>JNQ</td></tr>
<tr><td>JQQ</td><td>MMN</td><td>MMQ</td><td>MNN</td><td>MNQ</td><td>MQQ</td><td>NNQ</td><td>NQQ</td><td>QQQ</td><td></td></tr>
</table>
</td></tr>
</table>

TABLE XXXIV
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 3A

1 HIT 1 KILLING COMBINATION	Q Only

2 HITS

20 KILLING COMBINATIONS

	CODE	A L/H ENG	B R/H ENG	C L/H GEN	D R/H GEN	G UTIL	H HDEG	I MP-1	J MP-2	K MP-3	M δ_{ACT}^{-1}	N δ_{ACT}^{-2}	Q δ
A	L/H ENG	N											
B	R/H ENG	Y	N										
C	L/H GEN	N	N	N									
D	R/H GEN	N	N	N	N								
G	UTIL	N	N	N	N	N							
H	HDEG	N	N	N	N	N	N						
I	MP-1	N	N	N	N	N	N	N					
J	MP-2	N	N	N	N	N	N	N	N				
K	MP-3	N	N	N	N	N	N	N	N	N			
M	δ_{ACT}^{-1}	N	Y	N	Y	N	N	N	Y	N	N		
N	δ_{ACT}^{-2}	Y	N	Y	N	N	N	Y	N	N	Y	N	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

176 KILLING COMBINATIONS

AAB	AAN	AAQ	ABB	ABC	ABD	ABG	ABH	ABI	ABJ
ABK	ABM	ABN	ABQ	ACN	ACQ	ADG	ADH	ADK	ADM
ADN	ADQ	AGJ	AGN	AGQ	AHJ	AHN	AHQ	AIN	AIQ
AJK	AJM	AJN	AJQ	AKN	AKQ	AMN	AMQ	ANN	ANQ
AQQ	BBM	BBQ	BCG	BCH	BCK	BCM	BCN	BCQ	BDM
BDQ	BGI	BGM	BGQ	BHI	BHM	BHQ	BIK	BIM	BIN
BIQ	BJM	BJQ	BKM	BKQ	BMM	BMN	BMQ	BNQ	BQQ
CCN	CCQ	CDG	CDH	CDK	CDM	CDN	CDQ	CGJ	CGN
CGQ	CHJ	CHN	CHQ	CIN	CIQ	CJK	CJM	CJN	CJQ
CKN	CKQ	CMN	CMQ	CNN	CNQ	CQQ	DDM	DDQ	DGI
DGM	DGQ	DHI	DHM	DHQ	DIK	DIM	DIN	DIQ	DJM
DJQ	DKM	DKQ	DMM	DMN	DMQ	DNQ	DQQ	GGQ	GHQ
GIJ	GIN	GIQ	GJM	GJQ	GKQ	GMN	GMQ	GNQ	GQQ
HHQ	HIJ	HIN	HIQ	HJM	HJQ	HKQ	HMN	HMQ	HNQ
HQQ	IIN	IIQ	IJK	IJM	IJN	IJQ	IKN	IKQ	IMN
IMQ	INN	INQ	IQQ	JJM	JJQ	JKM	JKQ	JMM	JMN
JMQ	JNQ	JQQ	KKQ	KMN	KMQ	KNQ	KQQ	MMN	MMQ
MNN	MNQ	MQQ	NNQ	NQQ	QQQ				

TABLE XXXV
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 3B

1 HIT 1 KILLING COMBINATION	Q Only

2 HITS

17 KILLING COMBINATIONS

	CODE	A	B	C	D	G	H	I	J	K	M	N	Q
		L/H ENG	R/H ENG	L/H GEN	R/H GEN	UTIL	HDEG	MP-1	MP-2	MP-3	δ_{ACT}^{-1}	δ_{ACT}^{-2}	δ
A	L/H ENG	N											
B	R/H ENG	Y	N										
C	L/H GEN	N	N	N									
D	R/H GEN	N	N	N	N								
G	UTIL	N	N	N	N	N							
H	HDEG	N	N	N	N	N	N						
I	MP-1	N	N	N	N	N	N	N					
J	MP-2	N	N	N	N	N	N	N	N				
K	MP-3	N	N	N	N	N	N	N	N	N			
M	δ_{ACT}^{-1}	N	Y	N	Y	N	N	N	Y	N	N		
N	δ_{ACT}^{-2}	N	N	N	N	N	N	N	N	N	Y	N	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

159 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABG	ABH	ABI	ABJ	ABK
ABM	ABN	ABQ	ACQ	ADG	ADH	ADK	ADM	ADQ	AGJ
AGN	AGQ	AHJ	AHN	AHQ	AIQ	AJK	AJM	AJQ	AKN
AKQ	AMN	AMQ	ANQ	AQQ	BBM	BBQ	BCG	BCH	BCK
BCM	BCQ	BDM	BDQ	BGI	BGM	BGQ	BHI	BHM	BHQ
BIK	BIM	BIQ	BJM	BJQ	BKM	BKQ	BMM	BMN	BMQ
BNQ	BQQ	CCQ	CDG	CDH	CDK	CDM	CDQ	CGJ	CGN
CGQ	CHJ	CHN	CHQ	CIQ	CJK	CJM	CJQ	CKN	CKQ
CMN	CMQ	CNQ	CQQ	DDM	DDQ	DGI	DGM	DGQ	DHI
DHM	DHQ	DIK	DIM	DIQ	DJM	DJQ	DKM	DKQ	DMM
DMN	DMQ	DNQ	DQQ	GGQ	GHQ	GIJ	GIN	GIQ	GJM
GJQ	GKQ	GMN	GMQ	GNQ	GQQ	HHQ	HIJ	HIN	HIQ
HJM	HJQ	HKQ	HMN	HMQ	HNQ	HQQ	IIQ	IJK	IJM
IJQ	IKN	IKQ	IMN	IMQ	INQ	IQQ	JJM	JJQ	JKM
JKQ	JMM	JMN	JMQ	JNQ	JQQ	KKQ	KMN	KMQ	KNQ
KQQ	MMN	MMQ	MNN	MNQ	MQQ	NNQ	NQQ	QQQ	

TABLE XXXVI
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configurations 4 and 11A

1 HIT 1 KILLING COMBINATION	Q Only

2 HITS 15 KILLING COMBINATIONS	CODE	A L/H ENG	B R/H ENG	C L/H GEN	D R/H GEN	F PC-2	G UTIL	H HDEG	I MP-1	J MP-2	K MP-3	M $\delta_{ACT}-1$	N $\delta_{ACT}-2$	Q δ
A	L/H ENG	~~N~~												
B	R/H ENG	Y	~~N~~											
C	L/H GEN	N	N	~~N~~										
D	R/H GEN	N	N	N	~~N~~									
F	PC-2	N	N	N	N	~~N~~								
G	UTIL	N	N	N	N	N	~~N~~							
H	HDEG	N	N	N	N	N	N	~~N~~						
I	MP-1	N	N	N	N	N	N	N	~~N~~					
J	MP-2	N	N	N	N	N	N	N	N	~~N~~				
K	MP-3	N	N	N	N	N	N	N	N	N	~~N~~			
M	$\delta_{ACT}-1$	N	N	N	N	N	N	N	N	N	N	~~N~~		
N	$\delta_{ACT}-2$	N	N	N	N	N	N	N	N	N	N	Y	~~N~~	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

135 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABF	ABG	ABH	ABI	ABJ
ABK	ABM	ABN	ABQ	ACQ	ADQ	AFN	AFQ	AGQ	AHQ
AIQ	AJQ	AKQ	AMN	AMQ	ANQ	AQQ	BBQ	BCG	BCH
BCK	BCN	BCQ	BDQ	BFQ	BGI	BGM	BGQ	BHI	BHM
BHQ	BIK	BIN	BIQ	BJQ	BKM	BKQ	BMN	BMQ	BNQ
BQQ	CCQ	CDQ	CFN	CFQ	CGQ	CHQ	CIQ	CJQ	CKQ
CMN	CMQ	CNQ	CQQ	DDQ	DFQ	DGM	DGQ	DHM	DHQ
DIQ	DJQ	DKM	DKQ	DMN	DMQ	DNQ	DQQ	FFQ	FGQ
FHQ	FIN	FIQ	FJQ	FKQ	FMN	FMQ	FNQ	FQQ	GGQ
GHQ	GIQ	GJM	GJQ	GKQ	GMN	GMQ	GNQ	GQQ	HHQ
HIQ	HJM	HJQ	HKQ	HMN	HMQ	HNQ	HQQ	IIQ	IJQ
IKQ	IMN	IMQ	INQ	IQQ	JJQ	JKM	JKQ	JMN	JMQ
JNQ	JQQ	KKQ	KMN	KMQ	KNQ	KQQ	MMN	MMQ	MNN
MNQ	MQQ	NNQ	NQQ	QQQ					

TABLE XXXVII
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 5

1 HIT 1 KILLING COMBINATION	Q Only

2 HITS

14 KILLING COMBINATIONS

	CODE	A L/H ENG	B R/H ENG	C L/H GEN	D R/H GEN	G UTIL	H HDEG	I MP-1	J MP-2	K MP-3	M δACT−1	N δACT−2	O δACT−3	Q δ
A	L/H ENG	N												
B	R/H ENG	Y	N											
C	L/H GEN	N	N	N										
D	R/H GEN	N	N	N	N									
G	UTIL	N	N	N	N	N								
H	HDEG	N	N	N	N	N	N							
I	MP-1	N	N	N	N	N	N	N						
J	MP-2	N	N	N	N	N	N	N	N					
K	MP-3	N	N	N	N	N	N	N	N	N				
M	δACT^{-1}	N	N	N	N	N	N	N	N	N	N			
N	δACT^{-2}	N	N	N	N	N	N	N	N	N	N	N		
O	δACT^{-3}	N	N	N	N	N	N	N	N	N	N	N	N	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

162 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABG	ABH	ABI	ABJ	ABK
ABM	ABN	ABO	ABQ	ACQ	ADG	ADH	ADK	ADO	ADQ
AGJ	AGN	AGQ	AHJ	AHN	AHQ	AIQ	AJK	AJO	AJQ
AKN	AKQ	AMQ	ANO	ANQ	AOQ	AQQ	BBQ	BCG	BCH
BCK	BCO	BCQ	BDQ	BGI	BGM	BGQ	BHI	BHM	BHQ
BIK	BIO	BIQ	BJQ	BKM	BKQ	BMO	BMQ	BNQ	BOQ
BQQ	CCQ	CDG	CDH	CDK	CDO	CDQ	CGJ	CGN	CGQ
CHJ	CHN	CHQ	CIQ	CJK	CJO	CJQ	CKN	CKQ	CMQ
CNO	CNQ	COQ	CQQ	DDQ	DGI	DGM	DGQ	DHI	DHM
DHQ	DIK	DIO	DIQ	DJQ	DKM	DKQ	DMO	DMQ	DNQ
DOQ	DQQ	GGQ	GHQ	GIJ	GIN	GIQ	GJM	GJQ	GKQ
GMN	GMQ	GNQ	GOQ	GQQ	HHQ	HIJ	HIN	HIQ	HJM
HJQ	HKQ	HMN	HMQ	HNQ	HOQ	HQQ	IIQ	IJK	IJO
IJQ	IKN	IKQ	IMQ	INO	INQ	IOQ	IQQ	JJQ	JKM
JKQ	JMO	JMQ	JNQ	JOQ	JQQ	KKQ	KMN	KMQ	KNQ
KQQ	MMQ	MNO	MNQ	MOQ	MQQ	NNQ	NOQ	NQQ	OOQ
OQQ	QQQ								

TABLE ~~XXXVIII~~
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 6

1 HIT	
1 KILLING COMBINATION	Q Only

2 HITS

12 KILLING COMBINATIONS

	CODE	A L/H ENG	B R/H ENG	C L/H GEN	D R/H GEN	G UTIL	I MP-1	J MP-2	M δACT^{-1}	N δACT^{-2}	O δACT^{-3}	Q δ
A	L/H ENG	N										
B	R/H ENG	Y	N									
C	L/H GEN	N	N	N								
D	R/H GEN	N	N	N	N							
G	UTIL	N	N	N	N	N						
I	MP-1	N	N	N	N	N	N					
J	MP-2	N	N	N	N	N	N	N				
M	δACT^{-1}	N	N	N	N	N	N	N	N			
N	δACT^{-2}	N	N	N	N	N	N	N	N	N		
O	δACT^{-3}	N	N	N	N	N	N	N	N	N	N	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

116 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABG	ABI	ABJ	ABM	ABN
ABO	ABQ	ACQ	ADG	ADO	ADQ	AGJ	AGN	AGQ	AIQ
AJO	AJQ	AMQ	ANO	ANQ	AOQ	AQQ	BBQ	BCG	BCO
BCQ	BDQ	BGI	BGM	BGQ	BIO	BIQ	BJQ	BMO	BMQ
BNQ	BOQ	BQQ	CCQ	CDG	CDO	CDQ	CGJ	CGN	CGQ
CIQ	CJO	CJQ	CMQ	CNO	CNQ	COQ	CQQ	DDQ	DGI
DGM	DGQ	DIO	DIQ	DJQ	DMO	DMQ	DNQ	DOQ	DQQ
GGQ	GIJ	GIN	GIQ	GJM	GJQ	GMN	GMQ	GNQ	GOQ
GQQ	IIQ	IJO	IJQ	IMQ	INO	INQ	IOQ	IQQ	JJQ
JMO	JMQ	JNQ	JOQ	JQQ	MMQ	MNO	MNQ	MOQ	MQQ
NNQ	NOQ	NQQ	OOQ	OQQ	QQQ				

TABLE XXXIX
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 7A

1 HIT	
1 KILLING COMBINATION	Q Only

2 HITS

15 KILLING COMBINATIONS

	CODE	A	B	C	D	E	G	H	I	J	K	M	N	O	Q
		L/H ENG	R/H ENG	L/H GEN	R/H GEN	PC-1	UTIL	HDEG	MP-1	MP-2	MP-3	δACT−1	δACT−2	δACT−3	δ
A	L/H ENG	~~N~~													
B	R/H ENG	Y	~~N~~												
C	L/H GEN	N	N	~~N~~											
D	R/H GEN	N	N	N	~~N~~										
E	PC-1	N	N	N	N	~~N~~									
G	UTIL	N	N	N	N	N	~~N~~								
H	HDEG	N	N	N	N	N	N	~~N~~							
I	MP-1	N	N	N	N	N	N	N	~~N~~						
J	MP-2	N	N	N	N	N	N	N	N	~~N~~					
K	MP-3	N	N	N	N	N	N	N	N	N	~~N~~				
M	δACT^{-1}	N	N	N	N	N	N	N	N	N	N	~~N~~			
N	δACT−2	N	N	N	N	N	N	N	N	N	N	N	~~N~~		
O	δACT^{-3}	N	N	N	N	N	N	N	N	N	N	N	N	~~N~~	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

160 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABE	ABG	ABH	ABI	ABJ
ABK	ABM	ABN	ABO	ABQ	ACQ	ADG	ADH	ADK	ADO
ADQ	AEQ	AGJ	AGN	AGQ	AHJ	AHN	AHQ	AIQ	AJK
AJO	AJQ	AKN	AKQ	AMQ	ANO	ANQ	AOQ	AQQ	BBQ
BCO	BCQ	BDQ	BEQ	BGM	BGQ	BHM	BHQ	BIO	BIQ
BJQ	BKM	BKQ	BMO	BMQ	BNQ	BOQ	BQQ	CCQ	CDO
CDQ	CEQ	CGN	CGQ	CHN	CHQ	CIQ	CJO	CJQ	CKN
CKQ	CMQ	CNO	CNQ	COQ	CQQ	DDQ	DEQ	DGM	DGQ
DHM	DHQ	DIO	DIQ	DJQ	DKM	DKQ	DMO	DMQ	DNQ
DOQ	DQQ	EEQ	EGQ	EHQ	EIQ	EJQ	EKQ	EMQ	ENQ
EOQ	EQQ	GGQ	GHQ	GIN	GIQ	GJM	GJQ	GKQ	GMN
GMQ	GNQ	GOQ	GQQ	HHQ	HIN	HIQ	HJM	HJQ	HKQ
HMN	HMQ	HNQ	HOQ	HQQ	IIQ	IJO	IJQ	IKN	IKQ
IMQ	INO	INQ	IOQ	IQQ	JJQ	JKM	JKQ	JMO	JMQ
JNQ	JOQ	JQQ	KKQ	KMN	KMQ	KNQ	KOQ	KQQ	MMQ
MNO	MNQ	MOQ	MQQ	NNQ	NOQ	NQQ	OOQ	OQQ	QQQ

TABLE XL
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 7B

1 HIT 1 KILLING COMBINATION	Q Only

2 HITS 15 KILLING COMBINATIONS	CODE	A	B	C	D	E	G	H	I	J	K	M	N	O	Q
		L/H ENG	R/H ENG	L/H GEN	R/H GEN	PC-1	UTIL	HDEG	MP-1	MP-2	MP-3	$\delta\mathrm{ACT}^{-1}$	$\delta\mathrm{ACT}^{-2}$	$\delta\mathrm{ACT}^{-3}$	δ
A	L/H ENG	N													
B	R/H ENG	Y	N												
C	L/H GEN	N	N	N											
D	R/H GEN	N	N	N	N										
E	PC-1	N	N	N	N	N									
G	UTIL	N	N	N	N	N	N								
H	HDEG	N	N	N	N	N	N	N							
I	MP-1	N	N	N	N	N	N	N	N						
J	MP-2	N	N	N	N	N	N	N	N	N					
K	MP-3	N	N	N	N	N	N	N	N	N	N				
M	$\delta\mathrm{ACT}^{-1}$	N	N	N	N	N	N	N	N	N	N	N			
N	$\delta\mathrm{ACT}^{-2}$	N	N	N	N	N	N	N	N	N	N	N	N		
O	$\delta\mathrm{ACT}^{-3}$	N	N	N	N	N	N	N	N	N	N	N	N	N	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

142 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABE	ABG	ABH	ABI	ABJ
ABK	ABM	ABN	ABO	ABQ	ACQ	ADG	ADH	ADK	ADO
ADQ	AEQ	AGJ	AGN	AGQ	AHJ	AHN	AHQ	AIQ	AJK
AJO	AJQ	AKN	AKQ	AMQ	ANO	ANQ	AOQ	AQQ	BBQ
BCO	BCQ	BDQ	BEQ	BGQ	BHQ	BIO	BIQ	BJQ	BKQ
BMO	BMQ	BNQ	BOQ	BQQ	CCQ	CDO	CDQ	CEQ	CGQ
CHQ	CIQ	CJO	CJQ	CKQ	CMQ	CNO	CNQ	COQ	CQQ
DDQ	DEQ	DGQ	DHQ	DIO	DIQ	DJQ	DKQ	DMO	DMQ
DNQ	DOQ	DQQ	EEQ	EGQ	EHQ	EIQ	EJQ	EKQ	EMQ
ENQ	EOQ	EQQ	GGQ	GHQ	GIQ	GJQ	GKQ	GMQ	GNQ
GOQ	GQQ	HHQ	HIQ	HJQ	HKQ	HMQ	HNQ	HOQ	HQQ
IIQ	IJO	IJQ	IKQ	IMQ	INO	INQ	IOQ	IQQ	JJQ
JKQ	JMO	JMQ	JNQ	JOQ	JQQ	KKQ	KMQ	KNQ	KOQ
KQQ	MMQ	MNO	MNQ	MOQ	MQQ	NNQ	NOQ	NQQ	OOQ
OQQ	QQQ								

TABLE XLI
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 8

1 HIT 1 KILLING COMBINATION	Q Only

2 HITS

15 KILLING COMBINATIONS

	CODE	A	B	C	D	G	H	I	J	K	L	M	N	O	Q
		L/H ENG	R/H ENG	L/H GEN	R/H GEN	UTIL	HDEG	MP-1	MP-2	MP-3	MP-4	δ_{ACT}^{-1}	δ_{ACT}^{-2}	δ_{ACT}^{-3}	δ
A	L/H ENG	~~N~~													
B	R/H ENG	Y	~~N~~												
C	L/H GEN	N	N	~~N~~											
D	R/H GEN	N	N	N	~~N~~										
G	UTIL	N	N	N	N	~~N~~									
H	HDEG	N	N	N	N	N	~~N~~								
I	MP-1	N	N	N	N	N	N	~~N~~							
J	MP-2	N	N	N	N	N	N	N	~~N~~						
K	MP-3	N	N	N	N	N	N	N	N	~~N~~					
L	MP-4	N	N	N	N	N	N	N	N	N	~~N~~				
M	δ_{ACT}^{-1}	N	N	N	N	N	N	N	N	N	N	~~N~~			
N	δ_{ACT}^{-2}	N	N	N	N	N	N	N	N	N	N	N	~~N~~		
O	δ_{ACT}^{-3}	N	N	N	N	N	N	N	N	N	N	N	N	~~N~~	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

163 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABG	ABH	ABI	ABJ	ABK
ABL	ABM	ABN	ABO	ABQ	ACQ	ADG	ADH	ADK	ADO
ADQ	AGN	AGQ	AHN	AHQ	AIQ	AJO	AJQ	AKN	AKQ
ALQ	AMQ	ANO	ANQ	AOQ	AQQ	BBQ	BCG	BCH	BCK
BCO	BCQ	BDQ	BGM	BGQ	BHM	BHQ	BIO	BIQ	BJQ
BKM	BKQ	BLQ	BMO	BMQ	BNQ	BOQ	BQQ	CCQ	CDG
CDH	CDK	CDO	CDQ	CGN	CGQ	CHN	CHQ	CIQ	CJO
CJQ	CKN	CKQ	CLQ	CMQ	CNO	CNQ	COQ	CQQ	DDQ
DGM	DGQ	DHM	DHQ	DIO	DIQ	DJQ	DKM	DKQ	DLQ
DMO	DMQ	DNQ	DOQ	DQQ	GGQ	GHQ	GIN	GIQ	GJM
GJQ	GKQ	GLQ	GMN	GMQ	GNQ	GOQ	GQQ	HHQ	HIN
HIQ	HJM	HJQ	HKQ	HLQ	HMN	HMQ	HNQ	HOQ	HQQ
IIQ	IJO	IJQ	IKN	IKQ	ILQ	IMQ	INO	INQ	IOQ
IQQ	JJQ	JKM	JKQ	JLQ	JMO	JMQ	JNQ	JOQ	JQQ
KKQ	KLQ	KMN	KMQ	KNQ	KOQ	KQQ	LLQ	LMQ	LNQ
LOQ	LQQ	MMQ	MNO	MNQ	MOQ	MQQ	NNQ	NOQ	NQQ
OOQ	OQQ	QQQ							

TABLE XLII
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 9

1 HIT	
1 KILLING COMBINATION	Q Only

2 HITS

16 KILLING COMBINATIONS

	CODE	A	B	C	D	E	G	H	I	J	K	M	N	O	P	Q
		L/H ENG	R/H ENG	L/H GEN	R/H ENG	PC-1	UTIL	HDEG	MP-1	MP-2	MP-3	δ_{ACT}^{-1}	δ_{ACT}^{-2}	δ_{ACT}^{-3}	δ_{ACT}^{-4}	δ
A	L/H ENG	~~N~~														
B	R/H ENG	Y	~~N~~													
C	L/H GEN	N	N	~~N~~												
D	R/H GEN	N	N	N	~~N~~											
E	PC-1	N	N	N	N	~~N~~										
G	UTIL	N	N	N	N	N	~~N~~									
H	HDEG	N	N	N	N	N	N	~~N~~								
I	MP-1	N	N	N	N	N	N	N	~~N~~							
J	MP-2	N	N	N	N	N	N	N	N	~~N~~						
K	MP-3	N	N	N	N	N	N	N	N	N	~~N~~					
M	δ_{ACT}^{-1}	N	N	N	N	N	N	N	N	N	N	~~N~~				
N	δ_{ACT}^{-2}	N	N	N	N	N	N	N	N	N	N	N	~~N~~			
O	δ_{ACT}^{-3}	N	N	N	N	N	N	N	N	N	N	N	N	~~N~~		
P	δ_{ACT}^{-4}	N	N	N	N	N	N	N	N	N	N	N	N	N	~~N~~	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

146 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABE	ABG	ABH	ABI	ABJ
ABK	ABM	ABN	ABO	ABP	ABQ	ACQ	ADQ	ADH	ADK
ADO	ADQ	AEQ	AGJ	AGN	AGQ	AHJ	AHN	AHQ	AIQ
AJK	AJO	AJQ	AKN	AKQ	AMQ	ANO	ANQ	AOQ	APQ
AQQ	BBQ	BCQ	BDQ	BEQ	BGQ	BHQ	BIQ	BJQ	BKQ
BMQ	BNQ	BOQ	BPQ	BQQ	CCQ	CDQ	CEQ	CGQ	CHQ
CIQ	CJQ	CKQ	CMQ	CNQ	COQ	CPQ	CQQ	DDQ	DEQ
DGQ	DHQ	DIQ	DJQ	DKQ	DMQ	DNQ	DOQ	DPQ	DQQ
EEQ	EGQ	EHQ	EIQ	EJQ	EKQ	EMQ	ENQ	EOQ	EPQ
EQQ	GGQ	GHQ	GIQ	GJQ	GKQ	GMQ	GNQ	GOQ	GPQ
GQQ	HHQ	HIQ	HJQ	HKQ	HMQ	HNQ	HOQ	HPQ	HQQ
IIQ	IJQ	IKQ	IMQ	INQ	IOQ	IPQ	IQQ	JJQ	JKQ
JMQ	JNQ	JOQ	JPQ	JQQ	KKQ	KMQ	KNQ	KOQ	KPQ
KQQ	MMQ	MNQ	MOQ	MPQ	MQQ	NNQ	NOQ	NPQ	NQQ
OOQ	OPQ	OQQ	PPQ	PQQ	QQQ				

TABLE XLIII
KILLING KIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 10

1 HIT 1 KILLING COMBINATION	Q Only

2 HITS

16 KILLING COMBINATIONS

CODE		A	B	C	D	G	H	I	J	K	L	M	N	O	P	Q
		L/H ENG	R/H ENG	L/H GEN	R/H GEN	UTIL	HDEG	MP-1	MP-2	MP-3	MP-4	$\delta ACT-1$	$\delta ACT-2$	$\delta ACT-3$	$\delta ACT-4$	δ
A	L/H ENG	N														
B	R/H ENG	Y	N													
C	L/H GEN	N	N	N												
D	R/H GEN	N	N	N	N											
G	UTIL	N	N	N	N	N										
H	HDEG	N	N	N	N	N	N									
I	MP-1	N	N	N	N	N	N	N								
J	MP-2	N	N	N	N	N	N	N	N							
K	MP-3	N	N	N	N	N	N	N	N	N						
L	MP-4	N	N	N	N	N	N	N	N	N	N					
M	$\delta ACT-1$	N	N	N	N	N	N	N	N	N	N	N				
N	$\delta ACT-2$	N	N	N	N	N	N	N	N	N	N	N	N			
O	$\delta ACT-3$	N	N	N	N	N	N	N	N	N	N	N	N	N		
P	$\delta ACT-4$	N	N	N	N	N	N	N	N	N	N	N	N	N	N	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

146 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABG	ABH	ABI	ABJ	ABK
ABL	ABM	ABN	ABO	ABP	ABQ	ACQ	ADG	ADH	ADK
ADO	ADQ	AGQ	AHQ	AIQ	AJQ	AKQ	ALQ	AMQ	ANQ
AOQ	APQ	AQQ	BBO	BCG	BCH	BCK	BCO	BCQ	BDQ
BGQ	BHQ	BIQ	BJQ	BKQ	BLQ	BMQ	BNQ	BOQ	BPQ
BQQ	CCQ	CDG	CDH	CDK	CDO	CDQ	CGQ	CHQ	CIQ
CJQ	CKQ	CLQ	CMQ	CNQ	COQ	CPQ	CQQ	DDQ	DGQ
DHQ	DIQ	DJQ	DKQ	DLQ	DMQ	DNQ	DOQ	DPQ	DQQ
GGQ	GHQ	GIQ	GJQ	GKQ	GLQ	GMQ	GNQ	GOQ	GPQ
GQQ	HHQ	HIQ	HJQ	HKQ	HLQ	HMQ	HNQ	HOQ	HPQ
HQQ	IIQ	IJQ	IKQ	ILQ	IMQ	INQ	IOQ	IPQ	IQQ
JJQ	JKQ	JLQ	JMQ	JNQ	JOQ	JPQ	JQQ	KKQ	KLQ
KMQ	KNQ	KOQ	KPQ	KQQ	LLQ	LMQ	LNQ	LOQ	LPQ
LQQ	MMQ	MNQ	MOQ	MPQ	MQQ	NNQ	NOQ	NPQ	NQQ
OOQ	OPQ	OQQ	PPQ	PQQ	QQQ				

TABLE XLIV
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 11B

1 HIT	
1 KILLING COMBINATION	Q Only

2 HITS			A	B	C	D	G	I	J	M	N	Q
		CODE	L/H ENG	R/H ENG	L/H GEN	R/H GEN	UTIL	MP-1	MP-2	δ_{ACT}^{-1}	δ_{ACT}^{-2}	δ
13 KILLING COMBINATIONS	A	L/H ENG	~~N~~									
	B	R/H ENG	Y	~~N~~								
	C	L/H GEN	N	N	~~N~~							
	D	R/H GEN	N	N	N	~~N~~						
	G	UTIL	N	N	N	N	~~N~~					
	I	MP-1	N	N	N	N	N	~~N~~				
	J	MP-2	N	N	N	N	N	N	~~N~~			
	M	δ_{ACT}^{-1}	N	N	N	N	Y	N	N	~~N~~		
	N	δ_{ACT}^{-2}	N	N	N	N	N	N	N	Y	~~N~~	
	Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS										
95 KILLING COMBINATIONS	AAB	AAQ	ABB	ABC	ABD	ABG	ABI	ABJ	ABM	ABN
	ABQ	ACQ	ADG	ADN	ADQ	AGJ	AGM	AGQ	AIQ	AJN
	AJQ	AMN	AMQ	ANQ	AQQ	BBQ	BCG	BCN	BCQ	BDQ
	BGI	BGM	BGQ	BIN	BIQ	BJQ	BMN	BMQ	BNQ	BQQ
	CCQ	CDG	CDN	CDQ	CGJ	CGM	CGQ	CIQ	CJN	CJQ
	CMN	CMQ	CNQ	CQQ	DDQ	DGI	DGM	DGQ	DIN	DIQ
	DJQ	DMN	DMQ	DNQ	DQQ	GGQ	GIJ	GIM	GIQ	GJM
	GJQ	GMN	GMQ	GNQ	GQQ	IIQ	IJN	IJQ	IMN	IMQ
	INQ	IQQ	JJQ	JMN	JMQ	JNQ	JQQ	MMN	MMQ	MNN
	MNQ	MQQ	NNQ	NQQ	QQQ					

TABLE XLV
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 12

1 HIT	
1 KILLING COMBINATION	Q Only

2 HITS

13 KILLING COMBINATIONS

	CODE	A L/H ENG	B R/H ENG	C L/H GEN	D R/H GEN	E PC-1	F PC-2	I MP-1	J MP-2	M $\delta_{ACT}-1$	N $\delta_{ACT}-2$	Q δ
A	L/H ENG	~~N~~										
B	R/H ENG	Y	~~N~~									
C	L/H GEN	N	N	~~N~~								
D	R/H GEN	N	N	N	~~N~~							
E	PC-1	N	N	N	N	~~N~~						
F	PC-2	N	N	N	N	N	~~N~~					
I	MP-1	N	N	N	N	N	N	~~N~~				
J	MP-2	N	N	N	N	N	N	N	~~N~~			
M	$\delta_{ACT}-1$	N	N	N	N	N	N	N	N	~~N~~		
N	$\delta_{ACT}-2$	N	N	N	N	N	N	N	N	Y	~~N~~	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

100 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABE	ABF	ABI	ABJ	ABM
ABN	ABQ	ACQ	ADF	ADM	ADQ	AEQ	AFJ	AFN	AFQ
AIQ	AJM	AJQ	AMN	AMQ	ANQ	AQQ	BBQ	BCE	BCN
BCQ	BDQ	BEI	BEM	BEQ	BFQ	BIN	BIQ	BJQ	BMN
BMQ	BNQ	BQQ	CCQ	CDQ	CEQ	CFN	CFQ	CIQ	CJQ
CMN	CMQ	CNQ	CQQ	DDQ	DEM	DEQ	DFQ	DIQ	DJQ
DMN	DMQ	DNQ	DQQ	EEQ	EFQ	EIQ	EJM	EJQ	EMN
EMQ	ENQ	EQQ	FFQ	FIN	FIQ	FJQ	FMN	FMQ	FNQ
FQQ	IIQ	IJQ	IMN	IMQ	INQ	IQQ	JJQ	JMN	JMQ
JNQ	JQQ	MMN	MMQ	MNN	MNQ	MQQ	NNQ	NQQ	QQQ

TABLE XLVI
KILLING HIT COMBINATIONS FOR SURVIVABILITY STUDY
Configuration 13

1 HIT	
1 KILLING COMBINATION	Q Only

2 HITS

14 KILLING COMBINATIONS

	CODE	A	B	C	D	E	F	I	J	M	N	O	P	Q
		L/H ENG	R/H ENG	L/H GEN	R/H GEN	PC-1	PC-2	MP-1	MP-2	δ_{ACT-1}	δ_{ACT-2}	δ_{ACT-3}	δ_{ACT-4}	δ
A	L/H ENG	~~N~~												
B	R/H ENG	Y	~~N~~											
C	L/H GEN	N	N	~~N~~										
D	R/H GEN	N	N	N	~~N~~									
E	PC-1	N	N	N	N	~~N~~								
F	PC-2	N	N	N	N	N	~~N~~							
I	MP-1	N	N	N	N	N	N	~~N~~						
J	MP-2	N	N	N	N	N	N	N	~~N~~					
M	δ_{ACT-1}	N	N	N	N	N	N	N	N	~~N~~				
N	δ_{ACT-2}	N	N	N	N	N	N	N	N	N	~~N~~			
O	δ_{ACT-3}	N	N	N	N	N	N	N	N	N	N	~~N~~		
P	δ_{ACT-4}	N	N	N	N	N	N	N	N	N	N	N	~~N~~	
Q	δ	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y

3 HITS

115 KILLING COMBINATIONS

AAB	AAQ	ABB	ABC	ABD	ABE	ABF	ABI	ABJ	ABM
ABN	ABO	ABP	ABQ	ACQ	ADF	ADP	ADQ	AEQ	AFJ
AFN	AFQ	AIQ	AJP	AJQ	AMQ	ANP	ANQ	AOQ	APQ
AQQ	BBQ	BCE	BCO	BCQ	BDQ	BEI	BEM	BEQ	BFQ
BIO	BIQ	BJQ	BMO	BMQ	BNQ	BOQ	BPQ	BQQ	CCQ
CDQ	CEQ	CFQ	CIQ	CJQ	CMQ	CNQ	COQ	CPQ	CQQ
DDQ	DEQ	DFQ	DIQ	DJQ	DMQ	DNQ	DOQ	DPQ	DQQ
EEQ	EFQ	EIQ	EJQ	EMQ	ENQ	EOQ	EPQ	EQQ	FFQ
FIQ	FJQ	FMQ	FNQ	FOQ	FPQ	FQQ	IIQ	IJQ	IMQ
INQ	IOQ	IPQ	IQQ	JJQ	JMQ	JNQ	JOQ	JPQ	JQQ
MMQ	MNQ	MOQ	MPQ	MQQ	NNQ	NOQ	NPQ	NQQ	OOQ
OPQ	OQQ	PPQ	PQQ	QQQ					

TABLE XLVII
HIT SURVIVABILITY SUMMARY

Configuration	Number of Components	Total Hit Combinations ①			Killing Combinations ②			$P_{K/H} = ②/①$			$P_s = 1 - P_{K/H}$		
		Hits			Hits			Hits			Hits		
		1	2	3	1	2	3	1	2	3	1	2	3
1	9	9	45	165	1	25	125	.1111	.5556	.7576	.8889	.4444	.2424
2A	10	10	55	220	1	18	121	.1000	.3273	.5500	.9000	.6727	.4500
2B	10	10	55	220	1	15	103	.1000	.2727	.4682	.9000	.7273	.5318
2C	10	10	55	220	1	15	103	.1000	.2727	.4682	.9000	.7273	.5318
2D	10	10	55	220	1	16	109	.1000	.2909	.4955	.9000	.7091	.5045
3A	12	12	78	364	1	20	176	.0833	.2564	.4835	.9167	.7436	.5165
3B	12	12	78	364	1	17	159	.0833	.2179	.4368	.9167	.7821	.5632
4	13	13	91	455	1	15	135	.0769	.1648	.2967	.9231	.8352	.7033
5	13	13	91	455	1	14	162	.0769	.1538	.3560	.9231	.8462	.6440
6	11	11	66	286	1	12	116	.0909	.1818	.4056	.9091	.8182	.5944
7A	14	14	105	560	1	15	160	.0714	.1429	.2857	.9286	.8571	.7143
7B	14	14	105	560	1	15	142	.0714	.1429	.2536	.9286	.8571	.7464
8	14	14	105	560	1	15	163	.0714	.1429	.2911	.9286	.8571	.7089
9	15	15	120	680	1	16	146	.0667	.1333	.2147	.9333	.8667	.7853
10	15	15	120	680	1	16	146	.0667	.1333	.2147	.9333	.8667	.7853
11A	13	13	91	455	1	15	135	.0769	.1648	.2967	.9231	.8352	.7033
11B	10	10	55	220	1	13	95	.1000	.2364	.4318	.9000	.7636	.5682
12	11	11	66	286	1	13	100	.0909	.1970	.3497	.9091	.8030	.6503
13	13	13	91	455	1	14	115	.0769	.1538	.2527	.9231	.8462	.7473

TABLE XLVIII
SURVIVABILITY RATING BASED ON THE CONFIGURATION'S ABILITY TO SURVIVE, GIVEN 3 HITS ON THE SYSTEM ($P_S = 1 - P_{K/3H}$)

Order	Configuration	P_S	Rating
1	10	.7853	A
2	9	.7853	A
3	13	.7473	A
4	7B	.7464	A
5	7A	.7143	A
6	8	.7089	A
7	4	.7033	A
8	11A	.7033	A
9	12	.6503	B
10	5	.6440	B
11	6	.5944	B
12	11B	.5682	B
13	3B	.5632	B
14	2B	.5318	C
15	2C	.5318	C
16	3A	.5165	C
17	2D	.5045	C
18	2A	.4500	C
19	1	.2424	C

APPENDIX II

SAMPLE THERMAL CALCULATIONS

1. GENERAL

The calculations presented herein are excerpts from those performed in support of the degree of redundancy trade study. Some narrative elucidation has been added to help the reader understand why a particular approach was used, to more fully present the limitations of the approach, or to justify its use. The calculations presented herein pertain to the predicted temperatures, designated "unimproved package temperatures", in the degree of redundancy trade study presented in Section III. To develop thermal models of the numerous candidate configurations of the trade study, it was necessary to assume package design details which normally evolve during the gradual progress of a design toward production. The sample calculations presented in this appendix are based on configuration 12, Figure 84, and utilize both soft cutoff and servocontrolled pumps.

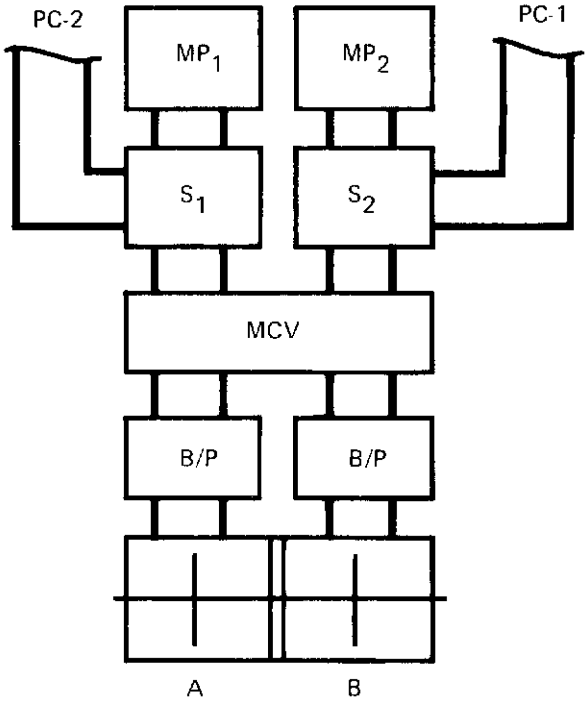


FIGURE 84
CANDIDATE SSAP CONFIGURATION NUMBER TWELVE

For thermal analysis purposes, the output rating was used to establish the electrical motor and pump efficiency. A common technique must be used to reduce the likelihood of selective errors appearing in the heat generation calculations. Several factors influence the required output rating for the pump and motor, the most important of which is the maximum hydraulic horsepower output required of the motor pump. This is a function of the peak flow, peak pressure, and cutoff pressure assumed.

Package size, shape, and surface area are used in calculating the convective and radiative heat transfer rate. The mass and equivalent specific heat of the package determine in part the package vulnerability to high temperature transients. The information used herein was derived by examination of the LTV-E Simplex package, the Vickers APU for the F-4, various preliminary configurations developed by GE and LTV-E during the AFFDL IAP Exploratory Development Program, and various scaling factors based on motor, pump, reservoir, and actuator piston area similarity and/or geometric relationships and estimates of what mounting bracketry for the motors, servovalves, Master Control Valve (MCV), etc., might be designed.

2. SOFT CUTOFF PUMP

a. Package Characteristics

Surface actuator piston area = 5.62 in^2 per chamber.

Maximum flow = 12.2 gpm per pump

Maximum pressure = 1600 psi

Peak hydraulic horsepower output per pump = 5.0, see Figure 85.

An electromechanical as well as an electrohydraulic secondary actuator is a candidate for use on the actual hardware. Its assumption for this thermal model would produce slightly different parameter values. It would not significantly change the comparative configuration ratings.

For initial sizing only, overall efficiency = 0.47 (same as Vickers APU for F-4). Maximum electric motor input HP = 5.0/0.47 = 10.6. Assuming that the peak electric motor output efficiency is 0.86 (see curve for Preco motor, Figure 43), the peak electric motor output power would be 0.86 (10.6) = 9.1 HP for thermal analysis purposes, i.e., for the thermal model.

There is an infinite number of combinations of peak output motor power and motor efficiency that could result in the same electric motor heat losses as used in this analysis. Therefore, if the peak output motor power of the thermal model used herein is subsequently found to be slightly different from the actual design, it does not necessarily follow that the motor heat losses themselves are in error.

To compute hydraulic pump losses, use the empirical equation developed by this thermal analysis to achieve consistency between soft cutoff and servocontrolled pump calculations, and between the pumps of the same type but of different sizes, and for the same pump at widely different operating points.

The power lost in the pump, in horsepower, is:

$$\text{Power Lost} = \frac{0.093\ QrP + 0.069\ Q\ Pr}{1714} \text{ when } P \geq 0.33\ Pr$$

$$\text{Power Lost} = \frac{0.031\ Qr\ Pr + 0.069\ Q\ Pr}{1714} \text{ when } P \geq 0.33\ Pr$$

where:

Pr = Rated pressure, psi

P = Pressure at operating condition where losses are to be computed

Qr = Flow rate, gpm

Q = Flow at operating condition where losses are to be computed, gpm

b. Calculation of Heat Generation

The following calculations are based on the assumption that a flow of .5 gpm continuous is used for the secondary actuator, servovalve, and leakage.

(1) Quiescent Surface Actuator

At 0.5 gpm, P = 1560 psi (from Figure 85).

$$\text{Pump Output Power} = \frac{0.5(1560)}{1714} = 0.5 \text{ HP}$$

$$\text{Power Lost} = \frac{0.093\ (12.2)(1560) + 0.069\ (0.5)(1600)}{1714} = 1.1 \text{ HP}$$

Pump Input Power = 1.1 + 0.5 = 1.6 HP per pump

$$\text{Percent of rated electric motor load} = \frac{1.6}{9.1} = 17.6\%$$

Motor efficiency = 0.75 (from Figure 42)

$$\text{Motor input} = \frac{1.6}{0.75} = 2.13 \text{ HP per motor pump}$$

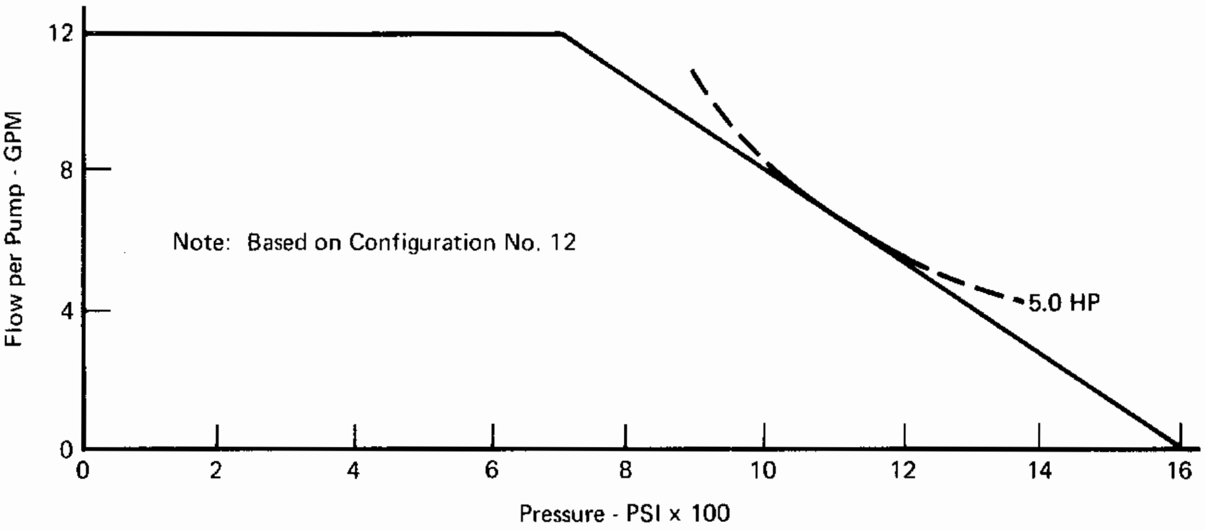


FIGURE 85
OUTPUT OF SOFT CUTOFF THERMAL MODEL

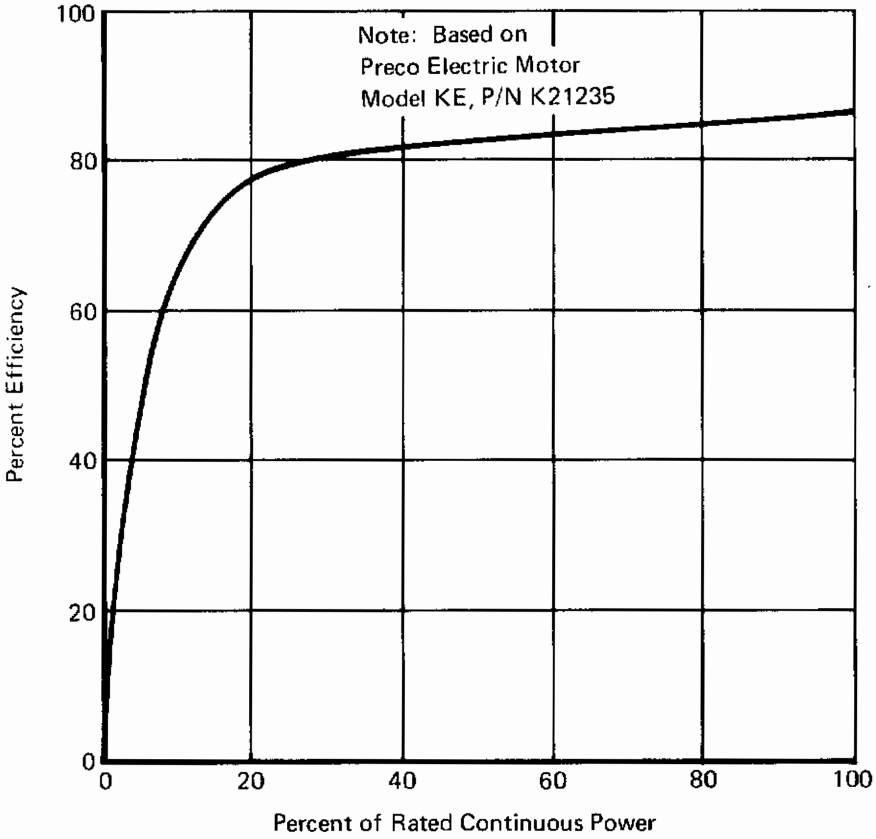


FIGURE 86
ELECTRIC MOTOR EFFICIENCY FOR THERMAL MODEL

(2) Gust Loading Condition

Assume a sine wave input from gusts with a peak-to-peak deflection of 1.2 inches and a frequency, f, of 0.8 cps.

$\omega = 2\Pi f = 6.28(0.8) = 5.03$ radians/sec

$x = Xo \sin\omega t$

$V = \frac{dx}{dt} = \omega Xo \cos\omega t$ = surface actuator velocity

Assume $\bar{v} = 0.707\ V\ max = 0.707\ \omega Xo$
$= 0.707\ (5.03)\ (0.6) = 2.13$ inches/sec = mean velocity

The Hydraulic flow rate, Q, expressed in gallons per minute (gpm), is:

$Q = 0.26\ ApV$; where Ap = piston area, in^2

$\bar{Q} = 0.26\ Ap\bar{v}$
$= 0.26\ (5.63)(2.13) = 3.12$ gpm = mean flow rate

Q Total = $3.12 + 0.5 = 3.62$ gpm

Pressure at 3.62 gpm = 1330 psi

Pump Output Power = $\frac{3.62\ (1330)}{1714} = 2.81$ HP

Power Lost = $\frac{0.093\ (12.2)\ (1300) + 0.069\ (3.62)\ (1600)}{1714} = 1.11$ HP

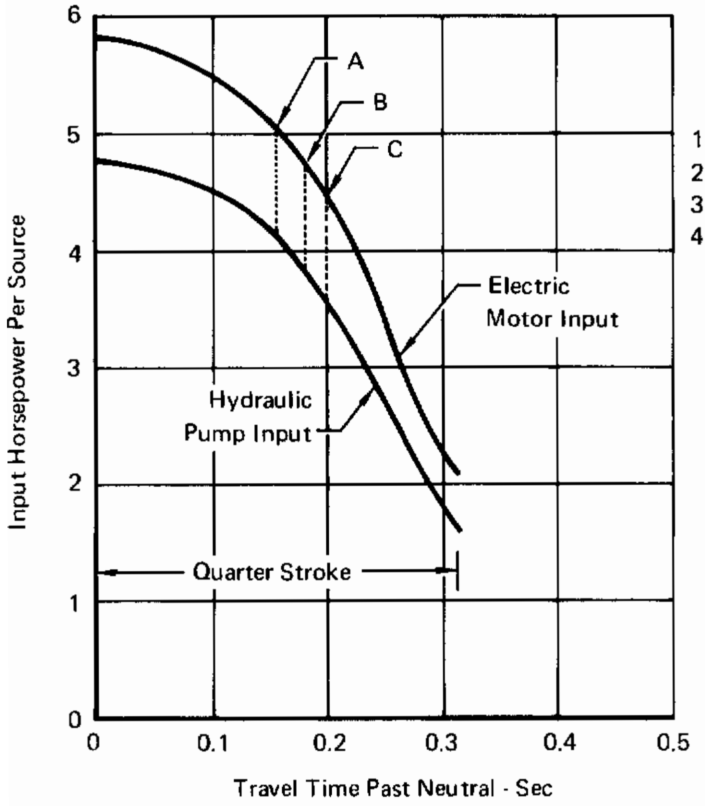
Pump Input Power = $1.11 + 2.81 = 3.92$

Percent of rated electric motor load = $\frac{3.92}{9.1} = 43.1\%$

Motor efficiency = 0.82 (from Figure 44)

Motor input = $\frac{3.92}{0.82} = 4.8$ HP per motor pump.

The gust cycle mean power computed by the foregoing relationships is slightly conservative, per Figure 87.



1. Soft-Cutoff Configuration 12 of Trade Study
2. Frequency of Actuator Motion = 0.8 CPS
3. Half Actuator Stroke = 0.6 In.
4. Actuator Simple Harmonic Motion Assumed

A - Assumed Mean for Thermal Analysis
B - Actual Mean Value for This Case
C - Mean if Power Varies Sinusoidally vs Time

FIGURE 87
GUST CYCLE MEAN POWER

(3) If the surface actuator is subjected to 1 minute of gusts and 4 minutes quiescent out of each 5 minute period, the power required is:

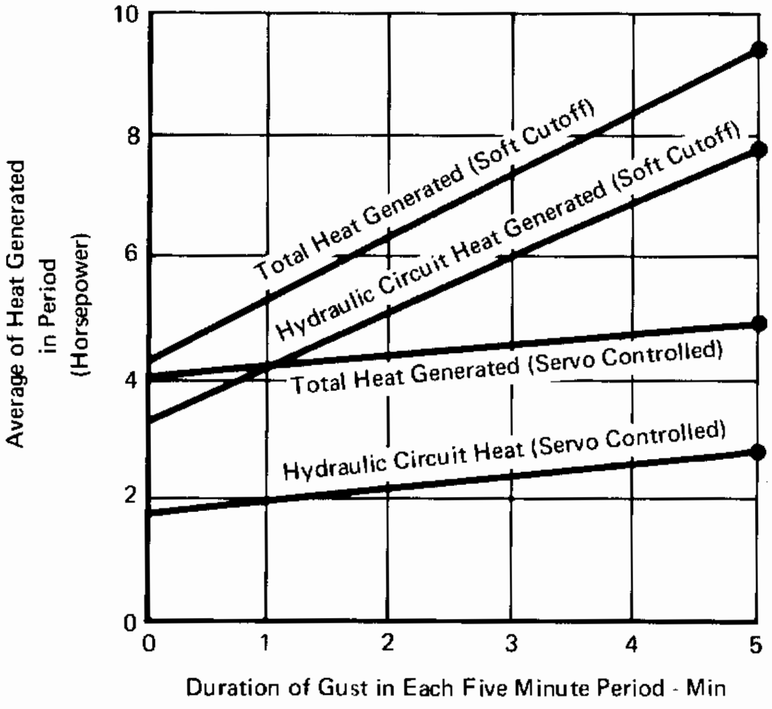
$$\frac{4\,(2.13) + 1\,(4.8)}{5} = 2.7 \text{ HP mean electric input power per motor}$$

$$= 5.4 \text{ HP electric input power total per package}$$

$$\frac{4(1.6) + 1(3.92)}{5} = 2.1 \text{ HP mean hydraulic circuit power per pump}$$

$$= 4.2 \text{ HP Hydraulic power total per package}$$

The effect of other gust durations is shown by Figure 88. A summary of the above calculations is shown in Table XLIX.



Notes:
1. Two motor pumps based on Configuration 12
2. Harmonic motion
    - Frequency = 0.8 CPS
    - Half Amplitude = 0.6 In.

FIGURE 88
EFFECT OF GUST DURATION ON HEAT LOAD

TABLE XLIX
SOFT CUTOFF HEAT GENERATION SUMMARY

	Hydraulic Circuit HP		Electric HP	
	Per Pump	Total	Per Motor	Total
Quiescent	1.6	3.2	2.1	4.2
Gust	3.9	7.8	4.8	9.6
1 Minute Gust +4 Minute Quiescent	2.1	4.2	2.7	5.4

Note: Harmonic actuator output motion at 0.8 cps and 1.2 inch peak to peak stroke

c. Operating Temperature Calculations

The unimproved fluid operating temperature is a function of the heat dissipated per unit area from the hydraulic circuits, the compartment temperature to which the heat is dissipated, and the convective and radiative heat transfer coefficients which determine the temperature difference between the hydraulic circuits and the compartment. The compartment temperature is a function of the sum of the heat dissipated by both the electrical and hydraulic components. It is also a function of the ram air flow and temperature.

(1) The surface area is used in calculating the heat dissipation per unit area and its geometry is used to estimate the convection heat transfer coefficients and the radiation view factors. The following is a breakdown of the surface area estimated for configuration 12.

Reservoirs*	604	in^2
Pumps	120	in^2
Filters	20	in^2
Switching Valve	131	in^2
Control Valve	101	in^2
Tubing	45	in^2
Actuator Cylinder	341	in^2
TOTAL	1362	$in^2 = 9.46\ Ft^2$

* Standard "unimproved" reservoir was assumed to have nominal finning or equivalent irregularities.

(2) Surface Heat Transfer Coefficients

(a) Convective Coefficient, h_c

The cross-sectional area of the compartment which, at the SSAP, is usable for airflow is approximately 400 in^2. At 1.46 at 50,000 feet pressure altitude the airflow through this area will be approximately 13 Lbm/min. The air density at 50,000 feet for a temperature of 150°F is .0075 Lbm/ft^3. Therefore, the average air velocity, V, will be

$$V = \frac{13/60}{(0.0075)(400/144)} = 10.4 \text{ ft/sec}$$

From Reference 4, for air flowing across a cylinder, Nusselt Number = (Reynolds Number)^n. Assume a diameter of .5 feet as the characteristic dimension of the thermal model. Evaluate the physical properties of the air at the average film temperature, t_f.

$$t_f = \frac{ts + t}{2}$$

t_s = surface temperature

t = air temperature

For a surface temperature of 574°F, the average film temperature is

$$t_f = \frac{574 + 150}{2} = 362°F$$

Using this temperature

$$\text{Reynolds Number, } N_R = \frac{VD\rho}{\mu} = 1950$$

and

$n = 0.446$
$C = 0.615$

and solving for h_c, we have

$$h_c = 0.615 \frac{k}{D}(N_R)^{0.446}$$

where

k = thermal conductivity at average film temperature
D = characteristic dimension.

Substituting, we calculate

$$h_c = 0.84 \text{ BTU/Hr Ft}^2 \text{ °F}$$

(b) Radiative Heat Transfer Coefficient, h_R

Assuming heat exchange between two non-black surfaces,

$$h_R = F_{AE}(0.174)(10^{-8})\left(\frac{T_p^4 - T_c^4}{t_p - t_c}\right)$$

t_p = Package radiating surface temperature, °F

T_p = Package radiating surface absolute temperature, °R

$= t_p + 460$

t_c = Compartment temperature, °F

T_c = Compartment absolute temperature, °R

$= t_c + 460$

F_{AE} = Combined emissivity and shape factor for package and compartment surfaces

Because of the complex shape of the package, only a portion of the area is fully effective for radiation heat transfer to compartment walls. Various package components could have emissivities significantly below 1.0. Therefore, F_{AE} is taken as .45 for the overall thermal model.

(c) Overall Package Heat Transfer Coefficient, h_T

Considering both convection and radiation heat transfer

$$h_T = h_c + h_R$$

$$= 0.84 + h_R, \text{ BTU/Hr Ft}^2\ ^\circ\text{F}$$

The actual overall heat transfer coefficient is envisioned as a composite coefficient created by thermal radiation coefficients and convection coefficients which individually vary a great deal over the package surface. The convection variation results from the fact that the airflow is variously over short-flat surfaces, and miscellaneous irregularities, along and across cylinders, and between actuator members which form passages of widely differing wetted perimeters, etc. Because of their proximity to each other, the irregularities and passages mutually disturb the airflow patterns of others. In addition, the immediate presence of aircraft structural members, wire bundles, tubing, etc., also disturbs the flow and tends to increase the rate of convection. The total package surface area is variously distributed among the resulting coefficients, tending to increase the importance of some and decrease the importance of others. The radiation variation results from the fact that some of the package surfaces have a poor view of the aircraft structure or "see" relatively hot structure. Therefore, locally, the benefit of radiation is reduced. It appears nonproductive to make inordinately complex assumptions to define explicit details of the foregoing areas and coefficients. Implicit in the thermal modeling herein is an allowance for the existence or use of any number (n) of combinations of surface areas (A) and heat transfer coefficients (h) that produces an overall coefficient equal to that in this analysis.

$$h_T = \frac{h_1A_1 + h_2A_2 + \ldots\ldots\ldots h_nA_n}{A_1 + A_2 + \ldots\ldots\ldots A_n}$$

It is possible, of course, that under these various influences, the local coefficients at different parts of the package may tend to be self equalizing. The following reasons could cause that tendency.

- o For large flat surfaces where parallel flow convection coefficients are usually low, the radiation coefficients are usually relatively high, if the emissivity is high.
- o For complex shapes and passages, having small characteristic dimensions, and where convection coefficients are usually high, the radiation coefficients may be low, because the complex shape produces a poor view factor.

Explicit detailed assumptions and exhaustive coefficient investigations can more productively be made when determining the heat transfer provisions required to operate at a specific fluid temperature. Then, the assumed details or their equivalent can be treated as required features, if beneficial, to be incorporated in the package design. For this prediction of unimproved package temperatures, however, the rationale for developing the thermal model is considered adequate.

(d) Compartment Temperature, t_c

The equilibrium compartment temperature is determined by means of energy balance equations which can be solved for tc as follows:

$$t_c = \frac{(\text{Total package horsepower})(2547) + UA(taw)W\,Cp60tr}{UA + W\,Cp60}$$

Total package horsepower = 5.41

Aircraft compartment thermal conductance, UA = 29.7 BTU/Hr°F

Adiabatic wall temperature, taw = 78°F

Ram air flow, W = 13 Lbm/min

Specific heat of air at constant pressure, Cp = 0.24 BTU/Lbm°F

Ram air temperature, $tr = 95^\circ F$

therefore

$$tc = \frac{5.41(2547)+29.7(78)+13(0.24)(60)(95)}{29.7 + 13(0.24)(60)}$$

$$= 156^\circ F$$

(e) Package Surface Temperature, t_p

The heat dissipated by the motor itself will be transferred directly to the compartment air and structure. Except for raising the compartment temperature, the motor dissipation will not significantly affect the temperature of the hydraulic components. The steady energy balance for the hydraulic components is

$$\text{Pump input power} = HP_{pi} = h_T A_S \ (t_P - t_c)$$

Rearranging

$$t_p = \left(\frac{HP_{pi}}{A_s}\right) \frac{1}{h_T} + t_c = \left(\frac{HP_{pi}}{A_s}\right)\left(\frac{1}{0.84 + h_R}\right) + t_c$$

A nomograph of this equation was made for various values of t_p, t_c and HP_{pi}/A_s for use in evaluating the unimproved package temperatures for all trade study configurations.

For the configuration 12 soft cutoff thermal model

$$\frac{HP_{pi}}{A_s} = \frac{4.21(2547)}{9.46} = 1132 \text{ BTU/Hr Ft}^2$$

Using the above value of heat dissipation per unit area, the value of t_p read from the nomograph was 574°F.

To check the temperature by calculation, we obtain:

$$h_R = 1.84 \text{ BTU/Hr Ft}^2$$

$$h_T = 1.84 + 0.84 = 2.68 \text{ BTU/Hr Ft}^2 \ {}^\circ F$$

and

$$t_p = (1132)\left(\frac{1}{2.68}\right) + 156$$

$$= 422 + 156 = 579^\circ F$$

This is close enough to the nomograph value to validate its use. Assuming the mean fluid temperature is 10°F higher than the surface temperature, the package mean fluid temperature = 574 + 10 = 584°F.

(f) Effect of Thermal Parameter Deviations

The thermal parameters for an actual package are expected to deviate somewhat from the surface area, convective heat transfer coefficient and the combined emissivity and shape factor that are used herein. Assuming a constant heat dissipation per unit area, the effect of deviations on unimproved package temperature is shown by Figure 89.

The convective coefficients used in calculating the unimproved package temperatures are probably low because a simple smooth cylinder was assumed in the analysis, rather than the complex arrangement of protuberances, shapes, and passages.

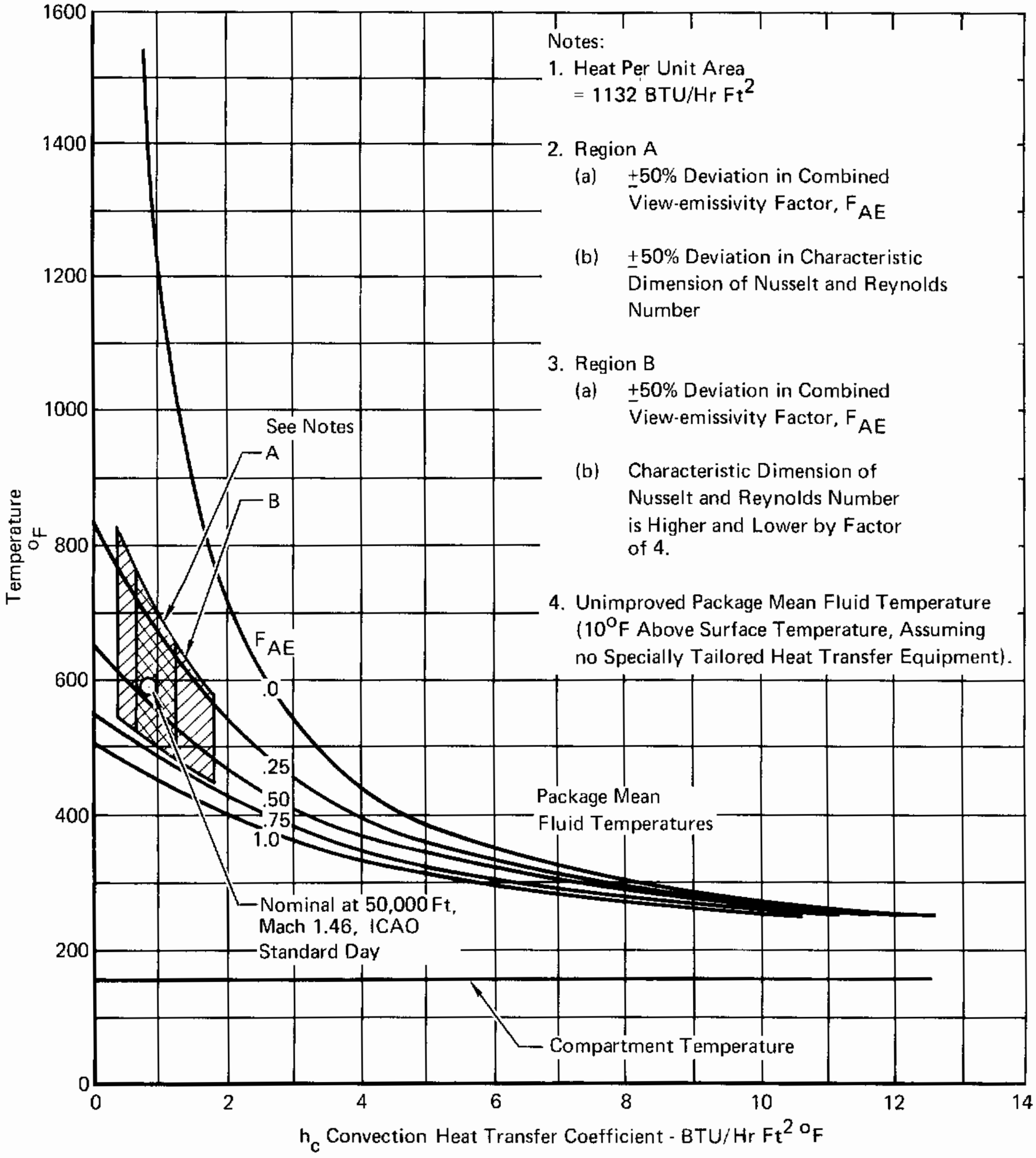


FIGURE 89
EFFECT OF THERMAL PARAMETER DEVIATIONS

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13. ABSTRACT

The Survivable Flight Control System (SFCS) Program is an advanced development program of which the principal objective is the development and flight test demonstration of an SFCS utilizing Fly-By-Wire and Integrated Actuator Package techniques. The studies and analyses conducted to date have sufficiently defined the system requirements to provide a definition of an approach to the implementation of the SFCS. The results of these studies and the definition of the approach are presented in the basic report. The details of the Control Criteria and Control Law Development studies are presented in report supplements 1 and 2, respectively.

The details of the Hydraulic Power and Actuation studies are reported in this supplement 3. The four major topics covered are:

- o Survivable Stabilator Actuator Package (SSAP) Degree of Redundancy Trade Study. Nineteen configurations were examined, leading to selection of a full time duplex package with an aircraft central hydraulic system backing up each integrated hydraulic system. Two electric motors of approximately eight horsepower each will drive two soft cutoff pumps. The SSAP will have a dual tandem piston, and will be designed for a maximum steady state fluid temperature of 450°F.

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13. ABSTRACT (CONTINUED)

- Hydraulic Power Supply Study. The SFCS requires addition of a fourth hydraulic system which will be based on the production F-4 APU.

- Fluid Selection Study. The SSAP and its backup central hydraulic systems cannot use MIL-H-5606 fluid, since it is limited to 275°F. MIL-H-83282, a new fire-resistant synthetic hydrocarbon fluid was selected. MIL-H-83282 is compatible with all the materials in the existing hydraulic systems of the F-4 test aircraft.

- Actuator Dynamic Analysis. The secondary actuator and the SSAP designs were analyzed to determine nuisance disconnect characteristics, stability, frequency response, and failure transients. A cross-element comparison technique has been selected for in-flight monitoring of both the quad-redundant electrohydraulic secondary actuators and the quad-redundant electromechanical secondary actuator of the SSAP.

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
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