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RESEARCH AND DEVELOPMENT OF AN INTEGRATED SERVO ACTUATOR PACKAGE FOR FIGHTER AIRCRAFT

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FOREWORD

The effort described in this document was performed by the Automatic Controls Section of LTV Electrosystems, Inc., Box 430, Arlington, Texas, 76010, under Air Force contract F33615-68-C-1387, "Research and Development of a Servo Actuator Package for Fighter Aircraft", The program was administered under the direction of the Air Force Flight Dynamics Laboratory (FDCL), Wright-Patterson Air Force Base, Ohio, 45433, with Vernon R. Schmitt, Project Engineer.

This report covers work performed between 1 April 1968 and 1 November 1969 by members of the Automatic Controls Engineering Department of LTV Electrosystems. W.G. Koch was the Project Engineer. Principal contributors to the design and testing effort and to the preparation of this report are J. McCurley, M.H. Post, D.H. Neill, R.D. Munsch, R.H. Marchell, and J.C. Huber. J.H. Hickerson provided the reliability analyses. In addition, assistance by various McDonnell Aircraft Company and Vickers Aerospace Division personnel is acknowledged in supplying F-4 system requirements and pump design information respectively.

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This technical report has been reviewed and is approved.

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ABSTRACT

A study was conducted to determine the feasibility of implementing present day fighter and attack aircraft with integrated actuator packages in the primary flight control system. The integrated package concept results in a control system having a considerably lower vulnerability to small arms fire than the conventional control system. A Simplex actuator, containing a single self-contained hydraulic supply was designed to meet the performance and structural interface requirements of the F-4 stabilator actuator. Three units were fabricated and functionally tested. One of the Simplex units was subjected to a comprehensive qualification test. The other two units were delivered to McDonnell Aircraft Company for flight testing. A Duplex integrated actuator, containing dual self-contained hydraulic systems and quadruplex electrical input channels, was designed, built, and tested. Test results demonstrate the basic feasibility of electrically signalled integrated actuator packages. Specific recommendations for further development of the integrated package concept are included.

This abstract is subject to special export controls and each transmittal to foreign governments or foreign nationals may be made only with prior approval of the Air Force Flight Dynamics Laboratory, FDCL, Wright-Patterson Air Force Base, Ohio 45433.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

SECTION I

INTRODUCTION

A. GENERAL

The work covered by this report concerns the establishment of designs and design techniques that represent a significant departure from those used in flight control over the past twenty-five to thirty years.

The conventional mechanization of an irreversible hydraulically powered flight control system consists of centralized hydraulic power supplies and actuation devices located at the control surfaces. The power supplies are variable displacement pressure compensated units mounted directly to the engine accessory pad. The hydraulic energy generated at the pump is then routed throughout the aircraft to the various flight control actuation devices at high pressure (usually at 3000 psig) via tubing optimized for overall weight and fluid frictional loss. Control of the surface actuation units is via the mechanical linkages and cables which route the pilot's command signals from the control column and rudder pedals to the surface actuators.

Recent combat experience in Southeast Asia indicated certain shortcomings in the ability of current USAF aircraft to survive intense ground fire. One of the major contributors to this vulnerability is the aircraft flight control system. Routing of the high pressure lines and their associated vulnerable area constitute a vulnerability aspect which is believed to be responsible for a significant percentage of aircraft losses due to ground and air fire. Modification of the conventional control system to remove or reduce this area should significantly improve the ability of current Air Force military aircraft to survive the Southeast Asia type of hostile ground fire environment.

B. OBJECTIVE

The objective of this program is to establish design techniques and the feasibility and advantages of integrated power control servo actuator packages for present day military fighter-bomber and attack aircraft. The primary motivation for this program is the general need for technology pursuant to decreasing the primary flight control system vulnerability in future aircraft designs.

The term Integrated Actuator Package (IAP) refers in general to a broad class of flight control actuators wherein the complete hydraulic power supplies are integrated directly into the actuator. Power is supplied electrically to the IAP's from the aircraft generators, and converted to hydraulic power within the package for control of the surface. The packages may be either mechanically signalled, as in conventional control systems, or be electrically signalled for use in a fly-by-wire (FBW) electrical primary flight control system, or a combination of the two. As the development of FBW system proceeds it is anticipated that the IAP devices will be all electrically signalled.

C. BACKGROUND

The use of integrated packages, either mechanically or electrically signalled, is not completely new. During World War II the Germans employed them in their V-2 rocket and in several of their military aircraft. Electrically powered and controlled IAP's were used as parallel surface actuators for autopilot directional control on the HE-11, JV-88, ME-110 and DO-17. The British IAP design which started some 15 years ago is used on the VC-10. It is equipped with integrated actuators on each of the split surfaces of all three axes. The initial block of Regulus I missiles built in this country by Chance Vought (now LTV Aerospace) for the U.S. Navy were equipped with electrically signalled integrated actuator packages built by Vickers. These IAP's used variable pressure, variable flow pumping units, with a loop closed around the pump displacement element (servo pump). More recently, the integrated package concept has been successfully applied to thrust vector control actuation of the Polaris missile.

Based on the above experience, it is obvious that the problem in designing IAP's is not one of developing a complelely new technology or inventing new components, but rather it is one of determining the feasibility of applying the concept to present and future high performance attack/fighter and fighter-bomber aircraft, and of up-grading component performance. For example, 1) methods and materials must be devised to overcome the heat dissipation problem; 2) problem of space limitation for installation of IAP's in the wings (lateral axis) must be resolved; and, 3) components having higher performance and temperature capability than those presently available must be designed.

D. DEFINITIONS

A number of relatively new terms are used in this report. For the sake of clarity these terms, as used in this report, are defined below.

1. Integrated Actuator Package (IAP)

IAP refers to a broad class of flight control actuators with self contained hydraulic power supplies. Each power supply consists of an electric motor driving a hydraulic pump with reservoir, check valves, filter, relief valves and associated hydraulic circuitry. Power for the IAP's is derived from the aircraft electrical system. Isolation from the aircraft hydraulic system is calculated to significantly reduce the vulnerability of the aircraft to small arms fire.

2. Simplex Integrated Actuator Package

A Simplex IAP is essentially a package containing a single non-redundant actuator and a single motor pump unit with associated hydraulic circuitry. However, the Simplex package as described in this report contains the components and functions required for the present F-4 stabilator actuator. In addition, it contains a non-redundant motor-pump, reservoir, and monitoring and switching functions integral with the actuator as an emergency or backup hydraulic power supply.

3. Duplex Integrated Actuator Package

The Duplex IAP has two independent electric motor driven hydraulic power supplies each supplying the hydraulic power for one-half of a dual tandem actuator. To meet the two fail operate, fail safe requirement for signal transmission, the package is equipped with quadruply redundant FBW input signal channels.

4. Triplex Integrated Actuator Package

The Triplex IAP reflects ideal redundancy for primary flight control actuation devices. It contains three independent and redundant electro-hydraulic power supplies and quadruply redundant FBW input channels. Two full-time power supplies provide primary hydraulic power for a dual tandem actuator, and a third unit provides emergency power for the actuator in the event of primary system failure.

5. Power-by-Wire (PBW)

Power-by-Wire refers to the transmission of power from the aircraft engine to the control surface actuator by electrical means rather than hydraulic. Instead of generating hydraulic power at the engine accessory pad, electrical power is distributed to IAP's located at the control surfaces, where the hydraulic power is generated.

6. Funk Strut

A funk strut is a bi-directional spring cartridge which is preloaded and acts as a solid link until the load in the link reaches the preload level at which point the spring collapses at its spring rate.

SECTION II

SUMMARY

The conflict in Southeast Asia has revealed a serious shortcoming on present day high performance military aircraft, namely the vulnerability of the aircraft flight control system to small arms or ground fire.

A potential solution to this critical problem appeared to be the use of the Integrated Actuator Package (IAP) concept. IAP designs use electrical power, thus the hydraulic lines are eliminated for flight control purposes. The electrical power and signal control lines can easily be made redundant. The threat of hydraulic fire is greatly reduced, thus permitting the aircraft to sustain considerable battle damage and still remain operational.

An R & D program was undertaken to determine if this concept together with the technical design and hardware problems, could be applied to the present and future operational fighter and fighter-bomber aircraft. Results of this program are contained in this report.

Early in the program it was determined that it was feasible to design and construct an IAP; however, feasibility of the IAP concept meant more than merely the feasibility of building a unit. It meant determining to what extent it could be applied to modern high performance aircraft. Because the pitch axis appeared to be the most critical and due to cost and time considerations, only the pitch (longitudinal) axis was investigated on this program. Studies of the application of the IAP concept revealed that problems in packaging, heat dissipation, and motor-pump efficiency which had not previously bothered the flight control designer were now of paramount concern. In order to properly size an IAP, i.e., one which would provide the proper stiffness, force output, response, etc., and yet use minimum power, have a high efficiency, and be minimum in terms of weight and size, it was necessary that the relationships between actuator and aircraft aerodynamics be thoroughly studied. In other words, establishment of precise and detail IAP design requirements required that the actuator output over the entire flight regime be given a close look. For example; actuator area as related to flutter, control surface slew rates as related to maneuvers and landing, electrical power vs hydraulic power, and component reliabilities, received special attention and study.

As an expedient measure, data on the A-7D was used in the study of actuator-aircraft functional relationships. The results of this study were not used per se in a particular design, but it did provide background experience and guide lines for an IAP design once a vehicle was selected.

Investigation on the application of the IAP to modern U.S. operational fighter-bomber aircraft was limited to the F-4 and F-111. This study was conducted in sufficient depth that a determination could be made as to the size, cost, ease of installation, maintainability, etc., of an IAP for use in the longitudinal axis of these aircraft. Based on the results of this study, the F-4 aircraft was selected. Hence, the experimental laboratory model IAP, which had been planned if the concept proved feasible, was designed to the F-4 requirements.

At this time, some five months after program start, there was an urgent need on the part of the Air Force to have flight test data which might be applicable to advanced aircraft such as the FX, now the F-15; therefore, it was decided to build a flight worthy Simplex package instead of the laboratory model. Although this did not change the basic objectives of the program, it added considerably to the amount of design detail and also required assurance of aircraft compatibility, complete environmental tests, and fabrication of additional units for laboratory and flight testing.

It should be kept in mind that after the initial phase, a number of investigations were being performed simultaneously on the program. As the decision had already been made at the start of this work that if any models were to be built, they would use start-of-the-art components, one group was looking at the hardware aspect while others were concerned with the analysis and advanced components and concepts.

One of the central problems in building IAP's of all types is that of heat generation of the power section, especially the heat generated continuously due to pump quiescent power losses. Investigation of pumps and discussions with pump manufacturers in the U.S. revealed that extensive development work would be required before an optimum design, e.g., servo-pump type would be available. This means that until such time as optimum pump designs become available, IAP's will, by necessity, use conventional type pump designs with the attendant heat problem and therefore will be relatively inefficient.

The problem of heat, both generation and dissipation, was thoroughly investigated. A thermal analysis was performed on the Simplex and Duplex designs. The Simplex design has successfully passed the high temperature requirement encountered in the F-4 aircraft, without any type of heat exchanger. The Duplex unit, however, has two blowers to provide forced air cooling to the package. Analysis shows that without cooling, excessive temperatures will be experienced.

Three Simplex packages were built on this program. One package was used for flight qualification and was subjected to all the functional and environmental conditions of the F-4 stabilator actuator, including life and endurance tests. Reliability studies were also performed. The two remaining Simplex packages were functionally tested with and without load, and delivered to the McDonnell-Douglas Aircraft Corporation for eventual installation and flight test in the longitudinal axis of an F-4 aircraft.

The Simplex IAP has a net weight of 88 lbs., is of the moving cylinder configuration, and has a steel body in place of an aluminum body as presently installed in the F-4. Command inputs are via the pilot's manual signal linkage, the stabilization augmentation system (SAS) and autopilot electrical inputs. System operation is in three distinct modes: manual, SAS, and autopilot. In the manual mode, the electrical unit is inoperative and spring loaded to center. In the SAS mode inputs from the aircraft motion sensors are summed in series with the pilot's input to position the actuator. SAS signals are converted to mechanical motion by the limited authority auxiliary ram which is a single channel electro-hydraulic servo. In the autopilot mode, the auxiliary ram has full actuator rate and position authority. Provisions are made in the mechanical summer to permit the pilot to override the autopilot by applying sufficient force at the stick.

The control valve is a dual tandem unit which meters fluid to the dual tandem main ram. Aircraft hydraulic power is supplied to the IAP in the form of P_1 and P_2. P_2 is ported directly to the dual tandem valve, and powers one half of the actuator. P_1 is ported through the switching valve to the dual tandem valve. In addition, it supplies the hydraulic power for the auxiliary ram, and for operating the mechanical summer through the pilot-operated valving.

The emergency hydraulic supply is integrated into the Simplex. The electrical motor is mounted rigidly to the actuator and contains its own air cooling circuit. The system is sized to provide emergency landing requirements of 10^{o}/sec of stabilator rate, and sufficient actuator output force for aircraft control up to Mach 0.9. During conventional operation, the emergency part of the system is inoperative. The emergency system can be initiated at the pilot's discretion, or it comes on automatically and to full power within one second in the event P_1 hydraulic power is lost (or drops below 500 psi).

As originally defined in the program, the Duplex was scheduled to be designed to accept only mechanical signals from the control system. A third actuator designated the Triplex was to be designed which would contain three integrated hydraulic systems, two primary units and one backup or emergency unit, and would accept only electrical input signals. The Duplex was to be an integrated actuator with two integrated full-time power supplies, but with manual control. The Triplex was to be an integrated actuator with three hydraulic supplies and electrical input channels (power-by-wire as well as fly-by-wire). The program was later re-directed to eliminate the Triplex and incorporate the fly-by-wire input channels on the Duplex. The Duplex is basically a fixed body dual tandem actuator with a redundant FBW signaling unit and two electrohydraulic power supplies with associated hydraulic circuitry. It is designed to meet the basic performance (in terms of power) and environmental requirements of the F-4 stabilator actuator. The basic pumping unit is a Vickers -044 in-line piston pump with load compensation control, driven by a 3-phase, 400 Hz, Preco motor mounted in line with the pump.

The signal conversion system on the Duplex consists of quadruply redundant electromechanical units which convert electrical command signals directly to a mechanical signal that operates the main control valve. Each electromechanical unit consists of a DC motor, the output of which is converted from rotational to linear motion by virtue of a highly efficient ball screw unit. Mechanical outputs of the four E/M units are force summed through individual funk springs onto a common torque tube which drives the main control valve. Failure of any channel results in the failed channel breaking out its funk spring, which removes electrical power to the failed unit.

The results of this R&D program are evidence that the Integrated Actuator Package concept can contribute significantly to improving the survivability of military aircraft when applied to the primary flight control system, including those equipped with fly-by-wire designs.

Two simplex packages were delivered in August 1969 to McDonnell Douglas for installation and flight tests in an F-4 aircraft currently being used as a test vehicle on the 680J ADP "Survivable Flight Control System Development Program".

SECTION III

INTEGRATED ACTUATOR PACKAGE FEASIBILITY STUDY

A. GENERAL

This study is concerned with evaluating the feasibility of implementing the primary flight control system of a current attack/fighter aircraft with integrated actuator packages.

The IAP concept as used in this study means dual actuation devices (Duplex units) at each control surface. Each package contains a dual tandem actuator, dual tandem control valves, two AC motor-pump units, and associated hydraulic components. The units accept both mechanical and electrical signals. The control surface loads, rates, duty cycles, and the hydraulic and electrical system configurations for the feasibility study were based on LTV's A-7 aircraft. Application of the IAP concept to an aircraft depends on the detail mechanization of the control surfaces, the overall mission of the particular aircraft, and availability of the required components. Consequently, extrapolation of the results of this study to other aircraft should be handled with care.

A conventional hydraulic power control system which reflects the existing A-7 control system size was configured for this study. It should be noted that all components in the existing control system may not be optimized. This system supplies only the primary flight control functions; lateral, directional, and longitudinal, including the SAS units in each axis and a feel isolation actuator (FIA) in the roll channel. It is assumed that all secondary and utility functions are supplied by an engine mounted pump or equivalent.

The IAP system is configured and defined to compare with the conventional system. Again, it is assumed that for purposes of this study, the secondary and utility functions are serviced by a conventional engine mounted pump, or perhaps a combination of remotely located electro-hydraulic power packages with conventional actuator devices.

The IAP power system configuration consists of two electrical power generators with Constant Speed Drives (CSD's). Electrical power is routed to the several IAP units. Each IAP unit is dualized, containing two motor pump units, dual tandem valves and actuators, and associated hydraulic components. The units accept manual inputs together with electrical inputs from the SAS and autopilot.

B. FEASIBILITY STUDY GROUND RULES

1. General

This study involves basically a comparison between a conventional and an IAP primary flight control system. The comparison is made for the A-7 control system requirements. Sophistication of a study of this nature can be extended to any degree desired. The results and the conclusions which can be drawn are dependent on the depth and completeness of the study. The validity of the results depends on the assumptions made and the ground rules established which define the thoroughness and completeness of the study. The ground rules defined in this section are designed to direct and control the scope of the study to stay within the allotted time and budget schedules.

2. Ground Rules

The following are general overall assumptions and ground rules are established for the Integrated Actuator Feasibility Study.

a. Aircraft Application

The study compares the conventional vs. IAP primary flight control systems as applied to LTV's A-7 aircraft series.

b. Flight Control System

Only the primary flight control system is considered in the comparison. The primary flight control surfaces are defined as the ailerons, spoiler/deflector, unit horizontal tail (UHT), and rudder.

c. Armor

No parasitic or integral armor is considered for either system. It should be recognized, however, that the IAP system could be more easily provided with armor; however, the implementation of armor plating and the evaluation of its effect on the study is beyond the scope of this study.

d. Hydraulic Pressure

Maximum hydraulic pressure for the conventional system is assumed to be 3000 psi. This is compatible with the energy levels of existing hydraulic systems. The IAP system is not limited to a constant pressure. Pump characteristics which provide the most efficient conversion of energy will be established.

e. System Mechanization

The weight, volume, cost, etc., of the electronic portion of the flight control system in terms of sensors, power supplies and amplifiers is assumed to be identical for both systems since the same function is performed; consequently, they are not considered in this study. In the mechanization of the conventional installation, the Stability Augmentation System (SAS) and autopilot signals are injected into the signal linkage upstream of the power control actuators. In the IAP configuration, they are injected into the signal channel at the IAP unit itself.

f. Evaluation Parameters

The initial evaluation parameters are weight, volume, cost, and vulnerable areas. These parameters are defined for the two systems and tabulated as required. After complete definition of the two systems by virtue of the above parameters, they are evaluated from the standpoint of reliability, maintainability, vulnerability and thermal aspects. The specific methods used in evaluation of these parameters are described in the following sections.

g. Vulnerable Area

The projected area of the flight control system is defined in the horizontal plane only. This is considered the most vulnerable area for assessing the probability of survival. Shielding of the flight control system is not considered except where this shielding was inherent; i.e., when part of the control system is shielded by itself.

h. Weight

Weight of the two systems reflects the total weight required to implement an aircraft with the respective system. The weight of the conventional system is obtained by systematically adding the weight of all components. These are obtained from available weight and balance records. The weight of the IAP system reflects state-of-the-art components.

i. Conventional System Definition

The conventional system definition is achieved with the aid of available controls and hydraulic design manuals and detail drawings and description of the A-7 flight control system. The system includes controls for the Unit Horizontal Tail (UHT), rudder, aileron, spoiler/deflector, and the associated Feel Isolation Actuator (FIA), SAS units, trim actuators, and all signal linkage. The hydraulic circuit is defined by using the existing A-7 PC-1 and PC-2 circuits as a guide. The utility functions and Ram Air Turbine (RAT) connections are eliminated.

j. IAP System Definition

The IAP system consists of engine-mounted CSD's and electrical inputs. The FIA, SAS units, and in some cases, the trim units, are eliminated. A total of seven IAP units are required, two ailerons, two spoiler/deflectors, two UHTs, and one rudder. IAP units are optimized to the specific aerodynamic requirements of rate, load, and duty cycle. Size and configuration of the units in the vertical tail and wings are restricted, where possible, to wing and stabilizer thickness. Normal pilot control of the IAP's is by conventional control linkage and cables. Consequently, much of the signal system for the IAP system is identical to the conventional system and only delta and total values for the initial evaluation parameters are required.

C. DESCRIPTION OF CONVENTIONAL CONTROL SYSTEM

1. General

The conventional primary flight control system configured for the Feasibility Study is patterned after the A-7 system. The mechanical control system linkage for the conventional system is identical to that currently employed on the A-7. The hydraulic power control actuators, FIA, and Automatic Flight Control System (AFCS) series actuators are also identical to the A-7 installation. Hydraulic power supplies (pump, reservoir, and associated components) are patterned after the A-7 Power Control System No. 1 unit which supplies only primary flight control functions. Definitions of the distribution system are based on extrapolation of the A-7 system. Sizing and routing of the hydraulic lines for the conventional primary flight control system reflects standard practice for this type of application.

2. Hydraulic Supply System

The power control hydraulic system defined for the conventional system is shown in block diagram form in Figure 1 and represents current design practice applied to the A-7 airplane. The system consists basically of two identical and independent hydraulic systems supplying pressure to the various actuation devices. The systems are referred to as Power Control System Number 1 (PC-1) and Power Control System Number 2 (PC-2).

Figure 2 is a detail schematic of the complete hydraulic system defined for this study. PC-1 supplies one-half of all the power control actuators, one-half of the FIA, and the yaw autopilot actuators. PC-2 supplies one-half of all the power control actuators, one-half of the FIA, and the lateral and pitch autopilot actuators. Components in Figure 2 are identified in Table I.

The two power control pumps are identical and represent the pump size presently used on the A-7 PC-1 system. This pump was sized on the basis of providing adequate flow to all actuation devices for the established control surface. Pump output can meet any aircraft maneuver requirements at engine speeds varying from military to approaching idle.

The conventional primary flight control hydraulic systems are provided with reservoirs which are of boot strap design. The reservoirs for PC-1 and PC-2 are identical, and are the same size as the A-7 PC-1 unit. System oil pressure enters into, and is felt by, a stationary piston within the reservoir (Figure 2). The large piston and shaft ride over the stationary piston and, because of a piston area ratio of 34.5:1, sets a reservoir return pressure of about 90 psi. The shaft of the large piston is also used as an oil level indicator.

The hydraulic systems are provided with the usual check valves, filters, relief valves, surge dampers-accumulators, etc., required for an operational hydraulic system. In addition, ground check connections are provided for both systems.

The distribution system was sized by flow requirements throughout the distribution system. The pressure and return trunk lines for PC-1 and PC-2 are assumed to be of equal length and are based on an average tubing size of 3/4 inch. Pressure and return lines from the trunk to the actuators are assumed to be equal and 3/8 inch diameter.

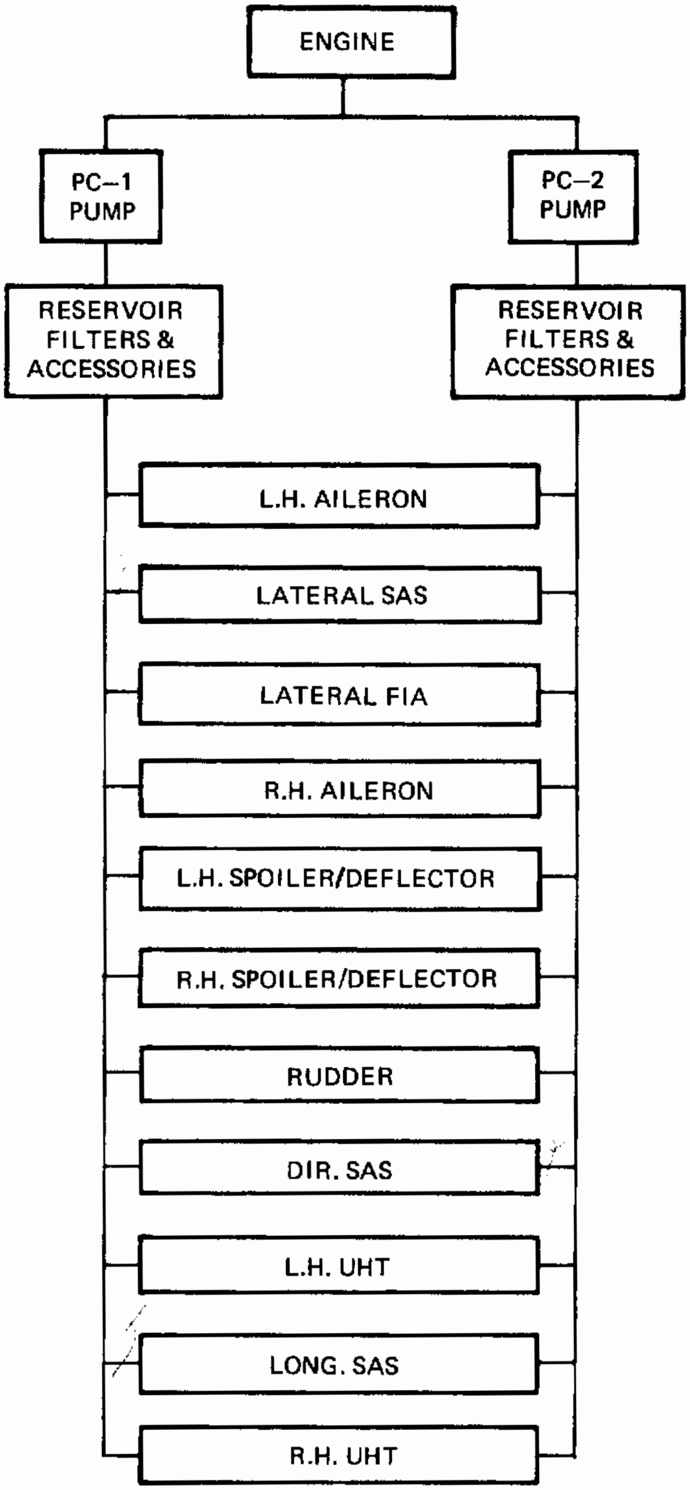


FIGURE 1. CONVENTIONAL HYDRAULIC SYSTEM FOR PRIMARY FLIGHT CONTROL SYSTEM

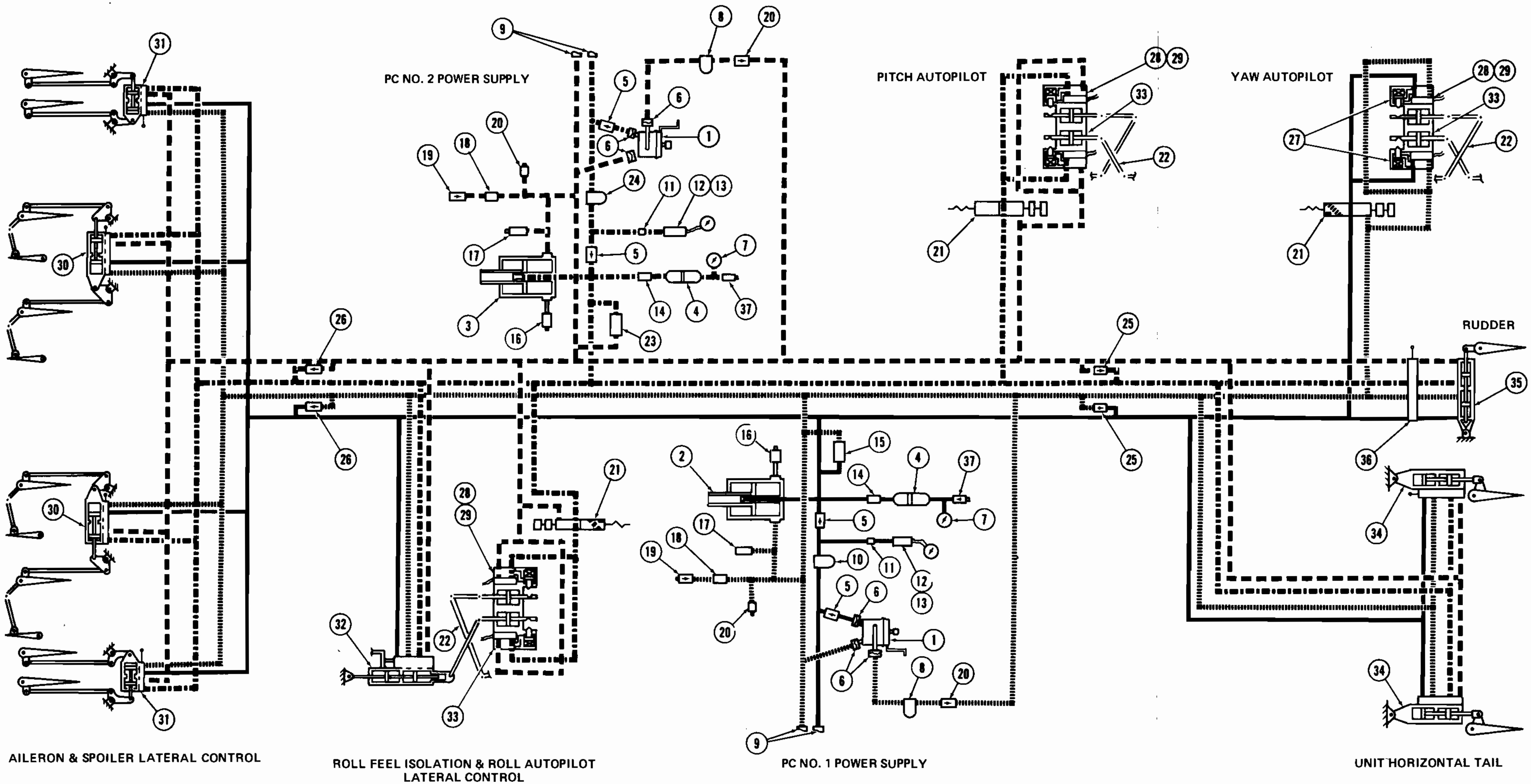


FIGURE 2. PRIMARY FLIGHT CONTROLS HYDRAULIC SCHEMATIC

TABLE I. HYDRAULIC SCHEMATIC COMPONENTS

Legend	Item	Legend	Item
1.	PC-1 & PC-2 Pumps	20.	Drain Valve - Reservoir
2.	PC-1 Reservoir	21.	Three Way, Two Position, Solenoid Valve
3.	PC-2 Reservoir	22.	AFCS Summing Linkage
4.	Surge Damper – Accumulator	23.	Relief Valve – PC-2
5.	Check Valve	24.	System Filter – PC-2
6.	Pump Quick Disconnects	25.	Check Valve
7.	Pressure Gage	26.	Check Valve
8.	Filter – Pump By-Pass	27.	Centering Locks
9.	Ground Test Connections	28.	Servo Valve
10.	System Filter – PC-1	29.	Transducer
11.	Pressure Snubber	30.	Spoiler Actuator
12.	Pressure Transmitter & Switch	31.	Aileron Actuator
13.	Cockpit Pressure Indicator	32.	Feel Isolation Actuator
14.	Restrictor	33.	Autopilot Actuator
15.	Relief Valve PC-1	34.	Horizontal Tail Actuator
16.	Bleed Valve	35.	Rudder Actuator
17.	Reservoir Relief Valve	36.	Rudder Servo Valve
18.	Filter – Reservoir Fill	37.	Filler Valve
19.	Check Valve		

3. Lateral Control System

a. General

The lateral control system for the Conventional Flight Control System is identical to the A-7 system. It consists of control stick and signal linkage, aileron trim and mixing linkage, Feel Isolation Actuator (FIA), autopilot units, and aileron and spoiler/deflector power control actuators located in both wings. Figure 3 is a schematic of the lateral system.

Stick motion strokes the linkage which deflects the closed loop FIA. FIA movement is transmitted to aileron and spoiler/deflector power control actuators.

A simple mechanical mixing linkage allows trim signals to be introduced into the aileron control linkage without disturbing spoiler/deflector position. This linkage also provides a spoiler/deflector deadband which allows 12^{o} of aileron deflection prior to any spoiler/deflector motion. Pilot trim signals are introduced via an electro-mechanical actuator.

Two mechanical springs constitute the lateral feel system. These springs provide stick forces proportional to stick displacement from neutral, and provide the linkage with a centering force.

b. Feel Isolation Actuator (FIA)

The FIA isolates the pilot from the high breakout forces of the spoiler/deflector load limiting links, inertia forces of the control linkage, and AFCS actuator feedback. The FIA is an irreversible dual-tandem hydraulic servo unit powered by PC-1 and PC-2.

c. AFCS Actuator

This actuator consists of two separate electro-hydraulic actuation units, centering mechanism, and follow-up sensors configured in parallel in a single housing. Location of the actuators downstream of the FIA prevents AFCS signal feedback to the pilot's stick. The dual actuator pistons move in equal and opposite directions (Figure 3) with each piston providing one-half of the total required stroke and rate. Net output of the unit is summed in series with the pilot's input. A comparator circuit compares the position of the two actuators and, in the event of the failure of either, supplies a signal which deactivates the hydraulic supply and locks the two units to neutral. This provides the AFCS system with the fail-safe feature.

d. Aileron Power Control Package

The aileron power control package contains a floating irreversible dual tandem actuator. This configuration permits use of a cyliner that is one-half the area of a

FIGURE 3. LATERAL (ROLL) SYSTEM

grounded unit. The result is a long stroke, small diameter cylinder that is best suited for installation in the thin wings of high performance aircraft. Removable four-way tandem hydraulic valves are mounted on the cylinder, and motion of the floating cylinder returns the valve slider to the off position when the surface deflection satisfies the input motion.

e. Spoiler/Deflector Power Control Package

The spoiler/deflector power control package contains an irreversible dual tandem hydraulic actuator. The package is similar in operation to the aileron package in that a floating cylinder, through bell cranks and drag links, drives the spoiler surface much as the aileron surface is driven. However, the spoiler/deflector package contains another set of drag links used to slave the deflector surface to the spoiler surface.

4. Longitudinal Control System

a. General

Longitudinal control for the conventional Flight Control System is identical to the A-7 system (Figure 4). It consists of control stick and linkage; a negative rate spring mechanism; a feel, trim, and autopilot package; linkage load limiting links; hydraulic power control packages; and horizontal stabilizers known collectively as the Unit Horizontal Tail (UHT). Stick motion is transmitted via the linkage system to the feel, trim and autopilot package located in the vertical fin. From this package, the signal linkage is split to the right and left power control valves which direct hydraulic pressure to the power control cylinders which deflect the surfaces.

b. Artificial Feel System

Feel forces are generated in the longitudinal control linkage by a combination of springs, bobweights, and viscous dampers operating collectively.

Forces proportional to normal and pitching accelerations are derived from a dual bobweight system mounted in parallel with the primary linkage system, one forward connected to the pilot's control stick (Figure 4) and one aft in the fin connected to the variable gain input. Under normal acceleration, the bobweights oppose each other; however, the forward bobweight has a larger mass and resultant greater effect. Pitching accelerations produce bobweight forces which are additive at the stick since the bobweights are on opposite sides of the aircraft center of gravity.

The viscous dampers are mounted in parallel with the signal linkage and provide stick forces proportional to rate of stick displacement, as well as providing damping for the bobweight mechanism. Motion of the damper piston pumps hydraulic fluid through a system of spring loaded poppets to yield an approximately linear relationship of force to velocity.

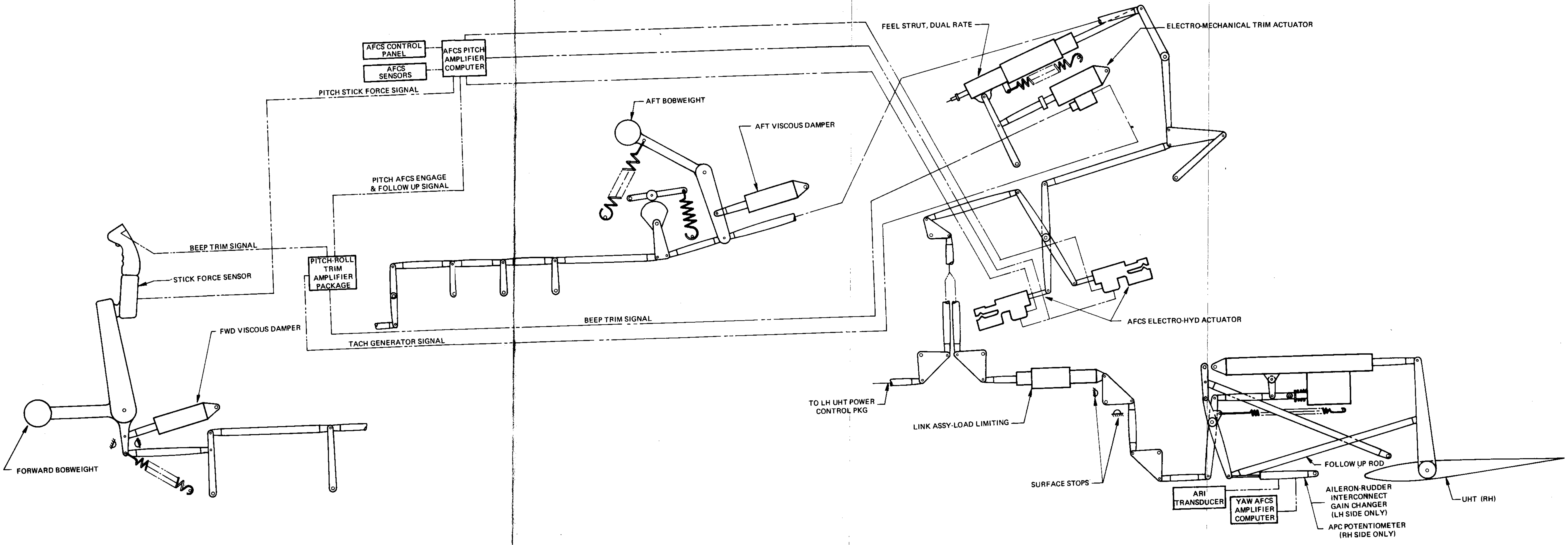


FIGURE 4. LONGITUDINAL (PITCH) SYSTEM

A double acting dual rate spring strut is located in parallel with the control linkage and grounded through the pitch trim actuator. Forces generated in the strut are porportional to stick displacement from trim position. The dual rate spring combines control linkage centering about the trim position with lighter operating forces for larger throws.

c. Trim Actuator

An electro-mechanical actuator provides parallel trim by beep command from the pilot. The extension or retraction of the trim actuator relieves pilot's stick forces for the surface trim position command. The trim actuator responds to the AFCS by acting as a bleed-off device for the AFCS actuators.

d. AFCS Actuator

The AFCS actuator in the longitudinal axis is similar to the unit in the lateral axis.

e. Power Control Package

The two sides of the UHT are powered by identical irreversible power control packages, each containing a dual tandem actuator and two servo valves. The valves are synchronized by interconnecting linkage. The units also have a structural support and corresponding structural feedback member. The structural feedback signal together with the actuator feedback signal are summed with the mechanical input signal to provide the valve displacement.

5. Directional Control System

a. General

The directional system defined for the Conventional Flight Control System is identical to that installed in the A-7 (Figure 5). It consists of rudder pedal and closed loop signal system, clean and landing condition feel springs, AFCS actuator, hydraulic power control, and rudder surface.

b. Feel Springs

The directional feel system is made up of two springs, the landing condition and clean condition feel springs. The landing condition feel spring operates in both landing and clean flight conditions, and is supplemented in the clean condition by the clean condition feel spring, thereby increasing the stick force gradient in the clean condition. The clean condition feel spring system is activated by the flap operation. Pedal neutral position is the same regardless of the trim, since trim is adjusted via the AFCS actuator whose output is in series with pedals.

FIGURE 5. DIRECTIONAL (YAW) SYSTEM

c. AFCS Actuator

The AFCS actuator configuration is identical to the units in the lateral and longitudinal axes. However, the actuators respond to trim and aileron-rudder-interconnect signals as well as AFCS signals. Output of the AFCS actuator is coupled in series with the primary control system by a scissors linkage, which by working against the feel spring, repositions the linkage aft of the actuator to displace the rudder surface via the power control package.

d. Power Control Package

The rudder power control package contains the rudder power control actuator, follow-up link, structural feedback link, and summing linkage. The actuator is a dual tandem unit. The control valve is mounted separately between the rudder power control package support cage and vertical fin structure. Control fluid is ported to the actuator. Structural deflections are monitored by and fed through the structural feedback linkage and summed with the input and follow-up signals to position the valve.

D. DESCRIPTION OF INTEGRATED ACTUATOR PACKAGE SYSTEM

1. General

The IAP system described in this section is designed to perform the same basic functions as the conventional primary flight control system. The IAP system is designed to power the primary flight control surfaces of the A-7 aircraft. As in the case of the conventional system, the aircraft secondary and utility functions are assumed to be supplied by other means.

Whereas the sizing of the conventional system reflects exactly the current installation on the A-7 aircraft, the IAP system sizing was based on actual performance requirements. These requirements are based on overall aircraft performance requirements as generated by A-7 Aerodynamic Section personnel.

2. System Mechanization

The IAP system mechanization for a primary flight control system consists of IAP units supplying control power to the three axes. A total of seven units control the rudder, two UHF surfaces, two aileron surfaces, and two spoiler/deflector surfaces. Figure 6 is a block diagram of the mechanization. The IAP units are of the Duplex configuration, a dual tandem actuator and control valve supplied by dual AC motor-pump units. Inputs to the package which modulate the control valve are both manual and electrical. Manual inputs result from the pilot's deflection of the control stick and pedals. Electrical inputs are supplied by the autopilot and SAS. Also, the trim actuator functions of the conventional system are considered integrated within each hydraulic package, except for the longitudinal system which employs parallel trim (an electro-mechanical trim actuator drives the feel spring ground point). Trim and SAS functions are not included in the spoiler/deflector IAP's.

Electrical power is supplied to the dual motor-pump units from two Constant Speed Drive (CSD) and generator units. The CSD's are assumed to be mounted to an accessory pad and driven by the engine. AC generators, control circuits, and distribution systems constitute the balance of the IAP system.

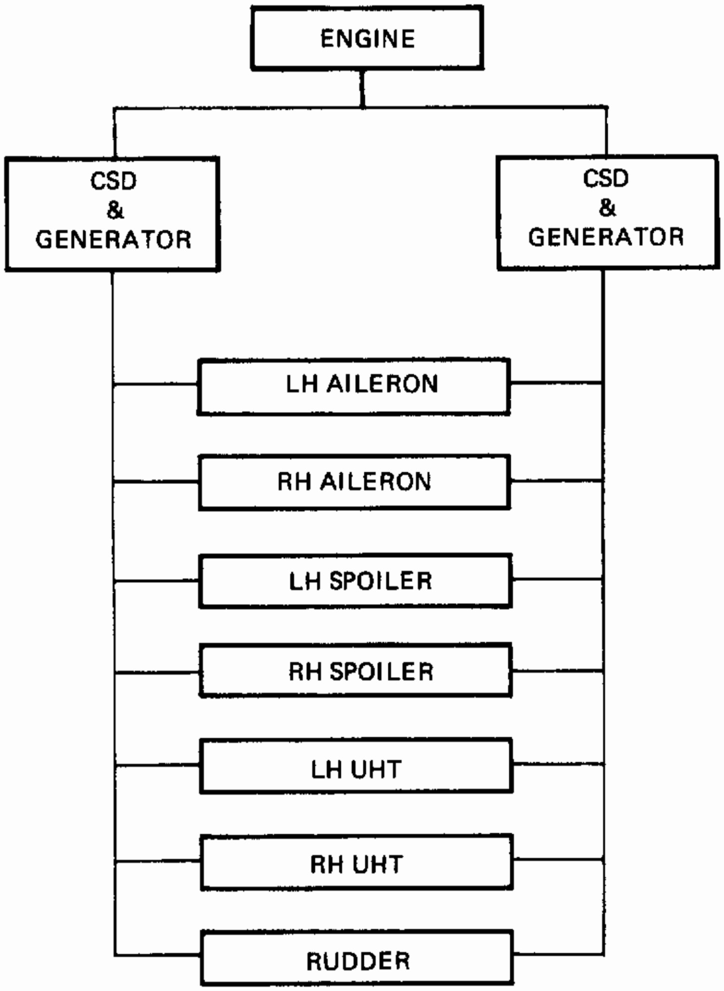


FIGURE 6. IAP CONCEPT FOR PRIMARY FLIGHT CONTROL SYSTEM

3. Integrated Actuator Package

a. General

The IAP contains basically two electric motor driven hydraulic power supplies, a dual tandem actuator, control valve, AFCS-trim actuator, and associated solenoids and hydraulic circuitry. Sizing of the hydraulic power supply is based on an ABEX 2 gpm power supply (P/N PQ8573A-2). This unit contains all the essential elements for the complete power system, i.e., motor, pump, reservoir, filter, relief valve, and fill, bleed, and monitor provisions. These elements are packaged in a modular fashion together with the dual-tandem actuator, servo valve, etc. Figure 7 is a hydraulic schematic of a single channel of the IAP units. Each channel supplies one-half of the dual-tandem actuator. The servo valve controls flow to and from the actuator, and is modulated by pilot's manual input together with the AFCS and trim inputs through the AFCS and trim actuators. The output of the AFCS and trim actuators is added and then summed with the manual input.

b. Pump-Servo Valve Description and Characteristics

A standard 3000 psi pressure variable displacement pumping unit with a soft cut-off feature is defined for the IAP's. The soft cut-off allows a proportional pressure reduction with flow to 25% rated pressure at full flow (Figure 8). A considerable reduction in valve pressure drop and corresponding increase in overall efficiency can be realized with this approach. Increase in overall efficiency is manifested in reduced IAP unit equilibrum temperature. The variable-pressure variable-flow pump characteristics result in a non-linear valve flow-stroke characteristic for a rectangular valve orifice. The relationship for orifice flow is:

$$Q = KA\sqrt{Ps} \text{ or } \frac{Q}{A} = K\sqrt{Ps}$$

Where:

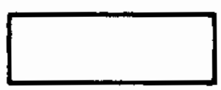
K = orifice constant

Q = flow, gpm

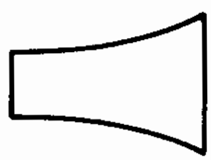
Ps = supply pressure, psi

A = orifice area, in^2

Consequently to obtain a constant flow gain for a linearly varying orifice area the supply pressure, Ps must remain constant, or conversely if Ps varies with flow Q, then the orifice area must increase nonlinearly as the valve is opened.



LINEAR ORIFICE AREA



NONLINEAR ORIFICE AREA

For a supply pressure variation of 4:1 for zero to full valve open the corresponding valve area must vary 2:1.

c. Drive Motors

The motor type chosen for use in the integrated package is a standard 3-phase

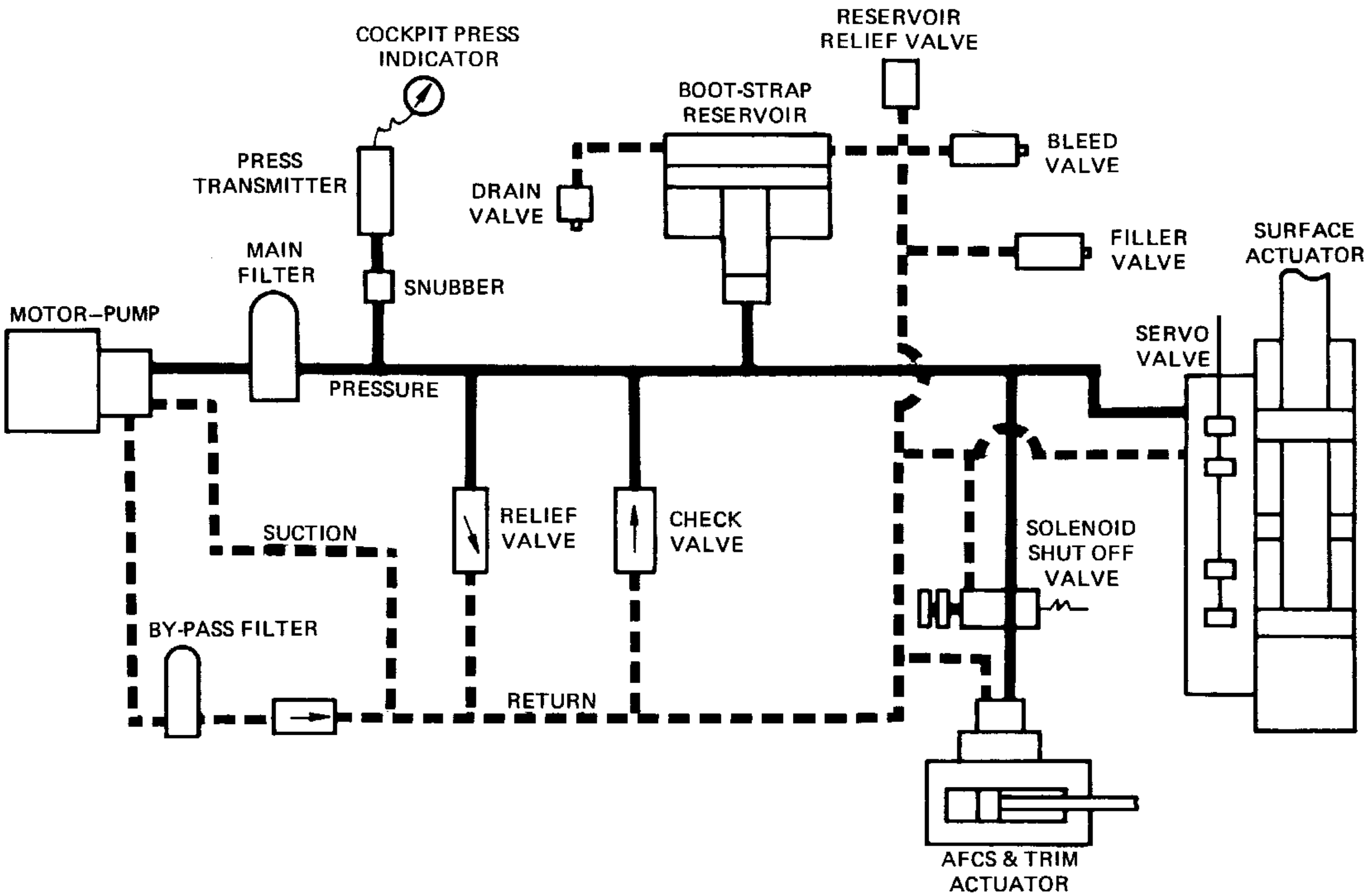


FIGURE 7. HYDRAULIC SYSTEM SCHEMATIC TYPICAL IAP PACKAGE
(SHOWING SINGLE PC CHANNEL)

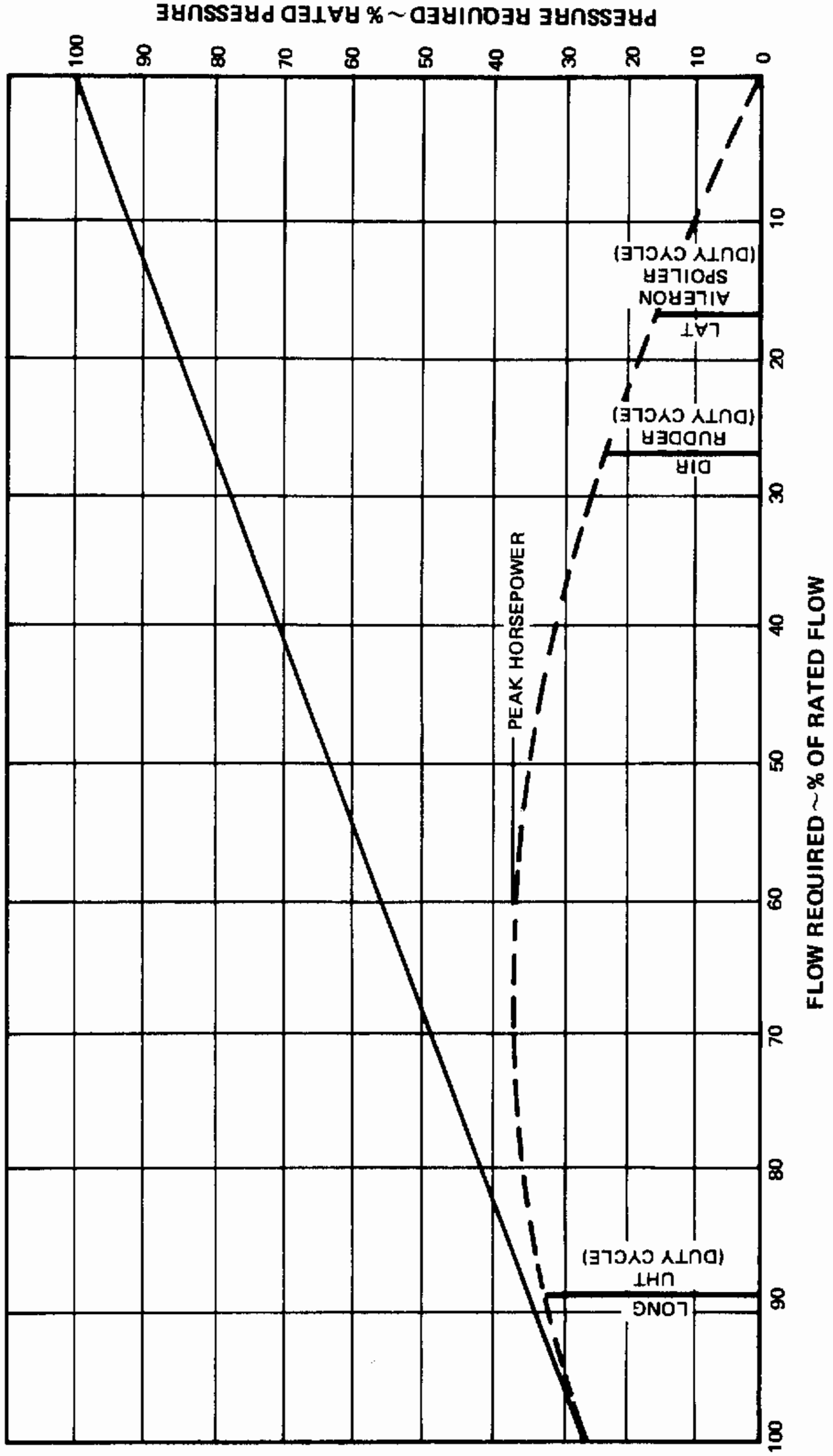


FIGURE 8. SOFT CUT-OFF PUMP CHARACTERISTICS

115 VAC, 400 cycle type designed to achieve flight weight hardware consistent with aircraft equipment reliability and life requirements.

The motors in each case are sized for the duty cycle required by the A-7 control surfaces. These duty cycles are based on statistical F-84F operational flight data reported in NASA Technical Note D-386. The duty cycle for each of the primary control channels for the A-7 is presented in Figures 9, 10, and 11 with the design points located to exceed the duty requirements for 95% of the total time. This allows an over-load situation to exist for 5% of the time, which is more than offset by under loading during the major portion of the duty period.

4. Power Generating System

The power generating system is a dual channel system, each channel being similar to that employed in the present A-7 electrical power supply system. The generation system is shown schematically in Figure 12. Sizing the generator was achieved by extrapolating the present A-7 generation system for IAP requirement.

Each power generating system includes a CSD unit, AC generator, trunk wiring, generator control circuit, and line contactor to the main power bus. The distribution system consists of individual circuit breaker elements and wiring to the motor leads. The auxiliary circuit functions, such as indicator circuits, stabilization, and trim amplifier channels, are not considered here because they are nearly identical to that of the conventional system in major respects affecting overall weight, area, volume, and cost.

For this study, the two generating systems required were considered identical and independent of any utility requirements. In an actual design incorporating integrated systems, consideration may be given to combining one control channel with utility power.

5. Signal Linkage

a. General

The control linkage from the pilot to the input of each of the seven IAP units is essentially the same as for the conventional system defined in III.C above. The difference is primarily in the areas where AFCS and trim functions are combined within the IAP units.

b. Lateral Control System

The IAP lateral control system is shown schematically in Figure 13. The spoiler/deflector IAP's differ from the other units in that they accept only manual inputs from the pilot; the roll AFCS and trim signals are supplied to the ailerons only. The FIA is eliminated from the linkage; no pilot isolation from the AFCS unit is required. A mechanical dwell linkage is provided in each spoiler linkage to prevent motion and force feedback from the down spoiler as the other is actuated,

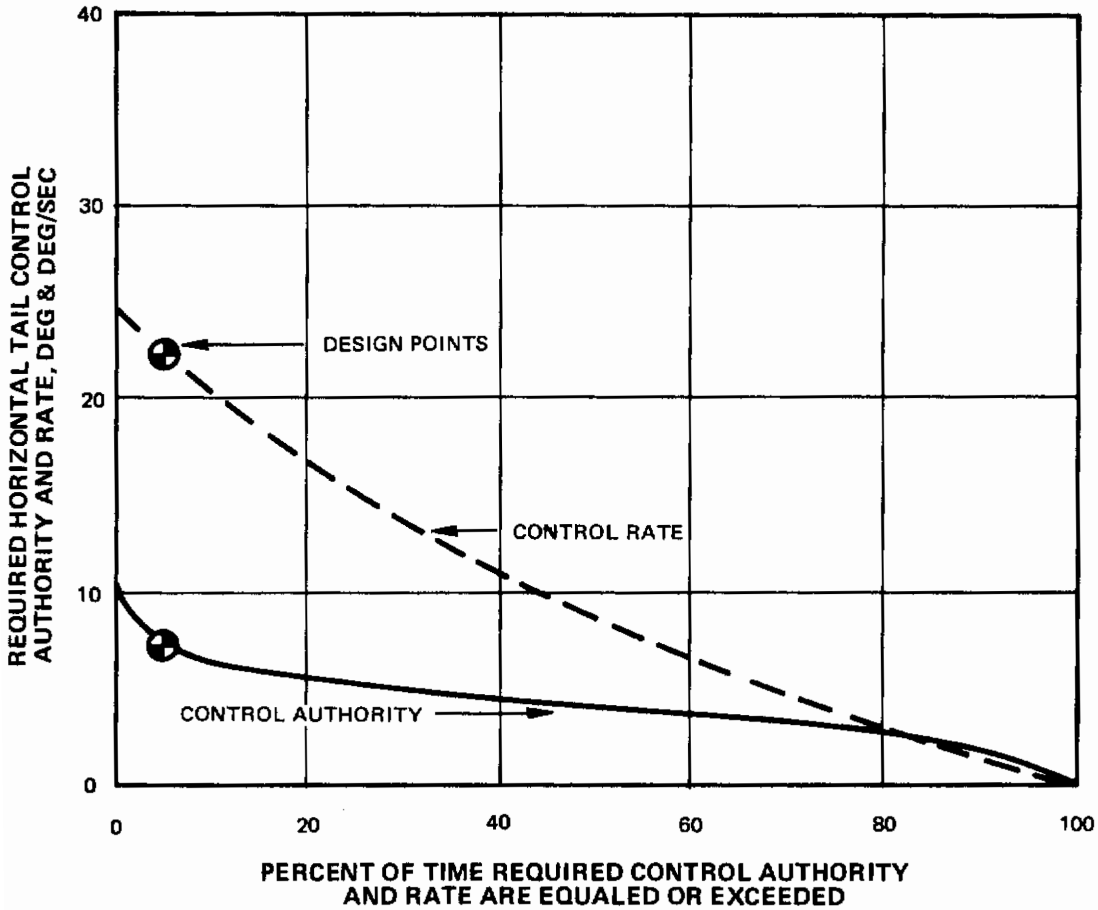


FIGURE 9. LONGITUDINAL CONTROL DUTY CYCLE

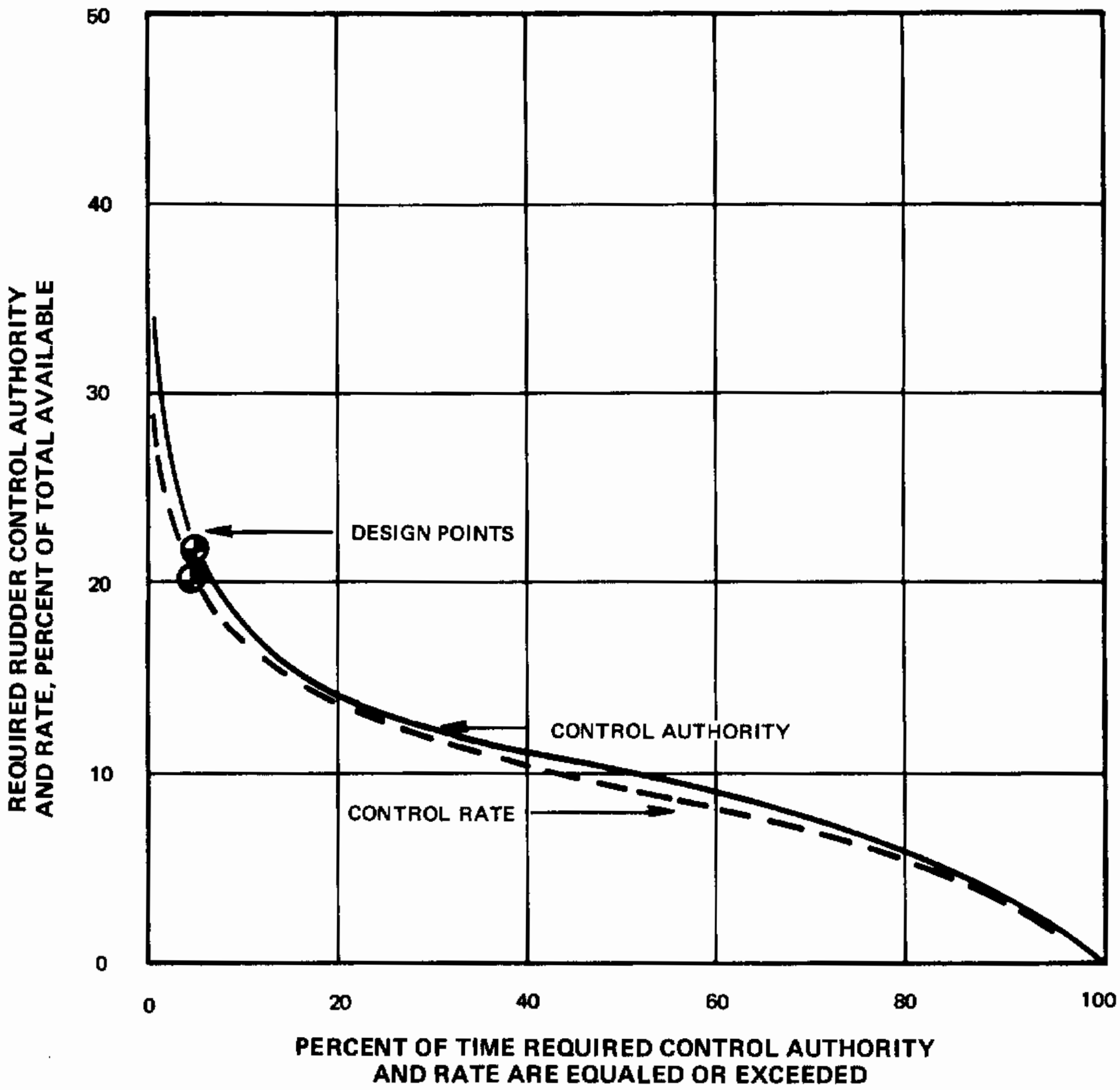


FIGURE 10. DIRECTIONAL CONTROL DUTY CYCLE

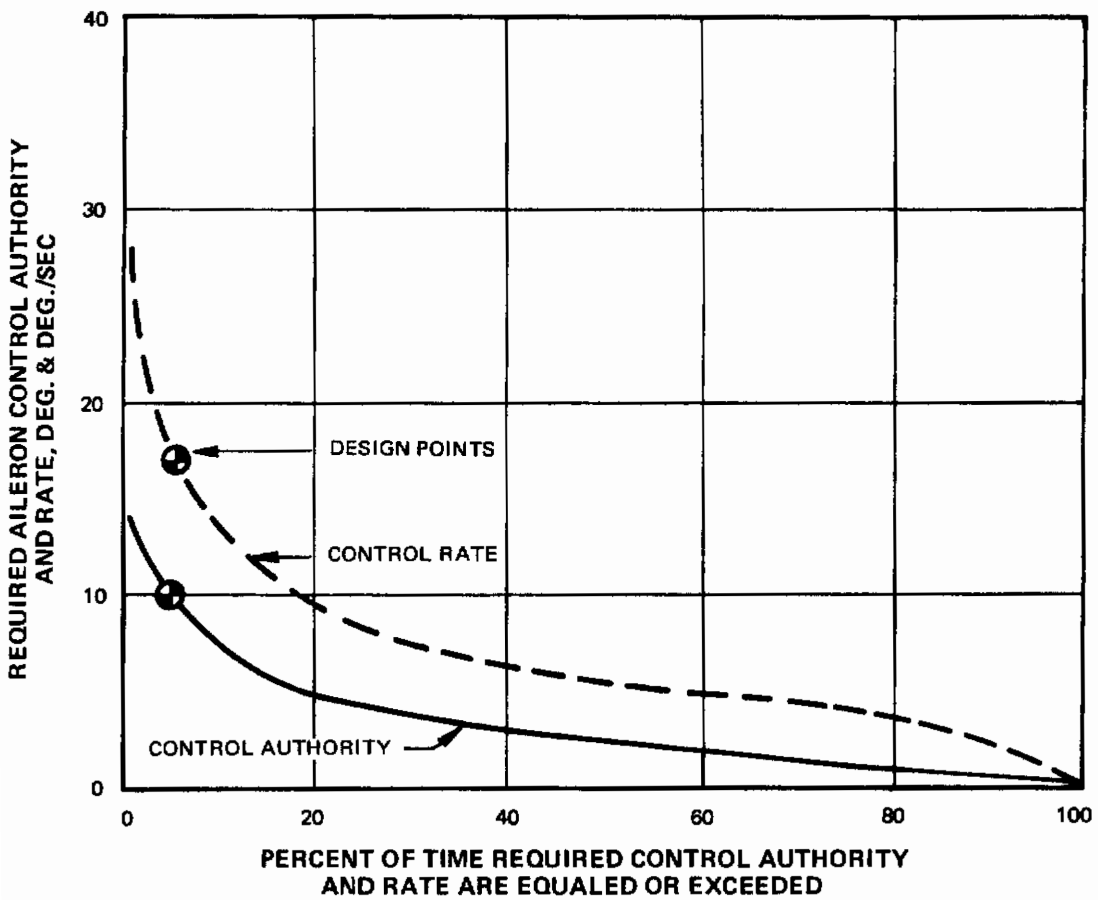


FIGURE 11. LATERAL CONTROL DUTY CYCLE

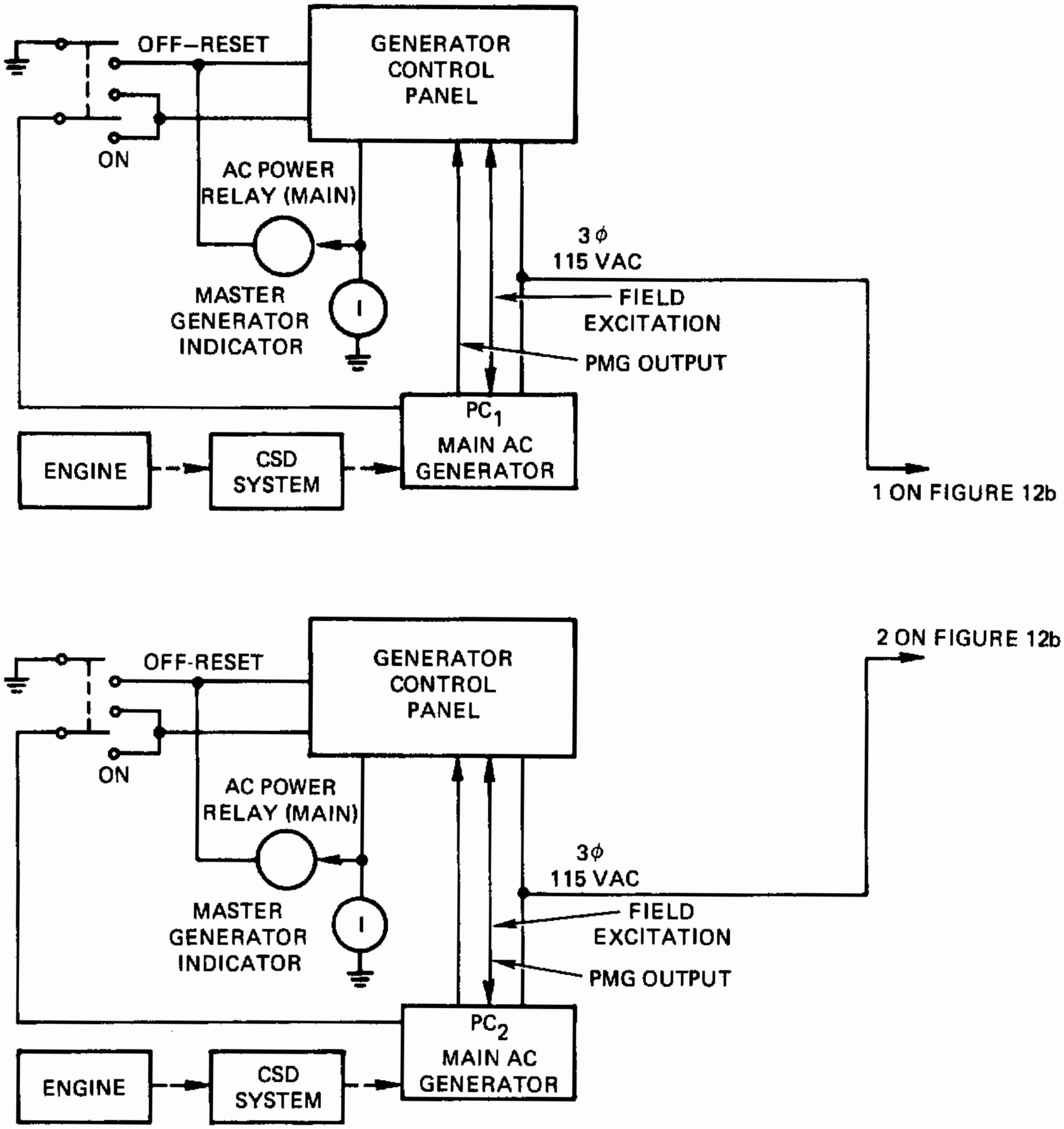


FIGURE 12a. GENERATION SYSTEM, ELECTRICAL SYSTEM SCHEMATIC, INTEGRATED ACTUATOR CONCEPT

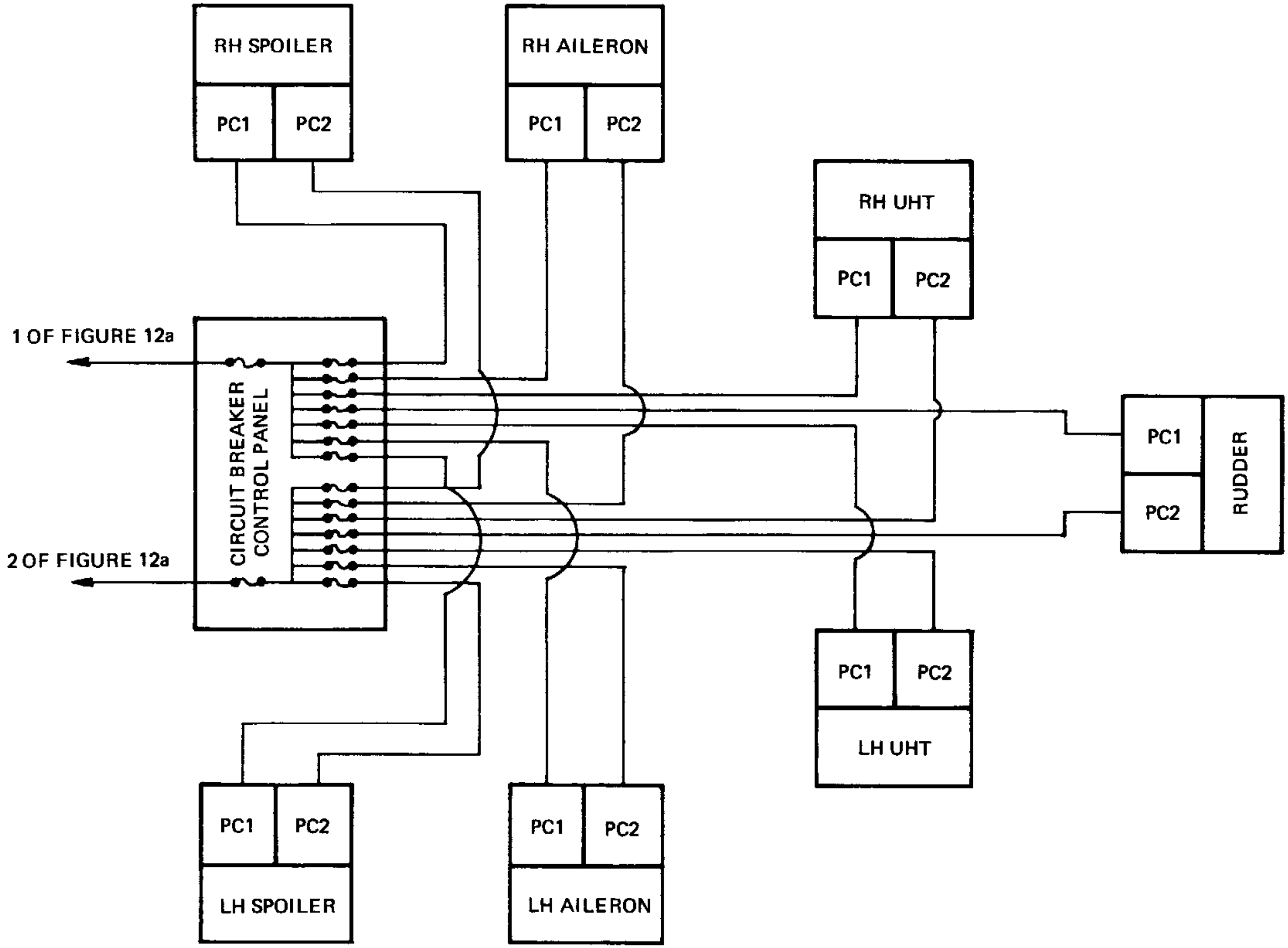


FIGURE 12b. DISTRIBUTION SYSTEM, ELECTRICAL SYSTEM SCHEMATIC, INTEGRATED ACTUATOR CONCEPT

FIGURE 13. LATERAL (ROLL) IAP SYSTEM

permitting a system mechanization without any hydraulic lines. The lateral AFCS actuator is integrated into the aileron IAP. The roll trim actuator is eliminated from the linkage and its function is incorporated into the aileron IAP.

c. Longitudinal Control System

The IAP longitudinal control system (Figure 14) is identical to the conventional system with the exception of the AFCS actuator which is incorporated into the IAP units. Two sets of AFCS actuators are used, one for each IAP. The trim function was not altered as it is a parallel trim system (stick motion with trim) which actuates the linkage feel spring ground and must remain at the feel spring location. However, since the trim actuator is an electro-mechanical unit, no hydraulic power is required.

d. Directional Control System

The IAP directional control system is identical to the conventional system (Figure 15) with the exception of the AFCS actuator which is integrated into the rudder IAP.

6. Packaging and Installation Considerations

The IAP approach involves considerably more packaging problems than the conventional approach. The space usually allocated for hydraulic surface control actuators must be utilized more efficiently to include complete IAP units. The space presently allocated in the A-7 compartments, with the exception of the longitudinal axis, is probably insufficient for integrated packages. Relocation of the packages or significant aircraft structure modification is required. Side-by-side packaging arrangement of the IAP's will minimize any structural changes.

The side-by-side arrangement consists of in-line motor-pump-hydraulic packages mounted side by side under the dual tandem actuator with the servo valves and AFCS trim module package mounted on top of the actuator. This arrangement consists of in-line motor-pump-hydraulic packages mounted side by side under the dual tandem actuator with the servo valves and AFCS trim module package mounted on top of the actuator. This arrangement minimizes the thickness of the total package to allow installation into relatively thin compartments. The UHT compartment is large enough to allow more flexibility in the desired configuration. The IAP units can be mounted side by side and under-slung beneath the actuator, allowing the servo valve and AFCS/trim functions to mount on top of the actuator.

The vertical fin compartment currently used by the A-7 rudder valve and actuator does not allow sufficent room for a compact integrated package design without serious compromises in component sizes and configurations. Structural modifications would be necessary to accommodate a side-by-side mechanization of the IAP unit. Custom packaging may permit the A-7 wing to accommodate the lateral IAPs. Alternate possibilities are to locate the lateral IAP's in the fuselage at the wing roots and route power linkage to the surfaces, or modify the wing structure with local bumps to accept the IAP's.

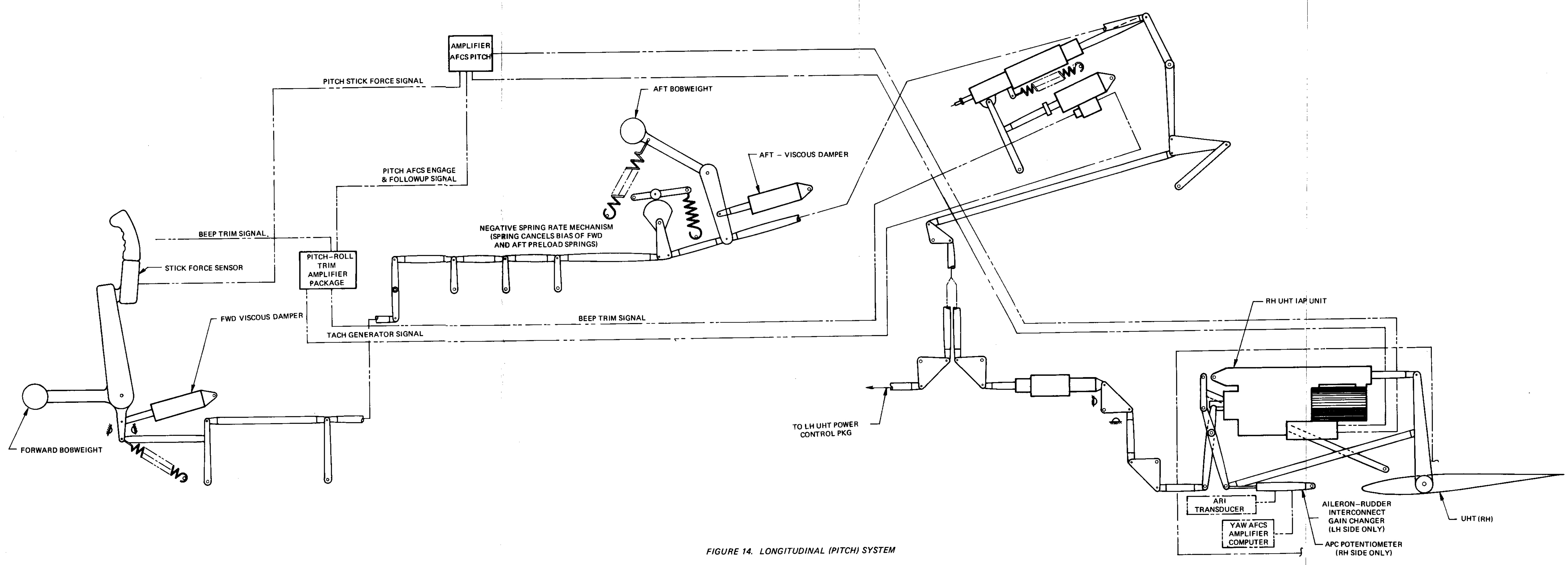


FIGURE 14. LONGITUDINAL (PITCH) SYSTEM

FIGURE 15. DIRECTIONAL (YAW) IAP SYSTEM

E. PARAMETER DEFINITION AND EVALUATION

1. General

The present A-7 conventional flight control system is not necessarily optimized for the A-7 aircraft from the standpoint of size and performance. However, it does represent the typical application of power control packages to present day high performance aircraft. The IAP units were sized to meet the actual aerodynamic requirements.

The general approach to defining the parameters of weight and volume, projected area, C.G. shift, and cost was to define the conventional unit completely with the aid of available documents and then define the IAP system relative to the conventional system. Vulnerability, maintainability, and reliability analysis was conducted on the two systems independently. Methodology and ground rules as well as calculations are included in subsequent sections of this report to illustrate components sizing methods and parameter definition.

2. Conventional System

Weight, volume, vulnerable area, and cost are defined for the conventional system. Appendix I contains a detail itemization of the complete system with sample calculations. The weight, volume, and vulnerable area of most components are taken directly from detail and installation drawings of the A-7. In areas where no details were available, such as parts of the hydraulic systems, conventional estimating methods, tempered by experience, were utilized. Weight, projected area, and volume estimating is based primarily on similarity to existing components. In addition to the components and tubing, weight estimates are made for system fluid and additional miscellaneous components.

Cost of components and, consequently, of the complete system is based on recurring cost (manufacturing or purchase) of the individual components. The cost estimates are based on purchasing history and records, cost estimating relationships derived from these records, and budgetary quotes from vendors. Costs are obtained for all components including pumps, reservoirs, plumbing, accumulators, filters, valves, actuators, control linkage, etc. A summary of the weight, volume area, and cost parameters is shown in Table II.

3. IAP System

a. General

Weight, volume, vulnerable area, and cost for the IAP system are based on actual power required at the control surface as established earlier. Appendix II describes the procedure that was used to size the IAP system. Calculations are provided showing analytically the methods and processes by which the IAP system was defined.

TABLE II. CONVENTIONAL SYSTEM PARAMETERS

SYSTEM	WEIGHT (lbs)	VOLUME (in^3)	PROJECTED AREA (IN^2)	COST (dollars)
Hydraulic System	245	4005.4	6549	25,941
Roll Control System	142	1424.6	1253	16,958
Directional Control System	59	601.05	466	8,412
Pitch Control System	186	1751.6	954	18,031
TOTAL	632	7786	9222	69,342

b. Actuator Definition

The dual tandem actuator size was extrapolated from the A-7 units based on the ratio of required hinge moment to actual hinge moment. Weight, volume, and cost were also established by ratioing to the known A-7 units.

c. Motor-Pump Units

Sizing the motor-pump units was based on delivering the required power for the duty cycles defined in III.D above. Pumping elements on all the Duplex IAP units are of the variable-pressure variable-flow configuration discussed in III.D and are sized accordingly. Aerospace pump vendor data was used to define the weight, volume, area, and cost parameters. For sizing the individual motors, 95% hydraulic pumping efficiency was assumed. The motor parameters were then derived from vendor supplied data.

d. Integrated Hydraulic Circuit

The balance of the IAP units; i.e., reservoirs, accumulators, filters, valves, etc. were defined in terms of weight, volume, area, and cost with the aid of vendor supplied data. A 25 in^3 reservoir was sized for all except the UHT IAP units. The UHT units contain 50 in^3 reservoirs.

e. Signal Linkage

The control linkage is essentially the same as the conventional system with the exception that all AFCS electro-hydraulic actuators are incorporated in the IAP units, the roll system FIA is replaced by a dwell mechanism, and the trim functions of the roll and yaw axes are incorporated into the IAP units. To define the parameter values, the conventional linkage system was taken as a base. Appropriate additions and subtractions were then applied to the base to define the IAP system.

f. Summary

A summary of the weight, volume, projected area, and cost parameters for the IAP Flight Control System are shown in Table III.

TABLE III. IAP SYSTEM PARAMETERS

SYSTEM	WEIGHT (lbs)	VOLUME (in^3)	PROJECTED AREA (in^2)	COST (dollars)
Electro-Hydraulic			3078	
Linkage			2252.36	
TOTAL	825	10,911	5330.36	94,263

4. Vulnerability/Survivability Evaluation

a. Introduction

The evaluation method described below is aimed at determining the relative probability of survival of the two types of primary aircraft flight control systems, conventional and IAP, as they apply to the A-7.

b. Program Description and Ground Rules

The two systems were divided into subsystems and a kill criteria determined for various combinations of subsystem kills. A digital routine was used to evaluate the probability of survival for the entire system, considering a certain number of hits per system. A detailed description of both the survivability methodology and the digital program used is provided in Appendix III.

The degree of control required of the aircraft in terms of ground landing, carrier landing, or bail-out depends on many factors such as pilot proficiency, meteorological conditions, etc. The basic assumptions presented here may be over-simplified. However, both the conventional and IAP systems are subjected to the same assumptions, so the results should be relatively accurate. The assumptions are:

- Sufficient lateral control can be maintained to recover the aircraft with only one of the four lateral surfaces operating with a single hydraulic system.
- The aircraft can be controlled without rudder control.

- The transient involved when one UHT surface control is lost is considered too severe for the pilot to overcome.

- The number of hits in the system is a relative number between the conventional system and the IAP system. In other words, if the vulnerable area of the conventional system is 40 ft^2, and four hits are assigned to the system, and the IAP system vulnerable area is 30 ft^2, then three hits are assigned to the system.

- Hits in the power system, generator, distribution, and IAP have the effect of neutralizing the surface. However, hits in the vulnerable area of the signal linkage will be assumed to drive the surface hard over.

- The effect of a projectile hitting more than one subsystem is not considered.

- Only vulnerable areas in the horizontal plane are considered.

- No subsystem shielding (such as by the engine or fuel cells) is considered.

c. System Description

Figure 16 is a block diagram of the conventional hydraulic system patterned after the A-7 installation. The system includes PC-1 and PC-2 hydraulic pumps, dual tandem actuators for all surfaces, dual SAS units for all axes, and a feel isolation actuator (FIA) in the lateral axis. The system is divided into 15 subsystems as identified in Figure 16.

Figure 17 presents a block diagram description of the signal linkage for the primary flight control system. The block diagram represents both the conventional and IAP systems. However, the vulnerable areas of some of the subsystems are different for the two systems due to removal of the SAS and FIA units. A total of 12 subsystems are used to describe the total linkage system.

Figure 18 shows the IAP system block diagram. The system is divided into 23 subsystems.

Tables IV, V, VI, and VII define the total area and effective area of each subsystem. Tables VIII, IX, and X define the killing criteria for both the conventional and IAP systems.

d. Survivability

Computation of the probability of survival is accomplished by using a binomial expansion and requires as input:

- number of hits on system

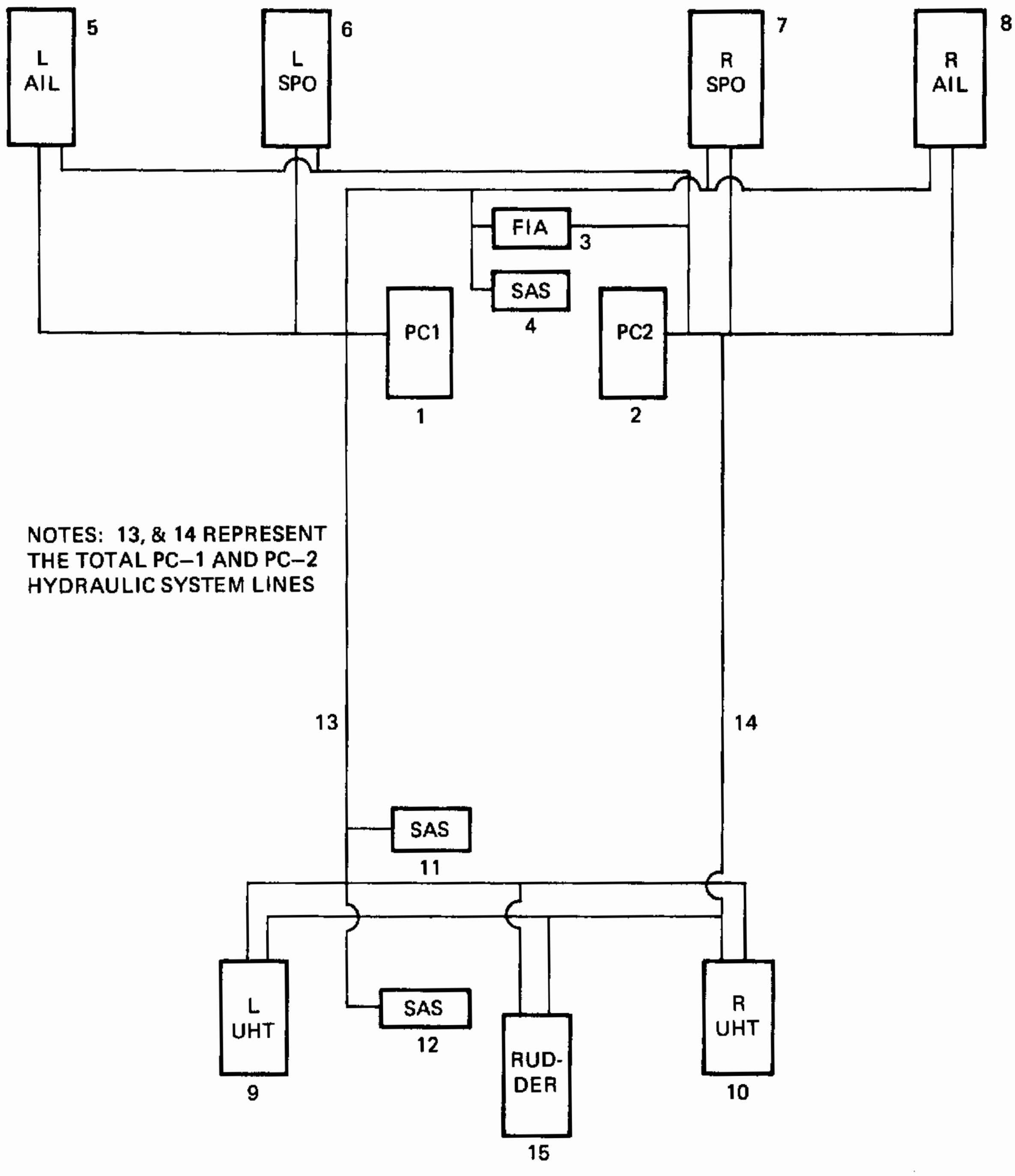


FIGURE 16. CONVENTIONAL HYDRAULIC SYSTEM

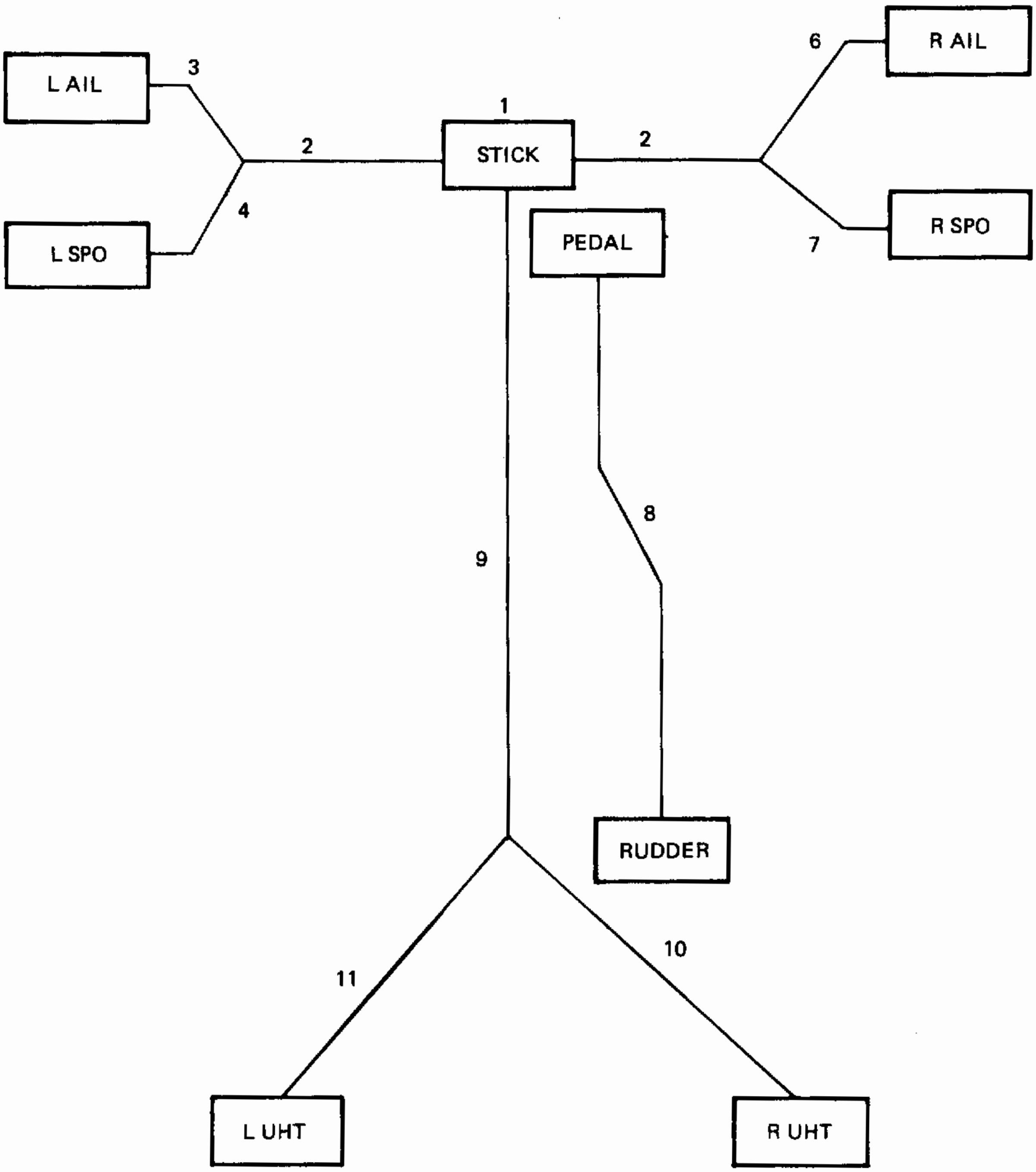
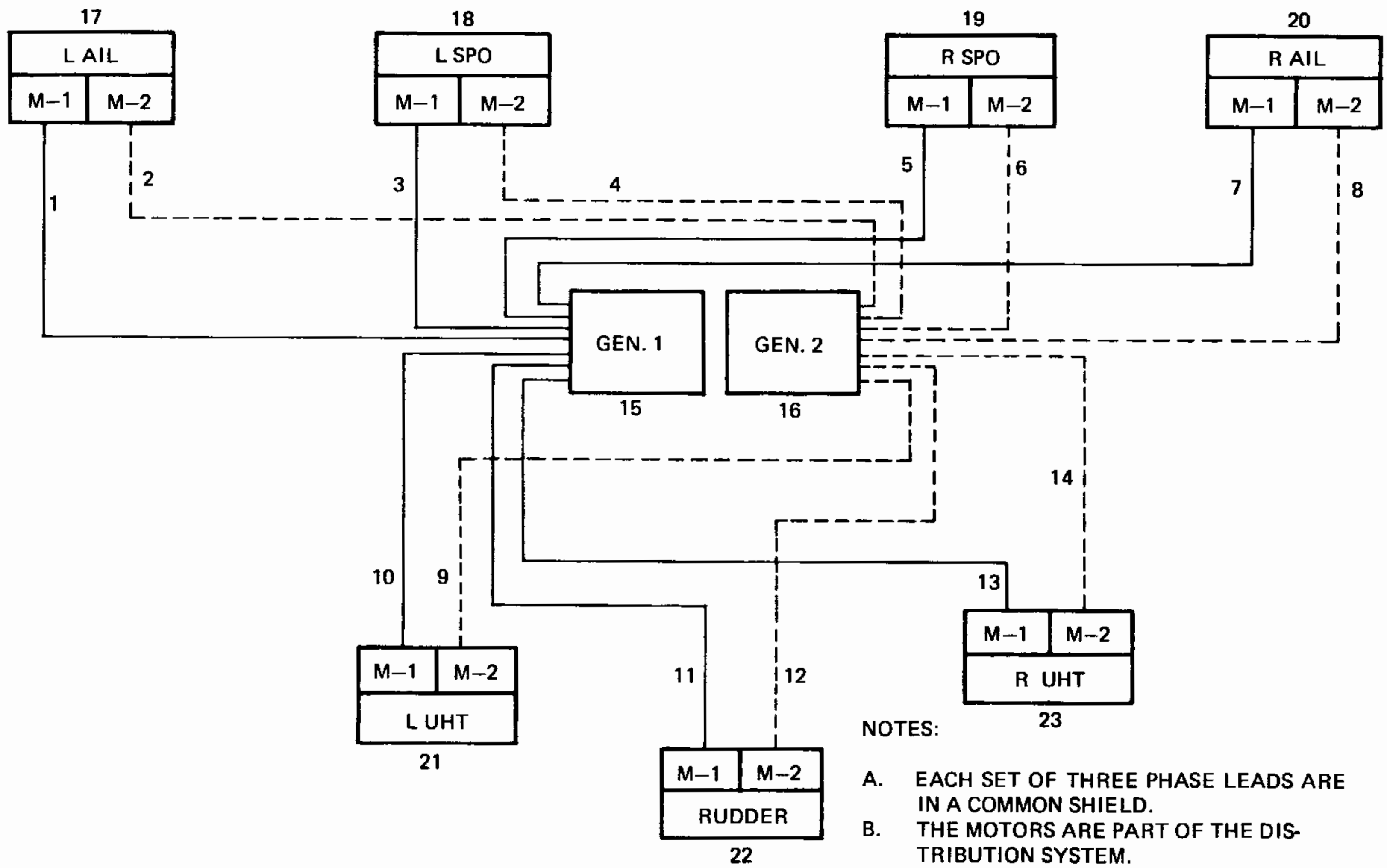


FIGURE 17. SIGNAL LINKAGE BLOCK DIAGRAM



NOTES:

A. EACH SET OF THREE PHASE LEADS ARE IN A COMMON SHIELD.
B. THE MOTORS ARE PART OF THE DISTRIBUTION SYSTEM.
C. THE PUMPS, RESERVOIRS, VALVES, AND ACTUATOR ARE COMBINED.
D. INDIVIDUAL POWER LINES ARE FUSED.
E. FUSES ARE PART OF DISTRIBUTION SYSTEM.
F. MOTORS ARE PART OF DISTRIBUTION SYSTEM.

FIGURE 18. IAP SYSTEM BLOCK DIAGRAM

- probability of subsystem kill
- System kill criteria

To permit a direct comparison of probability of survival for the systems considered, the number of rounds striking the aircraft is chosen as 80.

A sensitivity analysis was performed as described in Appendix III, the results of which are shown in Figure 19 and indicates that the relative positions of the systems will be maintained regardless of the number of aircraft hits.

TABLE IV. IAP LINKAGE SYSTEM VULNERABLE AREA

SUB-SYSTEM NO.	PK	AREA	EFFECTIVE AREA	SUBSYSTEM KILL PROBABILITY
1	0.500	104.930	52.465	0.023293
2	0.500	311.800	155.900	0.069216
3	0.500	233.400	116.700	0.051812
4	0.500	114.860	57.430	0.025498
5	0.500	0.0	0.0	0.0
6	0.500	233.400	116.700	0.051812
7	0.500	114.860	57.430	0.025498
8	0.500	344.040	172.020	0.076373
9	0.500	493.030	246.515	0.109447
10	0.500	103.050	51.525	0.022876
11	0.500	103.050	51.525	0.022876
12	0.500	95.940	47.970	0.021298
PK = SINGLE HIT PROBABILITY OF KILL				

TABLE V. CONVENTIONAL LINKAGE SYSTEM VULNERABLE AREA

SUB-SYSTEM NO.	PK	AREA	EFFECTIVE AREA	SUBSYSTEM KILL PROBABILITY
1	0.500	104.930	52.465	0.023910
2	0.500	333.680	166.840	0.076035
3	0.500	233.400	116.700	0.053185
4	0.500	74.860	37.430	0.017058
5	0.500	0.0	0.0	0.0
6	0.500	233.400	116.700	0.053185
7	0.500	74.860	37.430	0.017058
8	0.500	344.040	172.020	0.078396
9	0.500	493.030	246.515	0.112346
10	0.500	103.050	51.525	0.023482
11	0.500	103.050	51.525	0.023482
12	0.500	95.940	47.970	0.021862

PK = SINGLE HIT PROBABILITY OF KILL

TABLE VI. CONVENTIONAL HYDRAULIC SYSTEM VULNERABLE AREA

SUB-SYSTEM NO.	PK	AREA	EFFECTIVE AREA	SUBSYSTEM KILL PROBABILITY
1	1.000	221.200	221.200	0.031124
2	1.000	221.200	221.200	0.031124
3	0.500	0.0	0.0	0.0
4	1.000	27.000	27.000	0.003799
5	0.500	73.470	36.735	0.005169
6	0.500	73.470	36.735	0.005169
7	0.500	73.470	36.735	0.005169
8	0.500	73.470	36.735	0.005169
9	0.500	57.400	28.700	0.004038
10	0.500	57.400	28.700	0.004038
11	1.000	46.400	46.400	0.006529
12	1.000	48.100	48.100	0.006768
13	1.000	3054.000	3054.000	0.429719
14	1.000	3054.000	3054.000	0.429719
15	0.500	26.390	13.195	0.001857
PK = SINGLE HIT PROBABILITY OF KILL				

TABLE VII. IAP ELECTRO-HYDRAULIC VULNERABLE AREA

SUB-SYSTEM NO.	PK	AREA	EFFECTIVE AREA	SUBSYSTEM KILL PROBABILITY
1	0.800	118.000	94.400	0.030669
2	0.800	118.000	94.400	0.030669
3	0.800	81.000	64.800	0.021053
4	0.800	81.000	64.800	0.021053
5	0.800	81.000	64.800	0.021053
6	0.800	81.000	64.800	0.021053
7	0.800	118.000	94.400	0.030669
8	0.800	118.000	94.400	0.030669
9	0.800	172.000	137.600	0.044704
10	0.800	172.000	137.600	0.044704
11	0.800	61.000	48.800	0.015854
12	0.800	61.000	48.800	0.015854
13	0.800	172.000	137.600	0.044704
14	0.800	172.000	137.600	0.044704
15	0.500	301.000	150.500	0.048895
16	0.500	301.000	150.500	0.048895
17	0.500	182.000	91.000	0.029565
18	0.500	144.000	72.000	0.023392
19	0.500	144.000	72.000	0.023392
20	0.500	182.000	91.000	0.029565
21	0.500	78.500	39.250	0.012752
22	0.500	61.000	30.500	0.009909
23	0.500	78.500	39.250	0.012752
PK = SINGLE HIT PROBABILITY OF KILL				

TABLE VIII. CONVENTIONAL HYDRAULIC SYSTEM KILLING COMBINATIONS

COMBINATIONS	SYSTEMS	GENERAL CLASS.
3	FIA	Actuators
5	Spoiler	Actuators
6	Aileron	Actuators
7	Aileron	Actuators
8	Spoiler	Actuators
9	UHT	Actuators
10	UHT	Actuators
13 & 14	Hyd. Lines	Lines
4 & 14	SAS & Hyd. Lines	SAS
11 & 14	SAS & Hyd. Lines	SAS
12 & 14	SAS & Hyd. Lines	DAS

TABLE IX. IAP SYSTEM KILLING COMBINATIONS

COMBINATIONS	SYSTEMS	GENERAL CLASS.
15 & 16	Generators	Primary
1, 2, 3, 4, 5, 6, 7, 8	Lateral Power Lines	
17, 18, 19, 20	Lateral Systems	
21	UHT System	
23	UHT System	
1, 2, 3, 4, 5, 6, 20	Aileron Power Lines & One Aileron Combined	Ailerons
1, 2, 3, 4, 7, 8, 19		
1, 2, 5, 6, 7, 8, 18		
3, 4, 5, 6 7, 8, 17		
9, 10	UHT Power Lines	UHTs
13, 14		
15, 2, 4, 6, 8 16, 1, 3, 5, 7	Aileron Power Lines & One Generator Comb.	Ailerons
15, 10	UHT Power Lines & One Generator Combined	UHTs
16, 13		
15, 14		
16, 10		

TABLE X. LINKAGE KILLING COMBINATIONS

COMBINATIONS	SYSTEMS	GENERAL CLASS.
1	Stick	Stick
2	Aileron	Aileron
3, 4	Ail. & Spo/Defl.	Aileron
6, 7	Ail. & Spo/Defl.	Aileron
3, 6	Ail. & Spo/Defl.	Aileron
3, 7	Ail. & Spo/Defl.	Aileron
4, 6	Ail. & Spo/Defl.	Aileron
4, 7	Ail. & Spo/Defl.	Aileron
9	Longitudinal	Longitudinal
11	Longitudinal	Longitudinal
10	Longitudinal	Longitudinal
8	DIR.	DIR.
12	Ped.	Ped.

Results of the survivability analysis are shown in Table XI. It should be noted that the conventional and IAP systems have been broken down into power and linkage subsets.

5. Thermal Considerations of IAP System

a. General

The IAP package thermal aspects were evaluated by determining the average temperature based on convection and radiation from the surface of the IAP's. The amount of energy converted from mechanical or hydraulic to thermal power is dependent on the exact pump curve, the particular load, velocity, and neutral leakage.

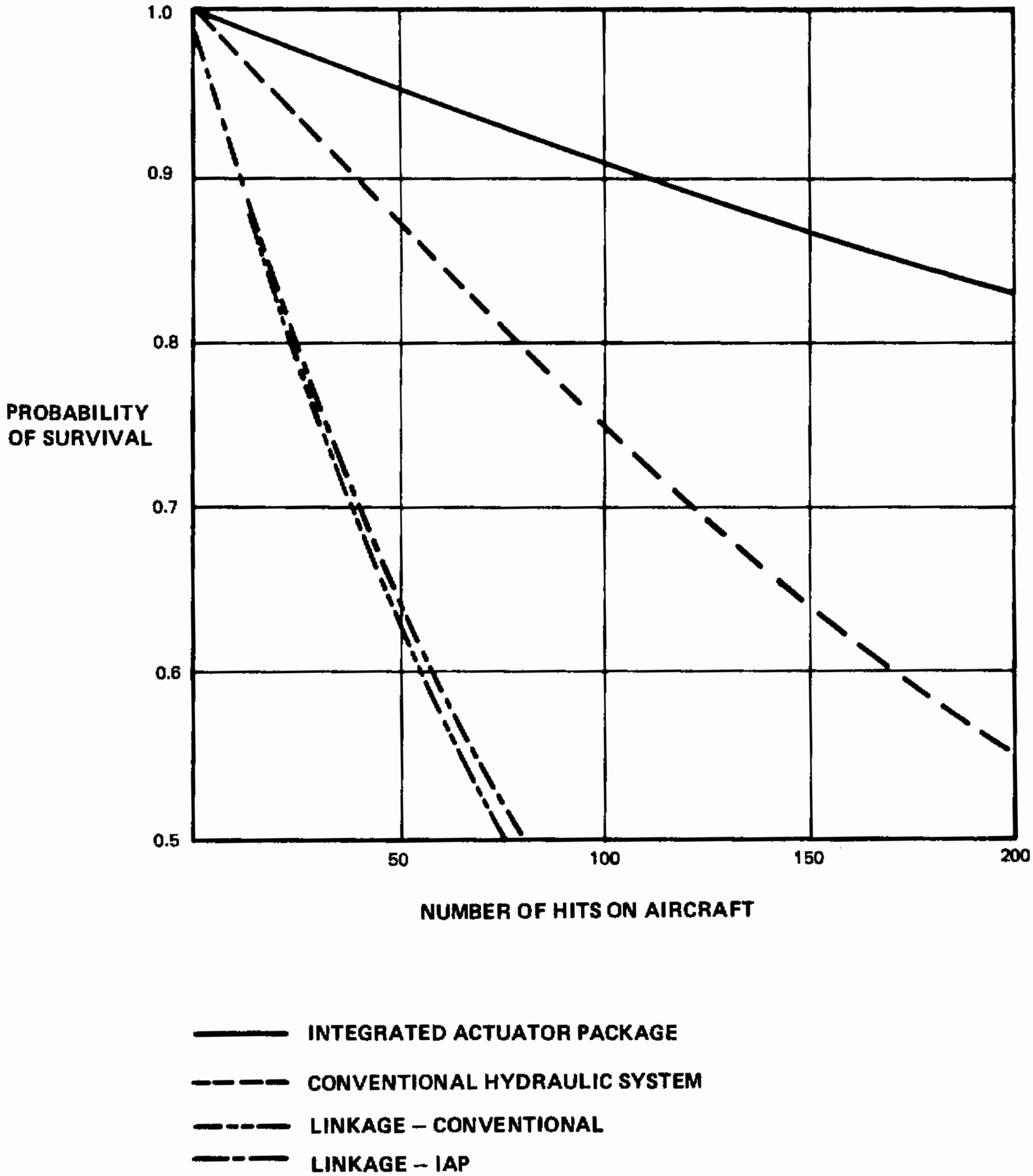


FIGURE 19. SENSITIVITY ANALYSIS

TABLE XI. PROBABILITY OF SURVIVAL SUMMARY
(80 hits on aircraft)

SYSTEM	PROBABILITY OF SURVIVAL
Total Conventional System	0.390286
Total IAP System	0.465915
Conventional Hydraulic System	0.798159
Integrated Actuator Package	0.926665
Linkage – Conventional	0.488983
Linkage – IAP	0.502787

For purposes of stabilized surface temperature, it is assumed that the worst condition is when the surface is stationary at zero hinge moment. The neutral leakage of the control valve, together with pump torque inefficiency constitutes the dissipated energy. The pump inefficiency can be readily determined and is assumed to be 5%. The valve neutral leakage can be controlled to a certain extent dependent on the overall servo analysis, deadband, threshold, etc. Past experience on the design of servo valves indicates that neutral leakage can be maintained at 4% of maximum flow.

The pumping inefficiency is a function of output pressure. The type of pump curve being considered for the IAP system is the variable pressure-variable flow unit previously described.

The electric motors have integral fans which provide sufficient air flow to remove all the steady state energy from the motor so that this study pertains only to hydraulic inefficiencies incurred.

A summary of the inefficiencies of the four different IAP units is shown below. These losses represent the losses for one-half of the Duplex units.

	HYDRAULIC POWER (Horsepower)				TOTAL LOSS Qg
SYSTEM	MAX. AVAIL.	PUMP INEFF.	VALVE LOSS	TOTAL LOSS	(BTU/HR)
Aileron	1.806	.091	.217	.308	790
Spoiler	1.68	.084	.202	.286	730
UTH	2.334	.116	.275	.391	995
Rudder	.827	.041	.100	.141	360

All heat from the IAP units is assumed to be removed by convection and radiation. The surface area of each half of the duplex IAP units available for heat transfer is:

Aileron	258.57 in^2
Spoiler/Deflector	233.4 in^2
UHT	286.37 in^2
Rudder	211.82 in^2

b. Thermal Calculations

The thermal energy balance equation can be stated as follows. The energy generated is equal to the energy removed by convection and radiation. Using conventional thermal symbols and units:

$Q_g = Q_c + Q_R$

Q_g = Hydraulic and mechanical energy converted to heat, BTU/Hr.

$Q_c = hA\ \Delta T$

$h = .25\ (\Delta T)^{.25}$ (for natural convection)

$Q_c = .25A\ (\Delta T)^{1.25} = .25A\ (T_a - T)^{1.25}$

$Q_R = \sigma\ A\ (T^4 - T_a^{\ 4})$ (for unity form factor)

h = Convection heat transfer coefficient, BTU/Hr^{o}F ft^2

A = Surface area, ft^2

T_a = Ambient temperature, ^{o}R

T = IAP surface temperature, ^{o}R

σ = Stefan-Boltzmann Constant, BTU/Hr ft^2 ^{o}R^4

$Q_g = .25A\ (T - T_a)^{1.25} + .171 \times 10^{-8}\ A(T^4 - T_a^{\ 4})$

The solution for T (IAP surface temperature) was obtained by use of a digital routine.

c. Results

The equilibrium surface temperatures of the IAP units is partly a function of the magnitude of surface convection coefficient and existing ambient temperature. The coefficient is a function of air velocity over the surface, which varies from one installation to another. Based on zero air velocity over the surface (natural convection) and an ambient temperature of 158^{o}F, the surface equilibrium temperature of the units is:

Aileron	297.7°F
Spoiler/Deflector	300.0°F
UHT	312.0°F
Rudder	245.7°F

These are the maximum expected equilibrium surface temperatures if the units are located in a stagnant atmosphere. In the event the units are not located in a stagnant atmosphere, and air flow over the units is provided at some velocity, an increasing convection coefficient and decreasing surface temperature results. Figure 20 is a graph of surface temperature as a function of convection heat transfer coefficient.

6. Reliability Evaluation

a. General

A reliability analysis and comparison for the conventional and integrated actuator package (IAP) concept was conducted. The probability of successful operation of each system is based upon a mean mission time of 1.85 hours, and an aircraft total life span of 4000 flight hours. Functional block diagrams are presented in Figures 21, 22, 23, and 24.

The aircraft electrical power system was considered in this analysis since it represents a link in the reliability block diagram. The IAP hydraulic power depends upon the two-generator systems while the conventional hydraulic power system depends solely on two pumps (Figures 21 and 22). The simplified functional block diagram for the conventional flight controls, Figure 21, shows that a single generator and two pumps are required in order to provide a completely functioning system. This generator provides power only for the AFCS and parasitic functions (indicators, etc.) within the conventional system, however. Figure 22, the functional block diagram for the IAP system, shows that the two-generator system provides all of the power functions required for a completely operational system as well as power for the AFCS and indicators. Figures 23 and 24 show that the main difference between the two systems from a simplified functional standpoint is that the conventional system is arranged in series as far as hydraulic lines are concerned, while the IAP system is arranged in parallel for its electrical circuits. If an electrical line in IAP is severed, only 50% power loss to one surface is effected; however, if a single hydraulic line in the conventional system is severed, a 50% power loss to all control surfaces is suffered. Interestingly, if a complete surface actuator is damaged so that both PC-1 and PC-2 are severed in the conventional system, the entire flight control system is inoperative. This is not the case with an IAP actuator (or even an entire single IAP system) where loss of the unit results in loss of control of only a single surface.

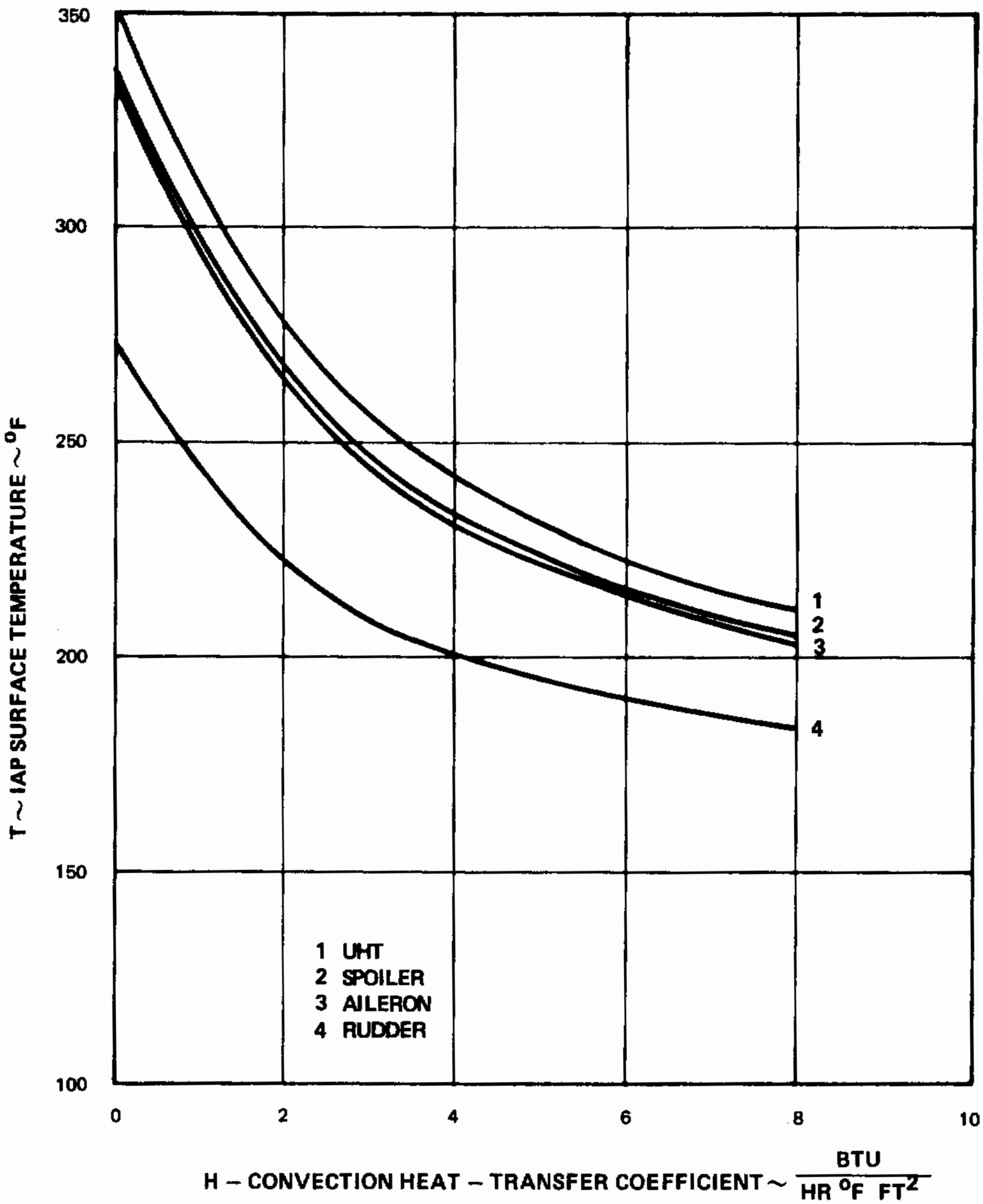


FIGURE 20. IAP SURFACE TEMPERATURE SOLUTION OF

$$Q_g = HA\,(T - T_a) + .171 \times 10^{-8} A\,(T^4 - T_a^4)$$

Q_g ~ *HEAT GENERATED BTU/HR*
A ~ *SURFACE AREA FT^2*
T_a ~ *AMBIENT TEMPERATURE 618.0 °R*

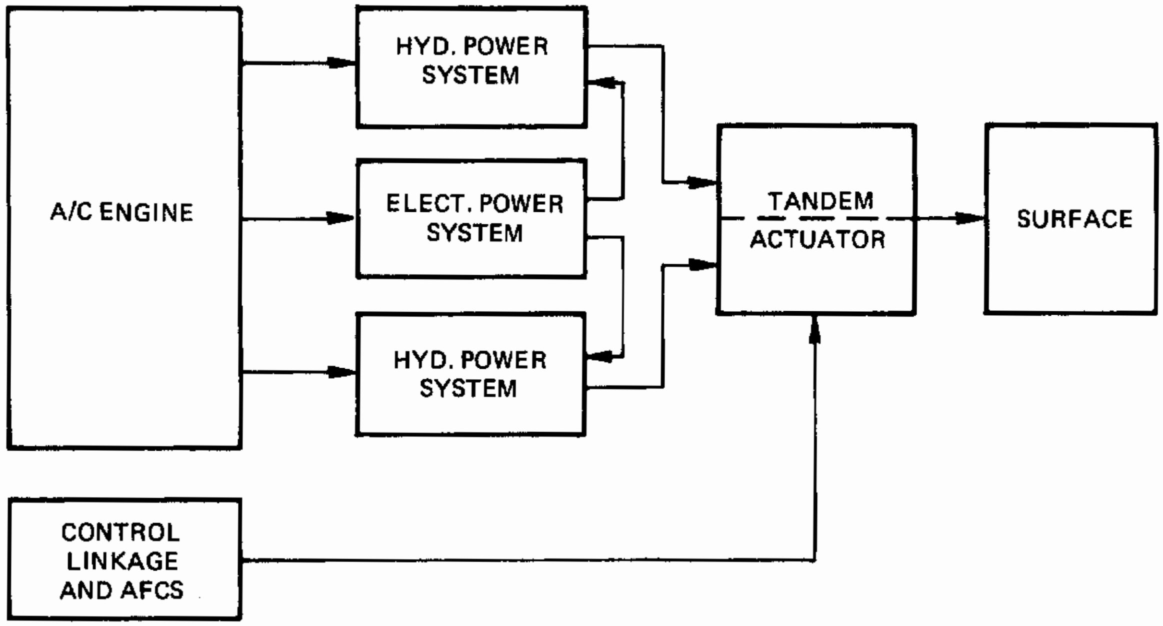


FIGURE 21. CONVENTIONAL SYSTEM BLOCK DIAGRAM (ANY SINGLE SURFACE)

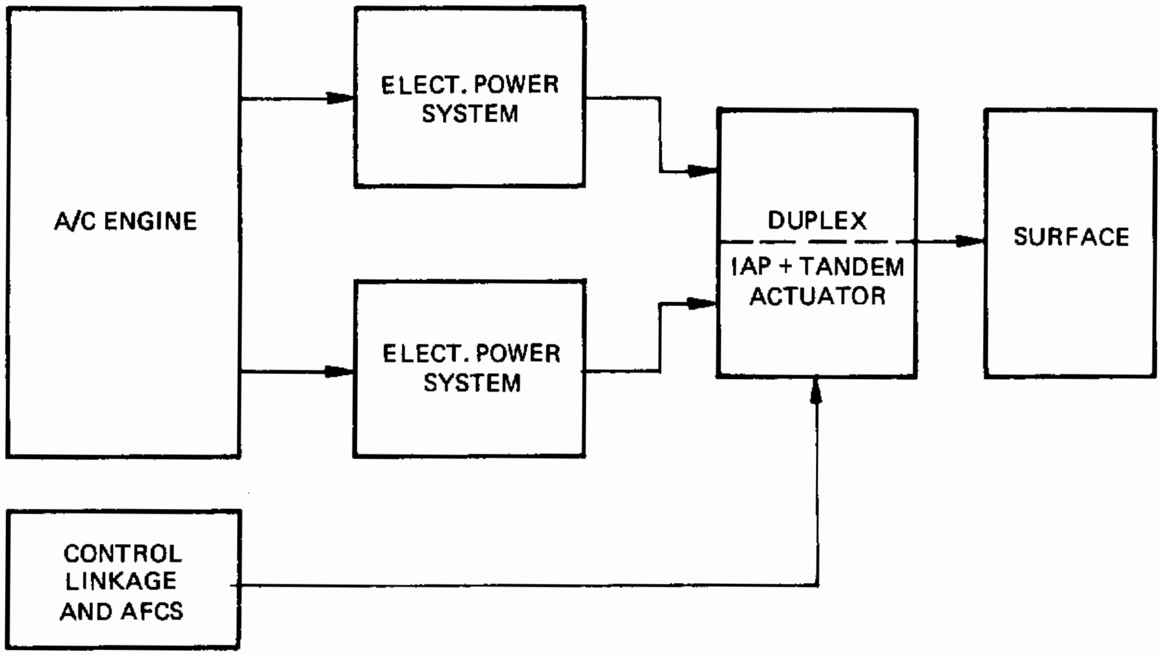


FIGURE 22. IAP SYSTEM BLOCK DIAGRAM (ANY SINGLE SURFACE)

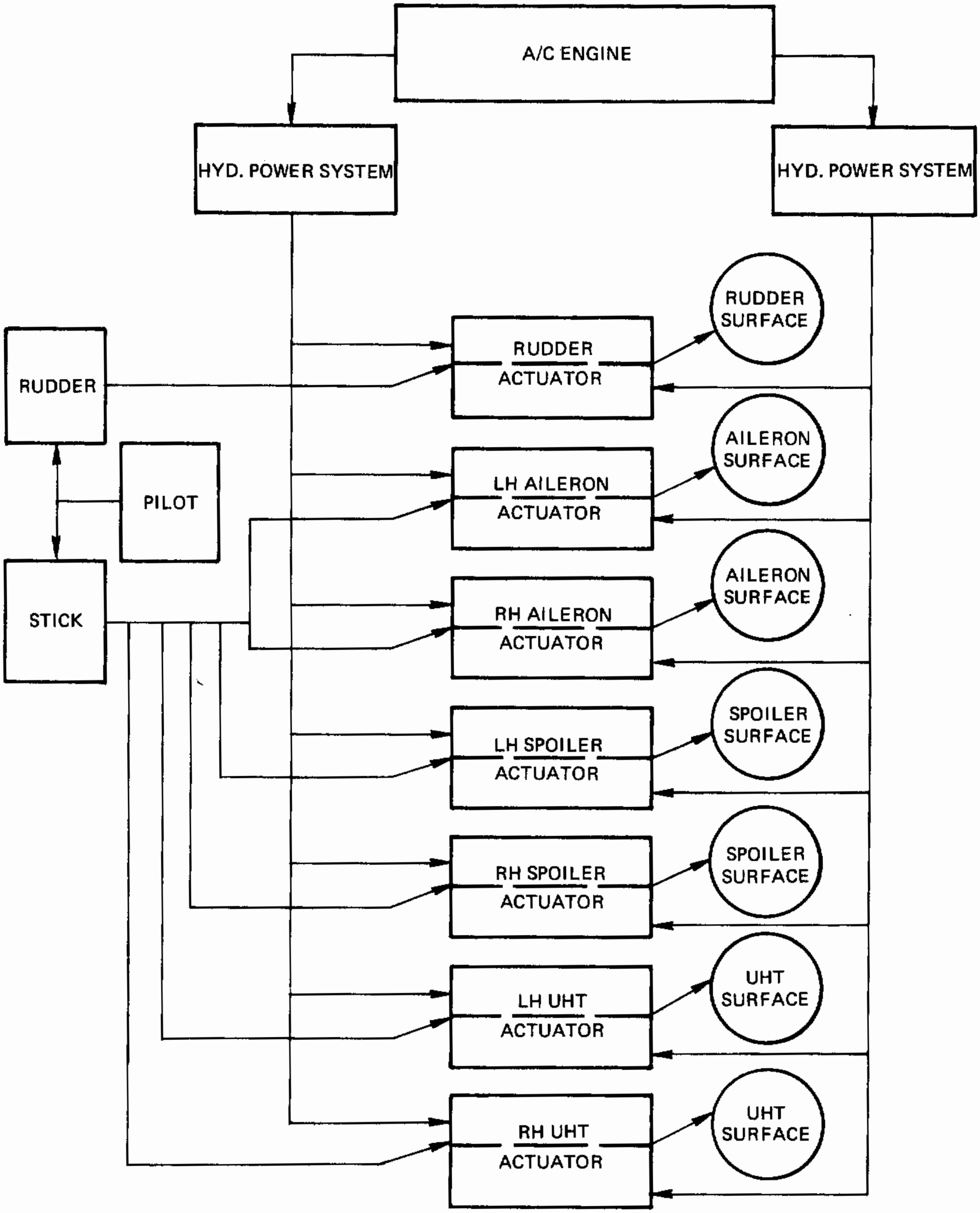


FIGURE 23. CONVENTIONAL SYSTEM FUNCTIONAL BLOCK DIAGRAM

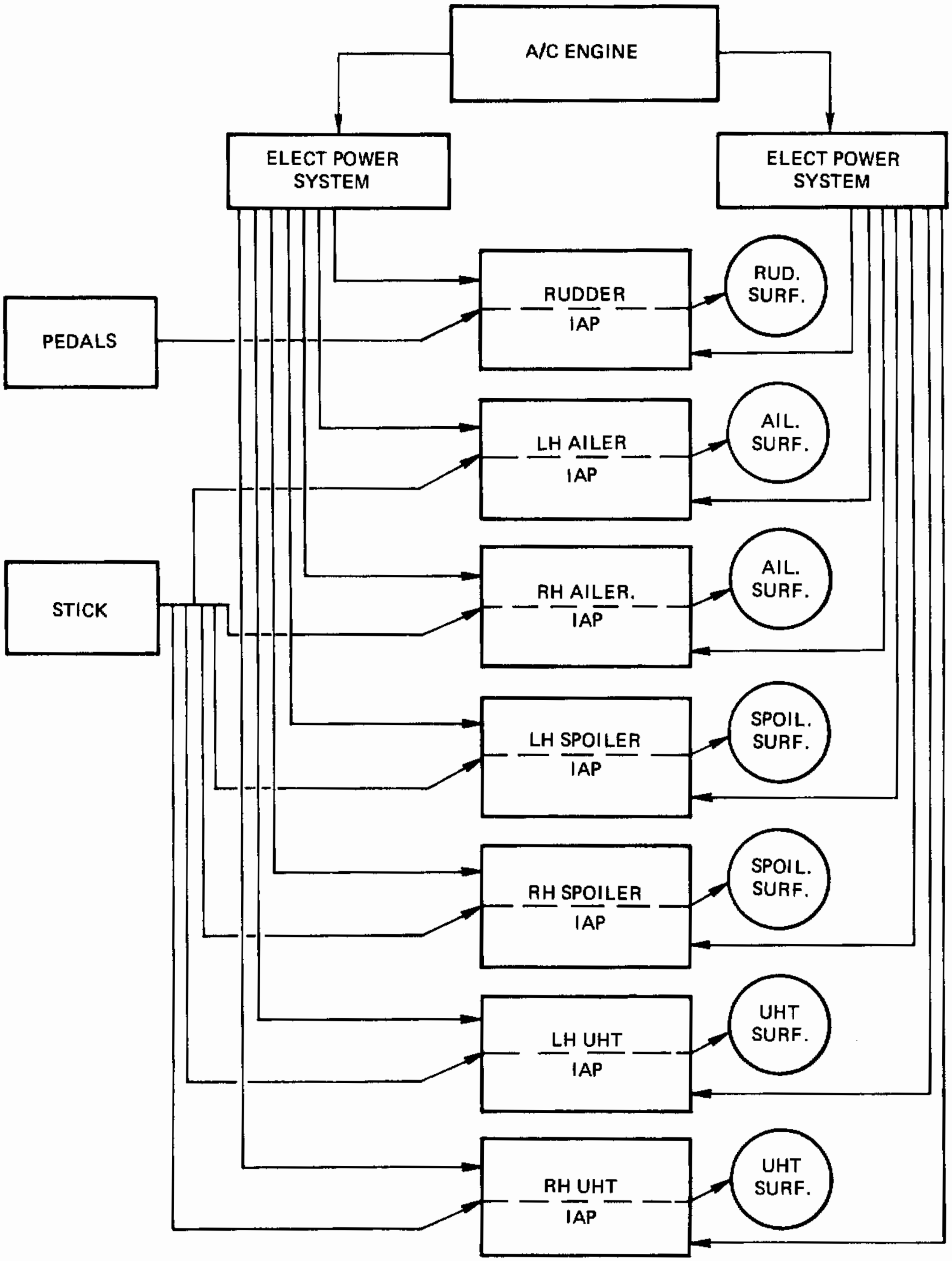


FIGURE 24. IAP SYSTEM FUNCTIONAL BLOCK DIAGRAM

b. Reliability Model

The reliability models and procedures are those standard within MIL-HDBK-217 A. Failure rate data was obtained from LTV failure data, MIL-HDBK-217 A, and FARADA (Failure Rate Data Handbook issued by Bureau of Naval Weapons, SP63-470).

Failure rates due to enemy action, early failures, and wear-out of components are not considered within this report. It is assumed that proper burn-in and/or systematic maintenance procedures will be used.

c. Summarized Results

Failure rates were assigned for each component and the reliability predictions for each subsystem were established. A failure mode and effects comparison was conducted and justification for the component failure rate estimate was established. The probability of success for the flight control systems studied for each concept (conventional and IAP) are itemized below:

Conventional Roll Controls	$R_{RC} = .99404$
IAP Roll Controls	$R_{RI} = .97197$
Conventional Rudder Controls	$R_{YC} = .99712$
IAP Rudder Controls	$R_{YI} = .99129$
Conventional UHT Controls	$R_{UC} = .99792$
IAP-UHT Controls	$R_{UI} = .98696$
Conventional Hydraulic Power (PC-1)	$R_{HC} = .99540$
Typical IAP System	$R_{IP} = .99352$
Electric Power System (Conventional)	$R_{EP} = .99636$
Electric Power System (IAP)	$R_{EI} = .99562$

Using the product law of reliabilities, i.e., $R_{system} = R_1 \cdot R_2 \cdot R_3 \ldots\ldots R_n$, the reliability values of each candidate flight control systems were combined to yield the following comparative values:

Conventional Flight Controls	$R_C = .97645$
IAP Flight Controls	$R_I = .93920$
Conventional System Failure Rate	R_C = 12,834 failures per million operat- ing hours ing hours
IAP System Failure Rate	R_I = 33,969 failures per million operat- ing hours ing hours

d. Summary

Based on the results of this study, the probability of partial failure of the IAP system is somewhat higher than the conventional A-7 system by a factor of 2.64:1. However, this study addresses itself only to the normal operational mode of the systems. Reliability ratio shows a marked improvement when intermediate and emergency operating modes are considered. In addition, the failure modes and effects study showed that the type failures experienced with the IAP are less catastrophic in general than failures with the conventional system. The 2.64:1 failure ratio is simply one indicator; a more complete analysis integrated with the entire aircraft for various operational modes is necessary to obtain the complete reliability picture.

7. Maintainability Evaluation

a. Introduction

A quantitative evaluation was conducted to determine the relative maintainability of the conventional flight control system and the integrated actuator package (IAP) system concept. The evaluation was based on the maintenance indexes described below and does not include common elements such as linkages, bellcranks, etc.

b. Maintenance Index Methods

To evaluate the maintenance characteristics of the conventional and IAP systems, three maintenance indexes were established. System components were then evaluated by these indexes. The indexes selected are frequency of maintenance action, type of maintenance evaluation action, and frequency of component replacement. These are considered to be the most significant indicators of inherent maintainability. The indexes with their weighting factors are presented in Table XII. The weighting factors reflect the degree of maintenance required – the larger the factor, the less desirable is the maintenance characteristic.

To determine the total weighted index for a component, a weighting factor is assigned to the component for indexes No. 1 and No. 2 of Table XII and the value for index No. 3 is determined. The sum of the above three valves is multiplied by the quantity of the particular component in the control system. The result is the Total Weighted Index. The index weighting factors for individual components are based upon LTV data, failure rates obtained in reliability evaluation, and individual experience in working with similar hardware.

c. Evaluation Results and Summary

From the results of this evaluation, it is determined that the IAP system will be less maintainable and will require more maintenance hours than the conventional system. The evaluation indicates that the maintenance action required of the IAP system is a factor of 1.75 (System index totals of 2925 to 5126 from Tables XIII and XIV) times that of the conventional system for the portion of the aircraft evaluated. However, this does not mean that 1.75 times more maintenance hours will be required to support the aircraft.

d. Evaluation Comments

The actual maintenance manhours incurred by an aircraft utilizing an IAP flight control system could probably be reduced to less than the indicated factor of 2.6. This judgement is based on the following considerations:

. Readily accessible and common servicing points should be provided for multi-usage components, i.e., hydraulic reservoirs, that require routine servicing.

. Present day state-of-the-art hardware will result in lower failure rates than presently depicted in available literature.

. Reduced contamination will result from the fewer components in an individual system.

. Modular installations should be provided to allow rapid replacement of an IAP system.

. Quick disconnect type connections should be utilized for servicing line connections to an IAP to facilitate removal.

. External electrical power only will be required of flight control system ground check-out operations.

- Reduced hydraulic power distribution system will result in fewer line leaks and tube assembly replacements.
- Reduced hydraulic plumbing will improve access to other system components.
- Troubleshooting for maintenance evaluation will be improved due to isolation from other systems.

TABLE XII. MAINTENANCE INDEXES

1. FREQUENCY OF MAINTENANCE ACTION

   Weighting Factors:

   0 – No Scheduled Maintenance
   2 – Iran (Inspect and repair as necessary)
   4 – Periodic or Forced Replacement
   6 – Post-Flight or Routine Servicing
   8 – Preflight Inspection

2. TYPE OF MAINTENANCE EVALUATION ACTION

   Weighting Factors:

   0 – General Appearance
   3 – Data Display (Provided on airplane)
   6 – Functional Checkout Without AGE
   9 – Functional Checkout With AGE

3. FREQUENCY OF COMPONENT REPLACEMENT

Definition: Unscheduled Replacement + Scheduled Replacement = Index

$$\text{Unscheduled Replacement} = \left(\frac{FR}{1000}\right)(8000)$$

$$\text{Replacement} = 1 \text{ I}_F \; FR \leqq .125$$

$$\text{Replacement} = 2 \text{ I}_F \; FR > .125 < .250$$

(FR = FAILURE RATE)

Assumptions: 1 Hour Ground Time Per Hour of Flight.

Design Life of A/C = 4000 Flight Hours.

4. Index totals are the summation of all those indexes indicated above.

TABLE XIII. CONVENTIONAL SYSTEM INDEX

SYSTEM COMPONENT	COMPONENT QUANTITY	TOTAL WEIGHTED INDEX
Motor Pump	2	62
Reservoir	2	20
Surge Damper and Accumulator	2	24
Pump Check Valve	8	80
Pump Quick Disconnects	6	18
Pump By-Pass Filter	2	118
System Filter	2	118
Pressure Snubber	2	20
Pressure Transmitter and Switch	2	112
Ground Quick Disconnects	4	4
Cockpit Indication	2	36
Restrictor	2	20
Relief Valve	2	68
Bleed Valve	2	20
Reservoir Relief Valve	2	68
Reservoir Filler Filter	2	112
System Check Valve	2	20
Reservoir Drain Valve	2	14
Pressure Relief Valve	2	68
Fittings and Joint Seals	150	1050
Servo Valve	6	246
Transducer	6	48
Roll Feel Isolation Actuator	2	22
Spoiler Actuator	2	24
Aileron Actuator	2	24
Pitch Autopilot Actuator	2	20
Horizontal Tail Actuator	2	24
Yaw Autopilot Actuator	1	10
Rudder Actuator	1	11
Constant Speed Drive	1	15
Motor Generator	1	27
Current Transformer	6	96
Main Generator Cockpit Panel	1	12
Main AC Contactor	1	8
Generator Control Panel	1	14
Circuit Breaker	15	180
Instrument Transformer	1	14
Relay Diode	1	10
Relay Racks	1	14

TABLE XIII. CONVENTIONAL SYSTEM INDEX (CONT)

SYSTEM COMPONENT	COMPONENT QUANTITY	TOTAL WEIGHTED INDEX
Junction Boxes	1	11
External Power Receptable	1	10
External Power Monitor	1	17
Monitor Reset Switch	1	10
External Power Arming Relay	1	10
	SYSTEM INDEX TOTAL	2925

The major disadvantages, as related to maintainability, inherent in the IAP concept are:

- Increased preventive maintenance, both routine servicing and component replacement.

- Increased corrective maintenance due to component failure.

Both of the above disadvantages are directly attributed to the increased number of components required in the IAP system.

F. SYSTEM RATING

1. General

The system definitions in this Section were generated to permit comparison of the two systems on a logical basis in terms of specific parameters. The parameters evaluated are weight, volume, cost, thermal characteristics, vulnerability/survivability, reliability, and maintainability. These parameters are not the only or total means by which control systems can be defined, but are considered representative of the overall system worthiness.

2. Rating

All parameters were assigned a rating of 10 for the conventional system simply as a base (Table XV). The IAP system rating reflects the difference between the two systems.

Ratings for the parameters of weight, volume, and cost are based on the effect they have on the total aircraft. The weight rating of the IAP system reflects the decreased available payload of the aircraft due to an increased weight of the control system. A current payload capability of 15,500 was assumed. Total cost of the aircraft was taken as $2,500,000.00.

Ratings for probability of survival, reliability, maintainability, and thermal aspects reflect the relative degree of worth or effectiveness of the two control system themselves rather than considering the impact on the parameter for the entire aircraft. Probability of survival reflects the impact of the complete IAP system, electro-hydraulic and mechanical linkage vs. the total conventional control system, hydraulic plus control linkage.

TABLE XIV. IAP SYSTEM INDEX

SYSTEM COMPONENT	COMPONENT QUANTITY	TOTAL WEIGHTED INDEX
Constant Speed Drive	1	15
Motor Generator	1	27
Current Transformer	1	16
Main Generator Cockpit Panel	1	12
Main Ac Contractor	1	8
Generator Control Panel	1	14
Circuit Breaker	5	60
Instrument Transformer	1	14
Relay Diode	1	10
Relay Racks	1	14
Junction Boxes	1	11
External Power Receptacles	1	10
External Power Monitor	1	17
Monitor Reset Switch	1	10
External Power Arming Relay	1	10
Motor Pump	14	434
Reservoir	14	140
System Check Valve	28	280
Pump By-Pass Filter	14	826
Filler Valve	7	70
Main Filter	14	826
Pressure Snubber	7	70
Pressure Transmitter	7	392
Cockpit Pressure Indicator	7	126
Reservoir Drain Valve	14	98
System Relief Valve	7	238
Reservoir Bleed Valve	7	70
Reservoir Relief Valve	7	238
Solenoid Shut-Off Valve	5	70
AFCS and Trim Actuator	5	125
Servo Valve	7	287
Servo Actuator	7	70
Electric Motor	14	518
	SYSTEM INDEX TOTAL	5126

TABLE XV. SYSTEM RATING

PARAMETER	CONVENTIONAL				IAP			
	Value	Rating	Weight	Weighted Rating	Value	Rating	Weight	Weighted Rating
Weight	632.0	10	9	90	825	9.87	9	88.8
Volume	7786 $in.^3$	10	2	20	10,911 in^3	9.9	2	19.8
Cost	$69,342	10	1	10	$94,263	9.9	1	9.9
Probability of Survival (Power System)	.80	10	10	100	.93	28.6	10	286.0
Probability of Survival (Signal System)	.49	10	10	100	.50	10.4	10	104.0
Reliability	.976	10	8	80	.939	9.64	8	78.1
Maintainability	2.6	10	3	30	1.0	3.85	3	11.5
Thermal Aspects	---	10	4	40	300^oF	8.0	4	32.0
TOTAL				470				630.1

The rating reflects the number of survived control systems. In other words, probabilities of survival of .80 and .93 mean that out of 100 systems, 20 and 7 respectively will fail, indicating roughly a 3 to 1 difference.

Reliability ratings reflect the calculated total normal system reliability rather than the number of failures per million operating hours. Reliability ratings would, of course, be different for operating modes other than normal.

Maintainability ratings is indicative of the relative maintenance time required of the two systems.

Rating of the thermal aspects parameter is based on judgement of the ultimate effects of operating at increased temperature and/or providing sufficient artificial cooling for the package. Obviously each individual IAP unit must be evaluated independently to accurately determine its thermal characteristic and assess the impact on the weapons system.

3. Center of Gravity Shift

Weight and C.G. calculations presented in Appendix II indicate a total shift in aircraft longitudinal C.G. location of approximately 1.64 inches. Calculation of C.G. shift in the lateral and vertical directions was not attempted. However, since the components are located symetrically about the lateral axis, no change is anticipated. Change in vertical C.G. location is considered to be minor.

4. Rating Summary

Table XV indicates that although there is a significant difference in the actual weight, volume, and cost of the two systems, when the impact on the overall airplane is considered there is very little difference in the two systems. The primary difference (for the particular rating factors applied) is in vulnerability and reliability areas. As pointed out earlier, the actual reliability number arrived at for the two systems does not tell the complete story. IAP system failures do not have near the catastrophic characteristic that the conventional system failures have. To assess completely the reliability picture of the two systems, a detailed analysis is required for intermediate and emergency modes. Intermediate and emergency modes assessments would be based on probability of completing a mission with various subsystems nonoperative due either to previous failure or small arms hit. Various means by which the IAP system survivability function would be improved are described in 5.a through 5.f below.

5. IAP Upgrading Considerations

a. General

In an attempt to further upgrade the survivability and other aspects of the IAP system, several considerations are offered under the following headings:

- Fly-By-Wire
- Fly-By-Wire Backup Trim System
- Optimized Aerodynamic Duty Cycle
- Servo Pump System for IAP
- Armor Plating

b. Fly-By-Wire

The weak link in the IAP vulnerability/survivability analysis is the mechanical signal linkage. A multiple redundant fly-by-wire scheme would eliminate the vulnerability and other attendant liabilities of the present signal linkage. The poor reliability aspects of a pure fly-by-wire approach necessitates multiple redundancy which is also very desirable from the vulnerability standpoint. Since multiple redundancy is already needed for improved survivability, the fly-by-wire approach is the most logical method to satisfy the sophisticated interfacing required for failure detection and correction.

Although fly-by-wire approaches have not been analyzed specifically for direct comparison to the IAP, this approach obviously offers much in the area of vulnerability due to the multiple redundancy, provided there is not an extremely adverse effect on system reliability. The fly-by-wire approach also offers interesting possibilities in upgrading system survivability by selective armor plate around critical IAP's, generators, etc.

c. Fly-By-Wire Backup Trim System

A very simple adjustment in the redefined signal linkage, allowing the feel spring activity to be transferred to each IAP unit, could yield a considerable benefit in the area of signal linkage vulnerability. With the linkage ground point located at each IAP, a broken linkage would not result in a hard-over condition. The system could still be operated through the full trim range. In this fashion, the trim system can be used as a fly-by-wire backup system with only a small weight and cost penalty.

d. Optimized Aerodynamic Duty Cycle

The duty cycle used for the study appears to be quite conservative, requiring continuous duty capacity of each system to exceed the power actually required 95% of the time. If the duty cycle requirements were based on average power expended, the motor-pump sizes could be decreased considerably. The power generation system could also be resized for approximately the reduced power requirement. It is probable that overloads imposed above the average duty cycle limit would be of sufficiently short duration to not adversely affect motor temperature limits.

e. Servo Pump System for IAP

A significant improvement in efficiency and reduction in neutral leakage of the system (resulting in thermal energy) may be achieved by use of a servo-pump system rather than the more conventional servo-valve system. The servo-pump system delivers flow directly upon demand by means of a compensator mechanism. Several versions of servo-pump system design have been advanced, all with various degrees of merit for particular applications. One shortcoming of servo-pump systems is low stiffness at neutral due to the desired low neutral pressure.

A reversible compensator servo-pump scheme requires an auxiliary pump in the system to keep a minimum pressure available for threshold flow requirements. A uni-directional pump with reversing valving can be pre-set for a minimum pressure as desired for both stiffness and response. The chief advantage of a servo-pump scheme is availability of full power (100% pressure at 100% flow rating) upon demand, while eliminating leakage losses at high pressure when flow demand is low. The servo-pump displacement lever itself can be driven by an electro-mechanical or electro-hydraulic actuator (the compensator mechanism on constant pressure pumps is hydraulically operated).

Using the same duty cycle as for the soft-cutoff pump system used for sizing the IAP, there would be very little difference in sizing between the two systems. Assuming a neutral pressure of 750 psi for the servo-pump system, the quiescent leakage would drop by a factor of approximately four, which would aid significantly in heat dissipation problems in the typical IAP.

f. Armor Plating

A fall-out of the IAP concept is its adaptability to armor plating. The compact integrated units can be readily armor plated to further improve the system survivability. The longitudinal IAP's would benefit the most from armor plating.

SECTION IV

AIRCRAFT SELECTION

A. GENERAL

As part of the contractual effort on the AFCS Integrated Actuator R & D program an investigation was conducted to determine the aircraft best suited as a testbed for the Integrated Actuator concept. It was necessary to complete the aircraft selection task prior to sizing of the IAP's since they are sized and designed to meet the performance requirements of the longitudinal (pitch) control surface of the selected aircraft. In addition, the physical configuration and component arrangement of the Simplex IAP should be interchangeable with the present pitch surface actuator of the selected aircraft with only minor modifications.

The three aircraft originally considered as likely candidates for flight testing the IAP concept are the A-7, F-4 and F-111. These aircraft are currently in the Air Force inventory and are expected to be active for a number of years. Due to different mission requirement and performance characteristics, each candidate has strong and weak points relative to serving as a testbed for flight testing the IAP concept. The F-4 and F-111 are supersonic fighter aircraft and the A-7 is a subsonic attack aircraft. After preliminary review of the three aircraft it was decided to eliminate the subsonic A-7 as a possible testbed candidate and to limit the investigation to the two supersonic vehicles so as to offer the widest potential application of the IAP.

Visits to the plants of the two aircraft suppliers were conducted. Data pertinent to evaluation of the vehicles as IAP testbeds were compiled.

B. OBJECTIVES

1. The general objective of selection task was to determine which one of the two candidate aircraft is best suited as a testbed for the Integrated Actuator Package (IAP) concept as applied to primary flight control surfaces.

2. Specific objectives in order of precedence are:

- Evaluate the two aircraft with respect to how readily the Simplex IAP unit can be adapted to the longitudinal axis.

- How readily can the Duplex IAP with quadruple redundant fly-by-wire (FBW) electrical input commands be adapted to the longitudinal axis.

- How readily can the quadruple redundant Triplex IAP unit or some modifications thereof be adapted to the Longitudinal, lateral (spoilers and aileron), and direction (rudder) channels of the two aircraft.

C. AIRCRAFT SELECTION CONSIDERATIONS

Numerous factors were considered in the selection of the testbed aircraft for the IAP control system. These factors have varying degrees of importance and their relative importance is dependent on individual judgement. In addition, the relative importance of various features varies depending on the application. The factors listed here and discussed below in relation to the candidate aircraft are considered the most important ones for use in selection of the demonstration aircraft.

- Performance requirements for the longitudinal simplex and duplex IAP units.
- Cost of design and fabrication of longitudinal IAP units.
- Cost of aircraft structure, signal linkage, and power system modification.
- Size and weight of IAP units.
- Reliability and design risk involved in IAP application to the aircraft.
- Availability of thermal circuits for cooling IAP's.
- Envelope restrictions for installation of lateral and directional IAP.
- Armor plating of longitudinal axis actuator, weight and volume.
- Availability of aircraft for flight test, and anticipated cost of flight testing effort.

D. DISCUSSION

The discussion which follows is basically a comparison of the F-4 and F-111 relative to the considerations in C above. In some cases the comparison is not as complete and thorough as might be desired due to inavailability of or access to required data.

1. Performance Requirements for the Longitudinal Simplex and Duplex IAP's

The power requirements in terms of flow rate and pressure for emergency conditions are well defined for the F-4 longitudinal and lateral axes. These requirements represent the results of a study to determine the control surface hinge moment and rate capabilities for adequate aircraft control. At the time of this evaluation, limited effort was expended in determining F-111 emergency landing requirements for the horizontal tail actuators. The approximate fully powered (performance) characteristics of the longitudinal surface control actuators for the two aircraft are:

	F-4	F-111
Total extend area	12.08 in^2	35.134 in^2
Total retract area	11.27 in^2	35.134 in^2
Actuator stroke	10.5 in.	6.74 in.
Moment arm	20.0 in.	8.625 in.
Max. surface rate required	25.0 deg./sec.	36.0 deg/sec.
Max. piston velocity required	8.8 in/sec.	5.6 in/sec.
Max. flow rate per system	14 gpm	26 gpm

The extend and retrack areas shown represent the sum of the two tandem piston areas. The F-111 is an equal area actuator while the F-4 actuator areas are slightly unequal.

Figure 25 shows the F-4 requirements for an emergency hydraulic supply suitable for the Simplex unit. These requirements are the result of aerodynamic and hydraulic studies. The graphs of Figure 25 are based on providing 10°/sec surface rate for landing, and 2.5°/sec rate for bailout. This results in a maximum hydraulic horsepower requirement of 1.76 and 1.2 respectively.

Figure 26 depicts the emergency pump requirements for the F-111 which were generated by General Dynamics. These requirements were based on emergency power sufficient to land the F-111 on land. A maximum horsepower of 16.2 is required. Although this may be a conservative requirement, it does indicate that considerably more power is required for the F-111 emergency system than for the F-4 emergency system.

2. Cost of Design and Fabrication of Longitudinal IAP Units

The F-4 stabilator actuator is a dual tandem actuator controlled by a flow control valve which is either mechanically and/or electrically signalled. The actuator performs in three distinct modes, manual, series electrically signalled, and parallel electrically signalled. Since the stability augmentation system is not a safety of flight feature, and since it has limited authority, the electrical channel is not redundant. In the autopilot mode, the pilot is provided with a manual override feature which permits the pilot to recover any electrical failure. For this study an approximate recurring cost of the present actuator was set at $3,000.00.

An auxiliary power supply for emergency control of the stabilator could be integrated into a unit similar to the existing stabilator actuator. Many of the same or similar components such as E/H valve, transducers, solenoids, etc. could be used in the Simplex construction.

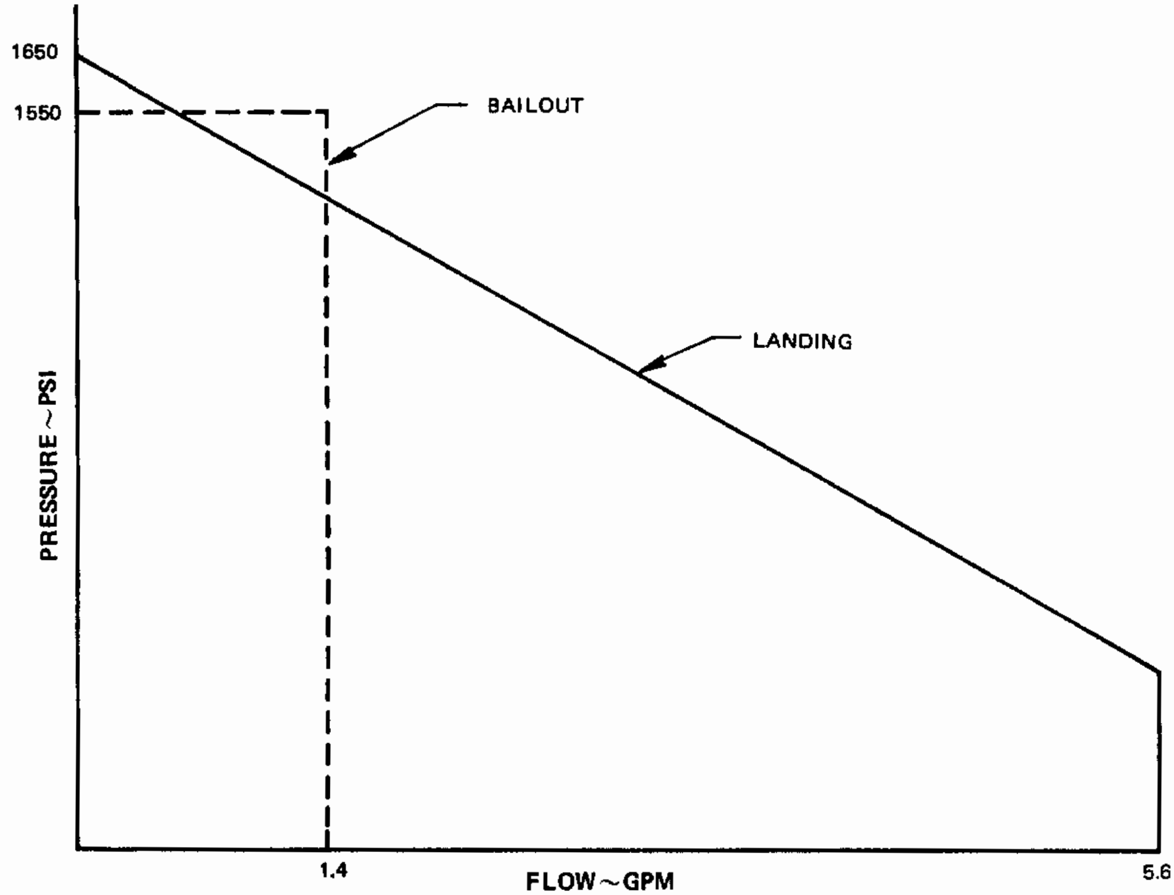


FIGURE 25. F–4 STABILATOR EMERGENCY PUMP REQUIREMENTS

The F-111 horizontal tail actuator is considerably larger and mechanically more complex than the F-4 unit. In addition, two units together with damper servos are required for the two horizontal tail surfaces. Each actuator contains a 2-stage flow control valve arrangement with associated hydraulic failure detection and correction features. The 2-stage control valve provides minimum mechanical input impedance for the longitudinal damper servo. The low impedance requirement is associated with the high gain self-adaptive damper system.

The damper servo is currently a triple redundant unit with hydraulic failure detection and correction features, and is located in the mechanical linkage upstream of the actuator input. The incorporation of the redundant damper feature into the actuator proper increases the complexity and consequently the cost of the actuator considerably. Estimated costs for the present actuator and damper servo are $5,000.00 and $6,000.00 respecitvely.

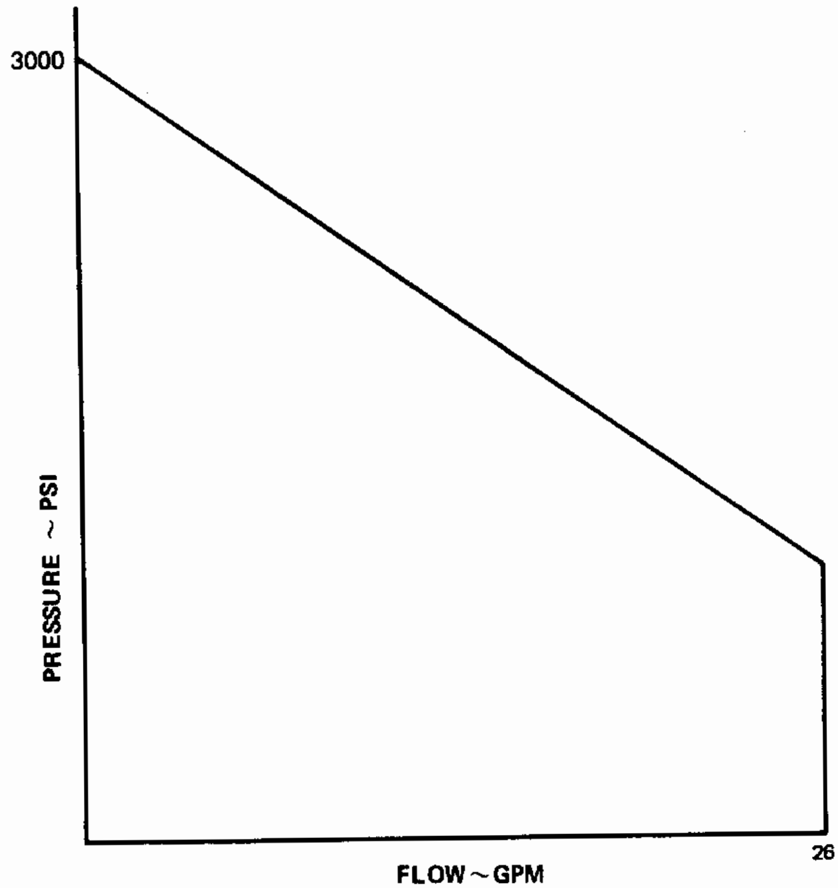


FIGURE 26. F–111 HORIZONTAL TAIL EMERGENCY PUMP REQUIREMENTS

3. Cost of Aircraft Structure, Signal Linkage, and Power System Modification

The F-4 stabilator actuator can be made Simplex or Duplex with very little airframe modification. Depending on the power requirements in the emergency mode, the Simplex unit may be able to be installed through the same access openings as the existing unit. Consequently, it is very probable that the F-4 Simplex unit could be designed as a retrofit unit with minor structural and signal linkage modifications. Also, due to the small amount of electrical power required, no major modification of the electrical system is anticipated. Electrical power would of course be supplied to the motor.

It is also possible that an electrically signalled Duplex can be fitted into the available space around the existing actuator. It would be necessary to remove a permanent panel in the aft fuselage for installation and removal of the actuator.

The F-111 horizontal tail actuator is located in a compartment which is presently filled completely by the actuator and linkage packages. The installation of a Simplex IAP would require significant structural modifications. The extent of the modifications were not completely evaluated. However, it is possible to install an emergency hydraulic supply with necessary reservoir, switching, and valving functions in the compartment just forward of the actuator. The rear section of a fuel saddle tank weighing about 100 lbs. could be removed to permit installation of the motor pump.

The incorporation of the damper servo function in the Simplex unit also requires linkage modification. Again, the extent of this required modification was not evaluated in detail.

4. Size and Weight of the IAP Units

The weight of the existing F-4 stabilator actuator is 38 lbs. The weight of the existing F-111 horizontal tail actuator is approximately 115 lbs. not including the damper servo function. Additional weight required for incorporation of emergency hydraulic supplies into the basic power controls depends on the various functions desired.

5. Reliability and Design Risk Involved in IAP Unit for the Two Aircraft

The F-4 stabilator actuator is a straightforward design. The single electrical signal channel can be easily mechanized. The power supplies for either emergency or full power operation are much smaller than those needed for the F-111. The F-111 horizontal tail actuator, aside from being much larger than the F-4, is also considerably more complex. A low mechanical input impedance is required for proper operation with the high gain electrical system. This requires a small pilot valve controlling a single power supply to minimize the hydraulic flow forces. A significant design risk is involved here.

Incorporation of the triple redundant damper servo in the actuator and the necessary mechanical summing of the manual and electrical signals is untried (whereas it is currently being done on the F-4) and constitutes a certain design risk. Further studies may prove the advisability of leaving the damper servo in the linkage rather than trying to incorporate it into the IAP.

6. Availability of Thermal Circuits for Cooling IAP's

A primary difficulty to be overcome in the implementation of IAP units is dissipation of the energy which is not converted to useful work. This thermal energy results from inefficiencies and the throttling or control action of the hydraulic servo.

The thermal energy must be removed by convection and radiation from the IAP unit to the aircraft structure and the surrounding ambient air. The F-4 has an air scoop located at the leading edge of the vertical stabilizer which ducts air to the general location of the stabilator actuator. This air can be used to cool the IAP by convection from the surface; or the air can be ducted across the affected areas to provide a maximum thermal convection coefficient. Availability of cooling air for the F-111 IAP could not be determined from accessible data.

7. Envelope Restrictions for Installation of Lateral and Directional IAP's

The available space in the vicinity of the F-4 rudder actuator may allow installation of a properly designed Simplex unit. Removal of the lower rudder surface damper and the lower mass balance should permit sufficient space for installation of Duplex and/or Triplex IAP's. The available access doors for installation of the rudder actuator as defined on appropriate structural drawings indicated that there is barely sufficient area for installing the existing actuator. The existing rudder actuator is a single hydraulic (utility) system, single actuator unit. The actuator contains a single electrical channel for summing AFCS and SAS signals in series with the mechanical input. In the event of utility system hydraulic failure, the actuator reverts to a solid link and permits the pilot to manually position the surface.

Direct installation of IAP units in the wing for the spoiler and aileron surfaces on either airplane does not appear to be feasible because of envelope restrictions. Only about six inches are available in the wing to install power supplies. Additional studies are required to optimize and design IAPS for the lateral axis. One approach for powering the lateral control surfaces may be to retain the engine mounted hydraulic pumps for the purpose of providing hydraulic power to the lateral control surfaces. If three aircraft hydraulic power supplies are still required for other hydraulic functions during any flight test program, the lateral control system could be made FBW with central power supplies. The hydraulic power to the lateral systems could be arranged such that loss of lateral control would result only after all three hydraulic systems fail. Removal of the mechanical linkage should provide sufficient space to permit incorporation of quadruply redundant electrical input channels into the actuator.

8. Armor Plating of Longitudinal Axis Actuator

Armor plating of the F-4 longitudinal axis Simplex can be achieved using a "V" shaped section to protect the IAP in the horizontal plane. Length of the plating will be such that it protects the package throughout its stroke. It is estimated that armor plating for the Simplex will weigh 86 lbs. and cost $2,400.00.

The armor plating material is Boron Carbide Composite suitable for protection of 50 calibre projectiles at 2750 ft/sec. The material made by Norton Company is comprised of boron carbide sections laminated to an inner core of woven resinate fiber. The boron carbide absorbs the initial impact of the projectiles and the inner liner stops the resulting fragments.

It is estimated that installation of armor plating on the F-111 horizontal tail Simplex units would be somewhat more difficult and, because of the dual units, would be approximately twice as heavy and costly.

9. Availability of Aircraft for Flight Test and Anticipated Cost of Flight Test

No attempt was made to assess the availability of a representative F-111 or F-4 which could be bailed specifically for a flight test program. However, it is assumed that a copy of either vehicle could be made available for any productive flight test.

The relative cost of flight testing is dependent on the exact nature of the implementation of the IAP and/or FBW concept. For instance, the F-111 can in a sense be considered to have a fly-by-wire lateral control system, although the spoilers are not quadruply redundant and are used only during low speed flight. The quadruply redundant electrical command modules could be detached from the stabilator and rudder actuators. The output of the modules could be a mechanical signal supplied to the existing actuators. This approach deviates considerably from fly-by-wire IAP units but does illustrate the many options available in implementing an IAP flight test program with or without FBW.

E. RECOMMENDATION ON AIRCRAFT SELECTION

Based on the findings discussed in this report, it was recommended that the F-4 aircraft be selected as the IAP test vehicle and that a Simplex unit be designed to meet the installation requirements and limitations of the aircraft.

SECTION V
SIMPLEX ACTUATOR

A. GENERAL

As indicated earlier, the idea of hydraulic control packages having their hydraulic power supplies integrated into the package is not new. However, the system considerations and implementation vary with each application depending on the desired performance. Of course, one feature that is always present in integrated packages and requires proper handling is the thermal aspect. The complete thermal path must be sufficiently low in impedance to permit operation of the package at a temperature acceptable to both the hydraulic oil and the mechanical and electrical components. Magnitude of the thermal problem is generally a function of package size. The package size is dependent on the output power requirement. It is, therefore, imperative that accurate and realistic power requirements for the various control surfaces be established. In a conventional control system, supplied by centrally located pumps, oversizing of power requirements simply results in a system which is heavier than optimum. Use of an existing actuator which is oversized can frequently be justified on the basis of cost. However, oversizing of power requirements for the IAP system is very likely to result in a package that is not only overweight, but either requires a large amount of forced air or liquid cooling of the hydraulic circuit to the point where weight and volume are unacceptable, or results in a stabilized package temperature in excess of tolerable levels.

Guidelines for the Simplex Integrated Actuator were established as follows. The unit should be interchangeable both physically and functionally with the present longitudinal control actuator of the selected aircraft. In addition, the Simplex should contain a complete backup or emergency hydraulic supply to be powered by the aircraft electrical system. The emergency supply should provide hydraulic power in terms of pressure and flow to permit recovery of the aircraft on land if the main hydraulic supplies are disabled. The emergency supply can be brought on-line automatically in the event of a failure or at the pilot's discretion. The Simplex also has the capabilities and contains the functions necessary for monitoring various performance parameters during flight test. The unit is designed with the idea of providing a considerable amount of versatility and flexibility during flight testing. The emergency motor-pump unit was to be a state-of-the-art design. Minor modifications would be permitted to optimize the integration into the Simplex. Duration of the program was insufficient to allow development of a special motor-pump unit for the emergency system.

A subcontract was negotiated with Vickers Aerospace to provide the motor-pump units with a modified valve plate for proper integration into the package. In addition to a variable pressure pump and electric motor, the emergency hydraulic supply comprises a spring loaded reservoir, filters, relief valve, check valves, and associated hydraulic circuit. A pressure switch provides system status information to the pilot.

Selection of the F-4 as the test bed vehicle for testing of the IAP prompted a detail study of its longitudinal control requirements and package envelope constraints. A sub-contract was negotiated with McDonnell Aircraft Company to assist LTV in determining overall requirements during this study. In addition, MCAIR provided guidance in the package design, mock-up evaluation, and evaluation of test procedures and results.

As a result of this study a layout of the package was achieved. A wooden mock-up was built and fitted into aft fuselage of an F-4 made available by MCAIR. The areas of interference were noted. Appropriate design changes were made to the layout to alleviate the areas of interference, and to ensure physical compatibility with the aircraft structure.

The following sections describe details pertinent to the analysis, design, and testing of the Simplex IAP's.

B. SYSTEM DESCRIPTION

1. General

The Simplex Integrated Actuator Package designed and built as part of the AFSC Integrated Actuator R&D Program meets the performance, environmental, and installation requirements of the F-4 stabilator actuator. The Simplex program, as initially defined, consisted of the design and fabrication of one unit. This unit was to be evaluated under simulated aerodynamic load conditions at room ambient. No environmental tests were contemplated. The SCD for the existing F-4 stabilator actuator was used as a guide for the design. However, after several months of effort the program was re-directed to include a total of three units. One unit was to be a qualification test article suitably qualified for flight test. The other two were to be made available to the aircraft manufacturer for flight test as part of the 680J Survivable Flight Contols System Program. The one unit was subjected to an extensive qualification test to simulate the actual life and environmental loads to which the Simplex was to be subjected during flight test. In addition, a detail specification for control of the Simplex actuator design and testing requirements was developed by the Air Force. This document reflected, in addition to the existing actuator requirements, features for the backup hydraulic system integrated into the actuator package, including a motor-pump unit with maximum output flow of 5.5 gpm, and output pressure of 1600 psi using MIL-H-5606 hydraulic fluid. In addition, the unit contains a switching valve to connect the emergency system to the actuator, and to isolate a failed hydraulic system, instrumentation, and actuation capability to effect emergency mode operation, either automatically or at pilot's discretion, and associated hydraulic circuits and components.

The approach taken in evolving the Simplex IAP design was to configure a system which would closely represent an eventual production configuration rather than to use an existing F-4 Stabilator actuator and simply strap on the necessary additional components. It was felt that in this way a better overall feel of the system's eventual merit, in terms of its use in a production program, could be assessed. Particular areas of concern which can be explored and evaluated with the LTV unit are system weight, total volume (both static and swept) required in aircraft, operating temperature and heat dissipation characteristics, and sensitivity to environmental (particularly vibration) inputs. In addition, the Simplex unit reflects current technology in power control actuator design: steel two-piece actuator barrel, rip stopper, minimum flow force control valve, etc.

2. System Description

a. Assembly

The Simplex Integrated Actuator is basically a moving body hydromechanical servo accepting both mechanical and electrical signals which are summed by an internal linkage arrangement. Figures 27, 28 and 29 define the Simplex Actuator Assembly. Figure 27 is a material and parts list of all components in the Simplex. Specification Control Drawings were prepared for the following components:

- Motor-pump
- Filter Element
- Solenoid valve, 4-way
- Solenoid valve, 3-way
- Filter
- Pressure Switch
- Servo Valve
- Linear Transducer
- Check Valve
- Linear Transducer

The actuator was designed to the AFFDL's "Simplex Servo Package Specification, F-4 Stabilator Control" revised 18 April 1969.

Due to the R&D nature of this program and the limited life expected of the unit, arrangement of components is such that they can be easily removed and reinstalled during checkout and shakedown of the unit. Electrical cabling is routed externally to allow easy troubleshooting. Components were selected which could be obtained within the time schedule. Primarily, electrical components with external connectors are used.

b. Hydraulic Circuitry

The Simplex hydraulic circuitry (Figure 30) is arranged to provide all the functions for proper operation of the package. The two ship's hydraulic systems are connected to the package at P_1 and R_1 and at P_2 and R_2. Filter screens (F_1 and F_2) are provided in P_1 and P_2 lines to prevent large contamination particles from entering the package. Check valves are installed in the two inlet lines to prevent back flow from the package in the event of a ruptured pressure line upstream of the package. System 2 powers the aft (lug) end of the actuator, and is controlled by the dual tandem main servo valve. System 1 powers the forward (rod) end of the actuator, and is also controlled by the main servo valve. System 1 also supplies the hydraulic power for the auxiliary actuator, including the electro-hydraulic servo valve, auxiliary ram, unlocking pistons, authority stops, and input linkage locking pistons. The emergency system circuit is arranged so that it can be brought on-line in place of P_1 to power the actuator's rod end. Position of the switching valve

-09	-08	-07	-04	-03	-02	-01	Zone	Item	Code Ident No.	Part or Identifying No.	Nomenclature or Description	Material and Finish or Note or Ref Des
						3				MS35649-284	NUT	
						8				MS35338-99	WASHER, LOCK	
						5				NAS602-6P	SCREW	
						2				CVC846-4	CLAMP	
						3				CVC846-6	CLAMP	
						3				AN938J6	TEE	
2	2	2								AN819-6C	SLEEVE	
2	2	2								AN818-6J	NUT	
						6				AN815-6J	UNION	
						8				AN960PD8L	WASHER	
						1				AN837-6	45° FITTING	
			2							MS21922-4C	SLEEVE	
				2						MS21922-6C	SLEEVE	
						1				AN833-6	90° FITTING	
			2							MS21921-4J	NUT	
				2						MS21921-6J	NUT	
						1				NAS1304-6H	BOLT	
						1				MS21250H06022	BOLT	
						1				NAS1304-8H	BOLT	
						2				NAS1304-5H	BOLT	
						1				MS9390-011	PIN	
						1				AN6289-6C	NUT	
						4				CV784-214-A10	WASHER	
						AR				MS20995C32	LOCKWIRE	
						AR				MS20995C20	LOCKWIRE	
					1					MS24266R-14T7	CONNECTOR	
						AR				GRADE N	PRIMER	MIL-S-22473
						AR				GRADE N	SEALING COMPOUND	MIL-S-22473
						1				NAS1595-8	PACKING	
						18				-6		
						9				-4		
						2				NAS1596-2		
						2				NAS1593-338		
						1				-228		
						1				-220		
						2				-218		
						2				-147		
						1				-129		
						2				-127		
						3				NAS1593-125	PACKING	

QUANTITY REQUIRED — LIST OF MATERIAL OR PARTS LIST

-09	-08	-07	-04	-03	-02	-01	Zone	Item	Code Ident No.	Part or Identifying No.	Nomenclature or Description	Material and Finish or Note or Ref Des
						1				NAS1593-124	PACKING	
						4				-121		
						2				-117		
						3				-114		
						2				-113		
						1				-111		
						6				-028		
						2				-025		
						2				-120		
						2				-021		
						14				-020		
						3				-019		
						1				-018		
						9				-017		
						13				-016		
						4				-015		
						15				-013		
						6				-012		
						2				-010		
						2				NAS1593-004	PACKING	
						4				NAS1352C06H8	CAP SCREW	
						4				NAS1352C06H6		
						4				NAS1352C04H6		
						8				NAS1351C3H8	CAP SCREW	
						2				NAS1304-1	BOLT	
						1				NAS1003-5	BOLT	
						1				NAS557-10C	GROMMET	
						1				NAS613-16	WASHER-ROD END LOCKING	
						1				NAS509-17	NUT	
						2				NAS464P3-12	BOLT	
						2				MS20002-4	WASHER	
						2				MS28774-147	RETAINER	
						1				-129		
						1				-127		
						2				-124		
						4				-117		
						5				-114		
						1				-111		
						6				-028		
						1				-025		
						2				-021		
						10				-020		
						3				-019		
						3				-017		
						24				-016		
						15				-013		
						4				-010		
						2				MS28774-004	RETAINER	
						3				MS24665-115	COTTER PIN	
						2				MS24391D2	BLEED PLUG	
						1				MS21902-6C	UNION	
						1				MS21250H06024	BOLT	
						2				MS21250H06008	BOLT	
						2				MS20002C6	WASHER	
						2				MS20002-6	WASHER	
						2				MS20002C4	WASHER	
					4					MS3106R10SL-3S	CONNECTOR	
					1					MS3102R28-11P	CONNECTOR	
					1					MS3102R22-16P	CONNECTOR	
						1				MS21902-4D-7206	UNION	
						1				MS21926-4C	ELBOW	
						5				AN960PD416	WASHER	
						6				AN960C10L	WASHER	
						8				AN814-6D	PLUG	
						3				AN814-4D	PLUG	
						3				AN320C3	NUT	
						1				AN924-4J	NUT	
						4				AN960PD6	WASHER	
						8				AN960PD10	WASHER	
						4				AN960PD4L	WASHER	
						2			97820	S-12563-333	DOUBLE DELTA CHANNEL SEAL, O.D.	
						1				S-12560-220-EA	CHANNEL SEAL	
						1				S-12068-226	GLYD RING, O.D.	
						2				S-12068-116		
						2				S-12068-018		
						2				S-12068-121	GLYD RING, O.D.	
						4				S-12066-120	GLYD RING, I.D.	
						4				S-12066-014	GLYD RING, I.D.	
						2				S-12542-218	DOUBLE DELTA CHANNEL SEAL, I.D.	
						1				S-11065-18	SCRAPER RING	
						2				S-12066-012	GLYD RING, I.D.	
						4			97820	S-12068-013	GLYD RING, O.D.	
						2			21335	MPLA1(MIL-G-21617)	BEARING	15
					2				71468	CA3106E-10SL-4S	CONNECTOR	
						AR			00141	83-1	SHIM, SPACER	
						8			56878	WP-6	WASHER, PLAIN	
						8				WC-6	WASHER, CSK	
						8				EWN22-626	NUT	
						8				EWB26-6-30	BOLT	
						2			56878	42FW624	NUT	

QUANTITY REQUIRED — LIST OF MATERIAL OR PARTS LIST

-09	-08	-07	-04	-03	-02	-01	Zone	Item	Code Ident No.	Part or Identifying No.	Nomenclature or Description	Material and Finish or Note or Ref Des
						4	C7-3			401-13887-01	SPACER, SPRING	
						2	C7-3			401-13819-09	SPRING, RESERVOIR (INNER)	
						1	D3-3			401-13811-02	PRESSURE SW	
						1	D5-3			401-13830-02	SOLENOID VALVE - 3 WAY	
						3	B2-3			401-13885-01	FITTING	
						2	B11-3			401-13882-01	RETAINER	
						1	F7-2			401-13871-01	MOTOR PUMP	
						1	B6-3			401-13832-01	FILTER ELEMENT	SPEC CONT DWG
						1	D6-3			401-13831-01	SOLENOID VALVE - 4 WAY	SPEC CONT DWG
						1	D4-3			401-13830-01	SOLENOID VALVE - 3 WAY	SPEC CONT DWG
						2	B6-3			401-13818-01	FILTER	SPEC CONT DWG
						2	D3-3			401-13811-01	PRESSURE SW	SPEC CONT DWG
						1	F7-3			401-13809-01	SERVOVALVE	SPEC CONT DWG
						1	H4-3			401-13808-01	LINEAR XDCR	SPEC CONT DWG
						5	B5-3			401-13807-02	CHECK VALVE	SPEC CONT DWG
						2	C5-3			401-13807-01	CHECK VALVE	SPEC CONT DWG
						1	F7-3			401-13806-01	LINEAR XDCR	SPEC CONT DWG
						1	C3-3			401-13880-01	SHAFT	
						1	C3-3			401-13879-01	BUSHING	
						2	B4-3			401-13878-03	CLIP	
						1	A3-2			401-13878-02	CLIP	
1							A8-2			-12	TUBING 3/8 O.D. x .028 WALL CRES 304 MIL-T-6845	
	1						A3-2			-11	TUBING 3/8 O.D. x .028 WALL CRES 304 MIL-T-6845	
		1					A3-2			-10	TUBING 3/8 O.D. x .028 WALL CRES 304 MIL-T-6845	
						1	A8-2			-09	TUBE ASSY	
						1	A3-2			-08	TUBE ASSY	
						1	A3-2			-07	TUBE ASSY	
			1				C7-2			-06	TUBING 1/4 O.D. x .028 WALL CRES 304 MIL-T-6845	
				1			A4-2			-05	TUBING 1/4 O.D. x .028 WALL CRES 304 MIL-T-6845	
						1	C7-2			-04	TUBE ASSY	
						1	A4-2			-03	TUBE ASSY	
						2	B3-3			401-13848-02	BAYONET	
						1	A5-3			401-13878-01	COVER	
						1	A5-2			401-13881-01	JUNCTION BOX	
						1	E7-3			401-13872-01	RETAINER, SLEEVE - AUX RAM	
						2	B4-3			401-13870-01	SHIM	
						1	D6-3			401-13869-01	SPACER, CHECK VA.	
						1	C6-3			401-13868-01	RETAINER, SPRING - RSVR	
						1	C7-3			401-13867-01	GUIDE, SPRING - RSVR	
						1	C8-3			401-13866-01	PISTON, RSVR	
						1	B8-3			401-13865-01	RETAINER, LOCKING	
						1	B8-3			401-13864-01	GUIDE, LOCKING	
						1	B8-3			401-13863-01	SLEEVE, LOCKING	
						2	B8-3			401-13862-01	PISTON, LOCKING	
						1	B8-3			401-13861-01	ARM ASSY, INPUT	
						2	B5-3			401-13860-01	RETAINER, CHECK VA.	
						2	D5-3			401-13859-01	RETAINER, CHECK VA.	
						1	C8-3			401-13856-01	LINK ASSY	
						1	B8-3			401-13855-01	CRANK	
						1	C8-3			401-13854-01	CAP, LINKAGE	
						1	C8-3			401-13827-01	PISTON, AUX RAM	
						2	C7-3			401-13826-01	SLEEVE, INTMD - AUX RAM	
						2				401-13853-01	RETAINER, BEARING	
						1	F7-3			401-13852-02	RELIEF VALVE ASSY	
						1	A4-2			401-13852-01	RELIEF VALVE ASSY	
						1	A7-2			401-13886-01	TUBE ASSY	
						1	D8-3			401-13849-01	RETAINER, FILTER	
						2	B3-3			401-13848-01	BAYONET	
						1	A8-3			401-13847-01	BUSHING	
						1	A8-3			401-13846-01	NUT (BUSHING & SEAL)	
						1	A6-3			401-13845-01	RETAINER TRANS.	
						1	A6-3			401-13844-01	PISTON	
						1	A8-3			401-13843-01	FITTING (TRANS. PROBE)	
						1	A8-3			401-13842-01	PISTON & ROD ASSY.	
						1	A7-3			401-13841-01	SEAL PLATE (CENTER DAM)	
						1	A9-3			401-13840-02	BARREL	
						1	B5-3			401-13840-01	BARREL ASSY	
						1	E7-3			401-13839-01	NUT, JAM, TRANS. AUX RAM	
						1	E7-3			401-13838-01	RETAINER, TRANS. AUX RAM	
						1	C7-3			401-13837-01	RETAINER, SLEEVE AUX. RAM	
						1	F7-3			401-13835-01	RETAINER, SERVO VA.	
						2	A6-3			401-13834-02	WASHER (TRANSDUCER)	
						1	A6-3			401-13834-01	WASHER (PISTON)	
						1	A6-3			401-13833-01	NUT (PISTON)	
						2	B8-3			401-13829-01	PISTON AUXILIARY RAM	
						1	C7-3			401-13828-01	SLEEVE, OUTER - AUX RAM	
						1	C7-3			401-13825-01	NUT, JAM - AUX RAM	
						1	C6-3			401-13823-01	RETAINER SWITCHING VA.	
						1	C6-3			401-13820-01	SWITCHING VA ASSY	
						2	C7-3			401-13819-06	SPRING, RESERVOIR (MIDDLE)	
						2	C7-3			-05	SPRING, RESERVOIR (OUTER)	
						2	C7-3			-04	SPRING, AUXILIARY RAM	
						1	D5-3			-03	SPRING, SEQUENCE VA.	
						1	C5-3			-02	SPRING, SWITCHING VA. (INNER)	
						1	C5-3			401-13819-01	SPRING, SWITCHING VA. (OUTER)	
						1	D5-3			401-13817-01	RETAINER, SEQUENCE VA.	
						1	D5-3			401-13815-01	SEQUENCE VA ASSY	
						1	B9-3			401-13812-01	ROD END ASSY	
						1	F6-3			401-13802-01	SERVO VALVE ASSY	
					REF					401-13873	WIRING DIAGRAM	
						REF				401-13801	SCHEMATIC (HYD)	
						1	A5-2			401-13800-01	HOUSING ASSY	
						1	B3-2	1		-02	WIRING HARNESS	
						1	A4-2	1		-01	INTEG ACT PKG	

QUANTITY REQUIRED

FIGURE 27. SIMPLEX ASSEMBLY – PARTS LIST

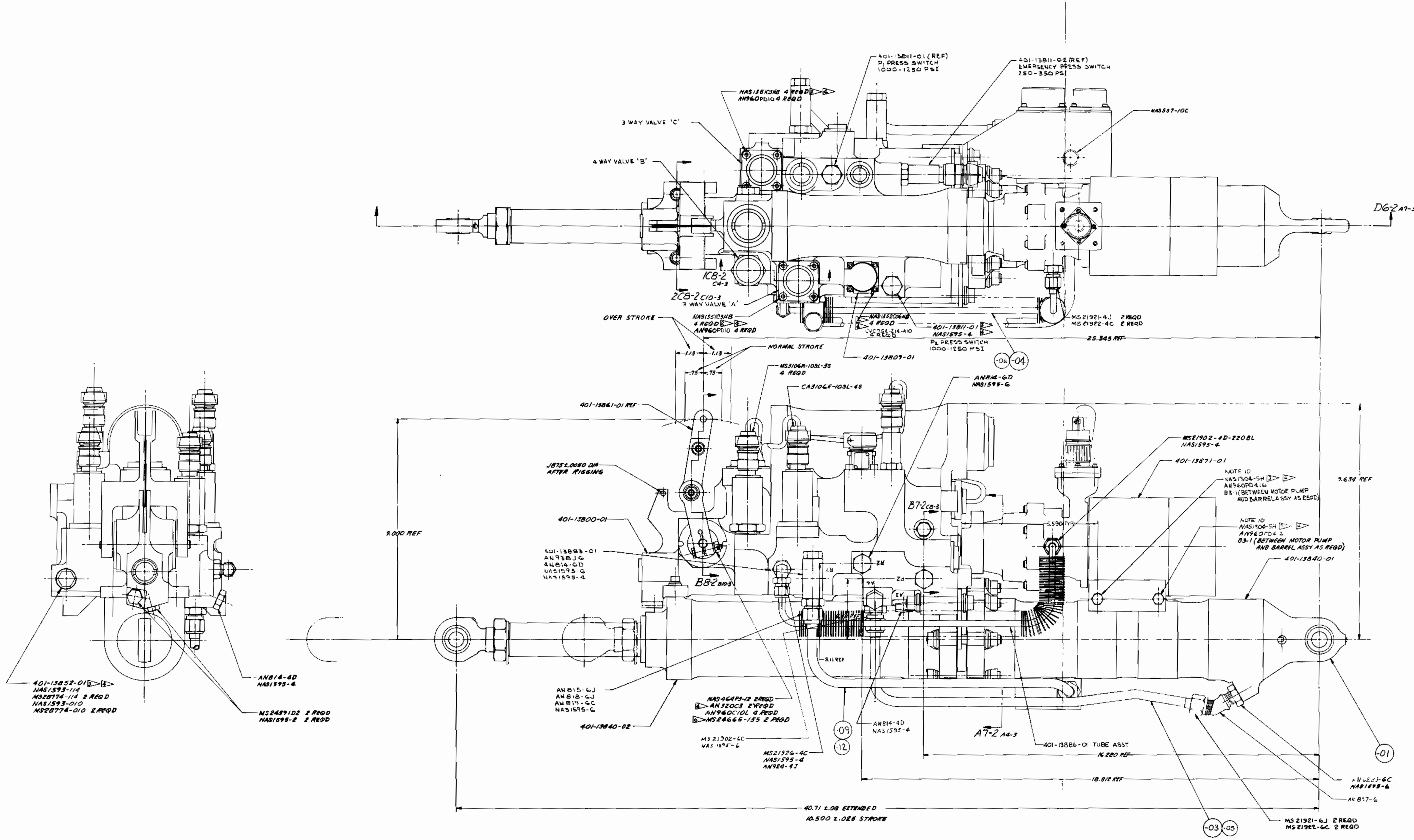


FIGURE 28. SIMPLEX ASSEMBLY

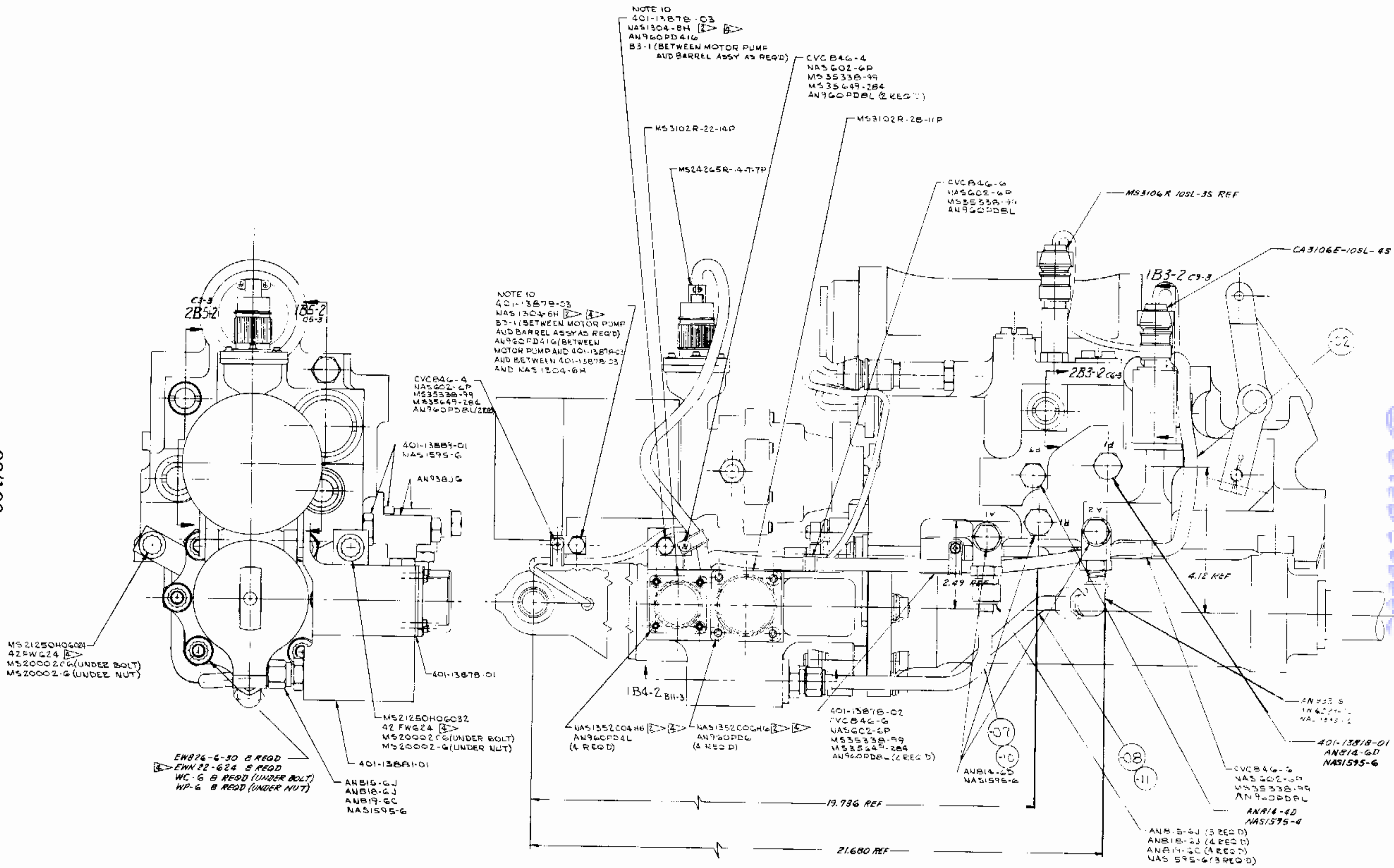


FIGURE 28. SIMPLEX ASSEMBLY (SHEET 2 OF 2)

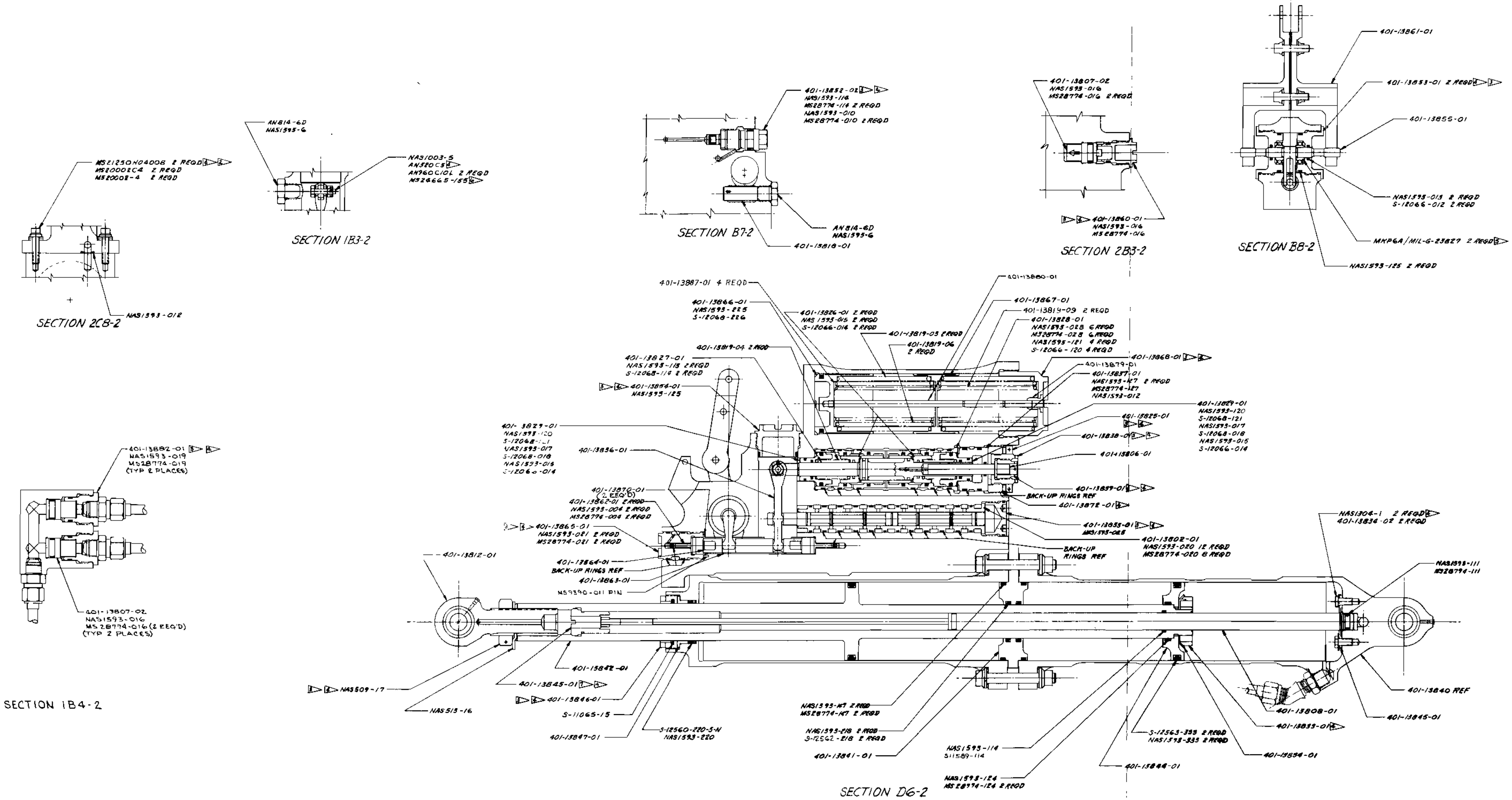


FIGURE 29. SIMPLEX ASSEMBLY (SHEET 1 OF 2)

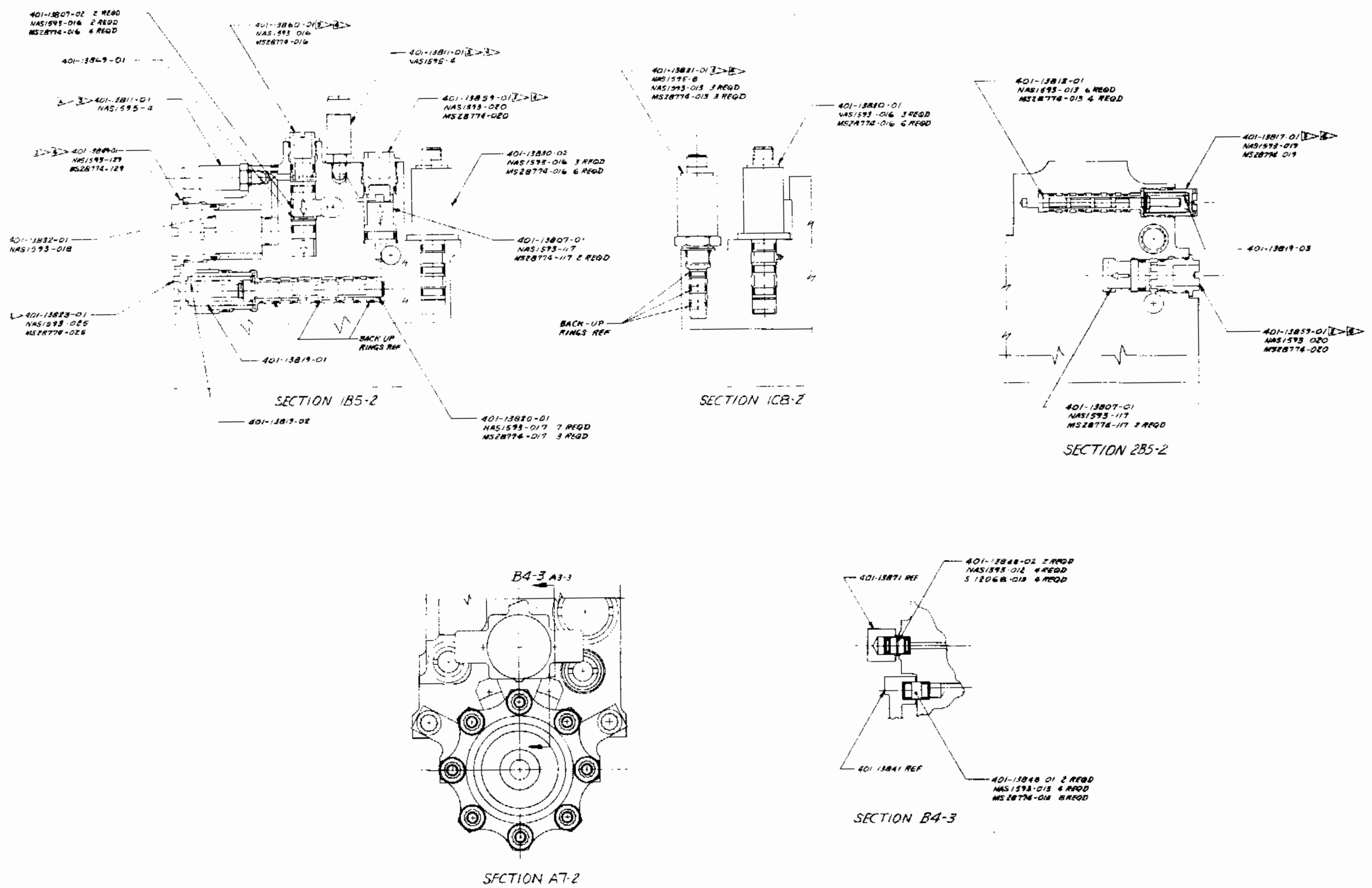


FIGURE 29. SIMPLEX ASSEMBLY (SHEET 2 OF 2)

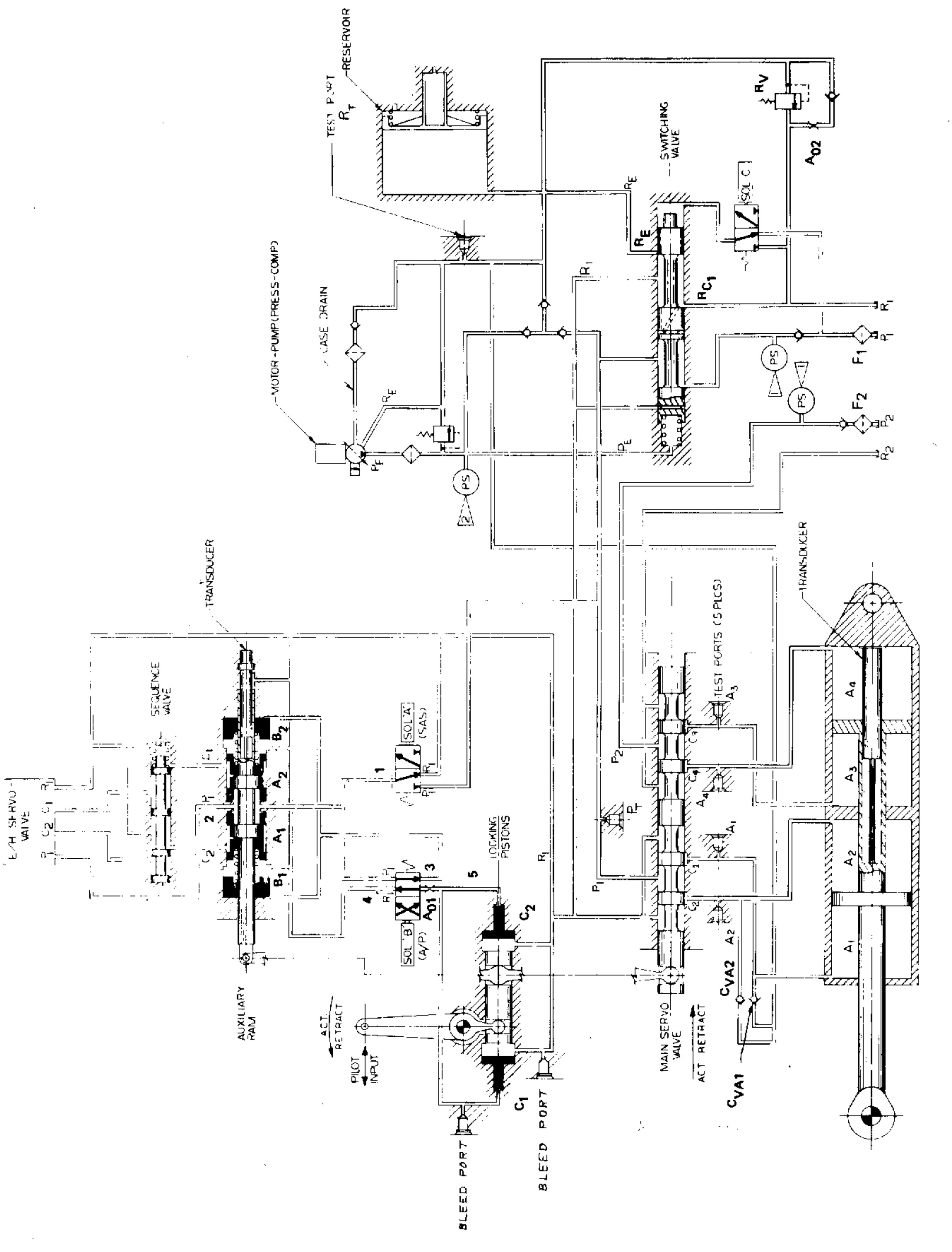


FIGURE 30. SIMPLEX HYDRAULIC CIRCUIT

dictates whether P_1 or the emergency system is on line. The emergency system reservoir is continuously and automatically filled by the return pressure of System 1 through an orifice and check valve arrangement. This arrangement eliminates the requirement for external filling of the emergency system.

c. Electrical Circuitry

Electrical wiring of the Simplex Actuator is according to Figure 31. The two system connectors which mate with the aircraft systems are MS3102E-22-14P and MS3102E-28-11P. The MS3102E-22-14P connector is identical to the connector on the F-4 production actuator. In addition, the connector on the Simplex is approximately in the same physical location as on the production actuator. Consequently, it can be serviced by the present aircraft cabling. The 3-way and 4-way solenoids contain individual connectors MS3106E-10SL-4S and MS3106E-10SL-3S for ease of checkout and maintenance. Electrical cabling between the solenoid connectors and the main connector is type E teflon per MIL-W-16878. The sleeving for the cables is a silicone type HA1 material per MIL-I-18057.

The cables for the main and auxiliary transducers are MIL-W-16878 and are routed directly to the connector from their internal terminal boards. The three leads from the servo valve are spliced and then connected directly to the connector.

All electrical functions associated with the emergency system operation are routed through the MS3102E-28-11P connector. These functions include the power leads for the emergency motor, the pressure switches probing main system pressure, emergency pressure switch, and the emergency system solenoid C. The motor driving the emergency system pump is a 3-phase, 400 cycle Wye connected unit with neutral and ground leads. The motor draws a nominal 10 amperes per leg at a power factor of 0.75, with a starting current of approximately 60 amps. Maximum motor efficiency is 82%. The pressure switches are all shown in the 0 psi condition. PC_1 and PC_2 pressure switches are open when the system is operating normally, i.e., when the ship's hydraulic supplies are functioning. When the PC_1 or PC_2 pressures drop below a nominal 1200 psi, the switches close to provide the failure indication. Depending on the external circuitry for monitoring the package performance, it may be desirable to reverse the polarity of the switches. This can be done simply by changing the one lead wire connection from B to C. The emergency pressure switch is normally open. The switch closes when emergency system is energized and pressure increased to 350 psi.

All electrical components are capable of withstanding 1000 volts AC for one minute between mutually insulated portions of a component part, or between insulated portions and ground, without disruptive discharge or deterioration. The equipment was designed to meet the electromagnetic interference requirements of MIL-I-6181.

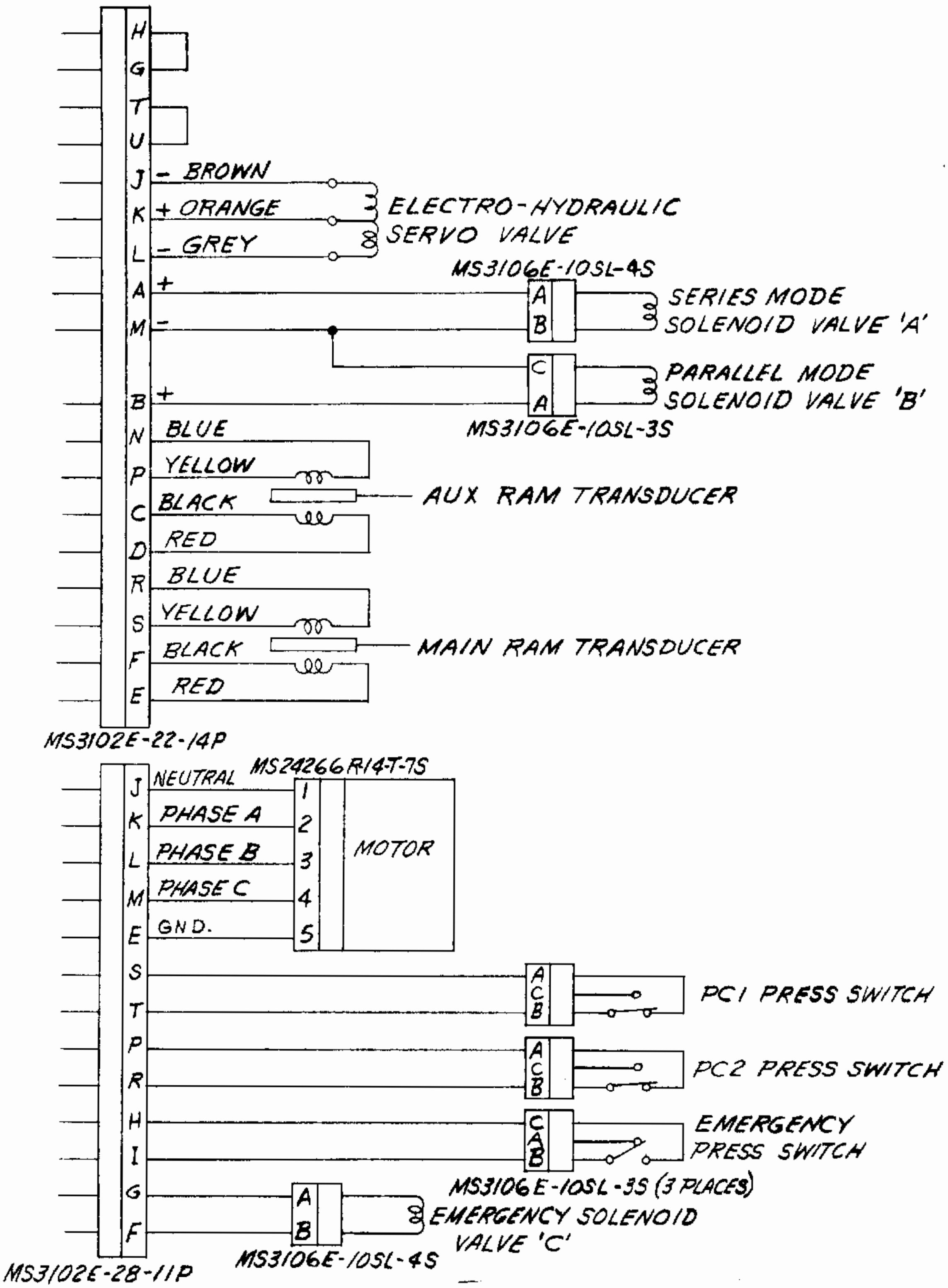


FIGURE 31. SIMPLEX ELECTRICAL SCHEMATIC

d. Linkage Arrangement

Figure 32 shows the Simplex mechanical signal linkage schematically. All the linkage is internal to the package. External linkage is avoided to eliminate the possibility of inadvertent jamming due to foreign objects. Pilot inputs are applied at the top of the input arm assembly. The linear inputs are converted to rotary motion through rotary seals and a pair of MKP6A/MIL-G-23827 ball bearings. The rotary signal is converted to linear displacement by a crank. Movement of the output arm of the crank strokes the locking slider which in turn displaces the end of the link assembly. Inputs from the auxiliary ram together with manual inputs are summed on the link assembly. The link assembly then positions the master control valve which valves the hydraulic fluid to the actuator.

The nominal stroke at the pilot's input is +0.75 inch. This constitutes a stroke of 0.105 inch at the main servo valve level, which is full effective stroke of the valve. However, an over-travel capability is built into the valve to prevent it from mechanically bottoming out when a hard over input signal is applied. The total mechanical travel of the main servo valve is a minimum of 0.262 inch. This corresponds to full output displacement of the auxiliary ram as well as a full displacement of the mechanical input in the same direction. External stops on the mechanical input are provided to limit its travel in both directions. These stops and the affected linkages and bearings are capable of transmitting sufficient pilot effort to produce a +1000-pound force along the master control valve center line for unjamming the valve. The auxiliary ram is designed to have 3.35% authority over the main ram displacement when operating in the series or stability augmentation system (SAS) mode, and 100% in the parallel or autopilot mode. Velocity authority of the auxiliary ram over main ram is approximately 25% in the SAS mode and 100% in the autopilot mode.

3. System Operation

The Simplex Integrated Actuator can be considered to be operable in four distinct modes. Selection of a particular mode is normally at the discretion of the pilot. The exception is the emergency system operating mode which, as well as being selectable by the pilot, can also be activated automatically in the event of primary hydraulic system failure. The four operating modes are:

Manual
SAS
Autopilot
Emergency

Manual is the primary operating mode. In this mode, the solenoids (Figure 30) are all de-energized. Therefore the auxiliary ram is inoperative, and the associated hydraulic circuitry is vented to return pressure. The supply pressure from P_1 is ported through the emergency solenoid valve C to the right end of the switching valve to keep the switching valve shuttled to

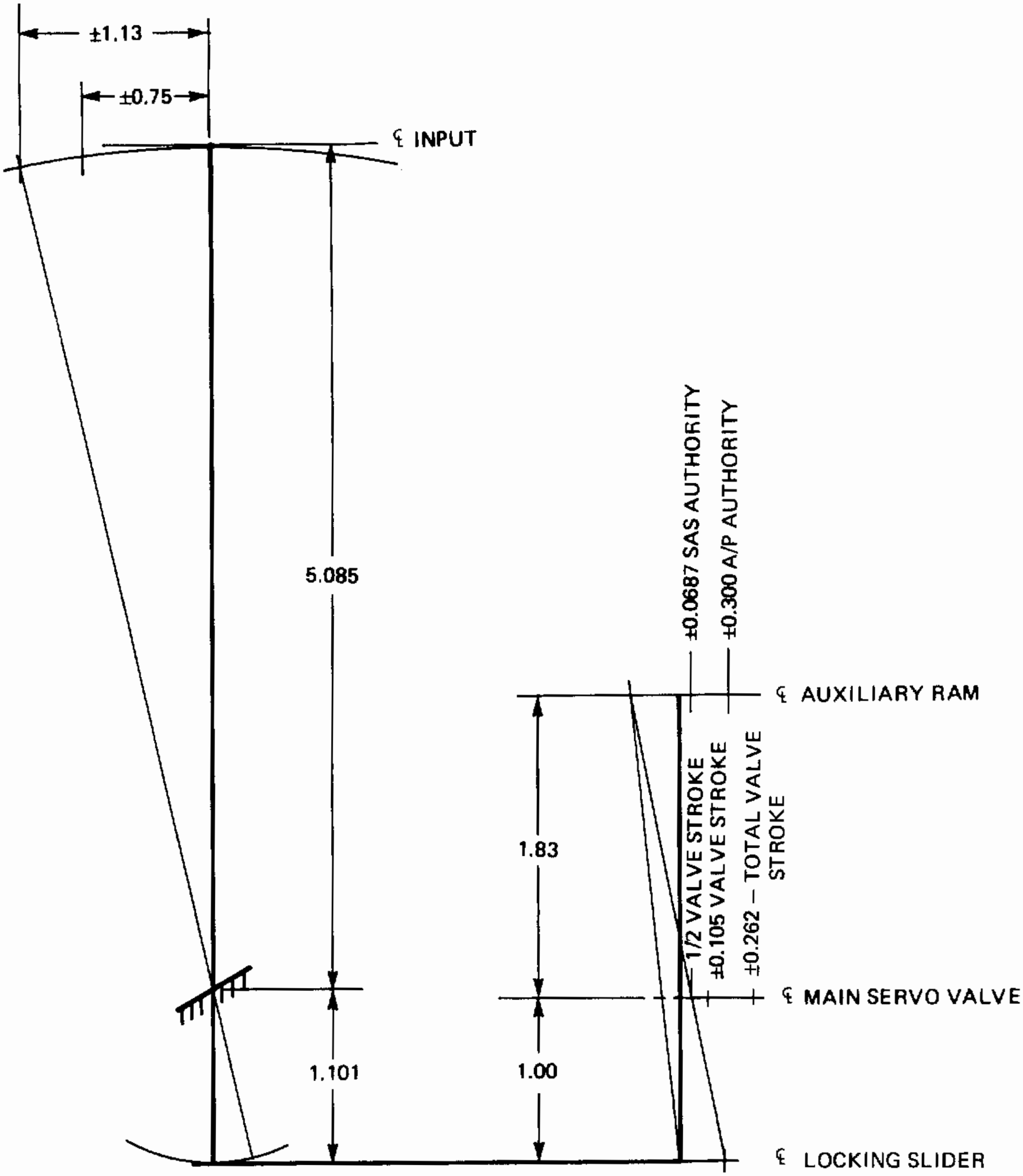


FIGURE 32. LINKAGE SCHEMATIC

the left. P_1 and R_1 are then ported through the switching valve to the main servo valve (MSV), and the emergency system is isolated from the actuator by the switching valve. P_2 and R_2 are ported directly to the MSV. With the auxiliary ram hydraulic circuitry vented to return, the auxiliary ram is grounded to center by the mechanical springs which are sized to ensure that the auxiliary ram remains at neutral during manual operation. Manual inputs position the MSV and cause flow to the actuator. The barrel of the actuator moves at a rate proportional to the valve opening. As the barrel moves, it moves the valve housing with it. The pivot point of the input linkage is grounded to, and moves with, the housing, thereby repositioning the input linkage (assuming the input remains fixed after the original displacement) until the MSV has returned to its neutral position, at which time the actuator stops.

Hydraulic system P_1 provides the power to operate the auxiliary ram and locking pistons. To transfer to the series or SAS mode, Solenoid A is energized applying pressure to line 1, the auxiliary ram, line 2, and to the E/H valve. Pressure in line 2 causes the sequence valve to stroke to the right, thereby connecting the E/H valve cylinder ports to the auxiliary ram cylinders. Pressure in line 1 causes the auxiliary ram locking pistons A_1 and A_2 to back off against the spring preloads to the limit stops B_1 and B_2, respectively, thereby uncaging the piston. At the same time pressure is ported to the 4-way solenoid valve B to line 3, and to the back side of the limit stops B_1 and B_2, causing them to stroke in approximately 0.2 inch, and shoulder out to limit the displacement of the piston to +0.0687 inch, (equivalent to 3.35% displacement authority). In the SAS mode the feel spring and linkage inertia normally maintain the pilot input point fixed. The auxiliary ram operates as an electrohydraulic servo. It is rigged so that with the ram spring loaded to center, the transducer has zero output. An electrical input to the SAS servo loop from the aircraft motion sensors signals the E/H valve to port fluid to the ram, causing it to move until the ransducer signal cancels the input signal from the motion sensors and returns the loop error signal to zero.

To shift the package to the autopilot or parallel mode, 4-way solenoid valve B is also energized. Pressure is then ported to the input linkage, locking pistons C_1 and C_2 which stroke inward to cause the linkage to be grounded to the package and move with it. In addition, the pressure in line 3 is vented to return line 4, allowing the limit stops to retract. The stroke of the auxiliary ram is now increased to +0.30 inch which can cause full opening of the MSV and full rate of the actuator. With the linkage grounded to the actuator package, the servo no longer has a mechanical closed loop. The loop is closed electrically via the main ram transducer. Consequently the actuator has full displacement authority over the control surface. Since the linkage is grounded to the actuator, the entire longitudinal control linkage, including the stick, moves with the actuator. The pilot can override the autopilot by simply applying a force to the input sufficient to back off the locking pistons C_1 and C_2 which are sized to provide a nominal centering force of 19 pounds to the linkage at the pilot's input point. Orifice A_{O1} provides damping to limit the overpowering rate. In the event of hard over failure of the autopilot, the pilot can override the autopilot input and reverse the direction of the hard over at a main actuator rate which is 50% of full rate. De-energizing solenoids A and B will have the effect of neutralizing a hard over auxiliary ram failure.

To revert to manual control of the package from the autopilot mode, solenoid valves A and B are de-energized. Lines 1, 2, 3, 4 and 5 are ported to return. This action removes pressure from the locking pistons, the auxiliary ram locking pistons A_1 and A_2, and the sequence valve. The spring loaded sequence valve moves to the left to the position shown. Auxiliary ram cylinder ports are connected to return line 6. The auxiliary ram centering springs then are able to rapidly center the auxiliary ram.

The return pressure of System 1 (R_1) is connected via a small resistor orifice, A_{O2}, and check valve arrangement to the emergency system reservoir. In the above operating modes the reservoir is always full. No external servicing is required. An external indicator is provided on the reservoir piston to permit visual indication of the amount of fluid in the reservoir. The orifice prevents pressure surges in the aircraft return system from entering the emergency system return. The check valve prevents loss of emergency system fluid in the event that system 1 return pressure is lost.

Emergency system operation can be initiated in a number of ways, depending on the manner in which the package functions are connected externally. The available emergency modes are:

Standby
Pilot activated
Automatically activated with P_1 failure
Automatically activated with P_1 and P_2 failure

The emergency system can be activated to the standby mode by simply energizing the motor. The output pressure of the pump increases to the pump cut-off pressure which is adjusted to 1600 psi. When the pressure exceeds 250 psi, the emergency system pressure switch closes to signal that the system is operating and is up to pressure. The check valves which are activated by P_1 system remain seated due to it's higher pressure and prevent the emergency system pressure from being transmitted to the MSV. A high pressure relief valve is provided to protect the emergency system in the event of a pump failure. Filters are provided in the case drain and output pressure lines for contamination control. The emergency pump case is rated at 600 psi proof pressure. The remainder of the emergency system return circuit is rated at 3000 psi.

The emergency system can be activated by the pilot at his discretion by energizing the motor and then energizing solenoid valve C to vent the right end of the switching valve to return. The spring then drives the switching valve to the right, thereby isolating P_1 and R_1 from the package and connecting the emergency system to the MSV through the switching valve. To revert to primary system operation, the pilot simply de-energizes solenoid C and removes power from the motor. The emergency system's electrical circuitry can also be arranged so that the system comes on-line automatically when either P_1 or both P_1 and P_2 fail. Failure of P_1 (or P_1 and P_2) as sensed by the pressure switch(s) activates a relay (not part of the Simplex Package) which applies power to the motor. The spring drives the switching valve to the right, connecting the emergency system to the MSV. Output pressure and flow of the emergency system is then ported to the MSV and provides sufficient power for control of the stabilator to permit landing the aircraft. It is designed to operate continuously for two hours in a typical environment which would be encountered when recovering the aircraft.

Relief valve RV is set at 330 psi. It's function is to ensure that the emergency return circuit pressure does not exceed the pump case capability. The following is an explanation of its operation.

If the emergency system is energized when the actuator is in the extended position, the excess fluid due to the unequal piston areas is relieved through the relief valve when the actuator is retracted. Check valves CVA_1 and CVA_2 are provided between system no. 1 cylinder ports and emergency return. When system 1 is inoperative and system no. 2 is operative, the check valves prevent cavitation in A_1 and A_2 cylinders, and prevent relief valve RV from relieving and dumping emergency system fluid.

4. Component Description

a. Actuator Barrel-Piston Assembly

The Simplex actuator barrel-piston assembly is an all steel dual tandem actuator (Figures 28 and 29). Piston areas are sized to provide the necessary dynamic stiffness to the control surface. The barrel is a two piece unit separated by a center dam. The two pieces are made from 4340 steel heat treated to Rc 43-46 (180,000 psi T.S.) and cadmium plated externally for environmental protection. The lug end portion contains strengthened sections for mounting the emergency system motor-pump, and provisions for mounting the main ram transducer. The lug end bearing is a KSBG-10SX unit suitable for transmitting the required loads. The center dam is made of 4340 heat treated to Rc 39-43. The dam contains dual seals with an over board vent located between the two seals. The rod and piston assembly as well as the rod end are made of 4340 heat treated for appropriate tensile strength. The rod and piston are chrome plated on all wear surfaces and cadmium plated for environmental protection. Rod and rod end are sized to meet the structural and vibration requirements.

b. Auxiliary Ram Linear Transducer

The auxiliary ram transducer (Figure 33), is a linear variable differential transducer (LVDT) having a nominal stroke of +0.30 inches with a null voltage not exceeding 15 mv. Excitation voltage is 26 volts, 400 Hz. Output voltage gradient is 18.6 volts/inch. The unit is phased so that the yellow and red leads are in phase when the core is retracted. Power requirement is less than two watts. The core is installed in the piston of the auxiliary ram and held in place by a Long Lok. Rigging and null adjustment of the transducer is performed by moving the LVDT body in or out by its adjusting and locking arrangement (Figure 29).

c. Main Actuator Tranducer

The main actuator transducer (Figure 34) is also an LVDT and is used to monitor the actuator position and close the actuator loop in the autopilot mode. The unit

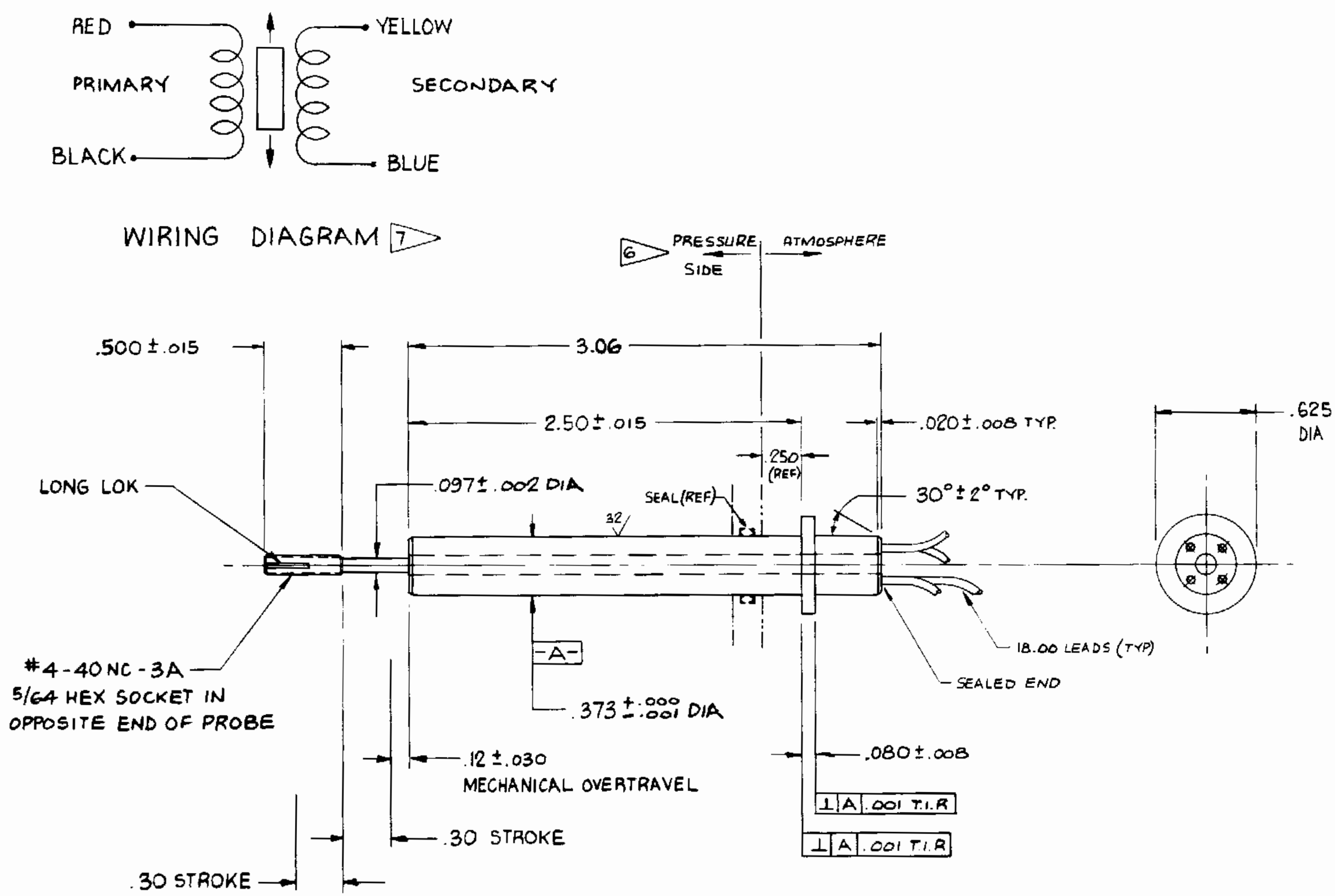
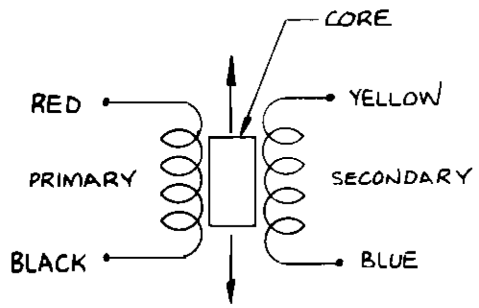


FIGURE 33. AUXILIARY RAM LINEAR TRANSDUCER



WIRING DIAGRAM

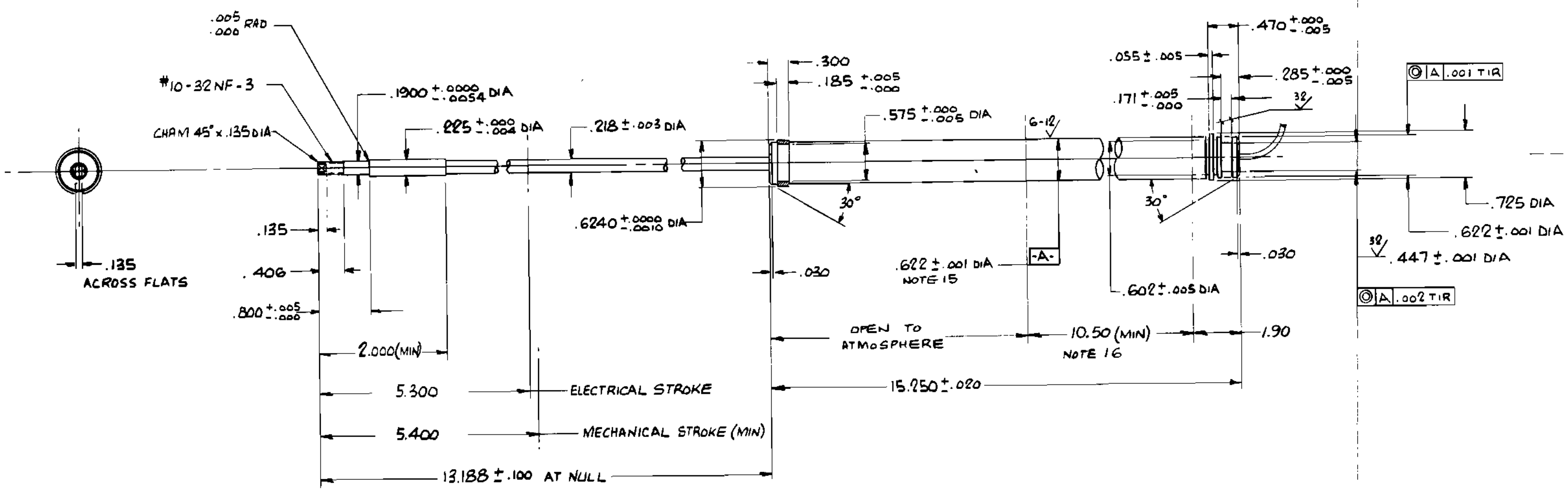


FIGURE 34. MAIN ACTUATOR TRANSDUCER

has an effective stroke of ±5.3 inches. Excitation voltage is 26 volts, 400 Hz, with a nominal output voltage gradient of 4 volts/inch. Electrical phasing of the unit is such that the yellow and red leads are in phase when the core is retracted. An internal seal in the piston rod seals around the outside diameter of the LVDT. The core is attached to the rod with a locking nut, and moves with it. No provision for rigging is included or required.

d. Electro-Hydraulic Servo Valve

The E/H servo valve, Model 25B manufactured by Hydraulic Research and Manufacturing Company, is used to drive the auxiliary ram on the Simplex IAP. The unit (Figure 35) is a standard 4-way valve, except that it contains high temperature packings for operation above 275°F. The operating pressure is 3000 psi. Operating current is ±15 ma with a proof current of ±25 ma. Maximum flow is a nominal 0.413 in^3/sec. Pressure gain is a minimum of 2750 psi/ma. The unit exhibits a first order break frequency of 30 Hz when operated at ±7.5 ma differential current. The threshold current of the unit is less than 0.05 ma peak-to-peak when it is defined as the total current differential change required about null current to produce 100 psi ΔP at the cylinder ports.

e. Pressure Switches

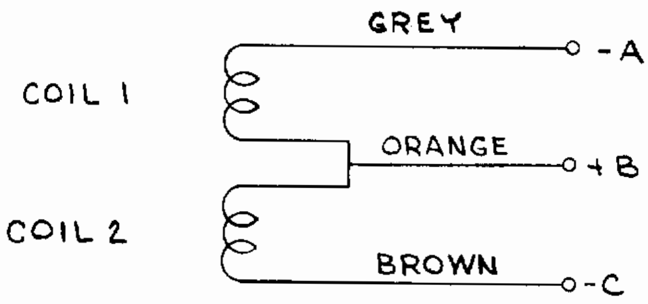
Pressure switches with two different settings are used on the Simplex, (Figure 36). They contain hydraulic pressure ports per MS33656E4 and electrical receptacles per MS33678-10SL-3P. Two -01 units are used in the inlet pressure lines of systems 1 and 2. The actuation point is between 1000 and 1250 psi. The -02 unit is used in the emergency pressure circuit and actuates between 250 and 350 psi. Both units are rated for 3000 psi operating pressure, 4500 psi proof pressure, and 7500 psi burst pressure.

f. Check Valve

Two sizes of check valves (Figure 37), are used in the Simplex. The -01 valves are used in the P_1 and P_2 inlets to prevent back flow of fluid from the actuator. The -02 valves are used in the emergency system circuit. The units are sized for minimum pressure drop at rated flow, and are designed to meet the environment of the Simplex actuator.

g. Filter

Figure 38 shows the design of the filter screens as well as an installation view. The screens provide filtration of 150 microns nominal and 200 microns absolute to prevent large particles from entering the package. The pressure drop rating is 50 psid maximum at 15 gpm at 72° ±10°F.



WIRING DIAGRAM

WHEN CURRENT IN BA IS GREATER THAN BC FLOW IS OUT PORT 2
WHEN CURRENT IN BC IS GREATER THAN BA FLOW IS OUT PORT 1

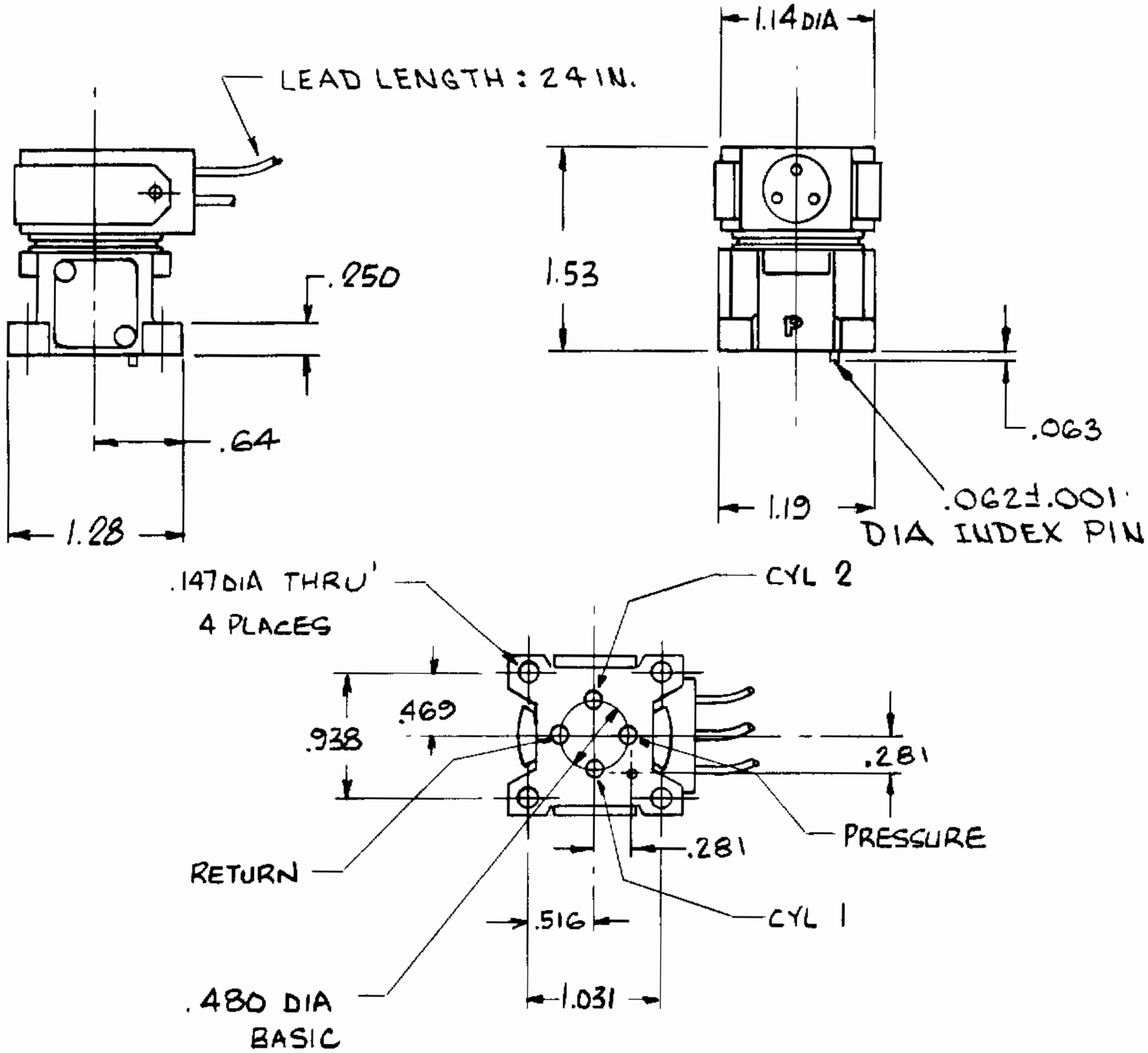
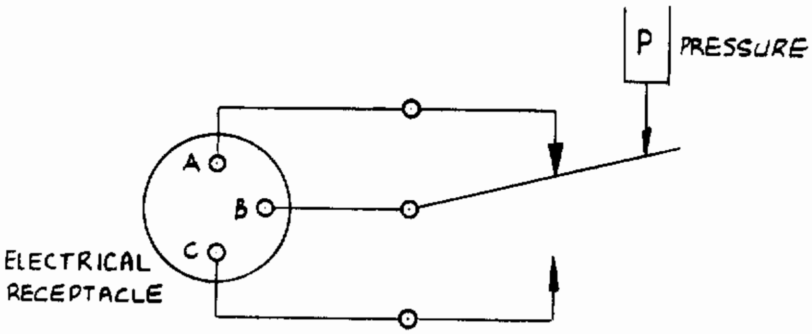


FIGURE 35. ELECTRO–HYDRAULIC SERVO VALVE



WIRING DIAGRAM

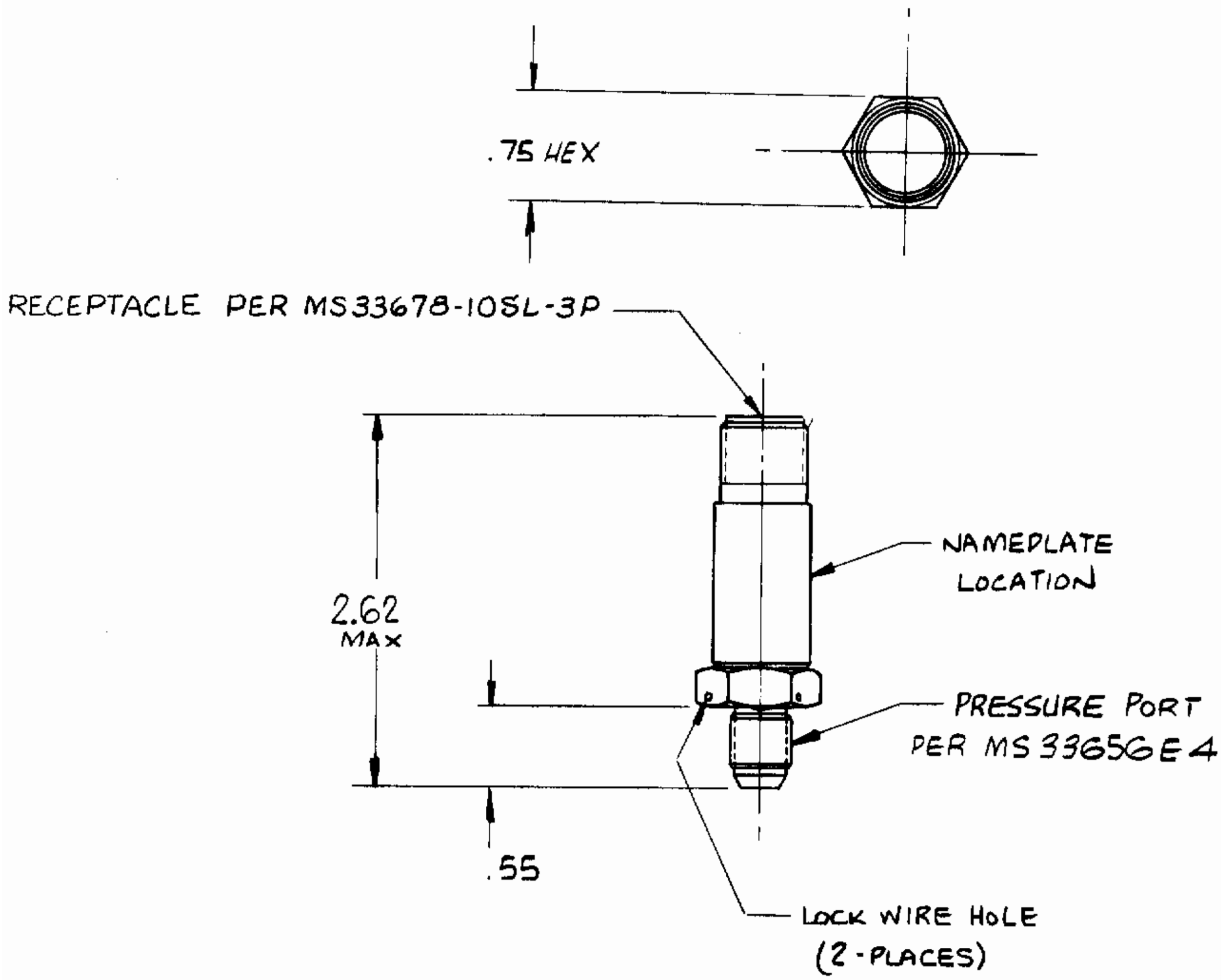
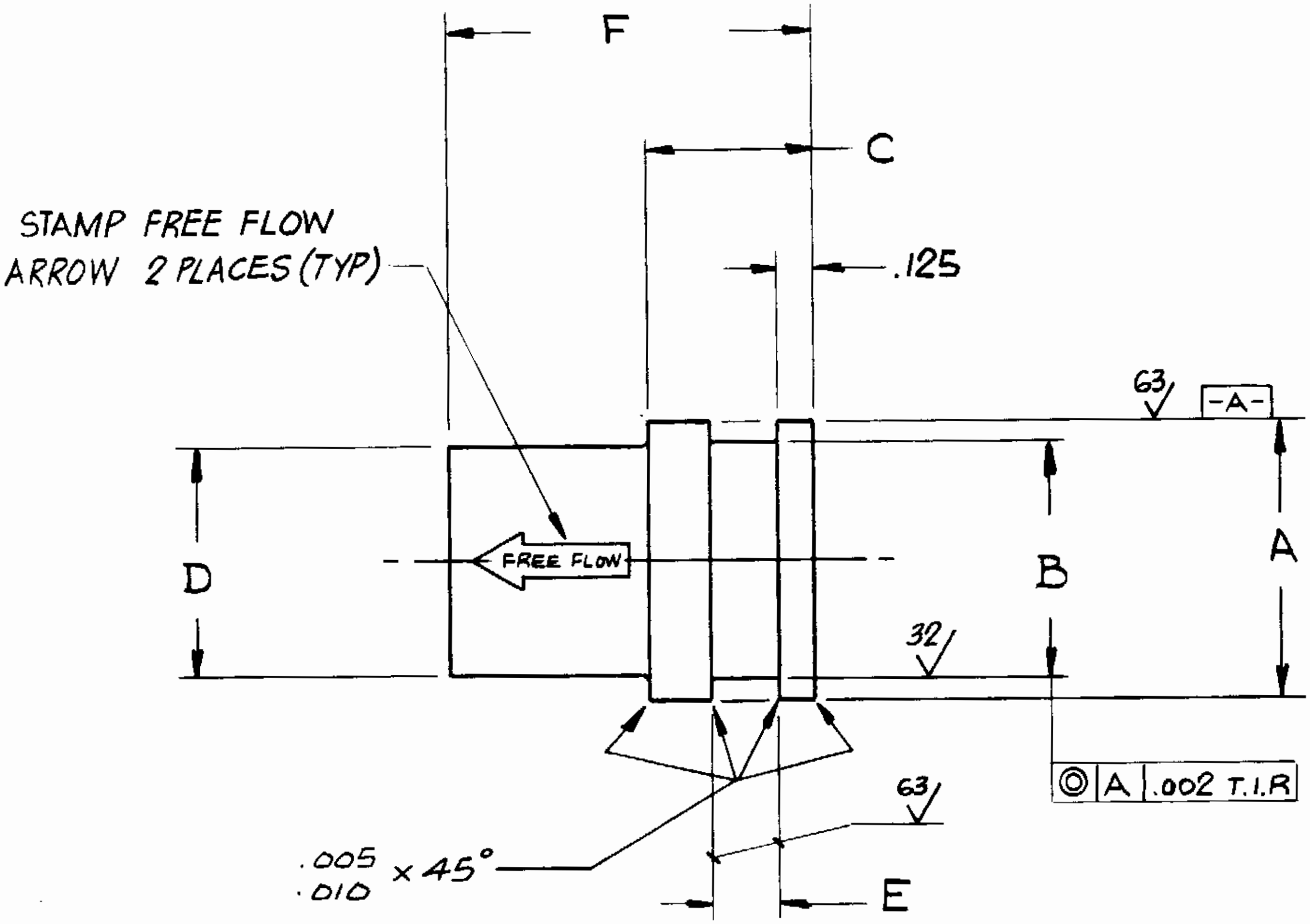


FIGURE 36. PRESSURE SWITCH



DASH NO	A	B	C	D	E	F	EST WEIGHT	CRACK PRESS	PRESSURE DROP
	+.000 -.001	+.000 -.002			+.010 -.000			MAX	AT ROOM TEMP
-02	.736	.626	.550	.625	.207	1.110	.06 LBS	2 PSI	4PSI AT 3GPM
-01	.991	.815	.585	.810	.245	1.300	.10 LBS	5 PSI	10PSI AT 10GPM

FIGURE 37. CHECK VALVES

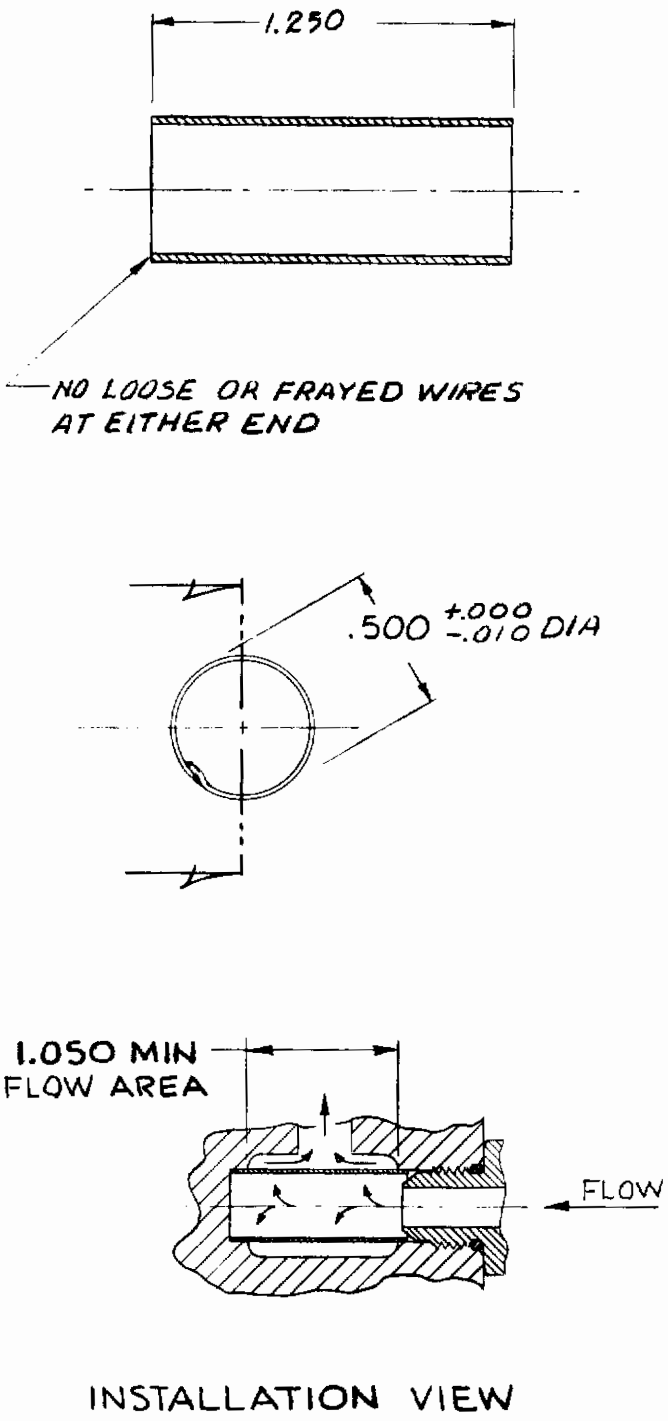


FIGURE 38. FILTER

h. Solenoid Valves

Three types of solenoid valves are used in the package. One of the 3-way valves is a ball valve configuration and (Figure 39) has the cylinder port connected to return in the de-energized condition. It is used to apply pressure to the auxiliary ram in the series and parallel modes. The other 3-way valve is of a slide valve configuration and has supply pressure connected to the cylinder port when de-energized. Its function is to apply the pressure signal for stroking the switching valve. The two units are similar in construction and performance. The 4-way valve (Figure 40) has pressure connected to C_1 cylinder and return connected to C_2 cylinder in the de-energized condition. Rated flow at 300 psid is 0.7 gpm. The unit is used to activate the autopilot mode.

i. Filter Element

The filter element (Figure 41) is installed in the emergency system output pressure line. All flow from the pump passes through the element. It is rated at 10 micron nominal and 25 micron absolute. Pressure drop of the unit is 12.0 psi. Design and construction of the unit is suitable for the operating environment.

j. Main Servo Valve

The main servo valve is a dual tandem valve as shown in Figure 42. Two 4-way valves are combined into a single assembly. This design eliminates the requirement for a synchronizing linkage. The two halves of the valve are synchronized by proper machining of the metering edges of the valve spool. The desired flow gain of the valve is obtained by configuring the orifices in the valve sleeve as shown in Figure 43. To ensure stable servo performance for the valve, a low non-linear flow gain around neutral is required. The flow versus stroke for the first 0.006 inch of stroke obtained with this orifice design with 500 psi pressure dropped across the orifice is shown in Figure 44. At full stroke the valve flow is a minimum of 15 gallons per minute with 500 psi dropped across the orifice. The valve flow forces are controlled by the use of narrow orifices angled 45^o to the valve axis.

k. Sequence Valve

The sequence valve (Figure 45) sequences the supply and cylinder pressures of the auxiliary ram to minimize transients during engagement and disengagement of the series and parallel modes. The valve reduces the auxiliary ram transient time to less than 0.1 second. Supply pressure at which the valve strokes is 400 +50 psi.

l. Switching Valve

The switching valve (Figure 46) switches the backup hydraulic power supply to the main servo valve when P_I supply pressure is lost, or when solenoid valve C is energized. The valve is designed to handle full system flow during normal system

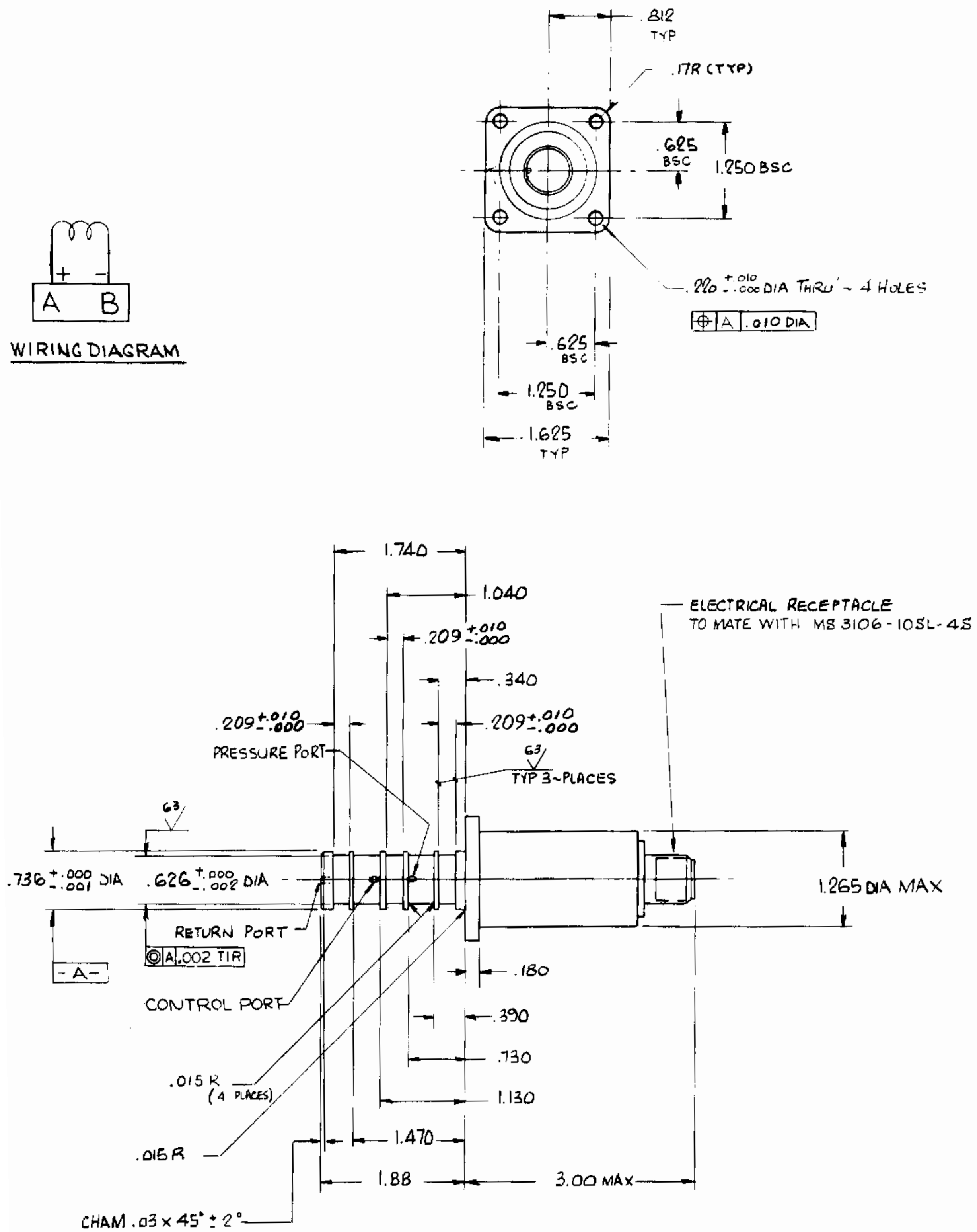


FIGURE 39. THREE–WAY SOLENOID VALVE

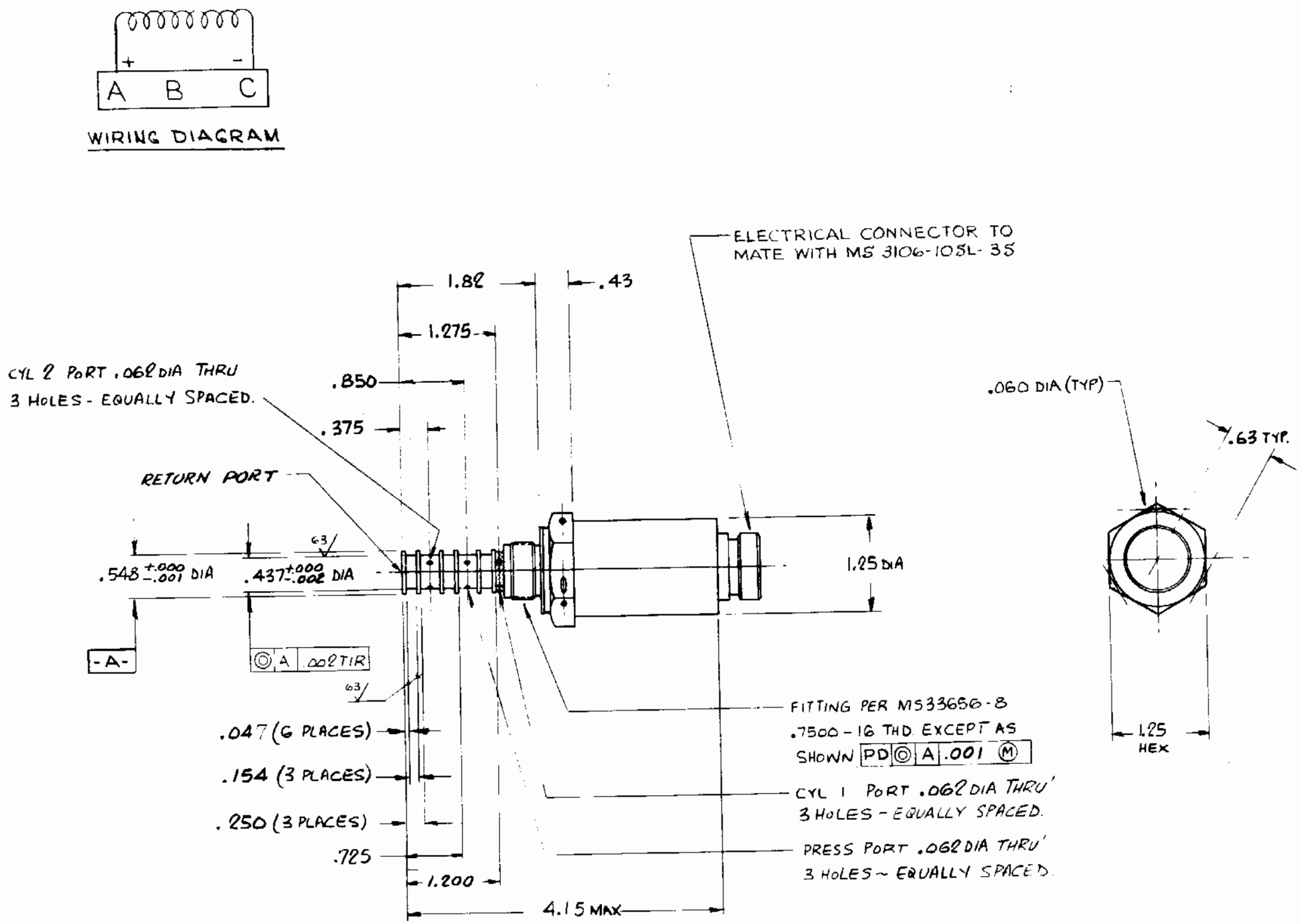


FIGURE 40. FOUR-WAY SOLENOID VALVE

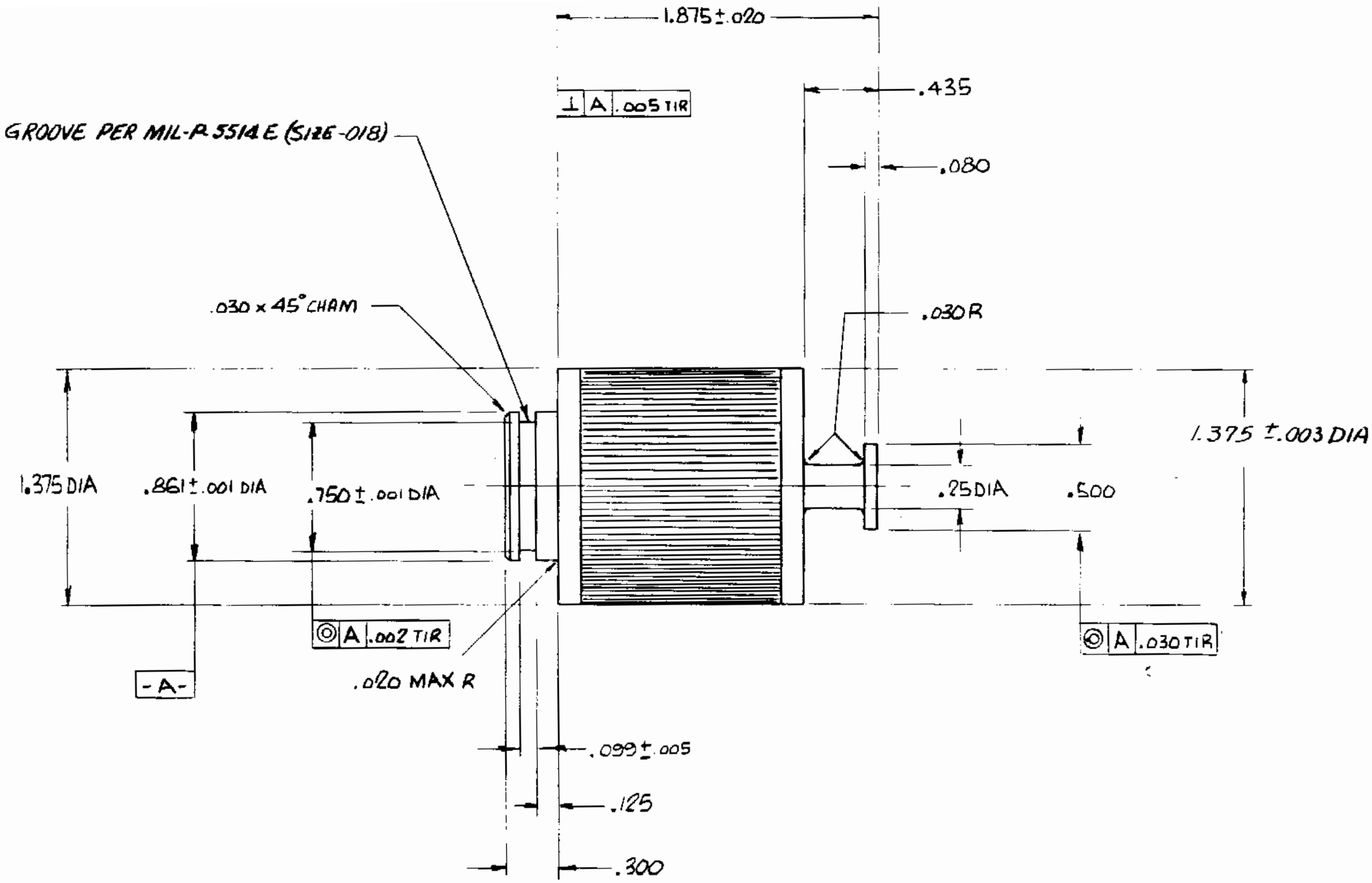


FIGURE 41. FILTER ELEMENT

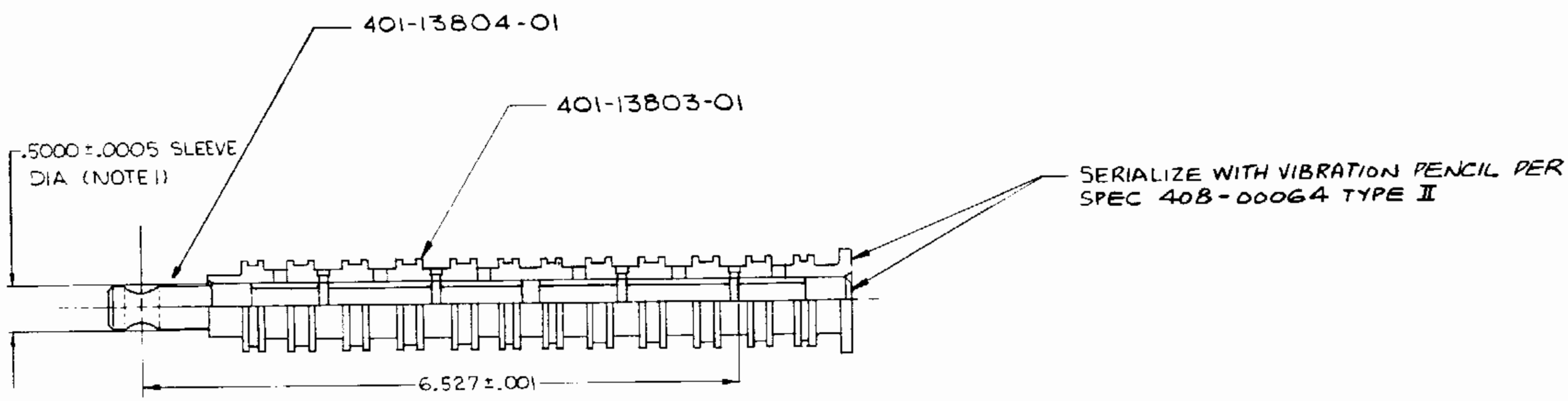


FIGURE 42. MAIN SERVO VALVE ASSEMBLY

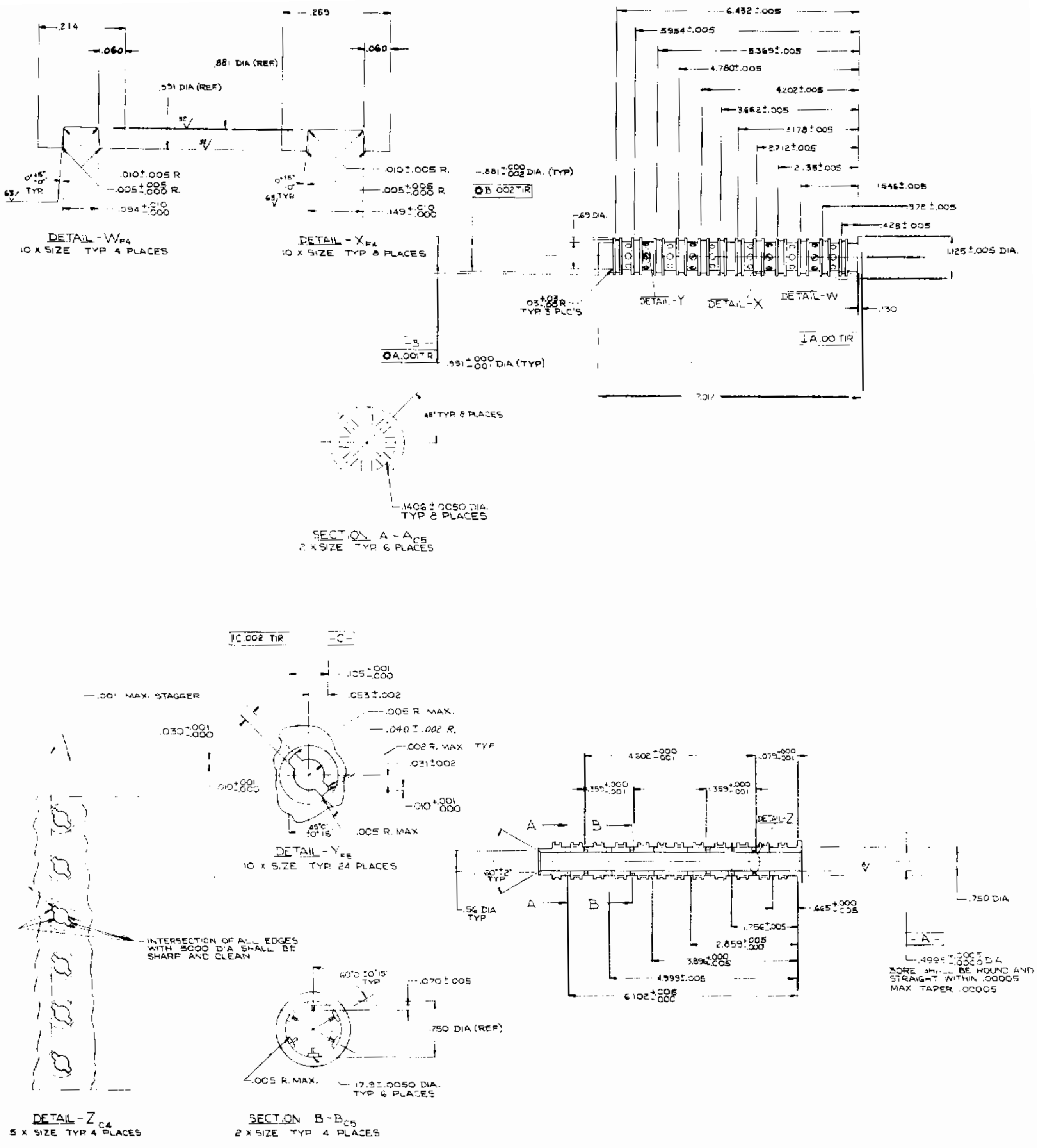


FIGURE 43. MAIN SERVO VALVE SLEEVE

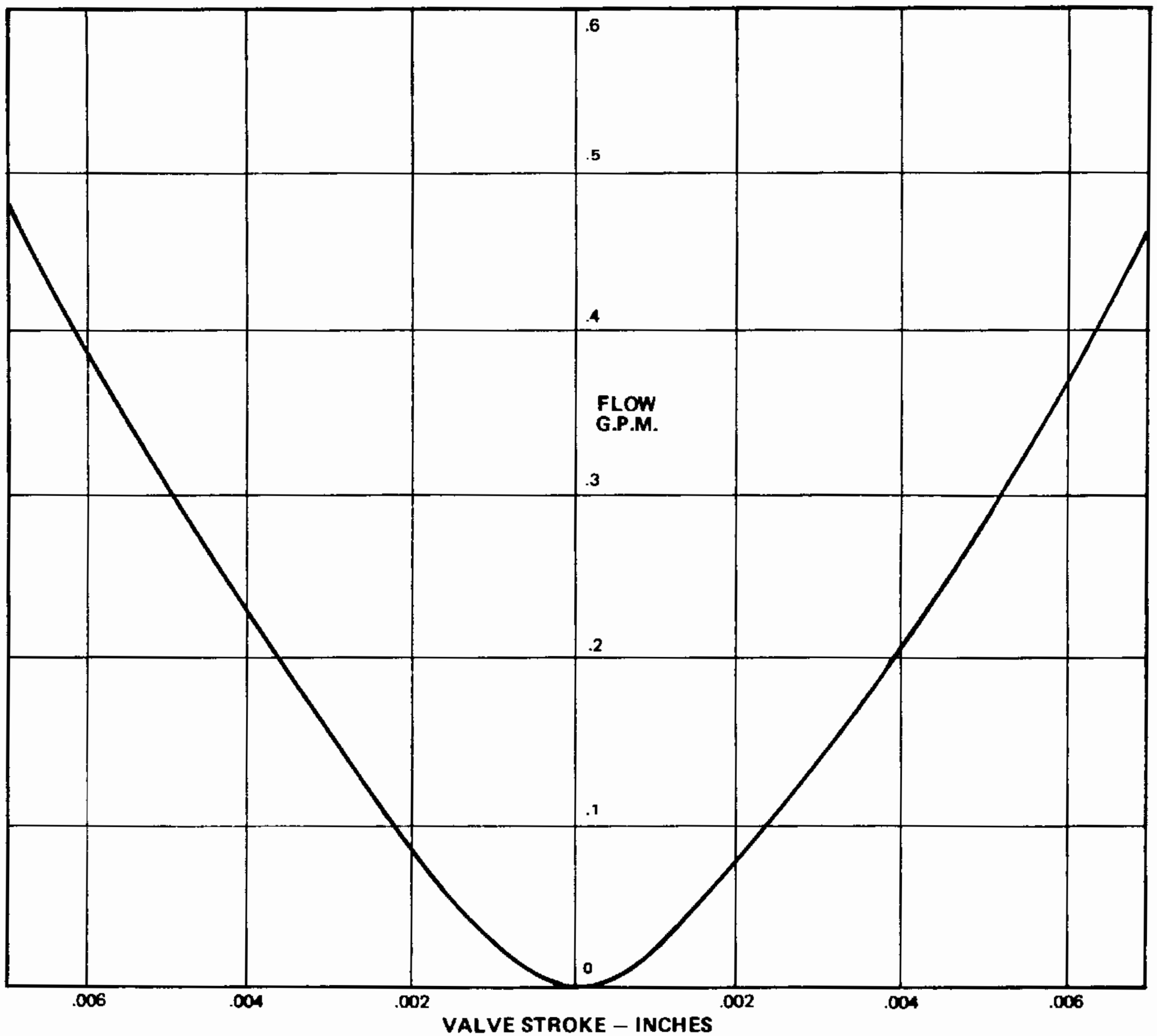


FIGURE 44. VALVE SHORT STROKE VS. FLOW CHARACTERISTIC

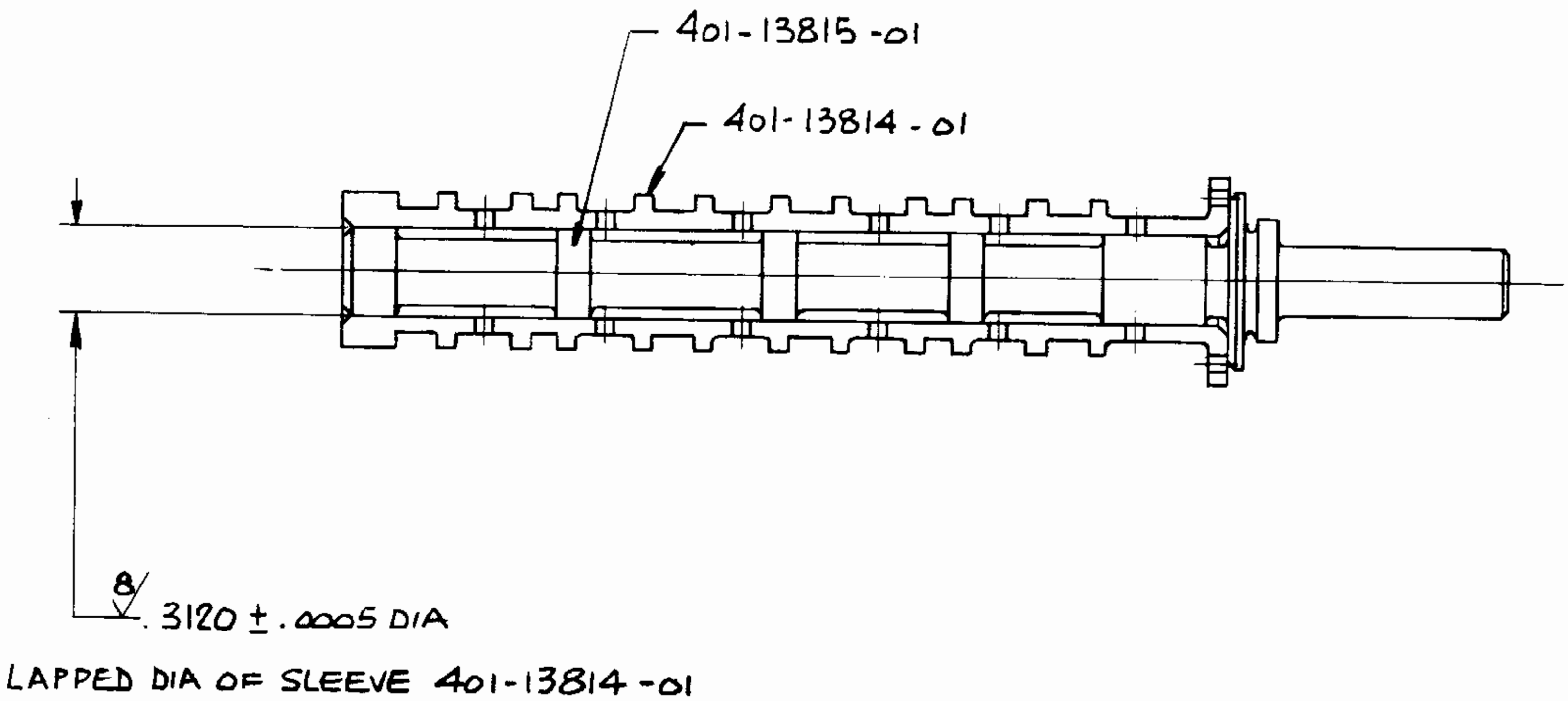


FIGURE 45. SEQUENCE VALVE

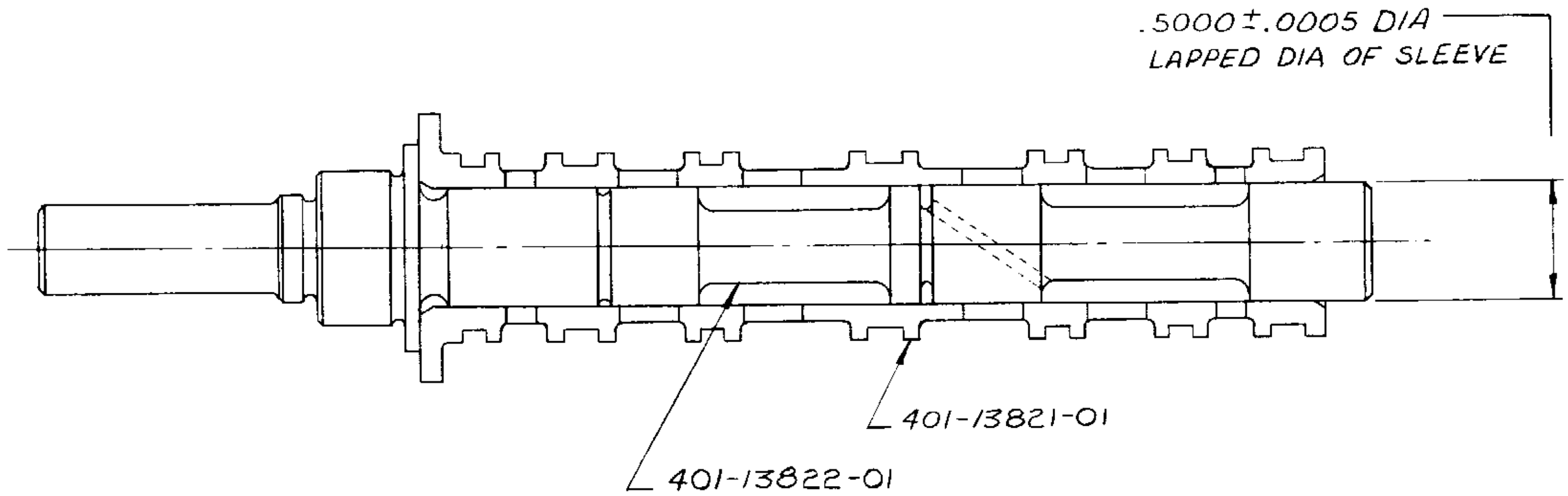


FIGURE 46. SWITCHING VALVE

operation, and a reduced flow during backup system operation. During emergency mode operation, leakage flow through this valve is very critical since it will gradually deplete the backup system reservoir. Leakage is minimized by controlling the diametrical clearance between the sleeve and slider. The valve is designed to allow a maximum leakage of 25 cc (approximately 13% of the backup system reservoir capacity) in two hours.

m. Motor Pump

The motor pump (Figure 47) is a Vickers Aerospace Division of Sperry Rand Corp design. The pump is a standard -011 Vickers pump modified to provide the soft cut-off pressure-flow characteristics shown in Figure 48. The soft cut-off is achieved by porting the pump output pressure directly to the yoke piston. Maximum pump displacement is 0.011 cubic inch per revolution. The motor is a Preco 3-phase, 400 Hz, 115/200 VAC unit with a nominal torque output of 15.4 in/lb at 11,550 rpm. It contains an explosion proof case and an integral cooling fan. Total weight of this assembly is approximately 12 lbs.

C. DESIGN

1. General

The Simplex Actuator Package was designed using standard techniques and practices. Stress analyses were conducted on parts to the degree required to ensure compliance with the specification and with the anticipated environment. Spring rate analysis was performed on the barrel assembly to ensure a stiffness compatible with the overall actuator spring rate requirements.

As indicated previously, the barrel assembly is an all steel 2-piece unit. The valve housing is 7075 aluminum unit mounted to the barrel assembly in such a way as to be sympathetic to structural flexing due to temperature and stress level changes. The valve housing design lends itself to numerical control machining. Configuration of the valve housing and the arrangement of components is such as to permit installation of the unit in the F-4. The emergency motor is rigidly mounted to the lug end half of the actuator. The pump is cantilevered from the motor and hydraulically coupled to the housing through bayonet fittings. Non-rigid attachment of the pump to the housing permits relative flexing between the pump and the housing during vibration, shock and acceleration environments.

In order to prevent the actuator areas A_1 and A_2 from cavitating when the actuator is being powered by P_2 alone, check valves in the cylinder lines permit free flow from R_1 return to the cylinders. This prevents pumping down of the reservoir due to the return relief valve action.

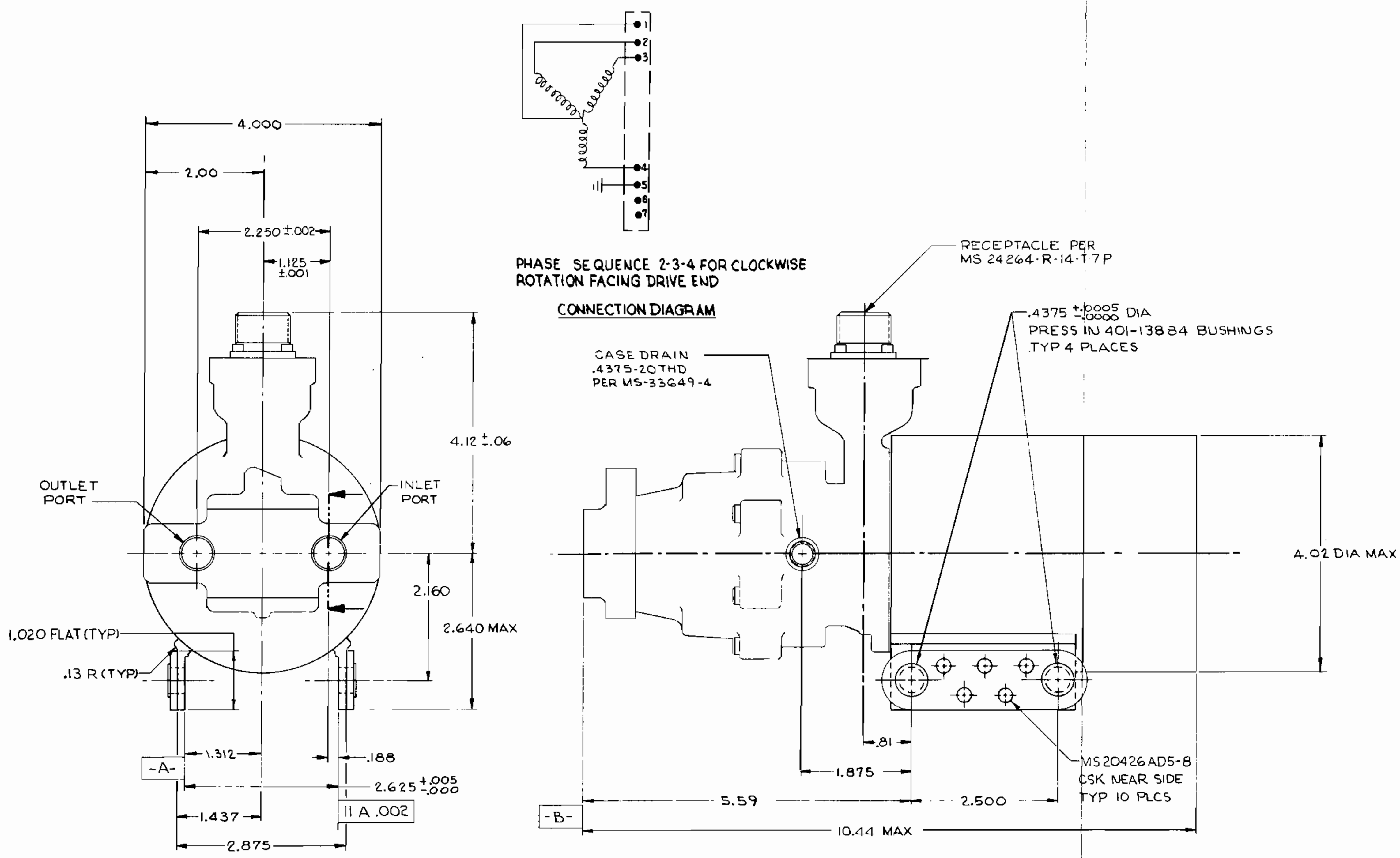


FIGURE 47. MOTOR PUMP

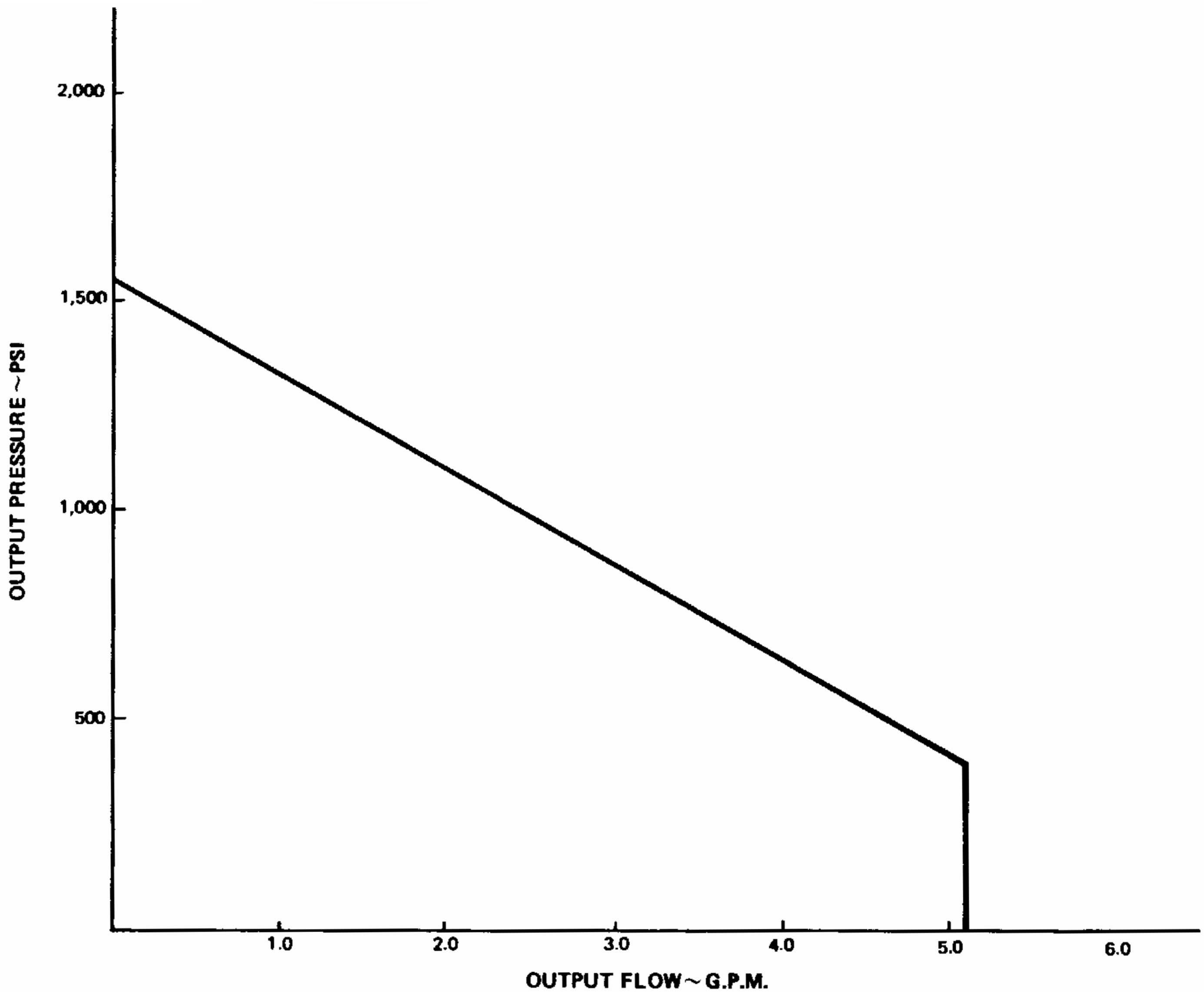


FIGURE 48. MOTOR–PUMP PRESSURE VS. FLOW CHARACTERISTIC

2. Detail Design Requirements

The Simplex actuator was designed to meet the requirements of the Air Force procurement specification. The detail design requirements and parameters for the four operating modes are listed below. Refer to Figures 27 through 48 for detail part nomenclatures and relationships.

a. Manual Mode

(1)	Area A_1	4.99 in^2
(2)	Area A_2	5.25 in^2
(3)	Area A_3	5.25 in^2
(4)	Area A_4	6.16 in^2
(5)	Ultimate actuator load, tension	29,518 lbs.
(6)	Ultimate actuator load, compression	29,518 lbs.
(7)	Ultimate load at main servo valve	±1,000 lbs.
(8)	Ultimate load at pilot's input	±482 lbs.
(9)	No-load actuator velocity with main servo valve full open and 1,000 psi source pressure	9.26 ±0.93 in/sec
(10)	Pilot input travel, normal	±0.75 in.
(11)	Pilot input travel, including over-travel	±1.13 in.
(12)	Main servo valve travel, normal	±0.105 in.
(13)	Main servo valve travel, including over-travel	±0.262 in.
(14)	Maximum flow forces at pilot's input	±1.75 lbs.
(15)	Travel ratio, input to main servo valve	7.14 in/in
(16)	P_1 and P_2 operating pressure	3,000 psi
(17)	P_1 and P_2 inlet filters, nominal rating	200 micron
(18)	Auxiliary ram centering spring pre-load	20 lbs.

b. Series Mode

(1)	Solenoid valve A (SAS) energized	28 vdc
(2)	Auxiliary ram stroke	±0.0696 in.
(3)	Series mode authority, main ram	0.352 in.
(4)	Travel ratio, auxiliary ram to main servo valve	2.83 in/in
(5)	Auxiliary ram transducer excitation	26.0 vac
(6)	Auxiliary ram transducer scale factor	18.48 ±0.18 v/in.
(7)	Maximum electro-hydraulic valve current	15.0 ma
(8)	Electro-hydraulic valve flow gain	0.106 in^3/sec/ma
(9)	Auxiliary ram area	0.304 in^2

c. Parallel Mode

(1)	Solenoid valves A (SAS) and B (A/P) energized	28 vdc

(2)	Auxiliary ram stroke	±0.2972 in.
(3)	Override force at pilot's input	19.0 ±2.0 lbs.
(4)	Travel ratio, pilot's input to locking pistons	4.63 in/in.
(5)	Main ram transducer excitation	26 vac
(6)	Main ram transducer scale factor	4.0 ±0.1 V/in.
(7)	Hole diameter of locking piston damper	0.006 in.

d. Backup Mode

(1)	P_1 pressure (or solenoid C energized)	0 psi
(2)	P_2 pressure	0 psi
(3)	Minimum actuator rate with main servo valve full open	3.5 in./sec
(4)	P_1 and P_2 pressure switch operating point	1,125 ±125 psi
(5)	P_E pressure switch operating point	300 ±50 psi
(6)	Motor voltage	3ϕ, 115/200 vac, 400 Hz
(7)	Motor current, per phase, at maximum power	12 amps
(8)	Maximum starting current	72 amps
(9)	Pump output pressure at no flow	1,550 ±50 psi
(10)	Pressure line filter, nominal rating	10 micron
(11)	Case drain line filter, nominal rating	200 micron
(12)	Reservoir capacity	11.83 in^3
(13)	Reservoir pressure	30 ±6 psi
(14)	Output pressure relief valve cracking pressure	2,050 ±200 psi
(15)	Return pressure relief valve cracking pressure	330 ±20 psi
(16)	Pump case proof pressure	500 psi
(17)	Hole diameter of reservoir filling restrictor	0.006 in.

3. Emergency System Reservoir Sizing

The reservoir volume is sized to provide sufficient fluid for a minimum of two hours of emergency system operation. The primary loss or use of fluid is due to the actuator differential area. If the actuator is completely extended when the emergency system is activated, a significant amount of fluid is drawn from the reservoir when the piston is retracted. In addition, fluid compressibility, switching losses, leakage, and thermal contraction effects must be considered. The following list makes up the reservoir volume requirement.

Switching valve shuttling volume displacement	1.00 in^3
Switching valve loss when shuttling	1.50 in^3
Differential cylinder volume	2.74 in^3
Fluid compression due to oil compressibility	0.10 in^3
Leakage through switching circuit during two hours	1.74 in^3
Thermal contraction of fluid from 330°F to 160°F	0.80 in^3
Total	7.98 in^3
50% safety factor	3.94
Reservoir volume required	11.82 in^3

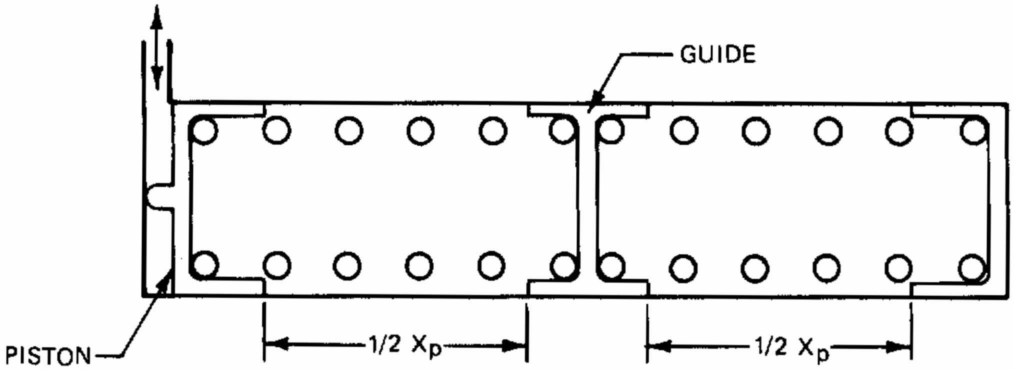
A reservoir diameter of 2.245 in. was selected to take advantage of a standard seal. The piston area is

$$A_P = (0.785)(2.245)^2$$
$$= 3.96 \text{ in.}^2$$

Required stroke is

$$X_P = 11.83/3.96$$
$$= 3.00 \text{ in.}$$

To prevent buckling of the reservoir springs a tandem arrangement as shown below was used.



To provide reservoir pressure of 24 psi at the piston extend position, each spring preload must be

$$F = (A_P) \text{ Pressure}$$
$$= (24)(3.96)$$
$$= 95 \text{ lbs.}$$

At full retract position the spring loads required for 36 psi pressure (assuming zero friction) are

$$F = A_P(\text{Pressure})$$
$$= (36)\, 3.96$$
$$= 142 \text{ lbs.}$$

Each spring has a rate of

$$K = \frac{142-95}{1.5}$$
$$= 31.5 \text{ lbs/in.}$$

and the free length is longer than the preloaded length by

$$Lf = 95/31.5$$
$$= 3 \text{ in.}$$

The tandem springs each consist of three concentric elements sharing the spring load. Operating at a relatively low stress level results in infinite fatigue life.

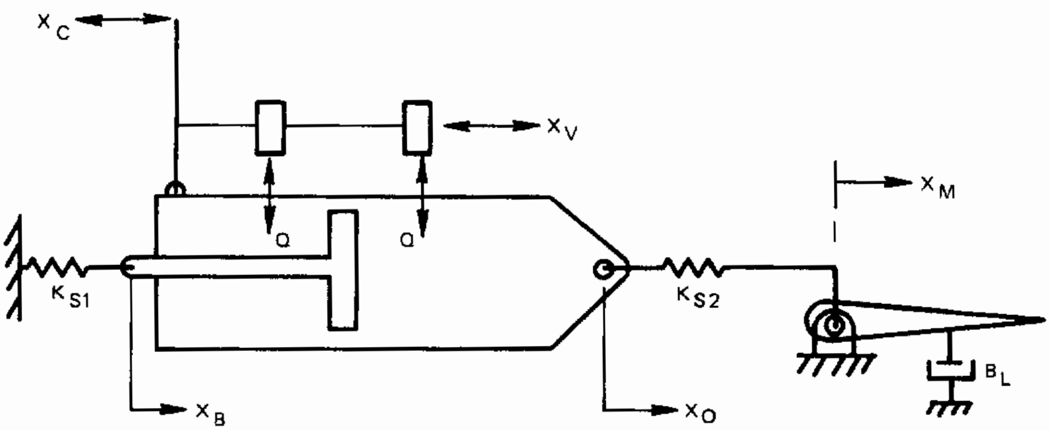
D. PERFORMANCE ANALYSIS

1. General

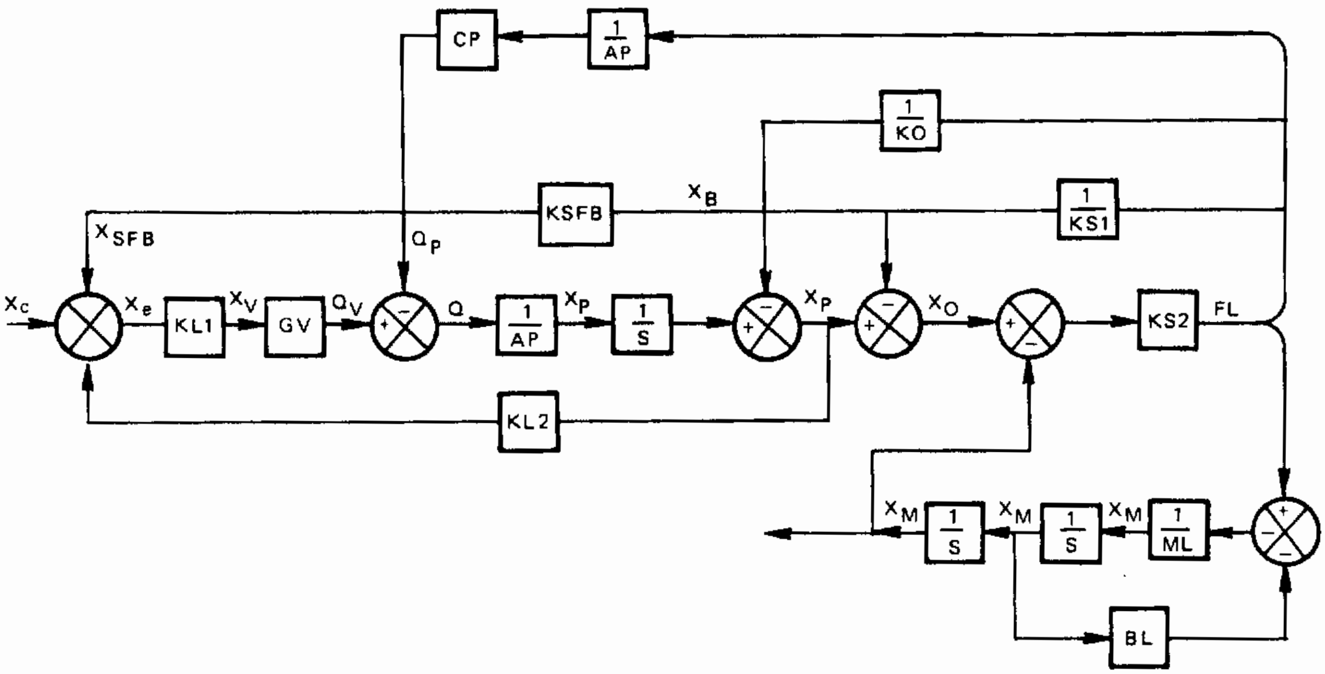
A servo dynamic analysis and a thermal analysis were conducted on the Simplex. Also a reliability study was conducted to establish the mean time between failure for the various Simplex operating modes. This analysis, included in Appendix IV shows the highest MTBF in the manual mode and the lowest MTBF in the emergency mode.

2. Stability Analysis

A limited analysis was conducted on the Simplex to determine the approximate degree of stability when installed in the F-4 and driving the stabilator surface. The actuator, when operating in the manual mode, can be represented by the following system schematic.



The system can be represented by the following block diagram:



where:

X_C	=	Input signal, in.
K_{L1}	=	Input linkage gain, in/in.
G_V	=	Valve flow gain, in^3/sec/in.
A_P	=	Piston area, in^2
K_{S2}	=	Spring rate-piston to surface, lbs/in.
F_L	=	Actuator force, lbs
M_L	=	Load Mass, lbs sec^2/in.
B_L	=	Inherent surface damping, lbs/in/sec
K_{S1}	=	Actuator structure spring rate, lbs/in.
K_O	=	Actuator spring rate, lbs/in.
K_{SFB}	=	Structural feedback gain, in/in.
C_P	=	Valve conductance coefficient, in^3/sec/psi
K_{L2}	=	Feedback linkage gain, in/in.

The total structural spring rate consists of the two springs in parallel and is described by

$$K_{S12} = \frac{(K_{S1})(K_{S2})}{K_{S1} + K_{S2}}$$

The equivalent system spring rate can then be described as

$$K_{EQ} = \frac{(K_O)(K_{S12})}{K_O + K_{S12}}$$

The above block diagram can be reduced, and the following open loop transfer function evolved.

$$G_{(S)} H_{(S)} = \frac{\dfrac{K_{L1} K_{L2} \cdot G_V \cdot K_{EQ}}{A_P \cdot K_{S12}} \left(S^2 + \dfrac{B_L}{M_L} S + \dfrac{K_{S12}}{M_L} \right)}{S\left(S^2 + \left(\dfrac{B_L}{M_L} + \dfrac{K_{EQ} \cdot C_P}{A_P^2} + \dfrac{K_{EQ} \cdot K_{L1} \cdot G_V \cdot K_{SFB}}{A_P \cdot K_{S1}}\right) S + \dfrac{K_{EQ}}{M_L} + \left(\dfrac{K_{EQ} \cdot C_P}{A_P^2} + \dfrac{K_{EQ} \cdot K_{L1} \cdot G_V \cdot K_{SFB}}{A_P K_{S1}}\right) \dfrac{B_L}{M_L}\right)}$$

The following parameters reflect the Simplex actuator characteristics. The surface damping coefficient reflects an inherent surface damping which results in a numerator damping ratio of 0.03. The value of valve conductance coefficient is based on nominal valve parameters. The valve flow gain value is the maximum anticipated.

K_{L1}	=	0.1400	C_P	=	0.0005000	K_{S2}	=	192500.00
K_{L2}	=	1.0000	K_O	=	454000.00	K_{EQ}	=	103630.06
G_V	=	500.000	K_{S12}	=	134281.13	K_{SFB}	=	1.0000
A_P	=	5.6350	K_{S1}	=	444000.00	B_L	=	38.0819
M_L	=	3.0000						

Substitution of these values into the open loop transfer equations results in the following transfer function:

$$G_{(S)}H_{(S)} = \frac{9.587\,(S^2 + 12.7S + 44760)}{S(S^2 + 11.43S + 34527)}$$

The roots are:

Numerator

$-6.347 + j\,185.7$
$-6.347 - j\,185.7$

Denominator

$-5.713 + j\,185.7$
$-5.713 - j\,185.7$

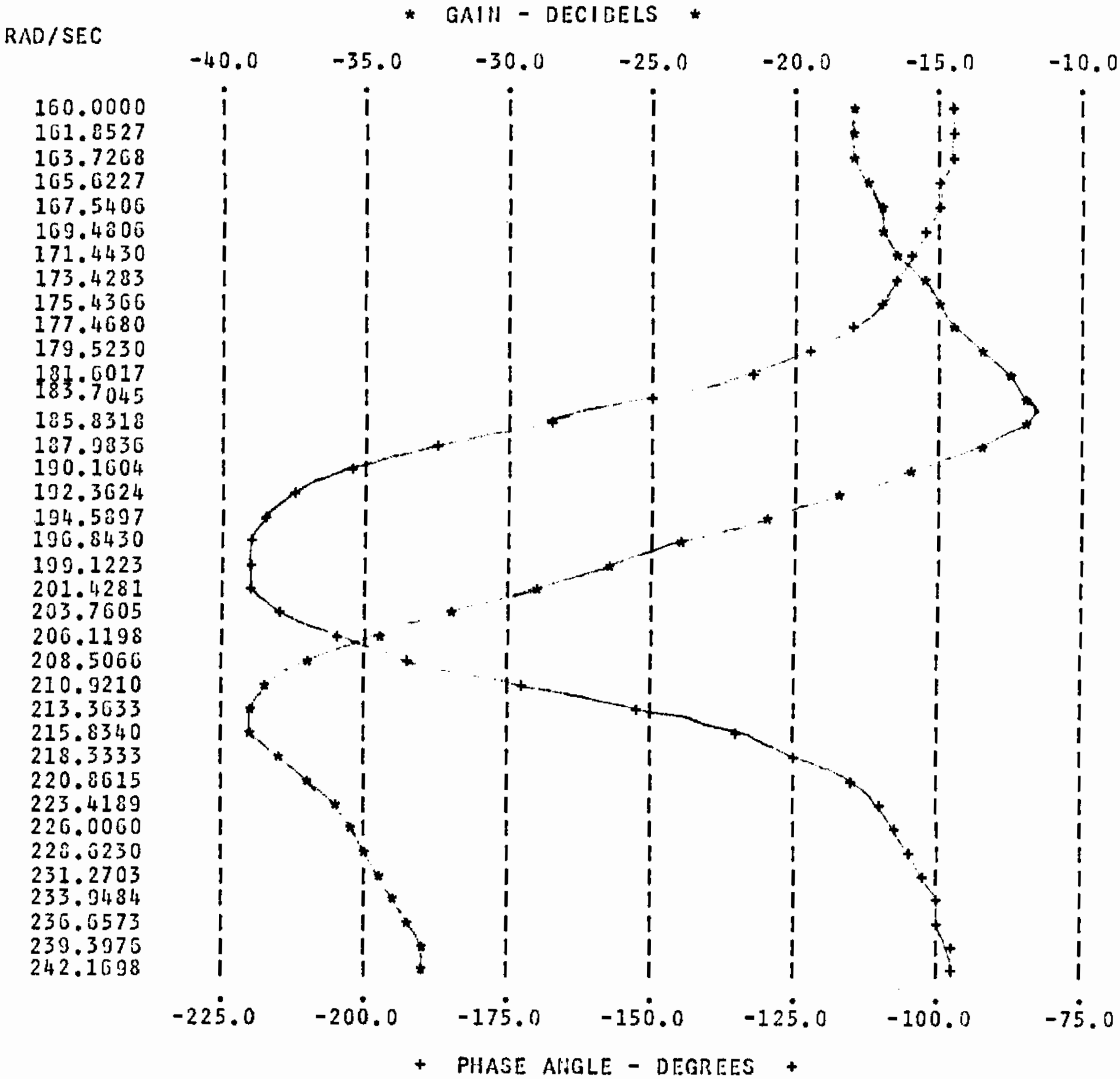
The closed loop relationship is:

$$\frac{9.5868\,(S^2 + 12.7S + 44760)}{S^3 + 21S^2 + 34649\,S + 429110}$$

having denominator roots

$$-12.4 + j\,0.0$$
$$-4.295 + j\,185.81$$
$$-4.295 - j\,185.81$$

A frequency response evaluation of the open loop transfer function in the frequency range of the denominator resonance indicates a gain margin of approximately 10 dB as shown below:



3. Thermal Analysis

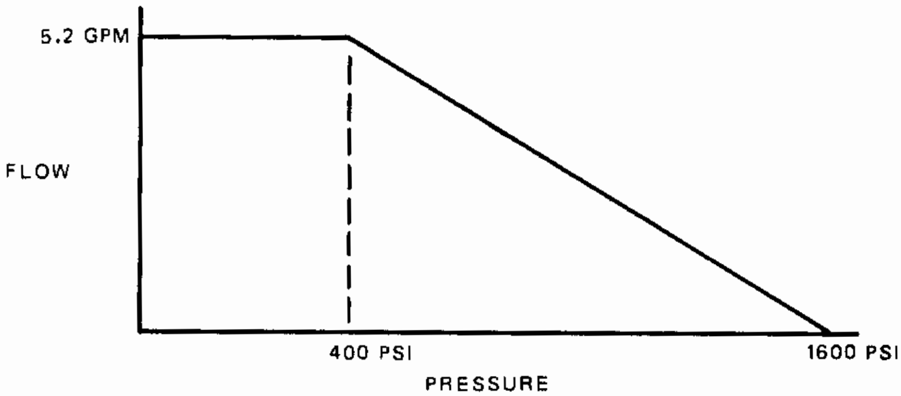
a. General

The fluid and package temperatures resulting from closed hydraulic circuit operation of integrated packages may be a limiting factor in their usage. Although the Simplex emergency system is not a continuous duty system, the transient thermal characteristics may be significant. A simplified thermal analysis was conducted to evaluate a nominal package temperature with the Simplex in the emergency mode. The operating temperature of the package is a function of the system duty cycle and operating conditions.

The specific actuator operating condition which was evaluated was at 5.2 gpm 400 psi pump output power. This condition corresponds to the maximum full flow pump pressure, and is consistent with anticipated Simplex operation.

b. Analysis

The Simplex pump characteristic curve is shown below. The operating point was taken at 5.2 gpm and 400 psi. It was assumed that all the resultant energy is converted to heat inside the package. Consequently,



the following thermal balance exists:

heat generated = heat absorbed + heat transferred.

$$H_g = H_a + H_t$$

where:

H_g = energy due to pressure and flow and pump inefficiency
= $(1.484)(5.2)(400) + 400 = 3490$ BTU/HR

$$H_a = W_{al}C_{pal}\frac{dT}{dt} + W_sC_{ps}\frac{dT}{dt} + W_oC_{po}\frac{dT}{dt}$$

W_{al}	=	weight of aluminum, lbs
W_s	=	weight of steel, lbs.
W_o	=	weight of oil, lbs.
C_{pal}	=	specific heat of aluminum, BTU/lbs^oF
C_{ps}	=	specific heat of steel, BTU/lbs ^oF
C_{po}	=	specific heat of oil, BTU/lbs^oF

Making some assumptions for the effective weight of stainless steel, aluminum, and oil, the heat absorbed relationship becomes:

$$H_a = [(20)(0.214) + (26)(0.107) + (2.5)(0.6)]\frac{dT}{dt}$$

The "heat transferred" relationship becomes:

$$H_t = \text{heat convected} + \text{heat radiated}$$

$$= H_1 A_1 (T - T_a) + H_2 A_2 (T^4 - T_a^4)$$

where:

H_1	=	convection coefficient which is a function of air velocity and is assumed to be 3.0 BTU/HR^oF ft^2
A_1	=	convection surface area (assumed 3.63 ft^2)
H_2	=	radiation coefficient
A_2	=	radiation surface area (2.81 ft^2)
T_a	=	ambient temperature, ^oR
T	=	surface temperature, ^oR

Combining:

$$3490 = 8.56\frac{dT}{dt} + 10.9(T - T_a) + 0.44 \times 10^{-8}(T^4 - T_a^4)$$

This nonlinear differential equation was then solved for the following conditions with the aid of a digital computer routine.

Condition	Initial Temperature	Ambient Temperature
I	620^oR	620^oR
II	660^oR	660^oR
III	735^oR	735^oR

c. Results

Figure 49 shows the results of the computer runs. For initial package temperatures of 160°F, 200°F, or 275°F, the surface temperatures after 30 minutes of operation will reach 292°F 328°F, and 398°F, respectively.

d. Conclusions

Based on the surface temperature resulting from the maximum full pressure operation, the Simplex unit will function satisfactorily for a minimum of two hours under normal emergency system operating conditions.

As part of the environmental tests during qualification, a test was conducted to evaluate the thermal characteristics of the Simplex package during emergency system operation. Figures 50, 51, and 52 show the results of these tests. It is noted that the results are similar to the predicted performance.

E. TESTING

1. Component Testing

All critical components of the Simplex Actuator were tested individually before they were assembled into the actuator. Due to cost and schedule considerations only informal component test procedures were prepared. Tests were conducted by engineering laboratory personnel and were witnessed by reliability quality assurance personnel and resident government inspector. Test data on each component were recorded and entered into the log book of each actuator. The components and the tests that were performed were as follows:

a. Linear Transducers

(1) Dimensional inspection

(2) Null voltage

(3) Phase shift

(4) Scale factor

(5) Electrical stroke

(6) Phasing

(7) Linearity

(8) Dielectric strength

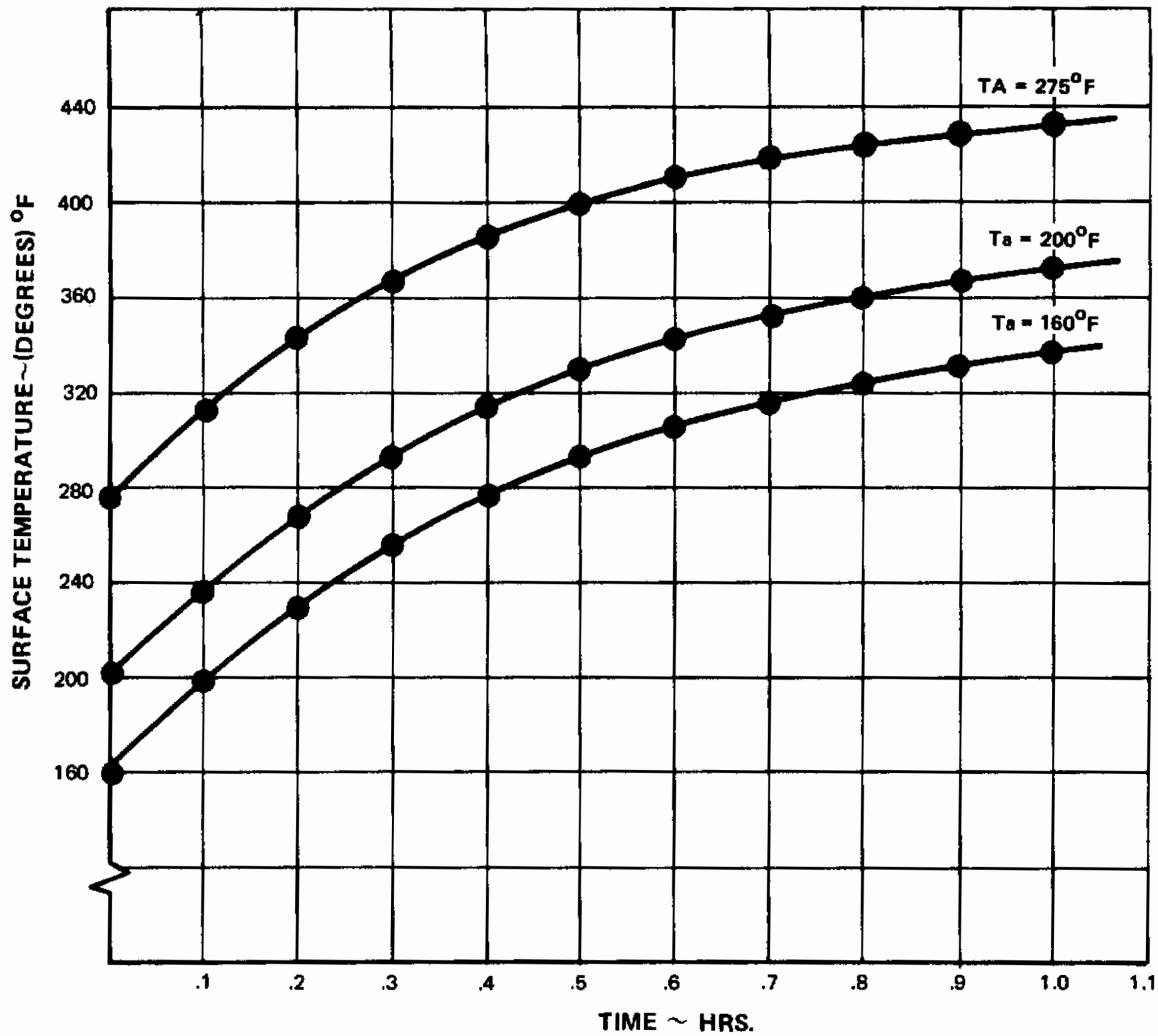


FIGURE 49
SIMPLEX THERMAL CHARACTERISTICS
PUMP OPERATING AT 5.2 GPM, 400 PSI
TA = AMBIENT & INITIAL SURFACE TEMPERATURE

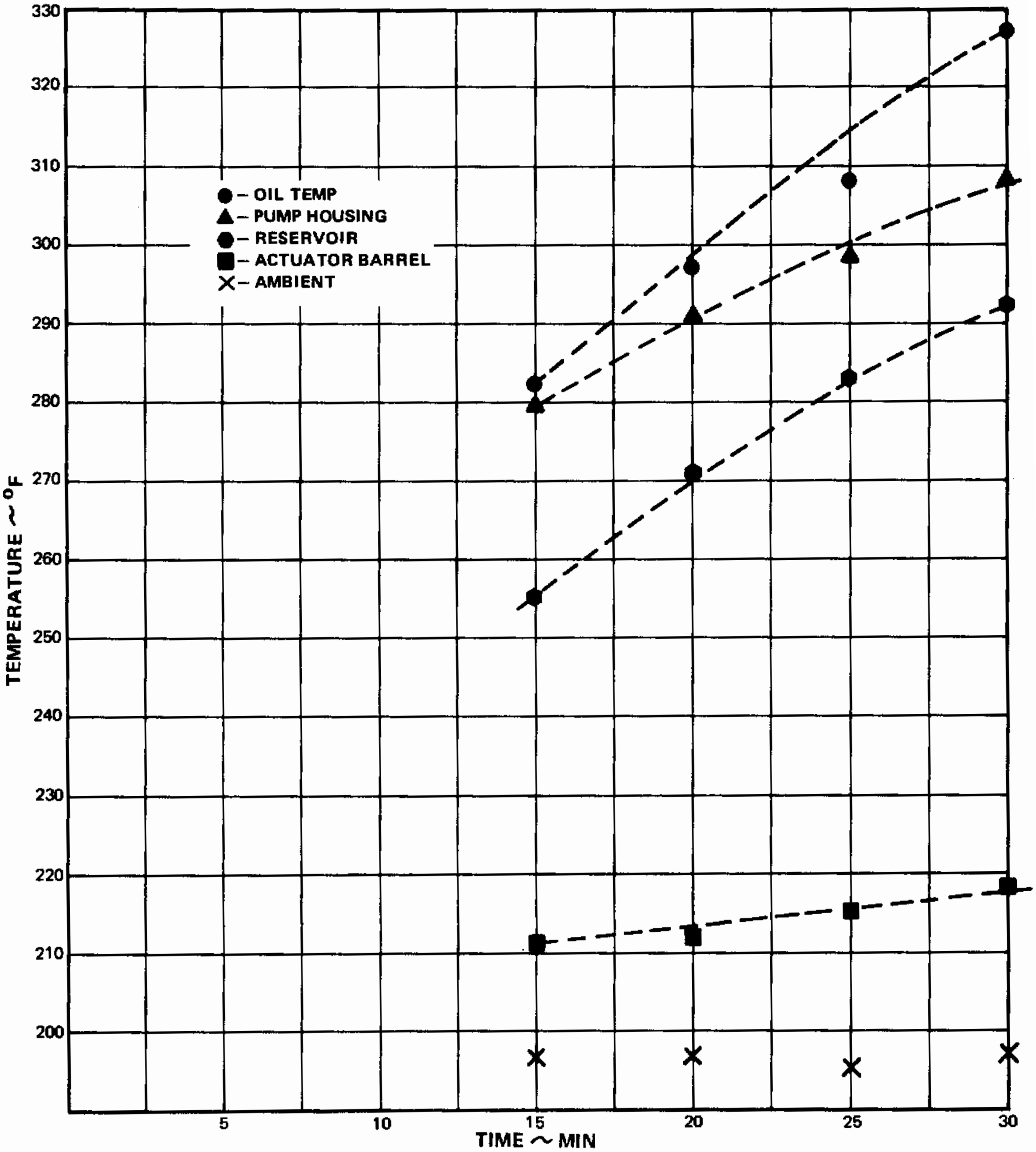


FIGURE 50
EMERGENCY SYSTEM THERMAL TEST – RATED SPEED AND FULL FLOW PRESSURE

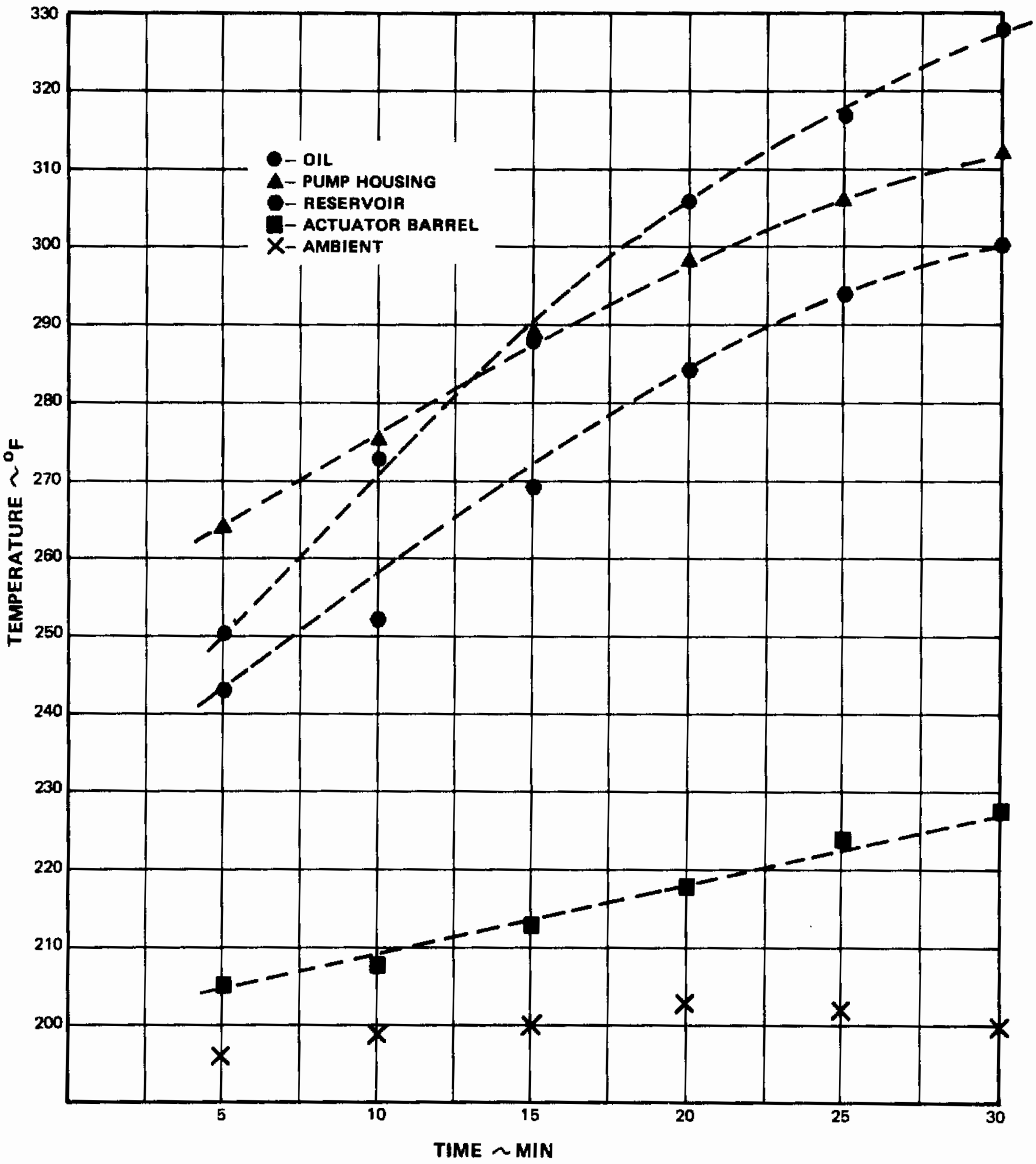


FIGURE 51
EMERGENCY SYSTEM THERMAL TEST – RATED SPEED CYCLING BETWEEN RATED PRESSURE AND MAX FULL FLOW AT 6 GPM

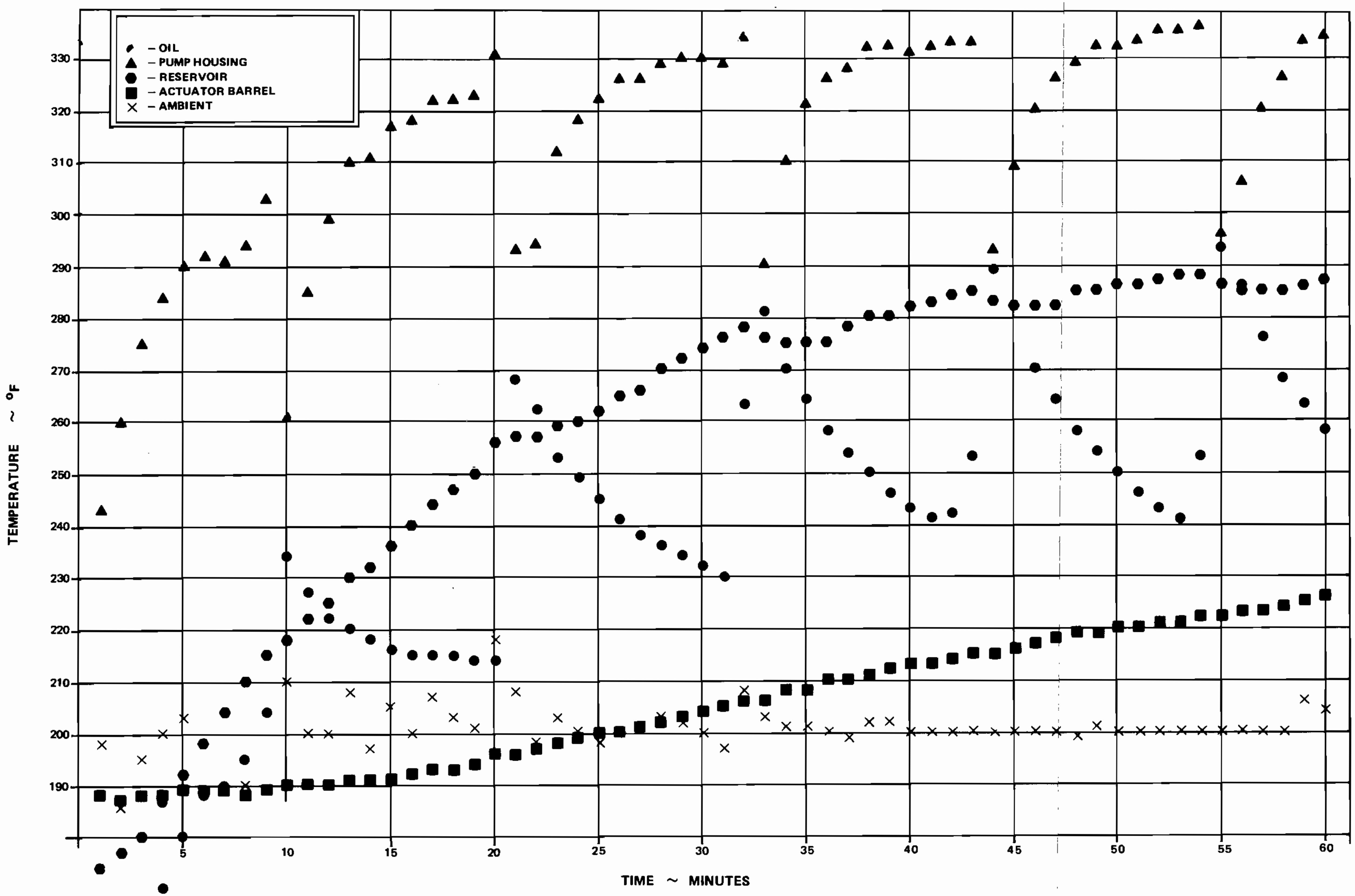


FIGURE 52 EMERGENCY SYSTEM THERMAL TEST – RATED PRESSURE FOR 10 MINUTES THEN MAX FULL FLOW PRESSURE FOR 1 MINUTE

b. Electro-Hydraulic Servo Valve

(1) Dimensional inspection

(2) Dielectric strength

(3) Coil resistance

(4) Proof pressure

(5) Polarity

(6) Pressure gain

(7) Flow gain

(8) Threshold

(9) Null current

c. Pressure Switch

(1) Dimension inspection

(2) Proof pressure

(3) Actuation pressures

(4) Wiring continuity

(5) Dielectric strength

d. Solenoid Valves

(1) Dimensional inspection

(2) Proof pressure

(3) Flow rate at rated pressure

(4) Solenoid current

(5) Pressure drop at rated flow

(6) Porting

(7) Internal leakage

(8) External leakage

(9) Dielectric strength

e. Check Valves

(1) Dimensional inspection

(2) Proof pressure

(3) Leakage in reverse direction

(4) Pressure drop at rated flow

f. Motor Pump

(1) Dimensional inspection

(2) Proof pressure

(3) Pressure – flow curve

g. Master Servo Valve

(1) Dimensional inspection

(2) Synchronization

(3) Neutral leakage

(4) Pressure gain

(5) Flow gain

(6) Friction

h. Sequence Valve

(1) Dimensional inspection

(2) Internal leakage

(3) Pressure drop at rated flow

(4) Friction

i. Switching Valve

(1) Dimensional inspection

(2) Internal leakage

(3) Pressure drop at rated flow

(4) Friction

j. Relief Valve

(1) Dimensional inspection

(2) Proof pressure

(3) Cracking pressure

(4) Reseat pressure

(5) Pressure drop at rated flow

2. System Functional Tests

An Acceptance Test Procedure (ATP) 406-10571 was prepared in accordance with the functional tests defined by the Government Procurement Specification. Each actuator was tested in accordance with the ATP. The tests were witnessed and verified by LTV reliability and quality assurance personnel and by resident government inspector. A detailed summary of the functional tests is presented in Appendix IV. The following functional tests were conducted on each actuator.

a. Actuator Stroke Measurement

b. Auxiliary Ram Rigging

c. Locking Piston Adjustment and Autopilot

d. Transient Velocity

e. Examination

f. Weight

g. Proof Pressure

h. Operation (Manual & Autopilot Modes)

i. Piston Velocity for Small Input Displacements

j. Structural Deflection

k. Servo Valve Synchronization

l. Overpower Forces

m. Capability of Pilot to Reverse Direction of Cylinder

n. Valve Travel and Flow

o. Auxiliary Ram Frequency Response

p. LVDT's and Servo Valve Phasing

q. Null Current and Threshold

r. P_1 and P_2 Pressure Switch Operating Test

s. Relief Valve Tests (Pressure drop, Cracking and Reseat)

t. Emergency Reservoir Fill Test

u. Emergency Pump Static Output Pressure

v. Emergency System Velocity Test at Normal and Electrical Power Extremes

w. Emergency System Switching Times

x. Loss of Reservoir Fluid during Switching

y. Internal Leakage – Neutral and Loaded

z. Valve Force (Static & Dynamic)

aa. Force Feedback in Series Mode

ab. Frequency Response – Emergency System

3. Qualification Tests

The qualification testing was performed in accordance with the applicable paragraphs of the Government specification. The life and endurance, backup system endurance, compatibility, full stroke cycle, impulse, high and low temperature environments, solenoid valve endurance, ultimate load, and burst pressure tests were performed at the Arlington plant by engineering laboratory personnel. Vibration, shock, humidity, EMI and acceleration tests were performed by the Garland Environmental Test laboratory. The Simplex actuator passed all qualifications test satisfactorily. A detailed summary of the qualification tests and results is given in Appendix VI.

SECTION VI

DUPLEX INTEGRATED ACTUATOR PACKAGE

A. GENERAL

The Duplex IAP developed as part of this program consists of a 2-piece dual tandem, steel actuator controlled by a redundant FBW signal converter and powered by dual self-contained hydraulic power supplies with associated hydraulic circuitry. The following guidelines were established for the Duplex design. The performance requirements were based on the F-4 stabilator actuator. The signal channels, or signal converter, are dual-fail operate; i.e., operate with possible small degraded performance after two failures. Power system is single-fail operate. Due to cost and time constraints, a ground rule similar to the one on the Simplex was established that components used in this package were to be state-of-the-art design.

The signal converter unit consists of four electro-mechanical actuators. The outputs are force summed on a torque tube which in turn operates the dual tandem servo valve for controlling hydraulic flow to the dual tandem actuator. Hydraulic power is provided by two load-pressure compensated pumping units driven by 3-phase, 400 Hz motors.

B. TRADE STUDIES

Consistent with the program goals to establish optimum designs and design techniques, trade studies were conducted in two areas to evolve the best over-all system. One trade study was conducted to determine the best redundant FBW signal converter mechanism to be used on the Duplex. The best signal conversion system should be simple and reliable and meet the Duplex performance and environmental requirements. Another trade study was conducted for the purpose of optimizing the electro-hydraulic power system. A significant consideration in the design of IAP's is the thermal load which is due to quiescent power losses and to the IAP operation. Consequently, the purpose of this study was to determine the most efficient system which would adequately perform the desired function to minimize the package thermal load.

1. Signal Conversion System Trade Study

There are almost an infinite number of combinations and arrangements of conversion devices, components, monitoring techniques, logic schemes, etc. that can be employed in a redundant FBW mechanization. The approach taken in this study was to conduct a preliminary investigation and reduce the conversion concepts to a reasonable number which could then be evaluated in detail. Nine systems were evolved. With the exception of one electro-hydraulic (E/H) mechanization in the active-standby category the nine systems are of an electro-mechanical (E/M) configuration. The preliminary evaluation indicated that for the Duplex IAP system a four channel E/H signal conversion system would not be optimum. The primary reasons are: only two hydraulic supplies are available for the four channels (quadruple redundancy would be compromised), and the quiescent power losses in E/H servo valves adds to the power losses in the package and further aggravates the temperature problem. Therefore for this particular application it was felt that E/M concepts were the best approach. The systems were grouped into three basic categories as Active-Standby, Force Summing and Displacement Summing. In addition the signal conversion systems can be mechanized in cascade with or as an inner loop inside the power servo loop (inside or outside the power servo loop). A listing of the systems is presented in Table XVI. A summary description follows:

TABLE XVI. SYSTEM DESCRIPTION SIGNAL CONVERSION MECHANIZATION

A Active-Standby	B Force Summing	C Displacement Summing
1. Electro-hydraulic Mechanization Inside the loop	1. Electro-mechanical 5 actuators Inside the loop	1. Electro-mechanical 4 actuators Inside the loop
2. Electro-mechanical 4 actuators Inside the loop	2. Electro-mechanical 4 actuators Inside the loop	2. Electro-mechanical 4 actuators Outside the loop
3. Electro-mechanical 4 actuators Outside the loop	3. Electro-mechanical 4 actuators Outside the loop	3. Electro-mechanical 4 actuators Inside the loop Non-limited disp.

a. System A-1 (Figure 53)

This system is basically composed of four electro-hydraulic pressure control valves, a main stage servo valve, four hydraulic comparators, and two switching valves. The main servo valve has piston areas with 2 to 1 ratio and is spring loaded to the center (neutral) position. During normal operation the four electro-hydraulic servo valves are operated and produce an output pressure which is a function of their driving signal. The comparators monitor the difference between the four pressures while the pressure from one of the valves controls the main stage servo valve position. In the event that a discrepancy occurs between the main stage servo valve control pressure and that of the comparator channels, the switching valve actuates and exchanges functions between the active channel and one of the standby channels. Three of the electro-hydraulic servo valves can be engaged in this manner while the fourth functions only as a monitor.

b. System A-2 (Figure 54)

The main dual tandem servo valve of this system is driven by electro-mechanical actuator No. 1 through a solenoid operated clutch. All E/M actuators operate

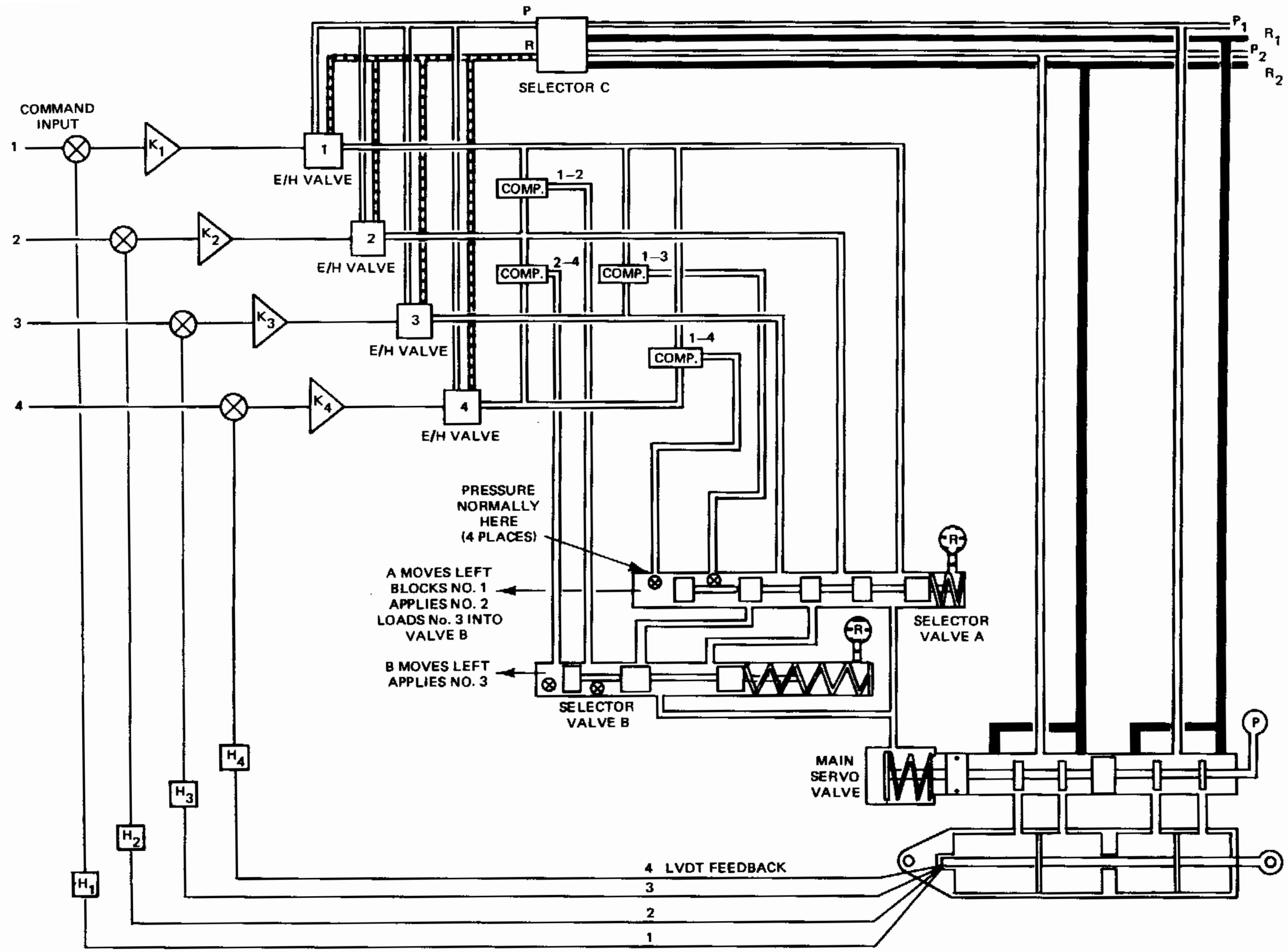


FIGURE 53. SYSTEM A–1 ACTIVE STANDBY WITH ELECTROHYDRAULIC VALVES

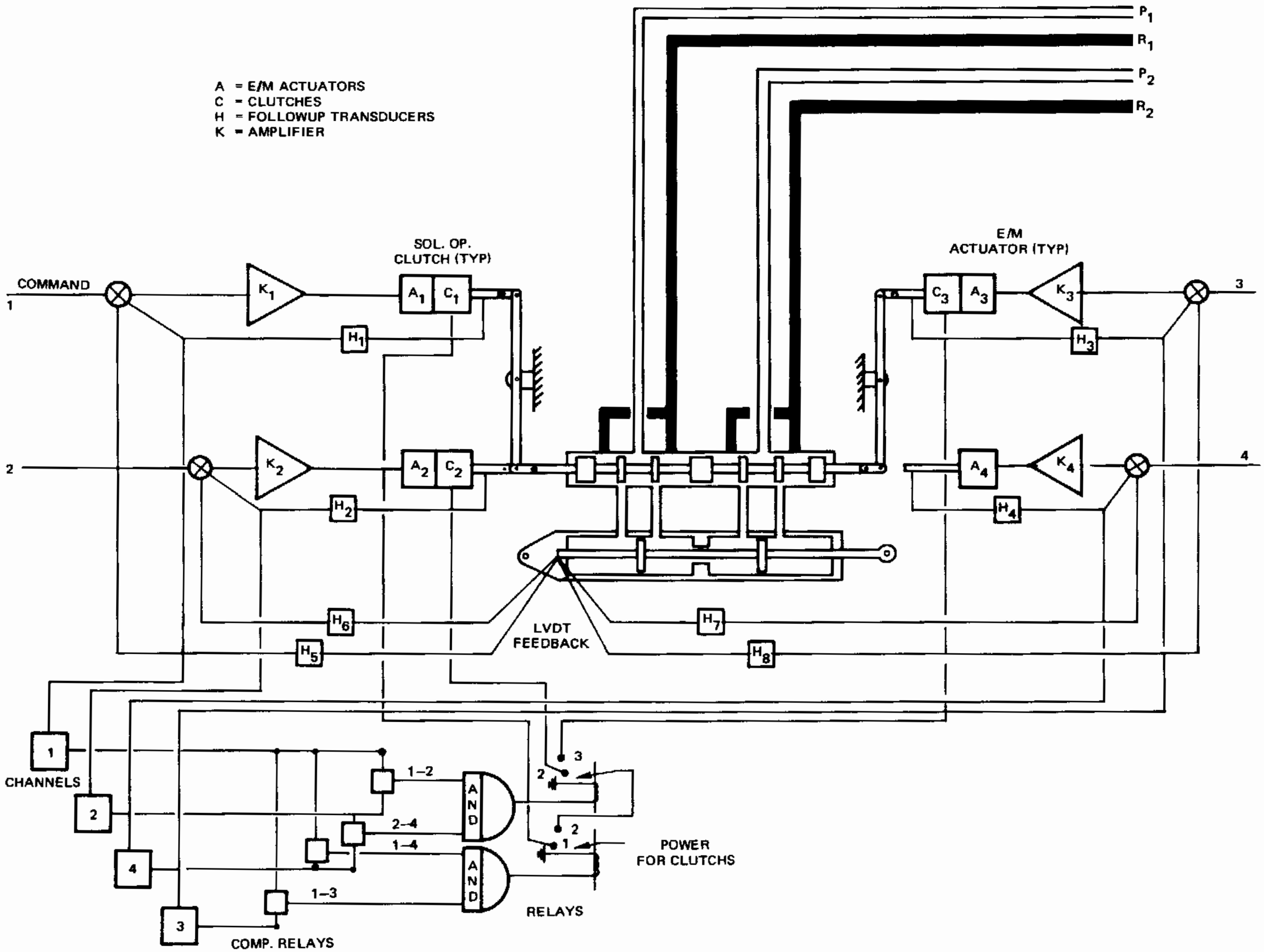


FIGURE 54. SYSTEM A–2 ACTIVE STANDBY WITH ELECTROMECHANICAL ACTUATOR INSIDE THE FEEDBACK LOOP

together, but clutches No. 2 and No. 3 are not engaged. A monitor actuator, No. 4 is used to provide an additional position signal for comparison with the other three. Main actuator position is fed back to each E/M actuator by means of a 4-channel LVDT.

The position feedback signal of each E/M control actuator is amplified in its respective monitoring channel for comparison in the following combinations: 1-2, 1-3, 1-4, and 2-4. Should actuator No. 1 lose power, jam, or fail in any way, the difference in its position feedback signal relative to those of actuators 2, 3, and 4 would cause the monitoring channels to trigger and lock comparator relays 1-2, 1-3 and 1-4. Relays 1-3 and 1-4 complete the circuit for triggering relay B which in turn switches power from clutch No. 1 to clutch No. 2. Relays A and B are wired such that they are self-locking and cannot be released except by means of a circuit breaking reset button. If actuator No. 2 fails, relay A is triggered and power is switched to clutch No. 3.

c. System A-3 (Figure 55)

The operation of this system is essentially identical to that of system A-2 except that the position feedback from the main actuator is mechanical, not electrical, and is summed down stream of the signal converters. The E/M actuator selection system is the same as that of system A-2, eliminates hydraulics in the logic, and tolerates two signal failures plus one hydraulic failure.

d. System B-1 (Figure 56)

In this system five electro-mechanical actuators drive the main valve simultaneously. The output of each actuator is transmitted to the main servo valve through a funk spring. Funk spring breakout force is equal for all actuators, but is made higher in one direction than the other to permit positive control under all conditions. Should any two of the actuators fail in any combinations or sequence of jams and opens, the remaining three combined have the capability of driving the main valve by breaking out the funk springs of the failed units. The difference in force required to break out a funk spring in one direction as opposed to the other assures that at least one of the remaining units is operating within its detent range. A transient effect is produced in reaching the new position where at least one spring will be in its detent range after a failure; however, the system retains its stiffness no matter what the failure.

In the event of a failure in one channel, the remaining channels control the output and the actuator in the failed channel causes its funk spring to collapse and the cut-out or failure switch to be activated. The switch in turn removes power from the failed channel and allows the failed motor to be back-driven by the remaining channels through the high-efficiency ball screw gearing. In the event of a jam in a motor, the remaining channels breakout the funk spring in the failed channel and simply over power the failed unit.

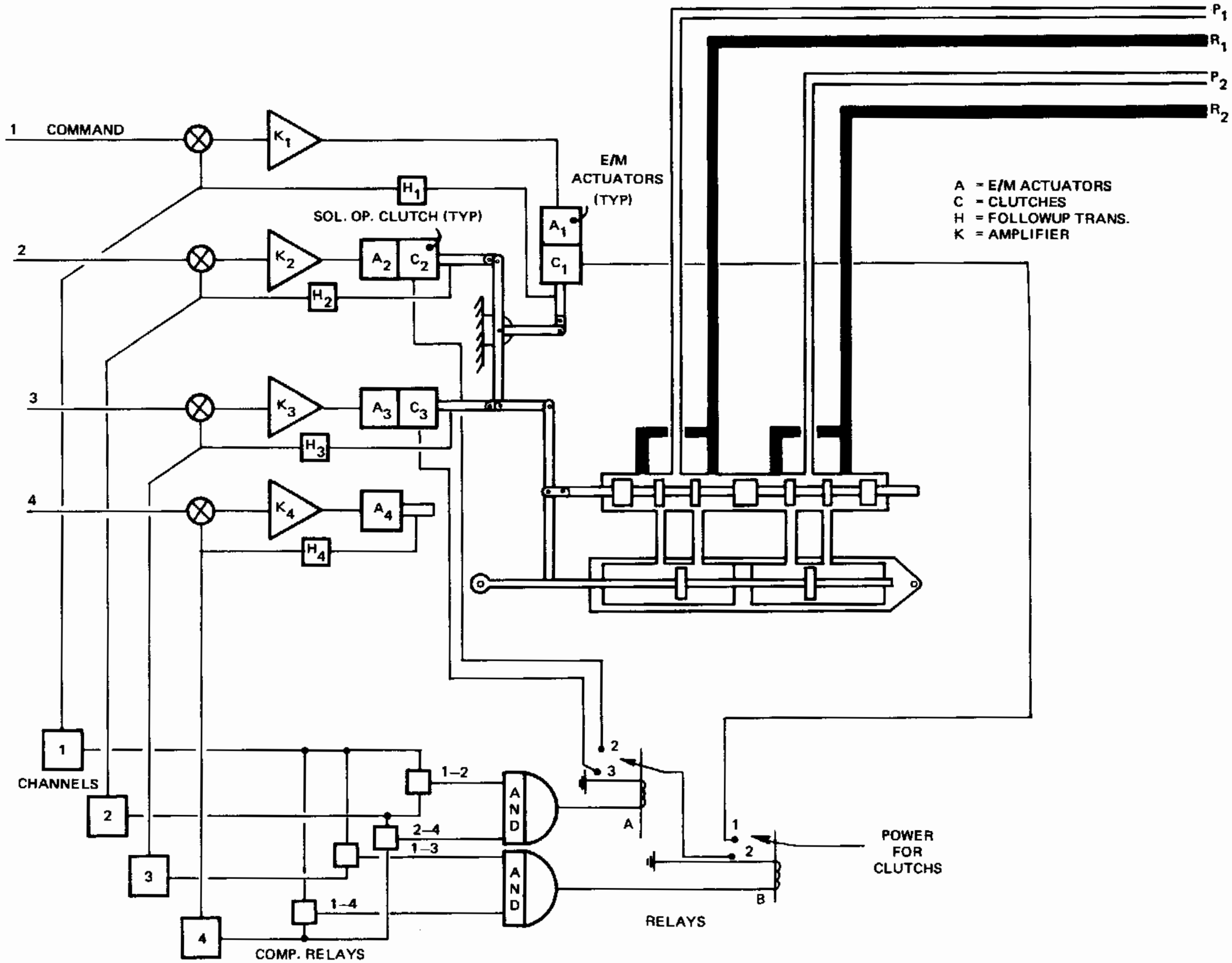


FIGURE 55. SYSTEM A–3 ACTIVE STANDBY WITH ELECTROMECHANICAL ACTUATORS OUTSIDE THE FEEDBACK LOOP

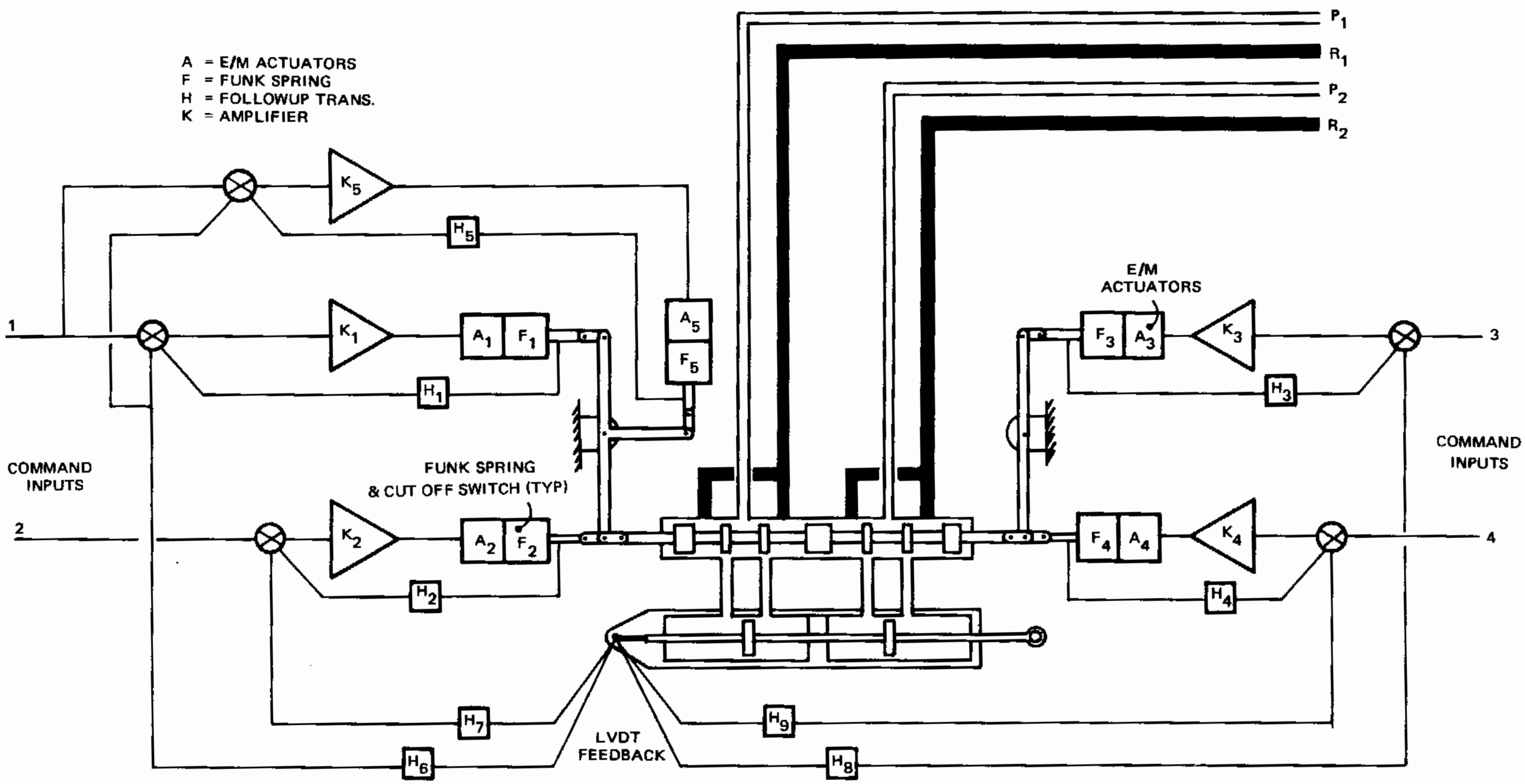


FIGURE 56. SYSTEM B–1 FORCE SUMMING WITH FIVE ELECTROMECHANICAL ACTUATORS INSIDE THE FEEDBACK LOOP

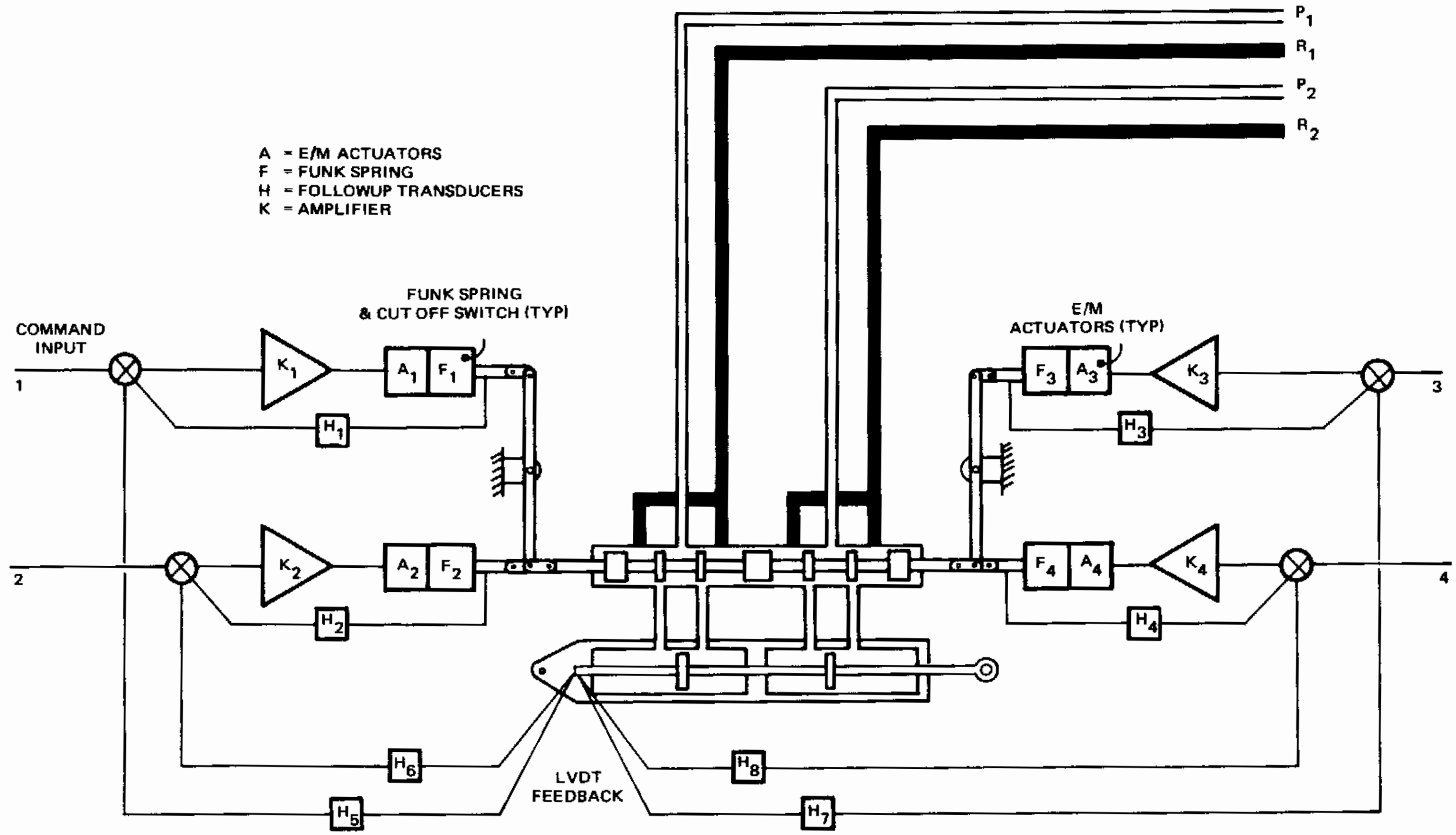


FIGURE 57. B–2 FORCE SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS INSIDE THE FEEDBACK LOOP

e. System B-2 (Figure 57)

This system operates identically to system B-1 with the exception that only one jammed E/M actuator can be tolerated. This is considered a realistic limitation and eliminates the requirement that all four signal channels must be good. Two hard over signal failures are still allowable as each channel can be back driven after electrical disconnection. The presence of only four E/M units (instead of five) allows for a greater funk strut breakout force differential between the two directions of operation. This results in a greater force margin for control of the main servo valve.

f. System B-3 (Figure 58)

System B-3 is identical to system B-2 with the exception that the feedback of the main actuator position is mechanical rather than electrical. This method of position feedback places the four electro-mechanical actuators outside the (main actuator) feedback loop, reduces the electrical feedback complexity, but introduces more severe transient and trim effects. In addition, long stroke E/M actuators become necessary.

g. System C-1 (Figure 59)

In this system all four electro-mechanical actuators drive the main valve through a displacement summing linkage. Two actuators are arranged to drive the valve slider and the other two drive the sleeve. The actuators are driven in opposite directions such that the result is a net relative displacement between sleeve and slider. A jam in any two actuators will not result in system failure unless they happen to be hard over jams in the same direction. A solenoid operated centering device is required to drive each failed actuator to center and lock. Electronic comparators similar to those used for Systems A_2 and A_3 are required to detect the position discrepancy and operate the required solenoid. Loss of two actuators results in loss of half the main servo valve output. Main actuator position, or LVDT output, is summed with the command signal of each E/M actuator.

h. System C-2 (Figure 60)

The conversion technique used in this system is the same as that used for system C-1. The difference between the two systems is that the signal converter in this system is outside rather than inside the loop. This, of course, results in the requirement for mechanical feedback of the main actuator position. In addition, when displacement summing is employed outside of the loop, any failure results directly in the loss of position authority. This means that loss of one of four channels results in the loss of 1/4 of the output position capability.

i. System C-3 (Figure 61)

This system can be considered velocity summing or a displacement summing scheme

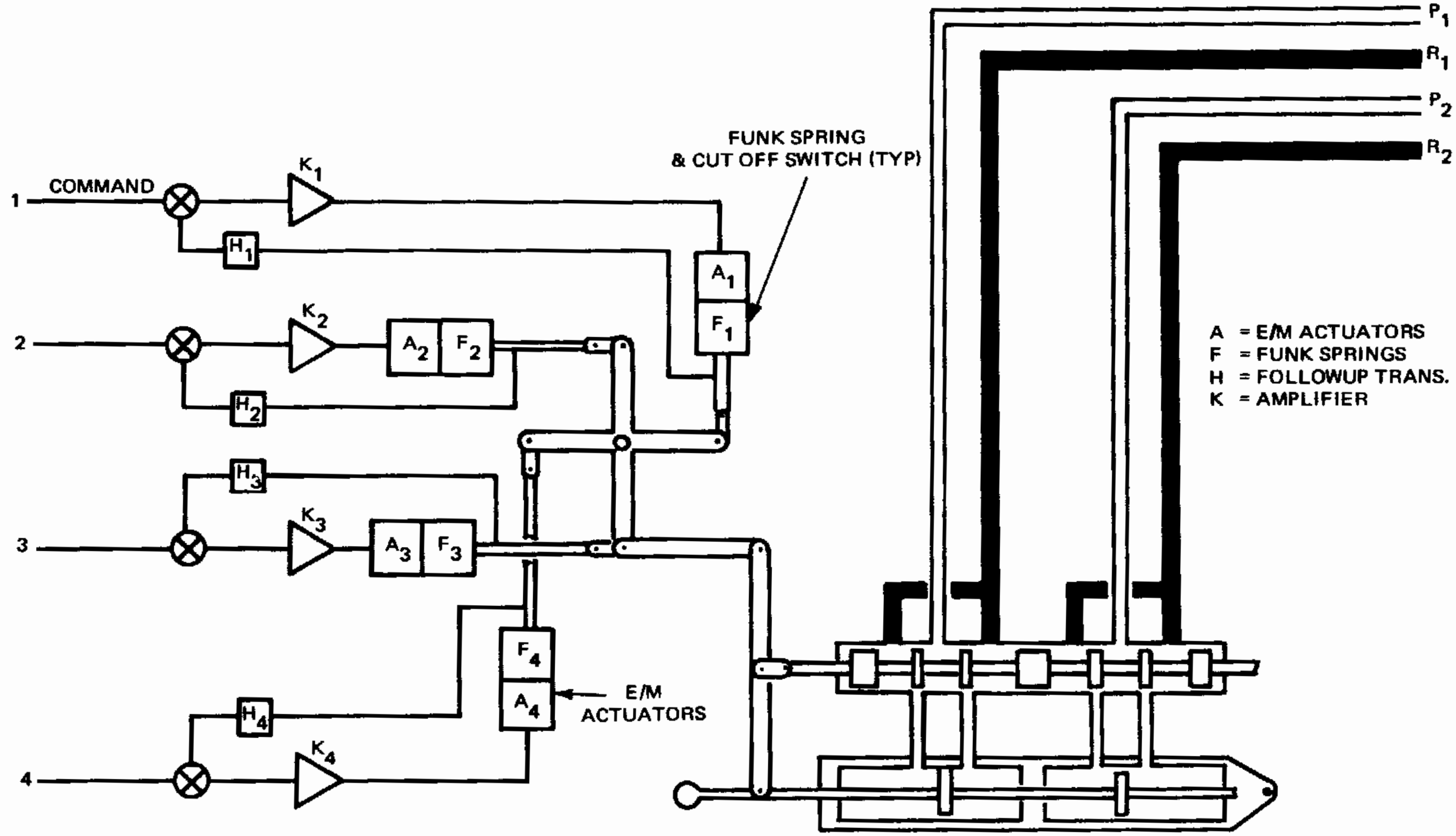


FIGURE 58. SYSTEM B–3 FORCE SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS OUTSIDE THE FEEDBACK LOOP

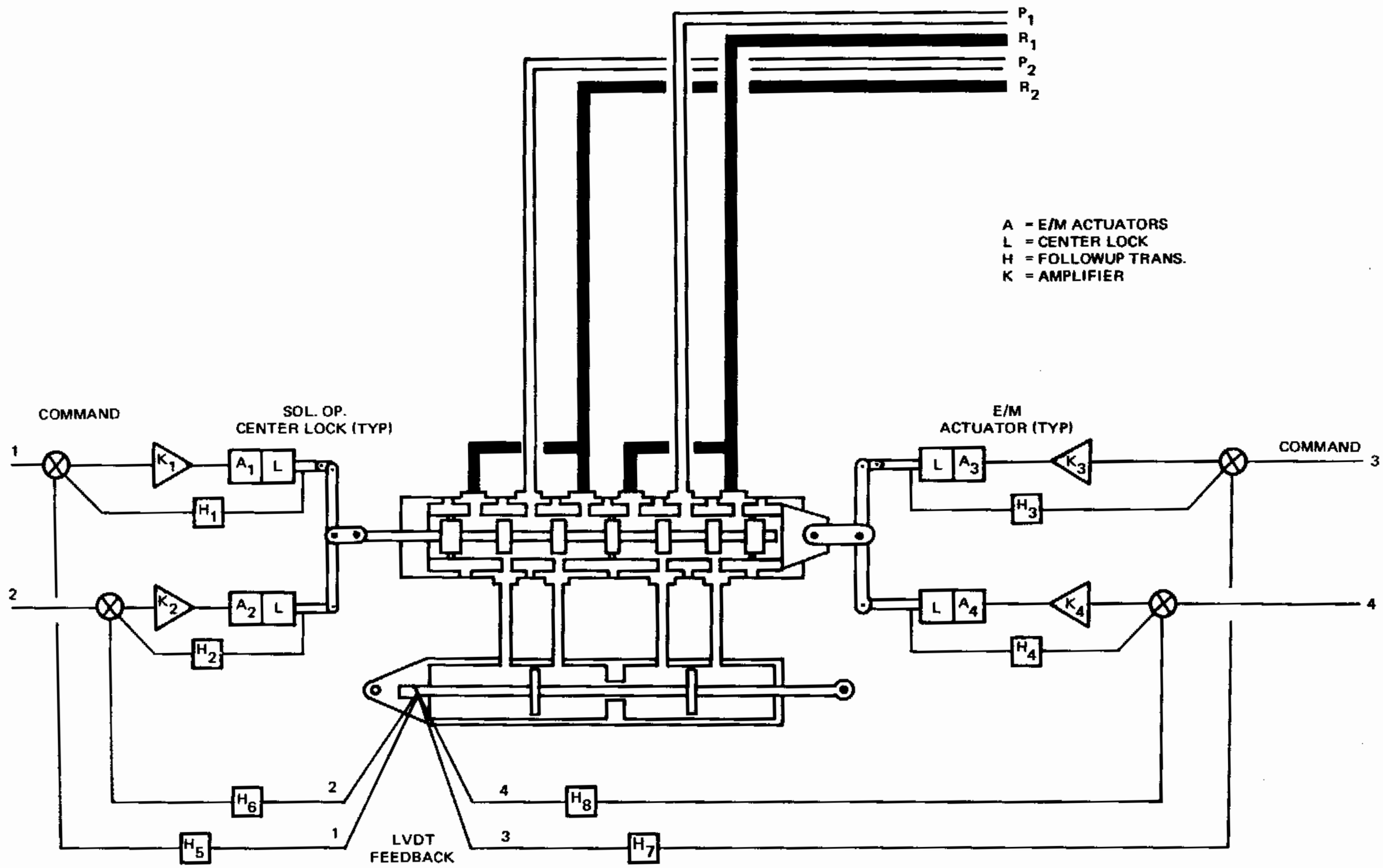


FIGURE 59. SYSTEM C–1 DISPLACEMENT SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS INSIDE THE FEEDBACK LOOP

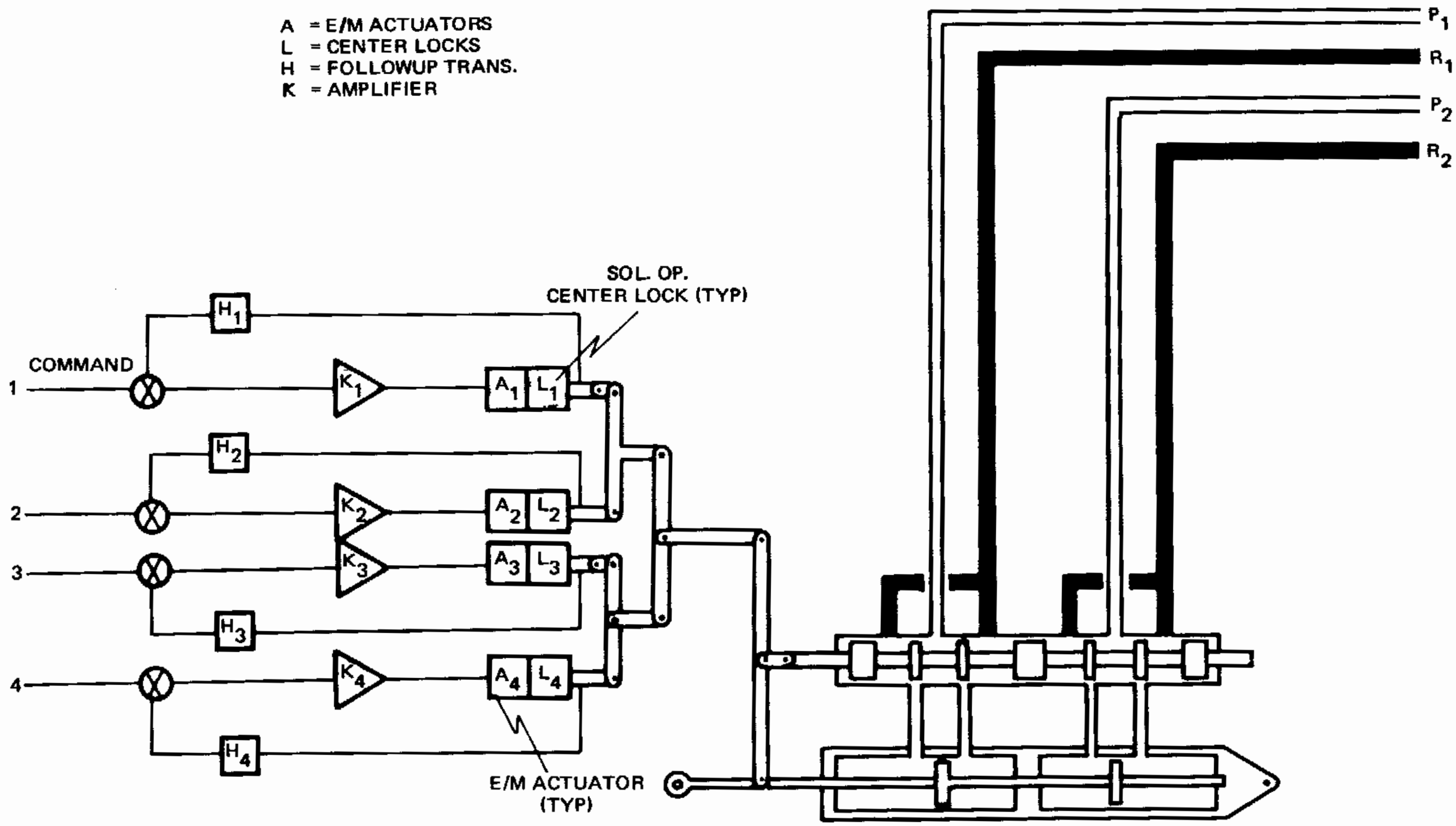


FIGURE 60. SYSTEM C–2 DISPLACEMENT SUMMING WITH FOUR ELECTROMECHANICAL ACTUATORS OUTSIDE THE FEEDBACK LOOP

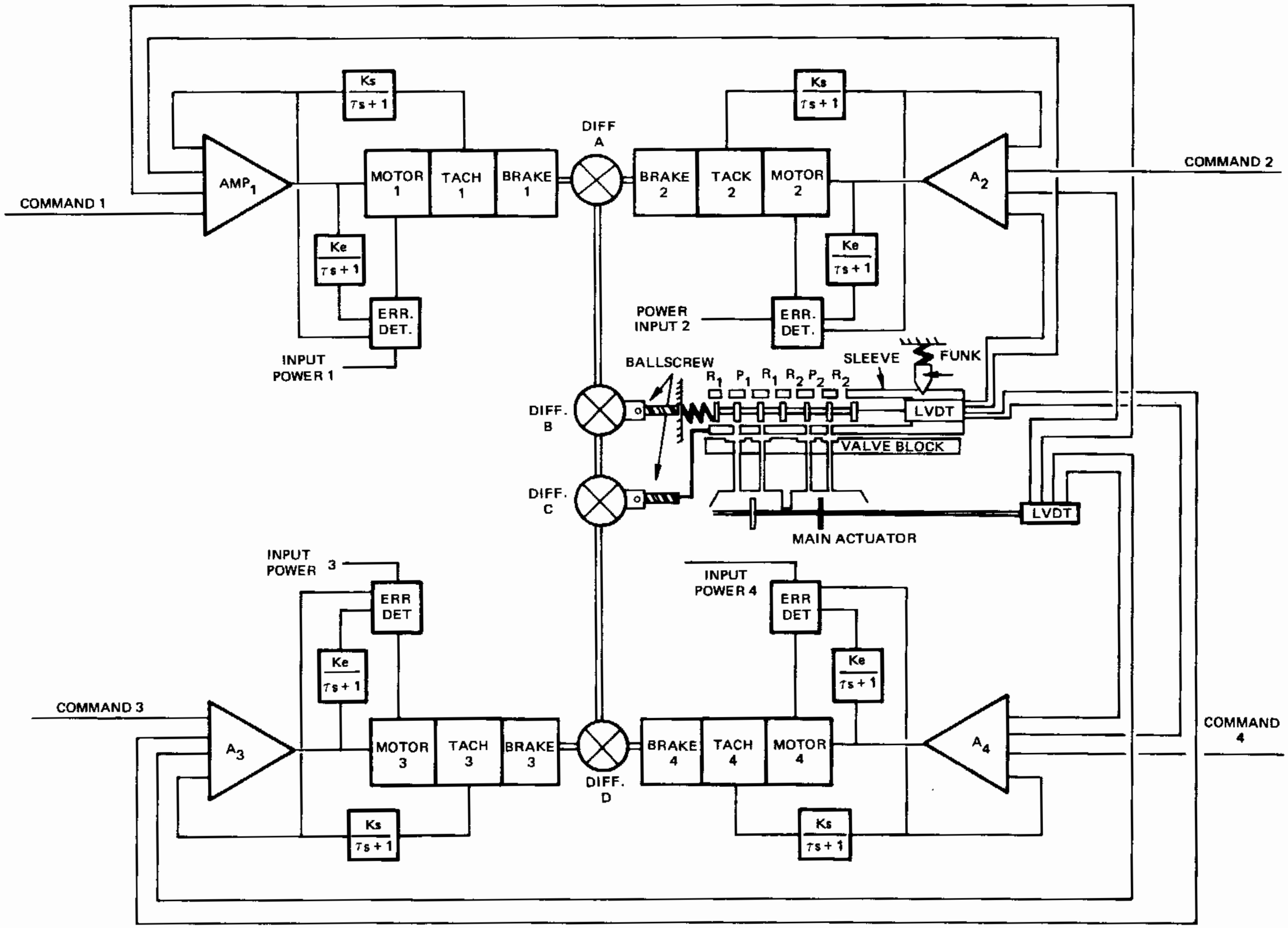


FIGURE 61. SYSTEM C–1 DISPLACEMENT SUMMING WITH DIFFERENTIALS AND ELECTRICAL MOTORS INSIDE THE LOOP

which is essentially without the normal displacement limitations. It is a rotational scheme which employs four electric motors whose velocities are summed through mechanical differentials and reversible gearing to drive the main valve. Under normal operating conditions, the valve sleeve is held stationary by a funk spring which restrains the output of differential C. Output sum of motors 1 and 2 are transmitted through differential A to one side of differential B. Similarly, the outputs of motors 3 and 4 are transmitted through differential D. However, because the output of differential C is restrained, the motion is transmitted across C to the other side of differential B to complete the summing action to the servo valve slider. Therefore, this system, unlike any linear displacement summing scheme, has an infinite stroke capability which has many advantages. One of these advantages is that full valve stroke can be achieved with as many as three of the four channels inoperative. Another important advantage is that failure monitoring and shut off can be achieved without electrical cross-coupling of the channels. This is possible because a constant velocity of the main servo valve is not germain to control of the aircraft, and can be washed out for normal operation and used for failure criterion. This is easily done in a rotational system.

Control of the motors is facilitated by an inner loop (4-channel LVDT measuring valve error) and an outer loop (4-channel LVDT on the main actuator piston). The inner loop is used to improve the system stability and to provide anticipatory failure information which reduces transient effects on the main actuators. A tach feedback closes the loop around each motor through a high gain lag circuit to keep the motors from circulating against each other due to mis-synchronization between the channels.

The main servo valve here is shown with redundant inputs. The normal action is for all displacements to be injected to the valve slider. (This is facilitated by the funk detent device on the sleeve.) If a jam occurs in differential B, enough force is generated to overcome the funk detent and drive the sleeve. Jams in differentials A, C, or D result in normal slider motion, but with reduced velocity. Therefore, a jam in any single element does not result in failure of the system, and the system is capable of withstanding many jams with no more than a reduction in slider velocity. Open links become critical in displacement summing systems. The output of differential B can be mechanically limited such that it can bottom out and permit force to be applied at the sleeve input when the slider input is open. The slider is spring centered to keep it from floating along with the sleeve under this condition.

Should any motor fail, the associated brake is energized to ground the motor output at the failed position. The tach feedback output which is modified by the time lag circuit is fed to an error detection circuit. The error circuit is designed to shut off electrical power to the motor if the tach output reaches a critical level for a specified length of time. This could occur with such things as a broken feedback element, malfunctioning amplifier, or gross mis-synchronization. To protect against a failure of the tachometer, the motor current is fed through a similar high gain lag circuit to the same error detector shut off. The outputs of the tach generators are available for comparison if it is considered necessary to detect an unresponsive motor.

j. Comments and Observations of System Tradeoff Scorings

(1) Minimum Transient Effects From Failure

The force summing systems were considered the best in regard to minimum transient effects due to the fact that at least one funk spring operates in its detent range, and negligible displacement is required to attain this condition after a failure. Force summing outside the loop was rated lower than inside the loop since any slight servo valve displacement due to a failure is more slowly compensated for by mechanical feedback. In the active standby systems, mis-sync results in considerable servo valve disturbance upon a change in command from one actuator to another. Mechanical feedback with its long strokes and slow response adds to this problem, and the system employing this method of control was given the lowest rating. The main servo valve in linear actuator displacement summing systems is disturbed upon action of the center locks. This disturbance may be somewhat less than that experienced in active standby systems.

(2) Minimum Susceptibility to Single Catastrophic Jam

Linkage is the determining factor here. Active standby systems provide the best protection against a single major jam in that position control is completely switched from the jammed unit to a good one. The servo valve may be made redundant by driving the sleeve with one group of actuators and the slider with another. Therefore, these systems were rated highest. Conversely, the force summing systems were rated lowest since a single jam in the servo valve drive linkage would be catastrophic. Also, a jammed actuator could force all funk springs to be broken out, resulting in loss of response.

(3) Minimum Susceptibility to a Single Catastrophic Open

As in the case of jams, linkage is the determining factor in protection against an open. The active standby systems were given the highest rating since position control is switched from the bad unit to a good one. (The assumption here is that the switching mechanism is relatively jam proof.)

Displacement summing systems are, in general, more susceptible to opens since an open linkage anywhere tends to result in system failure. Open actuators can be centered and locked, but the failure of a locking device associated with a bad actuator would have deleterious effects on the system. The differential gear method of displacement summing (System C-3) has advantages over linear actuator summation methods in that an open actuator need not be centered. However, a point where a single open would result in system failure can still be found, and the displacement summing systems received the lowest rating in this regard. (Opens can be minimized here just like jams can be minimized in force summing systems.)

(4) Minimum Trim Change Due to Failure

Trim change somewhat parallels transient effects, and due to the degree of servo valve disturbance caused by a failure, standby systems received the lowest rating, and force summing the highest.

(5) Maximum Overall System Reliability

Without considering reliability of each individual component and its relation to each particular system, it is rather difficult to determine overall system reliability. However, an estimation of the relative reliability of each system was attained by considering the type, number, and degree of failures each system would tolerate without total failure. The force summing systems were given the high ratings due to the fact that all actuators are tied together initially, and no switching, centering, or locking need be performed in the event of failures. The 5-actuator force summing system rated highest since it will tolerate an additional jammed actuator over the 4-unit systems. The standby systems were rated low because of complications inherent in clutches and switching techniques. The velocity summing system (System C-3) in the group of displacement summing methods does not require switching or locking, and would appear to approach the reliability of force summing systems.

(6) Lowest Risk

Due to simplicity in design and operation, the 4-actuator force summing system was given a high rating. Outside the loop force summing received the highest rating since mechanical feedback, discounting the need for longer strokes, is somewhat more successfully mechanized than electrical feedback. The lowest rating was given to the velocity summing system (System C-3, displacement summing) due to the anticipated development time required for this system. The standby systems received equally low ratings because clutches also introduce unknown complications.

The 5-actuator force summing design was considered undesirable in that funk spring design problems are inherent in such 5-unit systems. In order to maintain one funk in detent after a failure, spring tolerances must be held extremely close and the difference in breakout force in one direction as opposed to the other is extremely small. Also, with five actuators and only four electrical channels, two actuators must be on one channel; this places an additional requirement on the incoming signals; two of them cannot be grounded since complete stoppage will result if one of the two is related to the dual channel actuator. All of these problems are greatly reduced in the 4-actuator force summing systems.

(7) Minimum Size

All active standby systems are considered more complex due to the electronic and electro-mechanical devices required. Four electro-mechanical actuators outside the loop (System C) is the least desirable from this viewpoint since, in addition to the extra electronics, additional linkages and longer strokes are needed to permit mechanical feedback directly to the main valve. Active standby systems are rated low with inside the loop being rated more desirable due to short stroke requirements. The best systems were, in general, the force summing type (B-1, B-2, B-3) which have few electronic components and no centering or locking requirements. The smallest of these (and most desirable) is the 4-actuator system with short stroke (inside the loop).

(8) Minimum Standby Power Loss

The main contributor to standby power loss is mis-sync between the actuators. The standby type systems were rated high in this regard since mis-sync can result in no fighting between the actuators. Conversely, the force summing systems received a low rating. Although means can be found to minimize mis-sync in force summing, it still remains a substantial reason for low rating. The 5-actuator force summing (System B-1) received a lower rating than the 4-unit design due to the added complications of funk breakout design, and the probability that spring tolerances will cause four of the five springs to be broken out under normal conditions.

(9) Minimum Weight

Weight tradeoffs somewhat parallel size tradeoffs, and approximately the same scoring resulted for minimum weight, with the 4-actuator force summing method rated the highest.

(10) Maximum Temperature Capability

The displacement summing designs have the highest temperature capability in that there is no force fighting between the actuators. In the gear system all motors are less active. The actuators of force summing systems continuously fight each other in a mis-sync condition, making them the least desirable. (It is assumed that absolute synchronization will not be possible.)

(11) Maximum Number of Permissible Jams

The gear system will tolerate the most jams and was thus rated highest. The others follow according to the number of jams permissible and the degree of resulting system degradation. The displacement systems, of course, rate high, and the force systems rate the lowest.

(12) Maximum Number of Permissible Opens (Other than major)

The force summing systems, in general, can tolerate a greater number of opens than the displacement systems and, therefore, have the highest rating in this category.

(13) Minimum Gain Change Due to Failures

Negligible gain change occurs upon a failure in active standby systems due to the complete switch from a failed actuator to an identical but good unit. For this reason these systems received the highest rating. Conversely, the displacement summing systems suffer a gain change with each failure due to the linkage configuration. Force summing systems fall somewhere between the above extremes since failures can reduce force margins which can result in degradation of performance.

(14) Simplest Checkout Procedure

By sequentially imposing erroneous command signals on two actuators and awaiting the error indication, both the active standby and force summing systems can be checked. This was considered the simplest checkout procedure, and these systems were thus rated highest. In the displacement summing systems each actuator must be checked individually since an output from any one would operate the system. Working all the combinations of two failures will accomplish this task, but was considered to be more work than required in the standby systems or force summing systems.

(15) Minimum Cost

The simple 4-actuator force summing system was considered the least expensive. An additional actuator (5-unit force summing) complicates the force summing design as was explained above; tight spring tolerances and machining tolerances would prove expensive. Precision gears, tight tolerances, and considerable amount of electronics caused the gear system to be rated lowest. The standby systems were rated equally with the gear system due to the necessity of precision clutches and electronics.

k. Ratings

Results of any rating technique depends on the weighting factors applied to the system characteristics. For the particular factors used in this study the 4-channel inside the loop force summing and the velocity summing (C-3) systems ranked highest. It was felt that the velocity summing approach had potentially the best overall characteristics and features. However due to the development time and effort required for this system, the 4-channel force summing concept was chosen as the design to adapt for the Duplex integrated actuator package.

2. Hydraulic Power Supply Trade Study

The selection of a motorpump package to supply power for a specific fly-by-wire actuator involves two fundamental decisions. First, a basic motor assembly must be sized to provide the required flows and pressures for the actuator duty cycle. Second, a pump control system must be selected to optimize the system design. This tradeoff study is intended to provide the necessary calculated data for making a logical selection of a pump control system.

Three methods of control were considered; a load sensitive variable displacement control, a wide pressure differential cutoff displacement control, and a soft (very wide) differential displacement control. These techniques are described below.

a. Control System Description

(1) Soft Differential Control

In a soft differential pump control the displacement is varied by applying pump discharge pressure directly on the stroking piston of the yoke. The yoke return spring preload, which is adjustable, determines the pressure at which displacement will begin to reduce. Spring rate and piston area primarily determine the differential pressure between full flow and cutoff. To alter this differential requires a design change.

(2) Wide Differential Cutoff

In the wide differential pump control the yoke stroking pressure is provided by a high gain, 3-way valve or pressure compensator. The valve spool is operated by system pressure acting on one end of the spool opposed by an adjustable spring on the opposite end. The valve porting is approximately line-to-line to simultaneously meter system pressure into the control port and control pressure out to return pressure. The ratio of metering port areas determines the control pressure. A sharp cutoff unit results when the spring end of the valve spool is exposed to return pressure. For a wide differential, control pressure is fed into the spring chamber to slow spool movement relative to system pressure rise. A schematic of this approach is presented in Figure 62. The spring setting determines pressure setting at full flow. To change pressure differential requires a change in design.

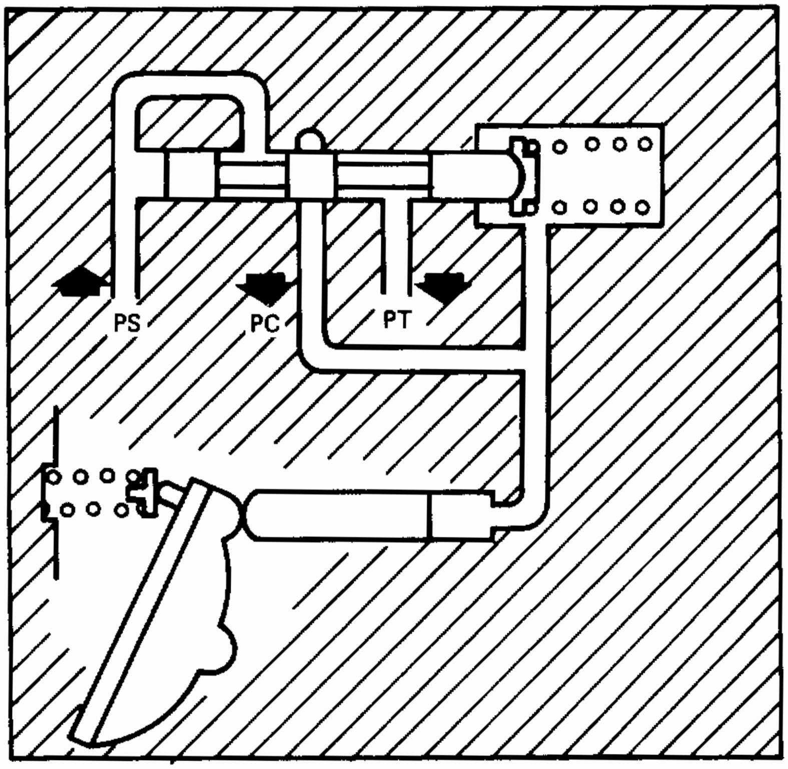


FIGURE 62. WIDE DIFFERENTIAL COMPENSATION

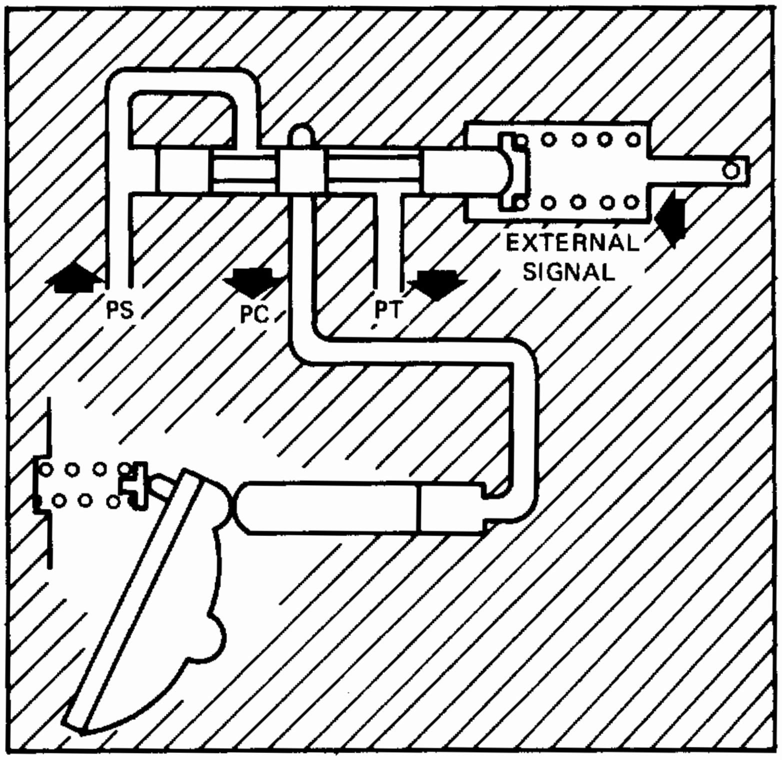


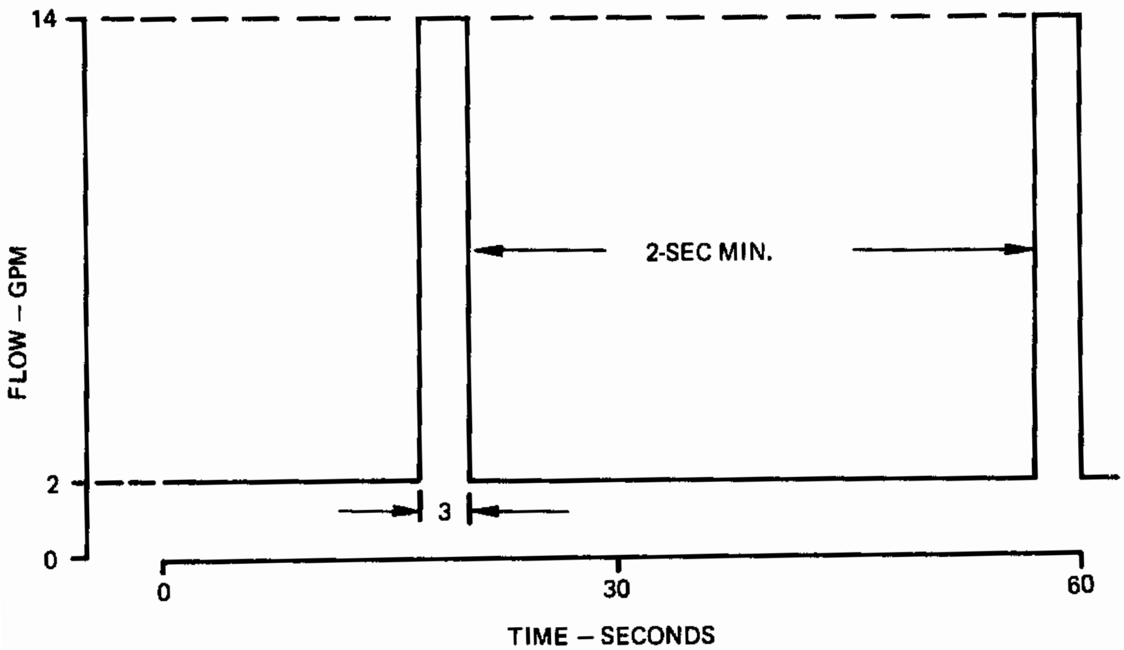
FIGURE 63. LOAD SENSITIVE COMPENSATION

(3) Load Sensitive Control

The load sensitive pump control is similar to the wide differential control except that load pressure is fed back from the actuator into the compensator spring chamber. The load pressure is picked up between the servo valve and the actuator. A shuttle valve must be provided to select the higher strut pressure for feedback. Figure 63 is a schematic showing the valving required for this scheme. The compensator spring adjusts the setting of system-to-strut pressure differential.

b. System Duty Cycle

The motorpump duty cycle was based on a 1-minute load interval with two gpm rated flow at 1550 psi discharge pressure for about 54 to 56 seconds, and 14 gpm for two to three seconds occurring not more than twice in a minute with at least two seconds off between periods of maximum 14 gpm flow demand. Since the worst case is for the condition when 14 gpm is on twice within an 8-second span, the duty cycle was assumed to be 54 seconds at 2 gpm, and six seconds at 14 gpm in each minute of flight or ground checkout, for the purpose of this tradeoff study.



c. Motorpump Package

A motorpump of 0.44 cubic inches/revolution at 7300 rpm full load speed is required for this application. The motorpump considered here is an in-line variable displacement, Vickers Model MPV3-044-XXX pump package. The pump is a

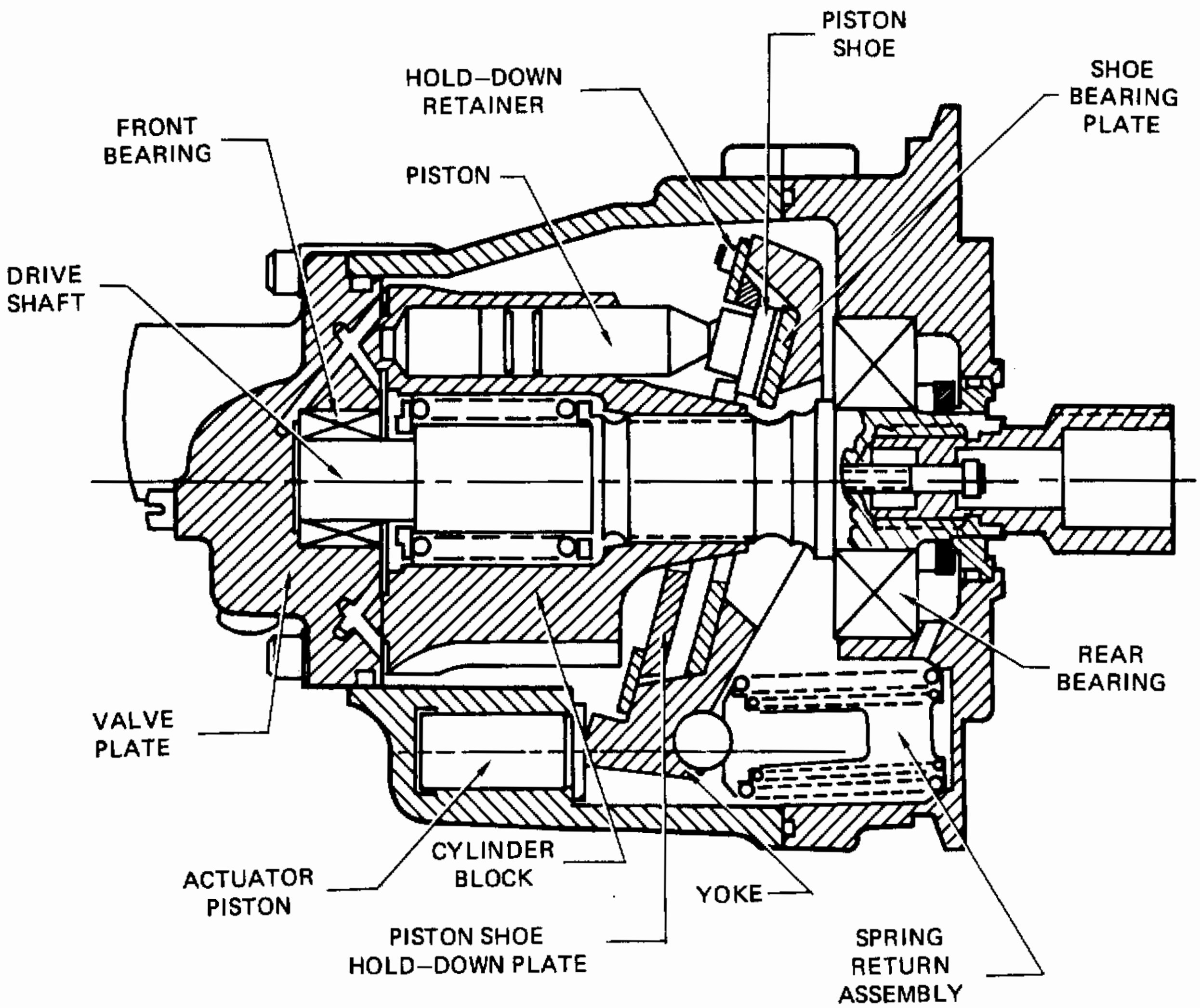


FIGURE 64. BASIC PUMP ASSEMBLY

7-piston unit in which the displacement is determined by an angled swash plate mounted as a pivoted yoke. The pistons with their shoes ride against the shoe bearing plate as they rotate and move axially to their bores parallel to the axes of the drive shaft. The yoke angle is controlled by a stroking piston and return spring. A cross section of the basic pump is presented in Figure 64.

d. Trade Study Results

The three different pump control techniques were evaluated using the duty cycle specified above. The criteria for comparison were:

(1) Weight and Envelope

(2) Efficiency

(3) Heat Rejection

(4) Dynamic Response

(5) Design Flexibility

(6) Reliability

In most cases only a relative value was assigned to these since exact absolute values were not available. The possibility of using an accumulator to reduce pump size with a comparable reduction in heat load was reviewed and rejected because of the large accumulator and reservoir required. A summary of tradeoff study results are shown in Table XVII.

It is evident from the results of the study that the load sensitive control system is clearly superior to the others on every count except reliability. The effect of this load sensitive control on servovalve size and response must be considered, since the servovalve drop is a constant in this case, whereas all other systems result in higher servovalve pressure drops at low flow rates. The load sensitive design was pursued for the Integrated Actuator Package based on the parameters considered in this tradeoff study.

C. SYSTEM DESCRIPTION

1. Requirements

The system was basically designed around the present stabilator actuator of the F-4 aircraft. The essential sizing parameters used were:

TABLE XVII. HYDRAULIC POWER SUPPLY TRADE STUDY RESULTS

Control Design	(a) Weight Lbs.	(b) Efficiency		(c) Heat Rejection BTU/MIN Hydraulic	(d) Dynamic Response at 100 CPS		(e) Flexibility	(f) Reliability	
		Rated Load	Full Load		Phase Lag	DB		Relative Ratio	MTBF
Load Sensitive	18.3	13.0	6.9	182	50^{o}	-1.5	A	.986	4705
Soft Differential	20.5	11.3%	7.4	206	80^{o}	-4.5	C	1	5000
Wide Differential	20.75	11.3	4.6%	225	80^{o}	-4.5	B	.995	4850

NOTES:

a) Weight differences are dependnet on electric motor NP requirements, duty cycle, and startup torque since control valves involved only weight about 1/4 pound.

b) Relative efficiencies are the best indicator of relative motor size and also of heat rejection.

c) Heat rejection is based on assumption that the sum of servovalve throttling and load feedback braking losses is proportional to pump output power. Electric motor losses are not included.

d) Response is open loop response of pump only.

e) Flexibility is rated in descending order from A, most flexible to C, least flexible.

f) Reliability values are estimated relative values only, MTBF are estimated values.

a. The package should be able to be installed in the present F-4 aircraft with minimum structural modification of the aircraft.

b. The actuator area should be essentially that of the present F-4 actuator.

c. The stall force may be reduced to approximately 1/2 of that achievable with the present F-4 actuator (present F-4 actuator is placarded for maximum pressure limit which results in this same reduced force).

d. No-load velocity should be equal to that of the present package.

2. General

The Duplex integrated package assembly is shown in Figure 65. Figure 66 is a photograph of the unit. The package as designed consists basically of the following subassemblies and components.

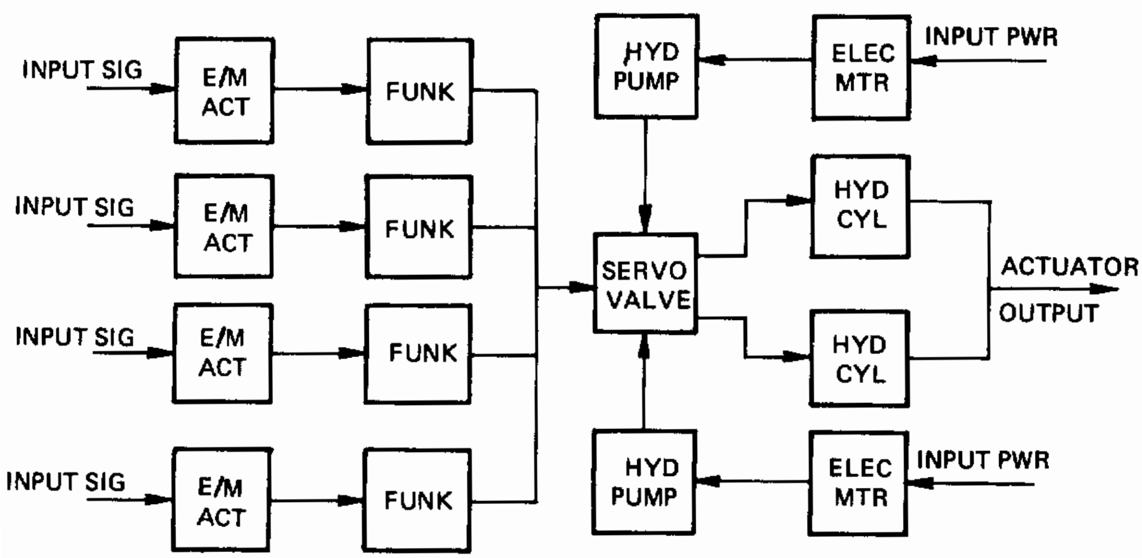
a. A dual tandem hydraulic cylinder.

b. Two electric motor driven hydraulic power supplies, one for each cylinder.

c. A dual tandem hydraulic servovalve which controls hydraulic fluid to the dual tandem cylinder.

d. Four electro-mechanical actuators which convert the electrical input signals into a hydraulic servovalve position.

A block diagram of the basic elements of the actuator is presented below.



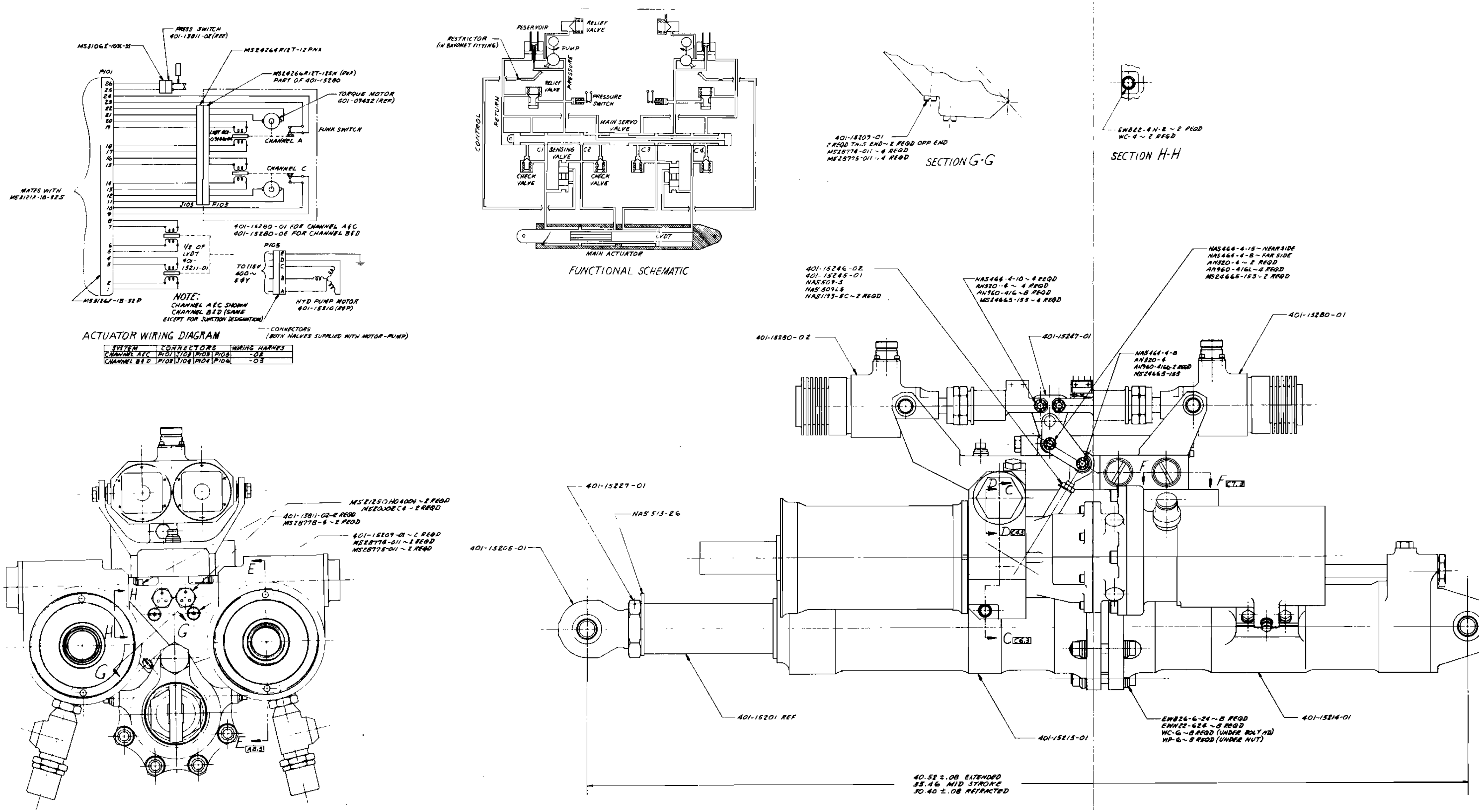


FIGURE 65. DUPLEX ACTUATOR ASSEMBLY (SHEET 1 OF 2)

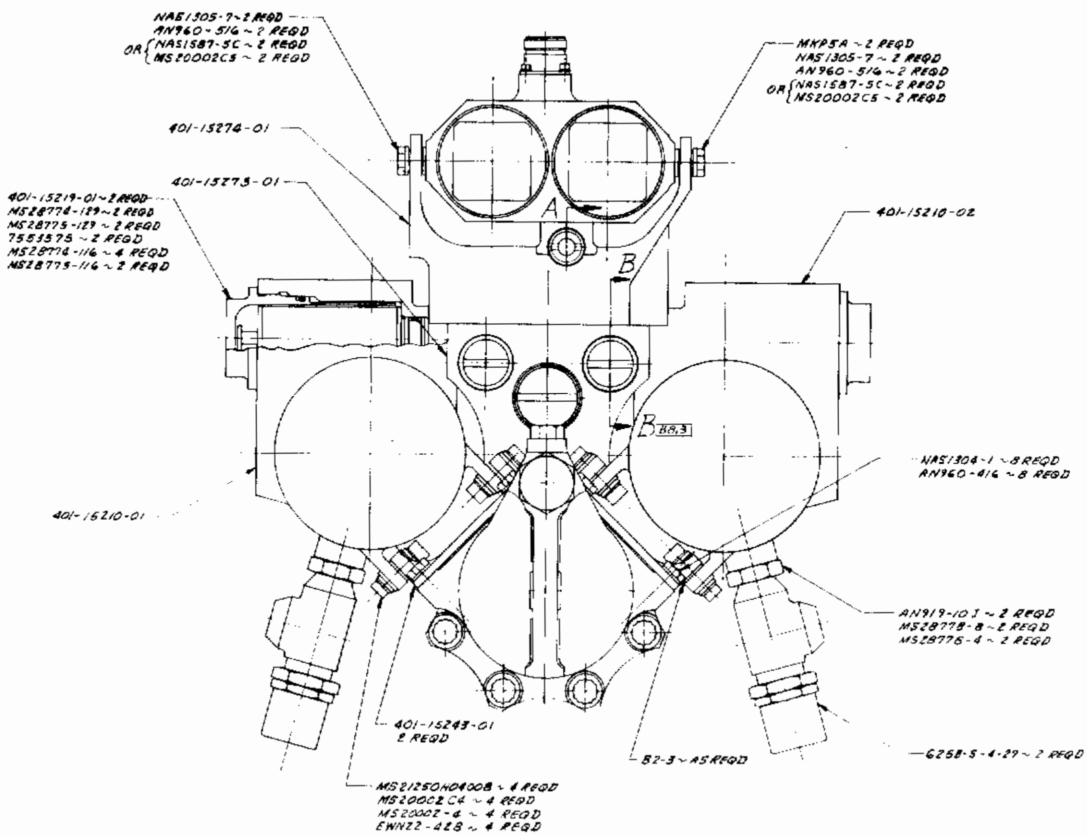


FIGURE 65. DUPLEX ACTUATOR ASSEMBLY (SHEET 2 OF 2)

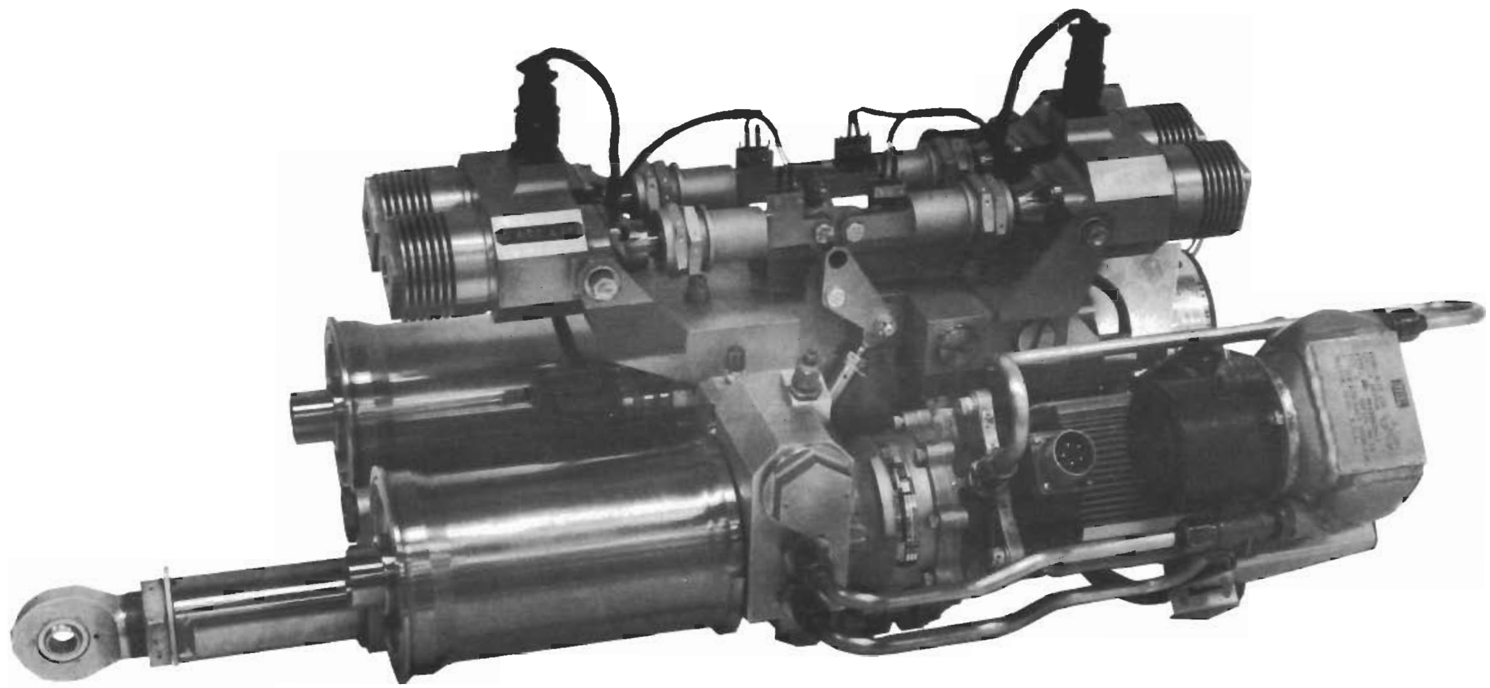


FIGURE 66. DUPLEX ACTUATOR

In operation, the signals from the four LVDTs (mounted in the main actuator) which monitor piston position are subtracted from the four independent input commands. These signals are in turn applied to four independent closed loop electro-mechanical actuators. The loop of the electro-mechanical (E/M) actuator is closed around the actuator position, hence the differential between the input command and the main piston position results in an E/M actuator position. The outputs of the four E/M actuators are force summed through the four funk struts, and the common output is applied to the dual tandem hydraulic servo valve. The hydraulic servo valve ports fluid to the hydraulic cylinders in the same manner as for any conventional dual tandem actuator.

The hydraulic power for the actuator is obtained from two separate hydraulic power supplies, one for the lug end cylinder and one for the rod end cylinder. Both supplies and their electric motor prime movers are integral to the actuator package. Each electric motor drives a variable displacement hydraulic pump which receives fluid from the boot strap reservoir and delivers it through a filter to the servovalve. Each system also contains a main system relief valve, a reservoir relief valve, a low pressure warning light switch, cylinder port anti-cavitation check valves, and a sensing valve.

The sensing valve serves as a 2-position switch which selects the highest pressure on either side of the piston and applies this pressure to the pump compensator. The pump is so configured that higher compensator pressures result in higher pump output pressures. In addition, a spring force equivalent to 300 psi is added to the compensator pressure which limits the minimum pump output pressure to 300 psi. A hydraulic schematic of the system is presented in Figure 67.

3. Component Description

a. Cylinder Assembly

The cylinder assembly is a conventional dual tandem cylinder design. The barrels and the piston and rods are made of steel. The rod end cylinder is an equal area configuration, whereas the lug end cylinder is an unequal area design. The unequal area is a result of burying the follow-up transducers inside the actuator piston rod.

b. Follow-up Transducers

The main stage follow-up transducer consists of four independent LVDT's ganged together and enclosed in a single package with their actuating axis parallel to each other (Figure 68). The transducer housing is attached to the lug end of the cylinder and supported inside the piston rod. Burying the transducer inside the actuator greatly reduces the susceptibility of the unit to handling damage, and decreases the vulnerability to small arms fire. The probe of each LVDT is attached to a common piece at the inside race of a duplex bearing. The outside race of the bearing is attached to the piston rod, and the bearing pair is axially preloaded by means of a spring. The purpose of the bearings is to allow piston rod rotation relative to the

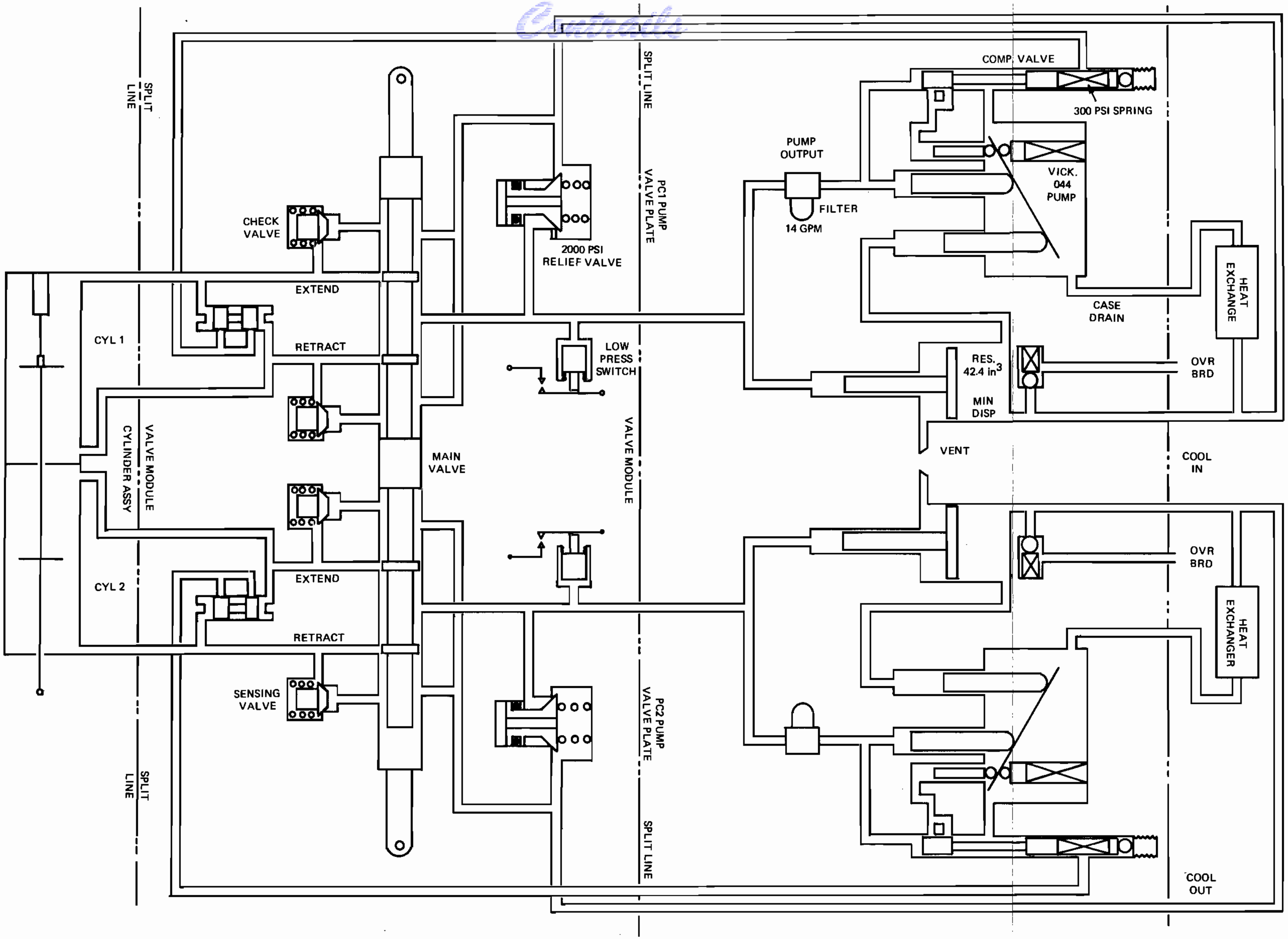


FIGURE 67. HYDRAULIC FUNCTIONAL SCHEMATIC DUPLEX ACTUATOR

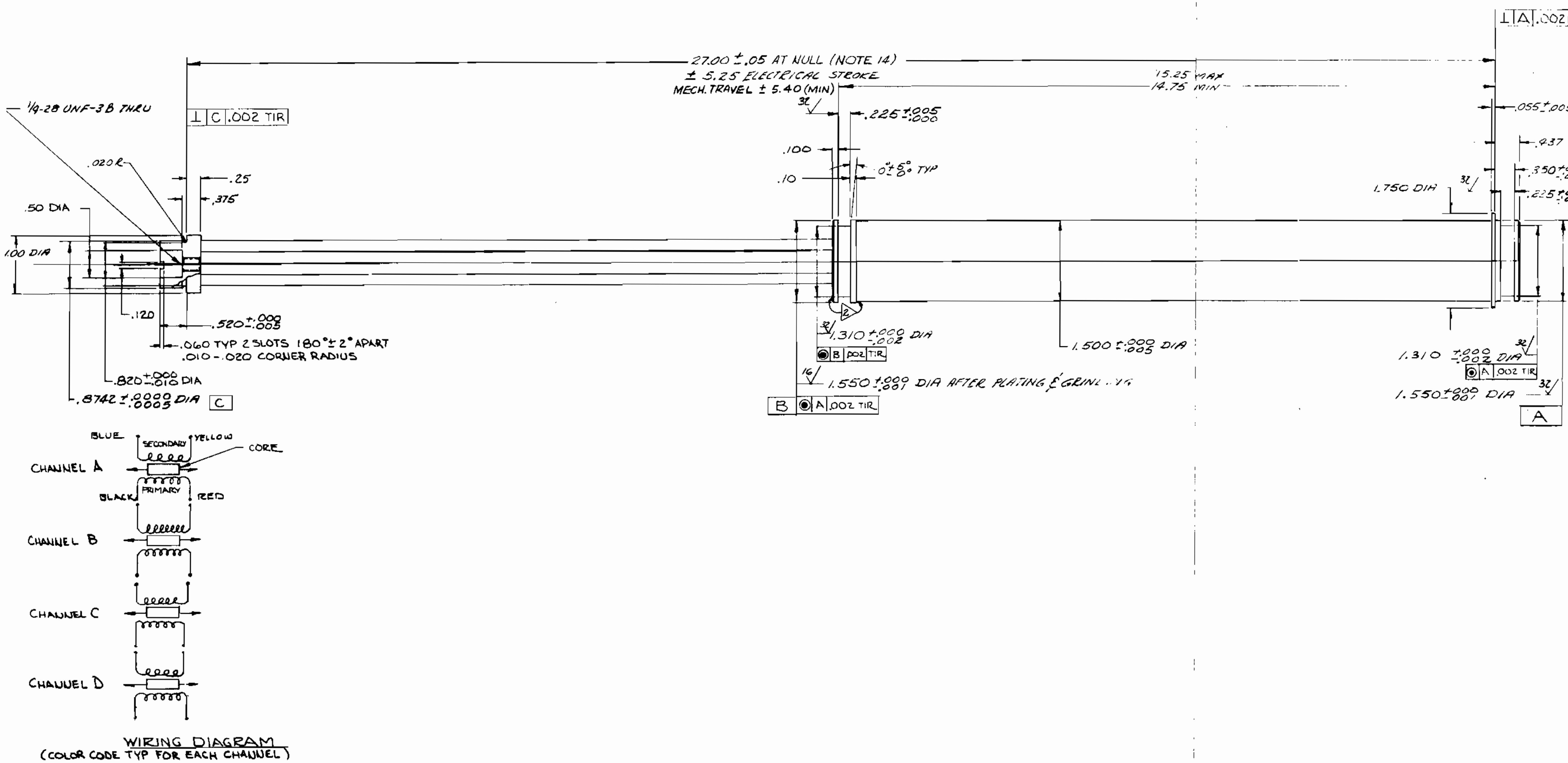


FIGURE 68. DUPLEX LVDT

actuator barrels without damage to the transducers. The preloaded duplex bearing reduces the unwanted axial backlash from the assembly without the adverse effect of excessive rotational friction.

c. Check Valves

Anti-cavitation check valves are installed in each cylinder port to allow free flow from return to the cylinder ports. During single hydraulic system operation, the passive cylinder is being driven as a pump. The check valves, by allowing free flow from return to cylinder, prevent high return pressure surges from causing loss of fluid through the overboard relief valve.

d. Sensing Valve

The sensing valves are small sleeve-slider type valves with the two cylinder ports connected to either end of the slider (Figure 69). The valve is naturally positioned at one end or the other of its stroke, depending on which cylinder pressure is higher. The valve action ports fluid from the cylinder with the highest pressure to the hydraulic pump compensator.

e. Motor Pump

The Duplex motor pump units supply hydraulic energy for Duplex unit operation. The two units are identical except one is a right hand unit and one is a left hand unit (Figure 70). The electrical motors are 3-phase, 400 Hz units sized to provide the necessary output power. The motors contain integral cooling fans and are designed to meet the requirements of MIL-M-7969.

The hydraulic pumps contain conventional rotating axial piston, variable volume units. The pump output volume is controlled by varying the angle of the piston yoke. However, to cause variable pressure in addition to variable flow, the compensator function is reversed from that of a normal pressure compensated pump, resulting in a load sensitive pump. In the pressure compensated pump, the pump displacement is spring loaded to the maximum position. The output pressure acts against this spring load to reduce the pump displacement. This results in an output pressure which is essentially a constant preset valve.

For the load sensitive pump, the spring load is in the direction to provide maximum pump displacement (Figure 63). The higher cylinder pressure, selected by the sensing valve, is applied to the compensator piston and causes an increase in pump output pressure corresponding to this compensator pressure. In addition to the cylinder pressure, a bias spring, equivalent to 300 psi, is applied to the compensator piston which limits the command pump output pressure to 300 psi. Thus, the pump output pressure will always be 300 psi higher than the highest cylinder pressure up to the relief valve setting of 1600 psi.

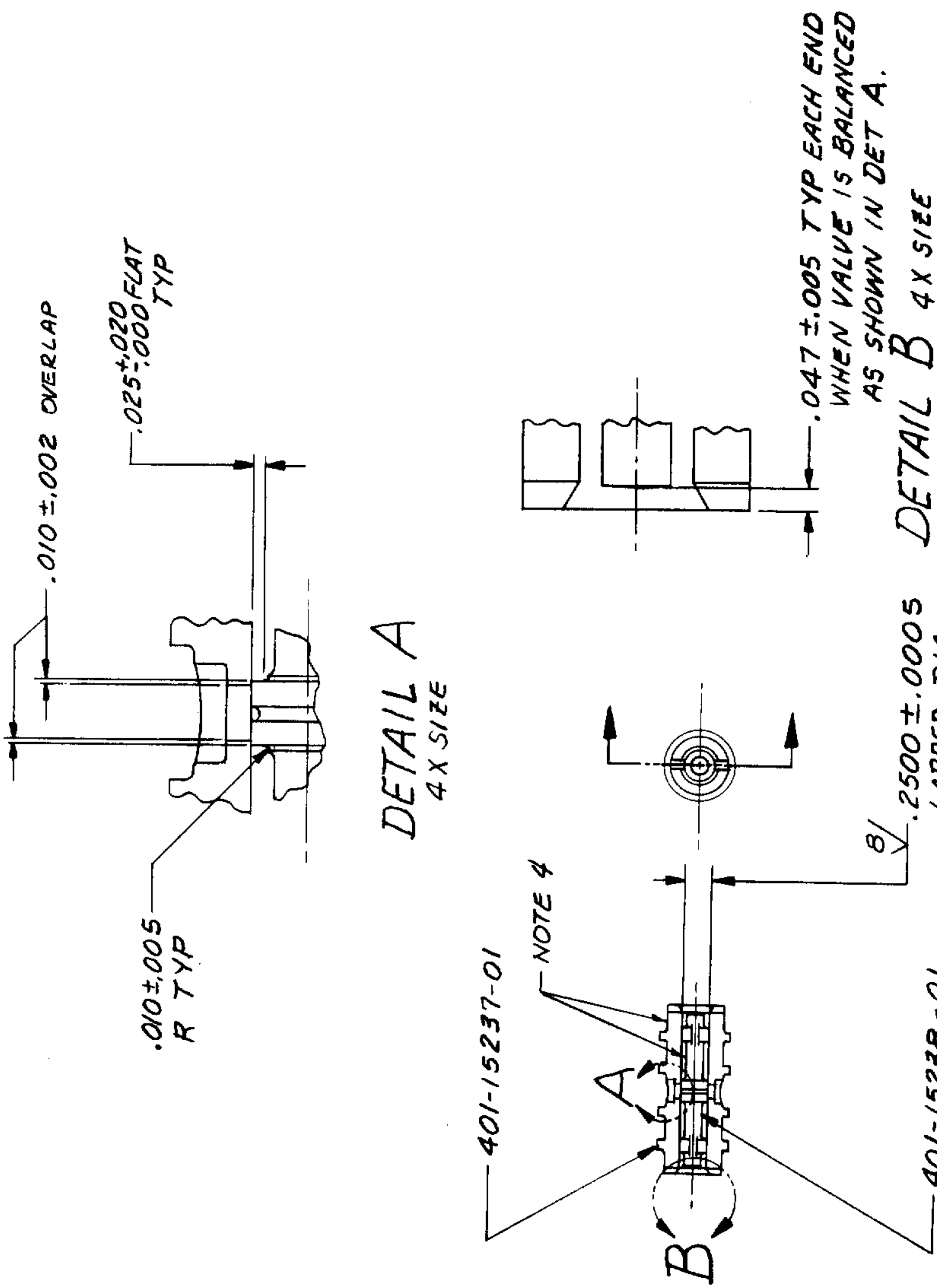


FIGURE 69. SENSING VALVE

INSTALLATION NO.	PART NO.	ORIENTATION	ROTATION	'X' PORT	Y PORT	MAX. THEORETICAL PUMP DISPL
	-01	AS SHOWN	RIGHT HAND	SEEP DRAIN	CASE DRAIN	.44 C.I.R
	-02	OPP. HAND	LEFT HAND	SEEP DRAIN	CASE DRAIN	.44 C.I.R

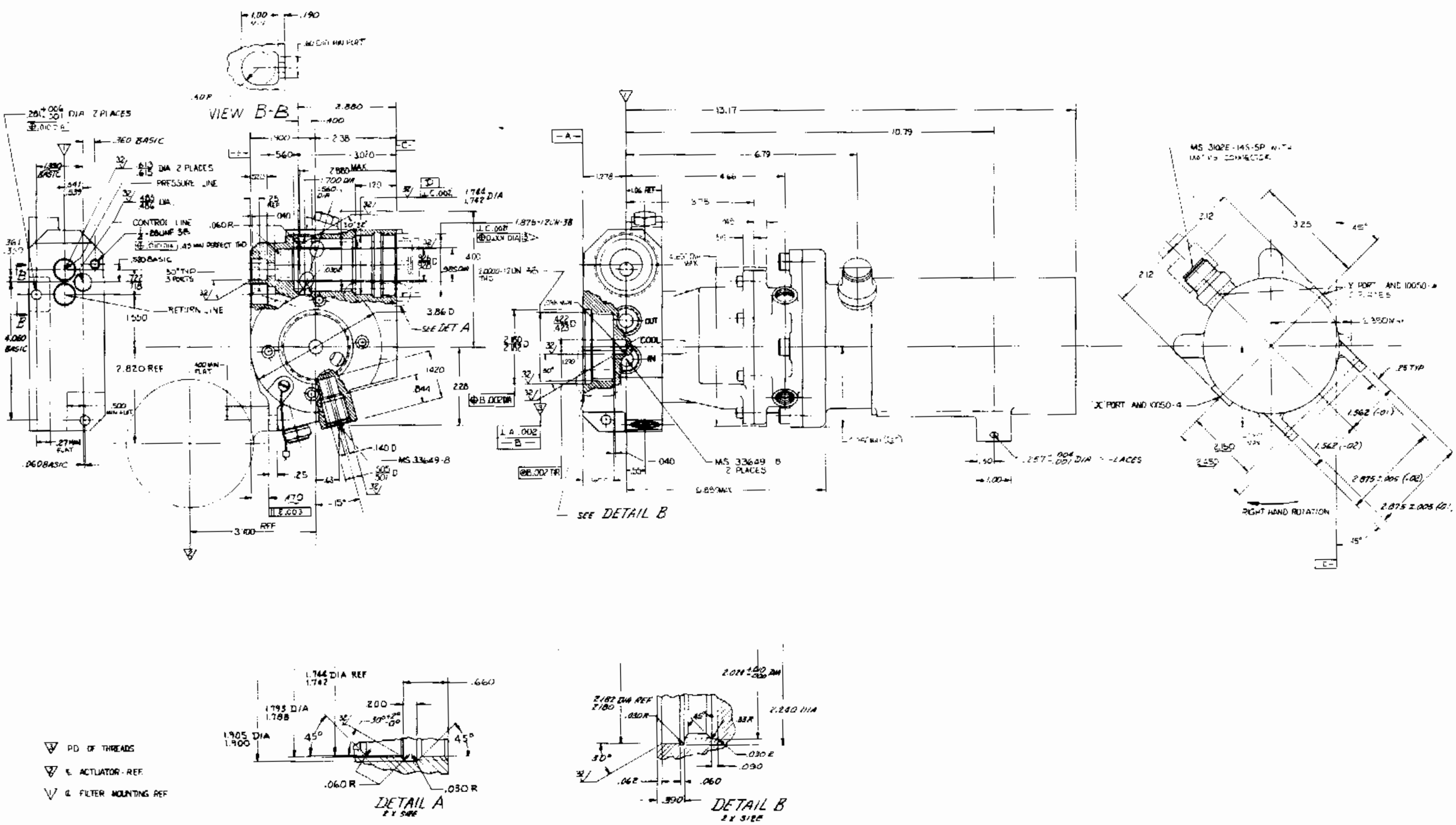


FIGURE 70. HYDRAULIC MOTOR PUMP

f. Reservoir

The reservoir is of the conventional boot strap design, where pressurization of the reservoir is accomplished by the pump output pressure acting on a reduced piston area. The bootstrap area ratio is 13.8 to 1. Therefore, the reservoir pressure will vary from a minimum of 39 psig to a maximum of 119 psig, depending upon the pump output pressure.

g. Relief Valves

Each hydraulic system contains two relief valves. One of the relief valves limits the supply pressure to approximately 1600 psi by by-passing fluid from pressure to return. This relief valve is necessary to prevent over-pressurization in the event of excessive external loads, or bottoming of the actuator which would command increased pump output pressure. The second relief valve is used to prevent over-pressurization of the return system in the event of overfilling the reservoir. This valve is set to relieve at a pressure of 350 psi.

h. Pressure Switch

Each hydraulic system has a pressure operated switch in the pump output pressure port. The switch is used to control a low pressure warning light in the cockpit.

i. Filter

A hydraulic filter is installed in the pump output line. The unit is a replaceable cartridge rated at 25 microns. It provides full flow filtration for the system.

j. Electro-mechanical Actuators

The four E/M actuators whose outputs are force summed to drive the hydraulic servo valve consist of a DC electric motor, a ball screw, a displacement transducer, and a funk strut (Figure 71).

The motor is a DC pancake type which consists of a permanent magnet field piece and a wound armature with conventional brushes and commutator. The motor has a high torque-to-inertia ratio which leads to exceptionally high dynamic performance capability. The motor rotor is mounted directly on the ball screw nut, thus minimizing inertia by eliminating additional parts.

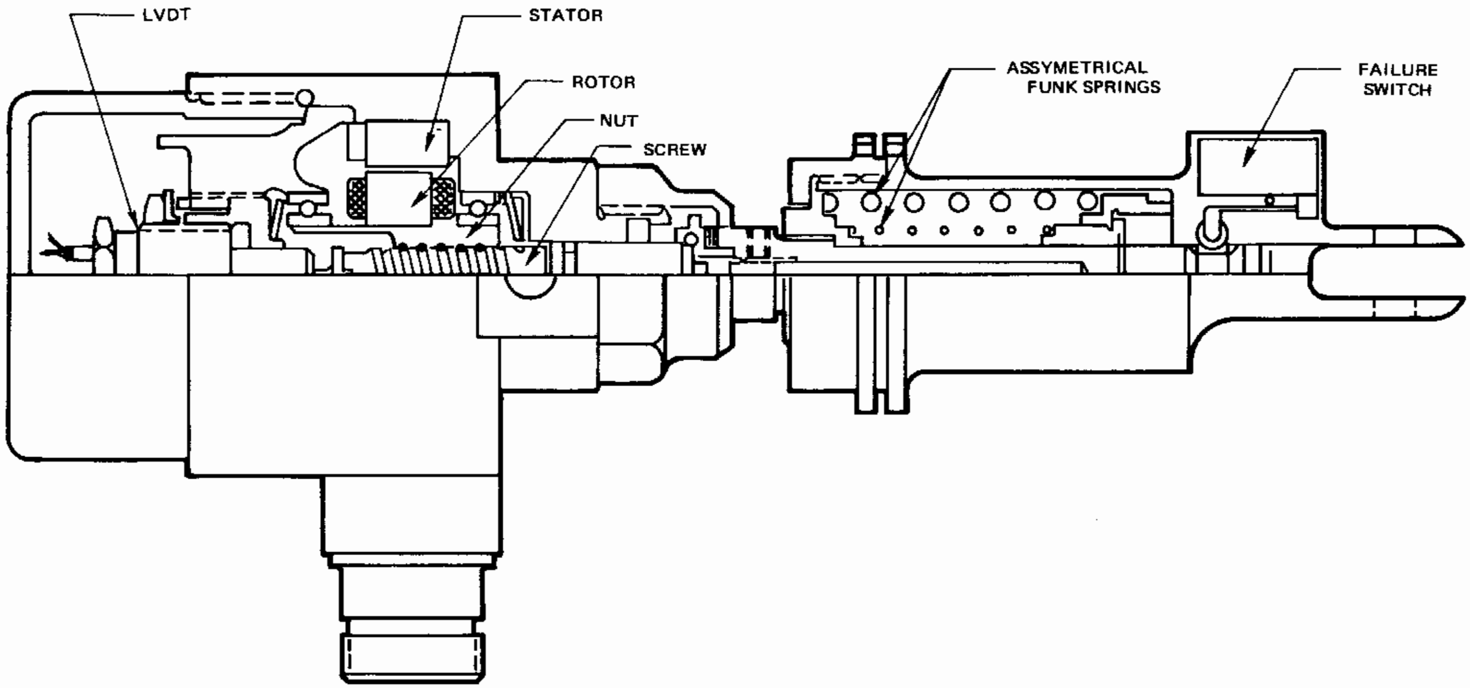


FIGURE 71. ELECTRO MECHANICAL ACTUATOR

The ball screw is a precision ball bearing, low backlash unit with high gearing efficiency. Its function is to convert the rotary output motion of the motor to a linear output. It has an internal, rather than external, ball crossover, so that a small motor armature diameter can be utilized. This small diameter motor helps to reduce the weight and inertia of the device. The ball screw stroke capability is +/-0.125 inches. The linear position of the ball screw is monitored by means of the follow-up transducer. This transducer is a conventional LVDT and provides the closed loop position signal for the E/M actuator. The funk strut is attached directly to the ball screw output, and is basically of the conventional dual concentric spring design. The difference between it and the conventional design lies in the asymmetrical breakout force of this unit. The asymmetry results from the engagement of both springs when breaking out in one direction, and engagement of only one of the springs when breaking out in the opposite direction. The breakout ratio provided is 2 to 3. The internal travel of the funk is 0.25 inches in either direction.

4. Failure Logic

The system is designed to remain operational after two signal channel failures and one hydraulic system failure.

a. Hydraulic System Failure

The integrated actuator package contains two functionally identical but separate hydraulic systems. In the event of the loss of one of the hydraulic systems, the stall torque of the actuator will be reduced by one-half with some reduction in its no-load velocity. The reduction in velocity will vary from 1°/sec to as much as 8°/sec, depending upon which system failed and the failure mode (passive cylinder empty or full of hydraulic fluid).

b. Signal Channel Failure

As previously described, the signal system contains four electro-mechanical actuators. Each actuator accepts a separate signal and provides an output which is a function of the input command. One reason that electro-mechanical approach was taken was to provide complete isolation between the signal channels and the power systems. Thus, a failure in the hydraulic power system will in no way affect the performance of the signal system, and conversly, a signal channel failure does not degrade the performance of either hydraulic system. The output of the four E/M actuators are force summed by connection to a common point (servo valve input linkage). This connection is made through funk struts which allow any actuator position to disagree with the position of any other actuator. During normal operation the four separate actuators are operating together and the funk struts are in detent.

c. Single Failure

If for any reason one channel takes on a position which is different from that of the remaining three channels, the funk strut of the maverick unit breaks out. If the disagreement (funk strut deflection) exceeds a preset value, the funk strut failure or monitor switch opens and disconnects the electrical power to the torque motor of the deviating unit. The removal of power from this discrepant unit allows the funk strut of that unit to drive it back to the detent (mean position) of the remaining actuators. The remaining three operational actuators function normally, and during operation, back drive the failed unit which is free wheeling. The only reduction in performance under this condition is the loss of the force capability of the failed actuator, and the additional force required to back drive it.

d. Second Failure

In the event of a second failure, where the position of one of the actuators disagrees with the mean actuator position by the preset amount, the second actuator will also be disconnected electrically, and the funk strut will drive this unit to the detent position. Thus, with two signal channel failures, the remaining two units drive the system in the normal manner. Again, the only reduction in performance of the package is that the valve driving force is reduced to that of two active actuators minus that force required to back drive the remaining two.

e. Third Failure

The system is designed so that voting can take place if there are more than two channels operating. If, however, two failures have already occurred, and there is a disagreement between the remaining two channels, the system is incapable of determining which unit is in error and should be shut down. For this reason, the electrical system is configured so that at least two channels are engaged at all times. A schematic of the logic circuit to perform this function is presented in Figure 72.

f. Locked Rotor

In the event that discrepancy of an actuator is a result of a locked motor rotor or a jammed ball screw, which destroys the ability to back drive the unit, the remaining channels are required to break out the funk strut of the discrepant channel in order to achieve the required operational function. The loss of one of the units in this manner results effectively in the loss of two channels because it requires the output of at least one of the good channels just to overcome the forces of the failed unit. Even though this failure results in the effective loss of two channels, a second failure where the channel is disconnected electrically, and can be back driven, can be sustained. Of course, after this failure, two channels are passive, one good channel is required to overcome the locked rotor channel, and the force output available to drive the valve is reduced to a value somewhat below that developed by a single actuator.

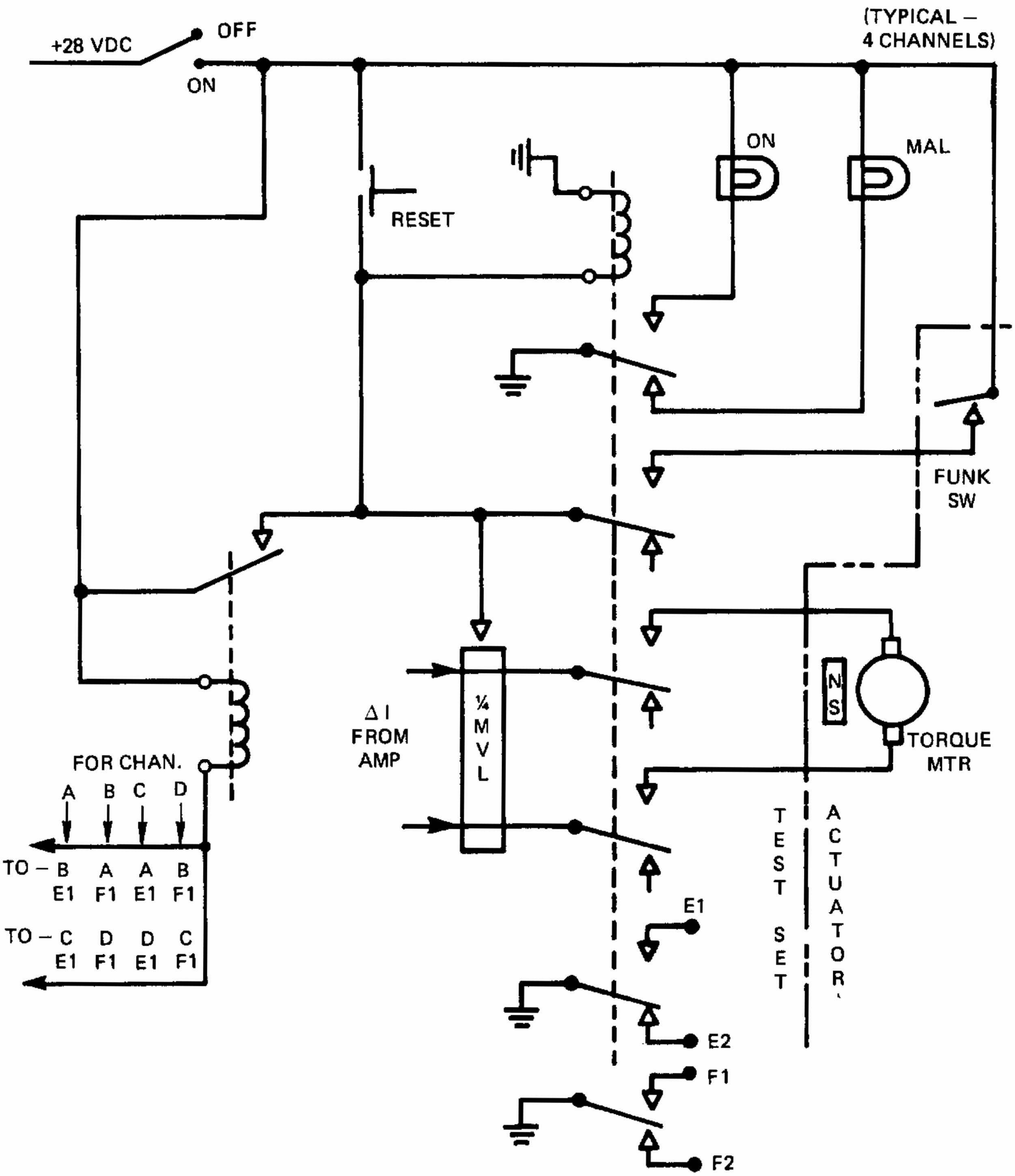


FIGURE 72. SWITCHING LOGIC FUNCTIONAL SCHEMATIC DUPLEX ACTUATOR

g. Jammed Funk Strut

In the event that one of the funk struts becomes jammed and cannot be deflected, the output of the signal system will be that of the unit with the jammed strut. Should a second failure occur in either of the three remaining channels, this unit will be disconnected and the system will operate just as before the failure, except for loss of the force capability of the disconnected channel. However, should the second failure be in the same channel that has the jammed funk strut, that channel cannot be disconnected, and system operation will depend upon the ability of the force developed by the three active channel funk struts being capable of overpowering the force developed by the discrepant unit.

In summary, the integrated package can, in general, sustain any two signal channel failures and remain operational as long as either or both hydraulic systems are functioning.

D. DESIGN ANALYSIS

1. General

Analyses were conducted as required to ensure that all components were designed to adequately meet their performance requirements. Specific attention was given the reservoir and funk springs. The following describes the sizing of these components in detail. In addition, informal stress analyses were conducted as required.

2. Reservoir Sizing

The reservoir volume required, and the reservoir pressure anticipated, were determined by the following design calculations:

a. Volume

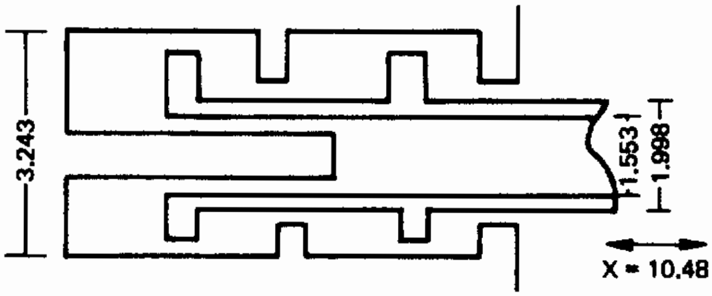
	Dia	Area (in^2)
Actuator cylinder bore =	3.243	8.260
Piston rod O.D.	1.998	3.135
Piston rod I.D.	1.553	1.894

The two actuator areas are,

$$A_1 = 8.26 - 3.135 = 5.125 \text{ in}^2$$

$$A_2 = 8.26 - 1.894 = 6.366 \text{ in}^2$$

and the differential area is



$$DA = 6.366 - 5.125 = 1.241 \text{ in}^2$$

while the differential volume will be,

$$DV = DA \times \text{actuator stroke}$$

$$= 1.241\,(10.48) = 13.005 \text{ in}^3$$

Assuming the total fluid volume is 1½ times that of the actuator, and adding 25% to allow for leakage, the total minimum fluid volume for full serviced condition will be

$$\begin{aligned} V_{min} &= 1.5\,(1.25)\,AP\,(XP) \\ &= 1.5\,(1.25)\,6.366\,(10.48) \\ &= 125.1 \text{ in}^3 \end{aligned}$$

Assuming that the temperature range is -65°F to +500°F, the fluid expansion can be determined as follows:

$$\rho = \rho_o\,(1 - \alpha\Delta\ T)$$

where:

$\rho =$ fluid density at temp.

$\rho_o =$ fluid density at ref. temp.

$\alpha =$ expansion factor

$\Delta T =$ change in temperature

$$\begin{aligned} \rho_{-65} &= 0.0312\,(1 - 4.26\,(10^{-4})\,(-65 - 70)) \\ &= 0.033 \text{ lbs/in}^3 \\ \rho_{500} &= 0.0312\,(1 - 4.26\,(10^{-4})\,(500 - 70)) \\ &= 0.0255 \end{aligned}$$

and the expanded fluid volume will be

$$\begin{aligned} V_{max} &= V_{min}\ \frac{\rho_{-65}}{\rho_{500}} = 125.1\ \frac{(0.033)}{0.0255} \\ &= 161.9 \end{aligned}$$

therefore the reservoir volume required will be

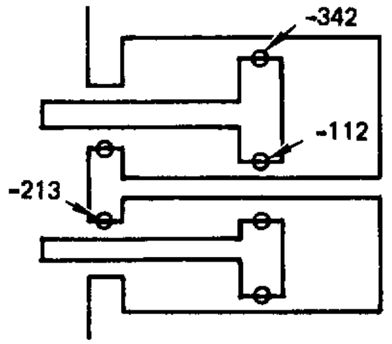
$$V_r = V_{min} - V_{act} + DV$$

$$= 161.9 - 125.1/1.25 + 13.05$$

$$= 74.9 \text{ in}^3$$

b. Pressure Calculations

The friction forces associated with O-ring seals are a function of O-ring area (Fh) and length (Fc).



To determine the approximate friction force, assume the supply pressure is 600 psi (minimum anticipated) and the reservoir pressure is 40 psia (min to prevent pump cavitation).

-No.	O-ring Area	Length
-341	2.248	12.58
-213	0.412	3.735
-112	0.166	1.568

fh = 11 lbs/in^2 @ P = 40 psi
   = 26 lbs/in^2 @ P = 600 psi

fc = 1 lb/in

$$F_f = fh\,(\text{area}) + fc\,(\text{length})$$

$$= 11(2.248) + 1(12.58) = 37.3 \text{ for } -342$$

$$= 26(0.412) + 1(3.735) = 14.4 \text{ for } -213$$

$$= 26(0.166) + 1(1.568) = 5.9 \text{ for } -112$$

TOTAL 57.6 lbs

with 600 psi supply and the calculated friction the reservoir pressure will be

$$Ps\,(Ap) - Ff = Pr\,(Ar)$$

$$Pr = (Ps\,Ap - Ff)/Ar$$

$$Ps = 600 \text{ psi}$$

$$AP = (1.178)^2\ \pi/4 - (0.498)^2\ \pi/4$$

$$Ff = 57.6 \text{ lbs}$$

$$Ar = (3.994)^2\ \pi/4 - (0.498)^2\ \pi/4$$

$$= 12.33 \text{ in}^2$$

$$Pr = (600\,(0.895) - 57.6)\,/12.33$$

$$= 38.88 \text{ psi}$$ or very nearly the design criteria of 40 psi.

In a like manner the maximum reservoir operating pressure can be determined. With a 1550 psi supply pressure and assuming a reservoir pressure of 125 psi the piston friction will be

$$Ff = fh\,(\text{area}) + fc\,(\text{length})$$

$$= 11.5\,(2.248) + 1(12.58) = 38.43 \text{ for -342}$$

$$= 4.9(0.412) + 1(3.735) = 23.92 \text{ for -213}$$

$$= 49(0.166) + 1(1.568) = 9.7 \text{ for -112}$$

$$\text{TOTAL} = 72.05 \text{ lbs.}$$

and the max operating reservoir pressure will be

$$Pr = (PsAp + Ff)/Ar$$

$$= (1550\,(0.895) + 72.05)/12.33$$

$$= 118.35 \text{ psi}$$

Thus the reservoir operating pressure will vary approximately from a minimum of 39 psi to a maximum of 118 psi.

3. Funk Strut Spring Sizing

Because the load sharing of the concentric springs of funk strut is determined from the asymetrical breakout performance required, rather than from the stress criteria, the approach used to size the springs was slightly modified from that normally used. The solution to the problem was obtained by use of a spring routine written for and performed on a digital computer. The problem was run for both a load distributions, which would utilize the inner of the concentric springs for the bi-directional load, and also for load sharing which would utilize the outer for the bi-directional loads. The results of the tests showed that for maximum efficiency in both size and weight, the bi-directional load should be developed by the outer spring.

The results of a typical computer run are presented in Table XVIII. As is shown by the table, the inputs for this design were:

a. Load distribution

b. Breakout force

c. Maximum force

d. Stroke

e. Maximum O.D.

Operating on the above input data the computer output produced the design shown in the table.

4. Electric Motor Sizing

The electric motors must be sized to provide sufficient power to the pump to produce the necessary flow and pressure output. Proper sizing of a motor for a production configuration requires precise definition of loads, duty cycles, and ambient and cooling air temperatures. The Duplex pump motors were sized to meet a maximum output flow of 14 gpm at 675 psi output pressure. Pump hydraulic output power required is

$$HP_{out} = (670)(14)/1715 = 5.47 \text{ HP}$$

The pump losses at rated speed are 25 in. lbs. Therefore, the pump power loss is

$$HP_{loss} = \frac{(25)\,(8000)\,(6.28)}{12\,(33000)} = 3.18$$

or the total power to the pump is

$$5.47 + 3.18 = 8.65 \text{ hp}$$

TABLE XVIII. COMPUTER RESULTS CONCENTRIC SPRING DESIGN

LOAD DISTRIBUTION OUTER = 0.750 INNER = 0.250

SPRING REQUIREMENTS

MATERIAL = CRES 302		LIFE = 1.00 TIMES 10 TO THE 6 CYCLES	
BREAK-OUT FORCE =	50.00	O.D. =	1.000
MAX FORCE =	60.00	I.D. =	0.0
STROKE =	0.250	MAX O.D. =	0.0
MAX BREAK-OUT LENGTH =	0.0	MIN I.D. =	0.0
SPRING RATE =	40.00 LBS/IN		

RADIAL CLEARANCE BETWEEN SPRINGS = 0.078

OUTER SPRING

BREAK-OUT FORCE =	37.50 LBS
MAX FORCE =	45.00 LBS
SPRING RATE =	30.00 LBS/IN

SPRING DESIGN

WIRE DIAMETER =	0.1120
OUTSIDE COIL DIA =	1.000
INSIDE COIL DIA =	0.776
TOTAL COILS =	11.16
SOLID HEIGHT =	1.250
FREE LENGTH =	2.875
LENGTH AT BREAK-OUT =	1.625
NATURAL FREQUENCY =	208.96 CPS
WEIGHT =	0.092 LBS
MAX STRESS =	85.887 KSI
MAX ALLOW. STRESS =	90.061 KSI

DESIGN IS SAFE IN BUCKLING

INNER SPRING

BREAK-OUT FORCE =	12.50 LBS
MAX FORCE =	15.00 LBS
SPRING RATE =	10.00 LBS/IN

TABLE XVIII. COMPUTER RESULTS CONCENTRIC SPRING DESIGN (CONT)

SPRING DESIGN	
WIRE DIAMETER =	0.0630
OUTSIDE COIL DIA =	0.620
INSIDE COIL DIA =	0.494
TOTAL COILS =	13.19
SOLID HEIGHT =	0.831
FREE LENGTH =	2.414
LENGTH AT BREAK-OUT =	1.164
NATURAL FREQUENCY =	245.48 CPS
WEIGHT =	0.022 LBS
MAX STRESS =	99.145 KSI
MAX ALLOW. STRESS =	99.886 KSI
DESIGN IS SAFE IN BUCKLING	

Based on a motor efficiency of 0.83, the required input power to the motor is

$$\frac{8.65}{0.83} = 10.4 \text{ hp}$$

The motor used on the Duplex has a nominal rating of 12 horsepower and synchronous speed of 8000 rpm. The motor/pump performance characteristics are shown in Table XIX.

5. Stress Analysis

A stress analysis for the integrated package was performed. The ground rules used for this analysis were:

a. Supply System Pressure

1600 psi operating

2400 psi proof

4000 psi burst

b. Return System Pressure

148 psi operating

500 psi proof

750 psi burst

TABLE XIX. PERFORMANCE REQUIREMENTS HYDRAULIC MOTOR/PUMP DUPLEX ACTUATOR

INPUT

1. Electric Motor - 3 ϕ class A/MIL-M-7969
2. Input - Category B AC/MIL-STD-704A
3. Input Power - 11.0 KVA max. (14 gpm @ 670 psi)
4. Max. Input Current - 600% of normal running current.
5. Line Current Balance/MIL-M-7969
6. Electrical connector - Both halves supplied with unit.
7. Dielectric Strength/MIL-M-7969
8. Radio Noise/MIL-I-6181

OUTPUT

1. Output pressure - 300 psi + sensing pressure (2000 psi - max)
2. Output flow - 0-14 gpm (press. comp)
3. Pump inlet press - Supply pressure/16
4. Fluid – MIL-H-5605
5. Output flow response - $\tau \leqslant 0.1$ sec.

ENVIRONMENT

1. The motor/pump assembly shall be capable of meeting all requirements specified during and after exposure to the following environments:
   a. Inertia Loads - 17 g's in all directions.
   b. Humidity - 100%
   c. Salt atmosphere - Procedure I/MIL-E-5272C
   d. Explosion Proof - Procedure IV/MIL-E-5272C

TABLE XIX. PERFORMANCE REQUIREMENTS HYDRAULIC MOTOR/PUMP DUPLEX ACTUATOR (CONT)

e. Sand Dust - Procedure I/MIL-E-5272C

f. Fungus - Procedure I/MIL-E-5272C

g. Vibration - Procedure XII/MIL-E-5272C

h. Ambient Press - S.L. to 60,000 feet

i. Ambient Temperature - -65° to $+160^{\circ}$F

j. Fluid Temperature - -65° to $+275^{\circ}$F

GENERAL

1. Life - 10 years @ 500 hours/year with normal maintenance MTBF - 1000 hours (min)

2. Duty Cycle - 14 gpm @ 670 psi for 6 sec/hr
   2 gpm @ 1070 psi for 54 sec/hr

3. Weight - 27.0 lbs (max)

4. Tolerances – Unit has capability of operating with aircraft power supply per MIL-E-7894.

c. Maximum Output Loads

16,400 lbs. tension

18,400 lbs. compression

d. Endurance

No. Cycles	Load	Stroke
150,000	100%	100%
750,000	50	50
2,100,000	10	10
12,000,000	2	2

The critical components were analyzed in detail. These included:

(1) Rod end

(2) Piston rod

(3) Piston

(4) Seal plate

(5) Cylinders

(6) Springs

(7) Valve housing

(8) Valve sleeves

(9) Filter cap

(10) Porting tube

(11) Manifold assy

(12) Reservoir assy

(13) Funk strut assy

(14) Servo valve linkage

(15) E/M actuator assy

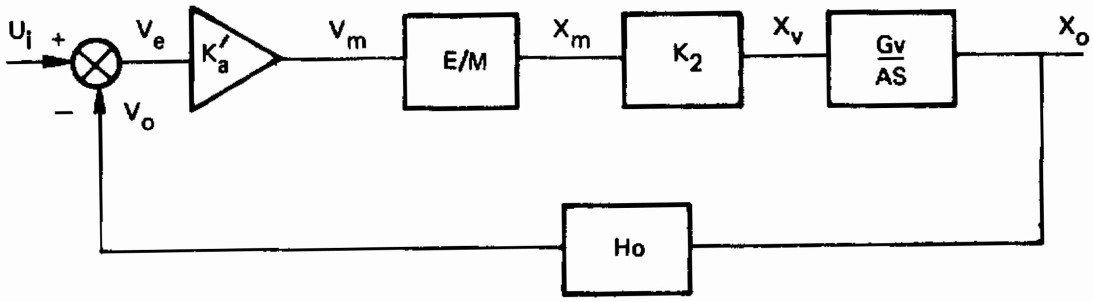
The analysis verified the ability of each component to meet the required performance under the ground rules stated above.

E. PERFORMANCE ANALYSIS

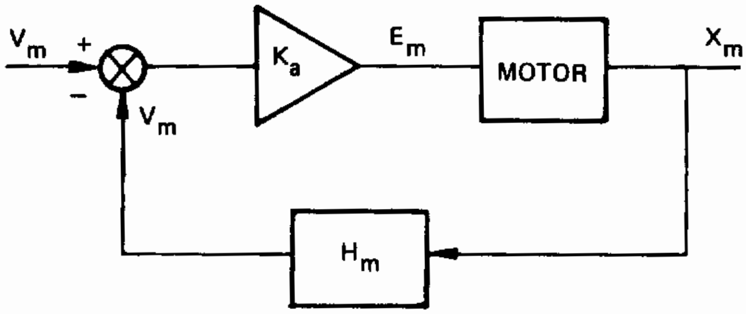
1. Threshold Analysis

High performance military aircraft are equipped with sensitive control surfaces. It is necessary to be able to accurately position these surfaces to gain the desired aircraft performance. One parameter which determines the positionability of the control surface is the threshold of the power control actuator. It is necessary that the actuator threshold be as small as possible. This section evaluates the threshold of the Duplex Integrated Actuator.

For the purpose of evaluating the duplex actuator threshold characteristics, the system can be represented by the following block diagram.



The electro-mechanical block, E/M, represents the following closed loop:



The open loop gain is defined as

$$G = K_a' \ (E/M)\, K_2\, (G_v/A)\, H_o$$

Solving this equation for the amplifier gain required for a loop gain of 21.2 rad/sec.

$$K_a' = \frac{G \ A}{E/M \ K_2 \, G_v \, H_o}$$

$$= \frac{21.2 \ (5.85)}{1/48 \ (1) \ 512 \ (4)}$$

$$= 2.9 \text{ V/V}$$

The force gain of the E/M actuators is

$$F_o/E_e = K_t \, N/R_t$$

where:

F_o	=	Force output
E_e	=	Torque motor input voltage
K_t	=	Torque gain constant
	=	0.865 in-lbs/amp
N	=	gear ratio
	=	50 radians/in
R_t	=	effective resistance
	=	12.5 ohms
	=	0.865 (50)/12.5
	=	3.46 lbs/volt

With an 8-volt motor hysteresis and assuming a main servo valve friction of 1 lb., the motor voltage required to produce valve motion will be,

$$8/2 + 1/3.46$$

$$E_e = 4.289 \text{ volts}$$

With a pressure gain of 3×10^6 psi/in and assuming 100 psi friction in the actuator, the valve motion required to produce actuator motion will be

$$X_v = 100/3 \times 10^6$$

$$= 33 \times 10^{-6} \text{ inches}$$

which (assuming X_v/X_m is unity) at the input level to the E/M actuator will be equivalent to

$$X_v H_m = (33 \times 10^{-6})\ 48 \text{ volts/in}$$

$$= 1.6 \times 10^{-3} \text{ volts}$$

Using the E/M amplifier gain K_a of 125 V/V to convert the stand-off motor voltage to the E/M actuator input level

$$4.289/125 = 0.0343 \text{ volts.}$$

Adding the two voltages and converting to the output level, the threshold will be

$(0.0343 + 0.0016)/(2.9 \times 4) = 3.09 \times 10^{-3}$ inches

2. Stability Analysis

A limited linear analysis was conducted on the Duplex to determine the approximate degree of stability when installed in the F-4 and driving the stabilator surface. With the gain adjusted to 90 rad/sec, the inner E/M closed loop performance can be described by the following transfer function:

$$\frac{19792.027}{S^2 + 219.911\ S + 19792.027}$$

Using parameters similar to those used for Simplex and a gain of 20.4 rad/sec, the complete open loop can be described by the following open loop transfer function:

$$G_{(S)}\,H_{(S)} = \frac{311719.8\ (S^2 + 12.7\ S + 44760)}{S(S^4 + 234.24\ S^3 + 57506\ S^2 + 7884570\ S + 684092672)}$$

The roots are:

Numerator:

-6.347 + j211.47

-6.347 - j211.47

Denominator:

-200 + j87.76

-200 - j87.76

-7.16 + j185.8

-716-j185.8

The closed loop transfer function is

$$G_{(S)}H_{(S)} = \frac{135530\ (S^2 + 12.7\ S + 44760)}{S^5 + 234\ S^4 + 57506\ S^3 + 8196290\ S^2 + 6.88 \times 10^8\ S + 1.39 \times 10^{10}}$$

The denominator roots are:

-27.9 + j 0.0

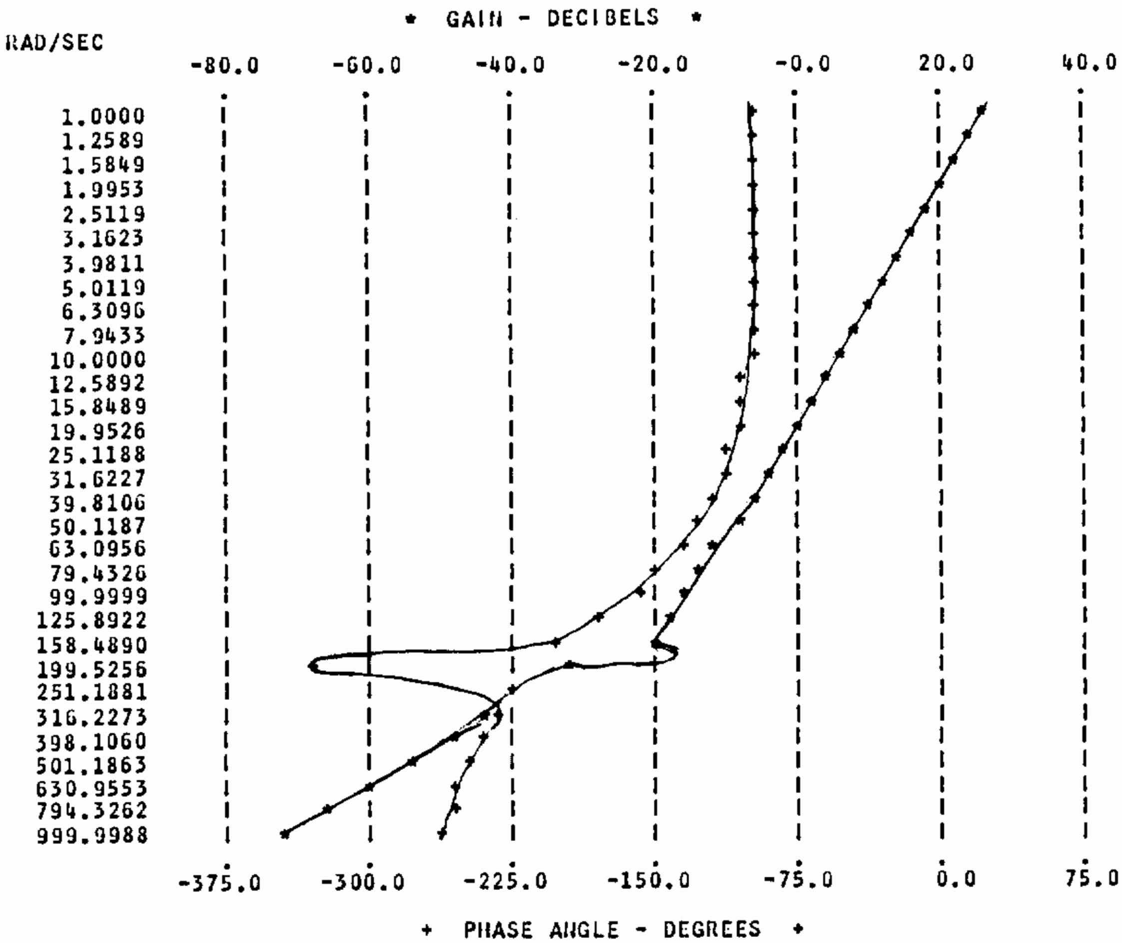
-95.4 + j 74.0

-95.4 - j 74.0

-7.7 + j184.7

-7.7 - j184.7

A frequency response evaluation of the open loop transfer function in the frequency range of the denominator resonance indicates a gain margin of approximately 18 db.



3. Funk Strut Analysis

When force summing the outputs of several position servos, it is necessary to sum them through compliant members because of the possible mis-synchronization of the servos. The servo mis-sync is due primarily to gain variation between the different feedback transducers. The funk struts are designed to take up the normal mis-sync between the four E/M position servos. In addition, the funk struts have asymmetrical breakout loads to ensure a detented position as the output of the four servos. During normal mis-sync operation as many as three of the four funk struts can be broken out at one time. A single strut remains in detent to provide the effective ground for the valve linkage. With four actuators operating, two can be broken out in the low force direction, one actuator can be broken out in the high force direction and the other positioning the linkage while holding a force within its detent.

With three actuators operating, one of the high force funks can be bucked out by one broken out in the low force direction with assistance from the detented actuator positioning the linkage. With two actuators operating, one can be broken out in the low force direction and restrained by the other which will be detented and controlling the linkage. The following analysis is conducted to ensure that the force margin, or difference between the load that the detented actuator carries and that at which it breaks out, is a maximum. For four operating units, the force margin ($F_{bo} - F$) will be

$$F + F_h = 2 F_e$$

or

$$F_{bo} - F = 2 F_h - 2 F_e$$

where

F	=	detented actuator force
F_{bo}	=	detented actuator breakout force
F_h	=	high breakout force
F_e	=	low breakout force

For three operating units, the force margin will be

$$F + F_e = F_h$$

or

$$F_{bo} - F = 2 F_e - F_h$$

For two units operating, the margin will be

$$F = F_e$$

or

$$F_{bo} - F = F_h - F_e$$

The optimum breakout ratio of the asymmetrical funk struts is that which results in equal and maximum force margin of the detented actuator for all three operating modes. Equating the margins of three and four units operating determines the best breakout ratio for those two modes. Therefore,

$$(F_{bo} - F)_3 = (F_{bo} - F)_4$$

$$2\,F_e - F_h = 2\,F_h - 2\,F_e$$

$$F_e = 3/4\,F_h$$

Equating the force margins when two and three systems are operating determines the best breakout ratio for those two modes. Therefore,

$$(F_{bo} - F)_2 = (F_{bo} - F)_3$$

$$2F_e - F_h = F_h - F_e$$

$$F_e = 2/3\,F_h$$

The above indicates that the optimum ratio is not the same for all three modes. Plotting the breakout ratio vs. force margin for all three modes shows the variation of force margin as the number of systems varies for any selected breakout ratio (Figure 73). It is evident that the funk will have the greatest minimum force margin for all conditions if the breakout force in one direction is 0.666 times the breakout force in the other direction.

4. Slew Rate Analysis

a. General

Actuator slew rate is a function of piston area, valve opening, and external load. The duplex actuator was sized to provide a nominal surface rate of 25^o/sec with a 20-inch moment arm. The actuator velocity is

$$V_A = (25/57.3)\,(20)$$

$$= 8.73 \text{ in/sec.}$$

The rod end cylinder has equal area of 5.15 in^2. The lug end cylinder has extend and retract areas of 6.38 in^2 and 5.15 in^2 respectively. The following is an analysis of the unloaded actuator slew rate under various operating modes, assuming adequate pump capability to sustain system flow, constant supply pressure, and balanced valve orifices.

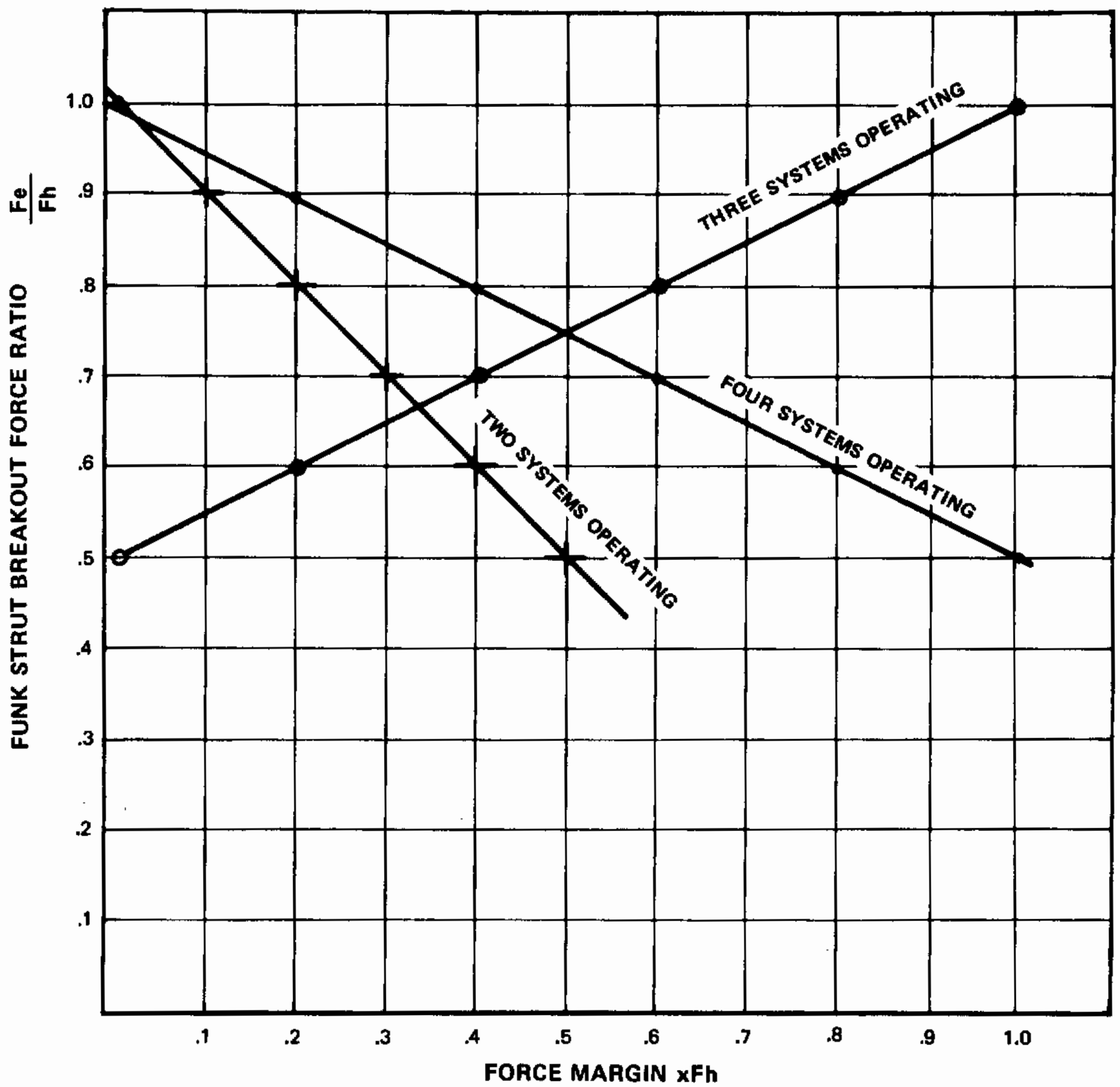
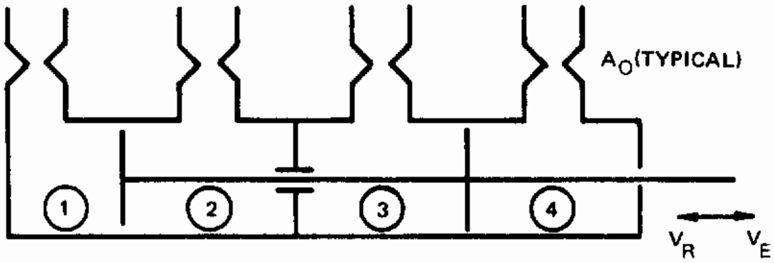


FIGURE 73. FUNK SPRING FORCE MARGIN

b. Single System Operation

The control valve-actuator combination can be described schematically as shown below.



The orifices A_O represent the control valve opening. For the equal area cylinder the actuator velocity can be described (assuming no loading by the lug end cylinder) as follows:

$$V = \frac{Q_3}{A_3}$$

$$= \frac{KA_O\sqrt{P_{S/2}}}{A_3}$$

$$V = 0.138\, KA_O\sqrt{P_S}$$

where:

K	=	orifice coefficient
P_S	=	supply pressure
A_3	=	cylinder 3 area

The slew rates resulting from the unequal area cylinder operation are different in the extend and retract direction. The following relationships exist:

$$P_1/A_2 = P_2/A_1$$

$$Q_1/A_1 = Q_2/A_2$$

$$V_E = Q_1/A_1$$

$$V_R = Q_2/A_2$$

Where:

P_1 = extend pressure

P_2 = retract pressure

A_1 = extend area

A_2 = retract area

Q_1 = extend flow

Q_2 = retract flow

V_E = extend velocity

V_R = retract velocity

The flow equation for extend motion is

$$Q_1 = K\ A_O \sqrt{P_S - P_1} = \frac{A_1}{A_2}\ K\ A_O\ \sqrt{P_2}$$

Solving these equations for V_E yields the extend velocity

$$V_E = 0.127\ KA_O\sqrt{P_S}$$

Also the flow equation for retract motion is

$$KA_O\sqrt{P_1} = \frac{A_1}{A_2}\ KA_O\sqrt{P_S - P_2}$$

Solving for V_R yields the retract velocity

$$V_R = 0.114\ KA_O\sqrt{P_S}$$

Therefore sizing the actuator to provide a minimum slew rate of 25°/sec when retracting with the lug end cylinder only, the extend slew rate is larger by the relative velocity ratios,

$$V_E = \frac{(0.127)\,(25)}{0.114} = 27.8^{\circ}/\text{sec.}$$

and the slew rates, both extend and retract, of the equal area cylinder are

$$V = \frac{(0.138)(25)}{0.114} = 30.2^{o}/\text{sec.}$$

c. Two System Operation

The unloaded slew rates of the actuator when both systems are operating can be determined in a manner similar to single system operation. The following relationships define the extend velocity.

$$Q_1 = KA_O\sqrt{P_S - P_1}$$

$$Q_2 = KA_O\sqrt{P_2}$$

$$Q_3 = KA_O\sqrt{P_S - P_3}$$

$$Q_4 = KA_O\sqrt{P_4}$$

$$Q_1 = A_1/A_2\, Q_2$$

$$Q_2 = Q_3 = Q_4$$

$$P_1 \frac{A_1}{A_2} + P_3 = P_2 + P_4$$

$$V_E = Q/A$$

Solving the above relationships results in an extend velocity of

$$V_E = 0.143\ KA_O\sqrt{P_S}$$

or:

$$V_E = \frac{(0.143)(25)}{0.114} = 31.3^{o}/\text{sec.}$$

Solving the corresponding equations for retract direction results in a retract velocity of

$$V_R = 0.124\ KA_O\sqrt{P_S}$$

$$= \frac{(0.124)(25)}{0.114}$$

or

$$V_R = 27.1^o/\text{sec.}$$

d. Single System Failure

With one hydraulic system passive, but full of fluid and loading the second system, the actuator velocity will be reduced. The greatest reduction occurs when the lug end cylinder is dormant and when the actuator is retracting (large area of lug end cylinder resisting motion).

The reduction in velocity under these conditions can be determined as follows:

$$P_2 = 0$$

$$P_1A_1 + P_3A_3 = P_4A_4$$

$$Q_3 = Q_4$$

$$KA_O\sqrt{P_3} = KA_O\sqrt{P_S - P_4}$$

$$V_R = \frac{Q_4}{A_4}$$

Solving these equations results in a retract velocity

$$V_R = 0.083\ KA_O\sqrt{P_S}$$

or an equivalent rate of

$$V_R = \frac{(0.083)\ (25)}{0.114}$$

or:

$$= 18.2\ \text{deg/sec.}$$

e. Summary

To ensure a minimum surface slew rate of 25^o/sec., with either or both systems operating normally, the actuator will exhibit a maximum rate of 31.3^o/sec. when both systems are operable. In the event of a failure in which the failed system is loading the operating system, the slew rate can be reduced to as low as 18.2^o/sec. The actual rate after a failure depends on the type of failure and the degree of loading that the failed system provides.

5. Thermal Analysis

a. Analysis

The Duplex thermal analysis is similar to the Simplex. The problem is to evaluate the thermal balance equation.

heat generated = heat absorbed + heat transferred

or

$$H_g = H_a + H_t$$

The heat generated is a function of pump inefficiency and the flow-pressure duty cycle. The duty cycle assumed for this analysis is 14 gpm at 670 psi for 6 sec/min. and 2 gpm at 1070 psi for 54 sec/min.

Pump output horse power is defined by

$$HP = Q \cdot P \cdot 3.86 \, \frac{in^3/sec}{gpm} \; \frac{1 \text{ ft}}{12 \text{ in.}} \; \frac{1 \text{ HP sec}}{550 \text{ ft lbs}}$$

For 6 sec.

$$HP = \frac{(14)(670)(3.85)}{(12)(550)}$$

$$= 5.46 \text{ HP}$$

For 54 sec.

$$HP = \frac{(2)(1070)(3.85)}{(12)(550)}$$

$$= 1.25 \text{ HP}$$

Based on manufacturer's performance data, the pump inefficiency is 1.825 HP and 1.68 HP for the 6 sec. and 54 sec. conditions, respectively. Therefore, for the 6 second condition

$$\text{Power loss} = 5.46 + 1.83 \text{ HP}$$

$$= (7.29)(42.44) \text{ BTU/min}$$

$$= 309 \text{ BTU/min } (6 \text{ min/hr})$$

$$= 1850 \text{ BTU/hr}$$

and for the 54 second condition

$$\text{Power loss} = 1.25 + 1.68 \text{ HP}$$

$$= (2.93)(42.44) \text{ BTU/min}$$

$$= (124 \text{ BTU/min})(54 \text{ min/hr})$$

$$= 6700 \text{ BTU/hr}$$

Total power lost = 1850 + 6700 = 8550 BTU/hr/system

Therefore the total heat generated neglecting input from the pump motor, E/M actuator motor, and LVDT's is 17,100 BTU/hr.

The heat absorbed

$$H_a = (W_s C_{ps} + W_o C_{po}) \frac{dT}{dt}$$

where:

C_{ps} = specific weight of steel, BTU/lbs ^oF

C_{po} = specific weight of oil, BTU/lbs ^oF

W_s = weight of steel, lbs

W_o = weight of oil, lbs

T = temperature, ^oR

t = time, hr.

The heat transferred,

H_t = heat convected + heat radiated

$$= H_1 A_1 (T - T_a) + H_2 A_2 (T^4 - T_a^4)$$

where:

H_1 = convection coefficient – $\dfrac{BTU}{HR\ ^oF\ ft^2}$

H_2 = radiation coefficient – $\dfrac{BTU}{HR\ ft^2\ ^oF^4}$

A_1 = convection surface area, ft^2

A_2 = radiation surface area, ft^2

T_a = ambient temperature, ^oR

Weight of Oil

Total oil volume @ $-65^{o}F = 100\ in^3$/system

W_o = vol (ρ-65) = 100 (.033) = 3.3 lbs.

or for both systems W_o = 2(3.3) = 6.6 lbs.

Weight of Package

Total weight of package = 184 lbs.

Less 2.0 lbs for funk struts

9.6 lbs for E/M actuators

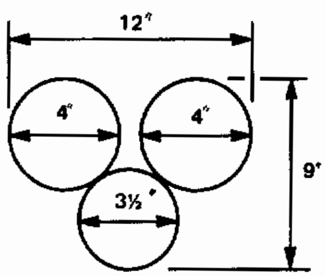
6.6 lbs for fluid

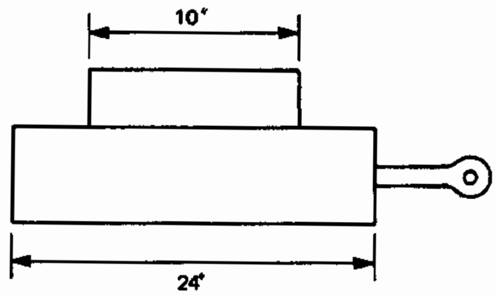
and 29.0 lbs for electric motors

results in a dry heat sink weight of

W_s = 137 lbs (assume all steel)

Area Estimates





Convection Area (A_1)

$$2(4\pi 10) + 3.5\ \pi 24 + (\pi/4)(4^2)2 + (\pi/4)(3.5)^2 \times 2$$

$$= 555\ in^2 \qquad = \underline{\underline{3.86\ ft^2}}$$

Radiation Area (A_2)

$$(12 + 9)\ 10 \times 2 + (3.5 + 3.5)\ 14 \times 2 + 2(9 \times 12/2)$$

$$= 724\ in^2 \qquad = \underline{\underline{5.02\ ft^2}}$$

Combining the above relationships, the surface temperature can be described by

$$T = \int \left[\frac{H_g - H_1A_1\,(T\text{-}Ta) - H_2A_2\,(T^4 - Ta^4)}{C_{ps}W_s + C_{pf}W_f} \right] dt$$

constants:

H_1 = 3.0

A_1 = 3.86

H_2 = 0.1548×10^{-8}

A_2 = 5.02

C_{ps} = 0.117

W_s = 137

C_{po} = 0.6

W_o = 6.6

b. Results

A computer routine was used to evaluate the transient temperature relationship. Figure 74 depicts the transient thermal response for three different generating loads at two different ambient temperatures. Cursory examination of the curves indicates steady state temperatures which are significantly above Type II hydraulic system limits.

F. TESTING

The unique features of the Duplex IAP offers many fertile areas for evalution testing and investigation. However, the tests which were conducted on the package were limited and were aimed at determining the basic characteristics of the Duplex. The package is a fly-by-wire unit having two-fail-operate capability. Therefore, a considerable amount of testing was conducted to determine the operating characteristics with various failures and combinations of failure. Specific tests which were conducted are:

1. Weight

2. Proof Pressure

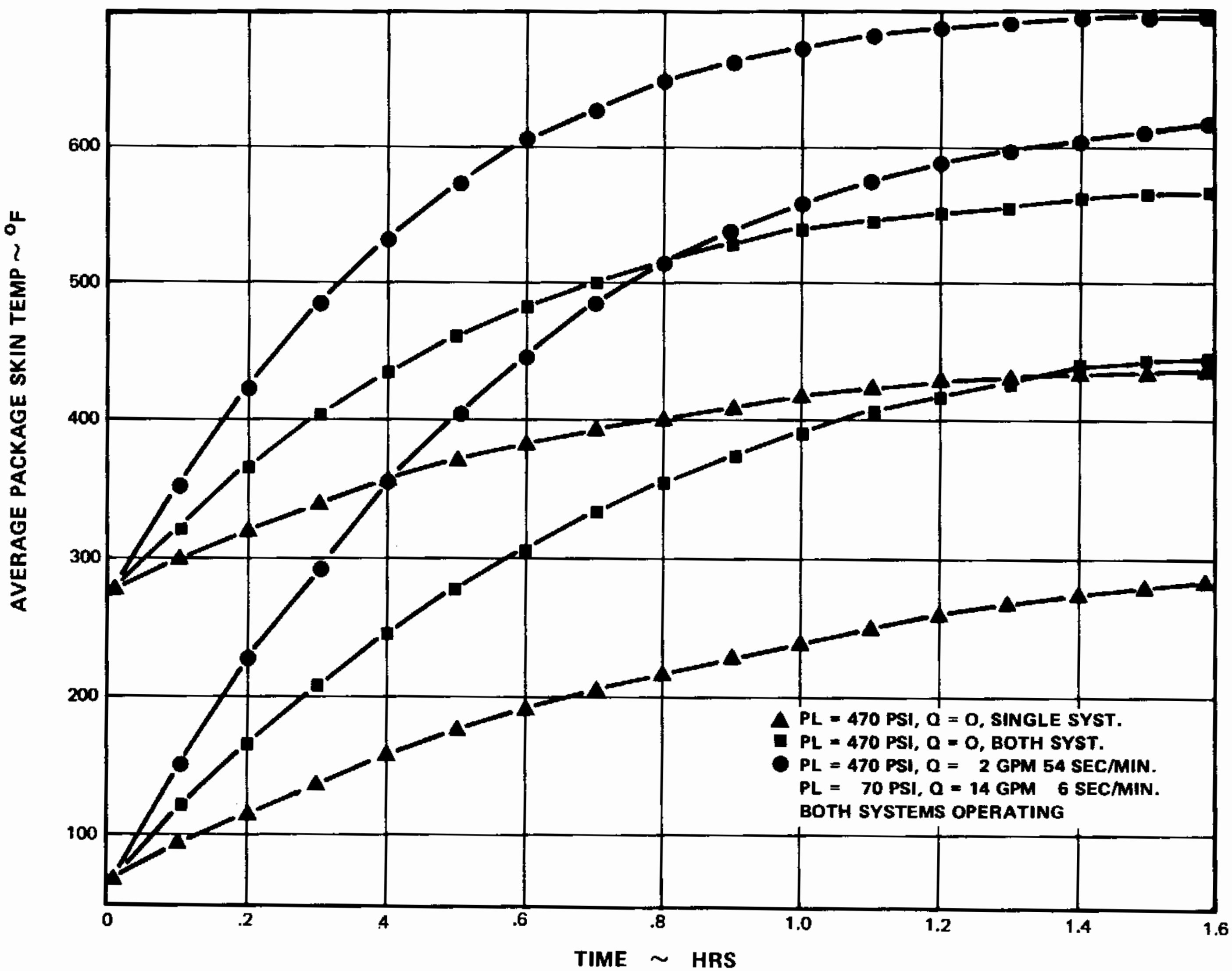


FIGURE 74. TEMPERATURE RESPONSE DUPLEX ACTUATOR

3. Frequency response

4. Failure transients

5. Slew Rate

6. Threshold

7. Load-temperature

8. Power Input

The results of these tests which are presented in Appendix VII indicate that the electro-mechanical fly-by-wire mechanization used on the Duplex shows excellent promise for use in a production fly-by-wire control system. The results also point up the requirement of external cooling requirements for integrated packages.

SECTION VII
CONCLUSIONS

1. General

The program described in this report has resulted in the establishment of primary flight control actuator designs and design techniques which offer the potential of significant small arms fire survivability improvement for present and advanced fighter/attack aircraft. A flight test model of a Simplex integrated actuator was designed for the F-4 stabilator control. Three of these units were built. Functional and environmental tests performed on the units indicate that the units are suitable for aircraft installation and flight test evaluation. A laboratory model of a fly-by-wire self-contained Duplex integrated actuator package was designed, fabricated, and tested. The unit demonstrates that the basic concept of fly-by-wire integrated packages is sound from performance and structural aspects.

2. Feasibility of Integrated Packages

Feasibility of the Duplex integrated actuator package concept was evaluated. Application of the package concept indicates that, although it does accrue a cost and weight penalty when compared to a conventional system, the aircraft surivability improvement it offers is of significant importance. Use of this concept as a candidate system in trade studies for flight control systems of future military aircraft is warranted.

The Duplex actuator developed as part of this program is signalled electrically. To accept the electrical signals, the unit contains quadruplex electro-mechanical signal converters (Electro-RAM) to provide dual fail/operate capability for the signal channels. The outputs of the converters are force summed to collectively position the main control valve. Evaluation of the signal converters as well as the complete Duplex package indicates that the electro-mechanical force summing arrangement offers characteristics which are superior to other conversion methods. These are primarily due to:

- Complete isolation of signal and power channels
- Power level and dynamic response requirements of signal converters are ideally suited to Electro-RAM concepts
- Minimum contribution to thermal loads

Dynamic response of the Fly-by-Wire Duplex was evaluated. Adequate system performance after one and two signal channel failures was demonstrated.

3. Problem Areas

During the concept feasibility study, it became obvious that successful application of the IAP concept to high performance military aircraft required careful analysis and evaluation of the control surface power requirements. Cost and weight of the IAP system is much more sensitive to output power requirements than is the conventional system. Not only does the basic hydraulic actuator have to be sized to meet the power requirements, but the pump and electric motor must be sized to provide the flow and pressure at the specified duty cycles. Also, the package structural members must be sized to support the weight of the unit under worst environmental conditions. Consequently, it is imperative that the actual surface power be properly determined. Location of integrated packages in the wing and vertical stabilizer will require, in addition to minimizing the input power, judicious arrangement of components and/or localized deviations to the air foils.

Another problem area in the utilization of integrated actuator packages is removal of the thermal energy generated in the package. Essentially all the electrical energy supplied to the package is converted to heat. This thermal energy must all be exchanged to the ambient air at a sufficiently low enough impedance to achieve a package temperature low enough to be compatible with the components' allowable operating condition.

REFERENCES

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2. "Maintenance Instructions; Flight Control Systems"; USAF Series: F-4C, F-4D, F-4E, and RF-4C Aircraft; McDonnell Douglas; NOs (A) 63-0032-i, N00019-67-C-0171.

3. "Tradeoff Study of Duplex Fly-By-Wire System Motor Pump Package"; C.J. Mali, Sr. Project Engineer; Vickers Aerospace Division, Sperry Rand Corporation, Troy, Michigan.

4. "Fly-By-Wire Techniques"; F.L. Miller and J.E. Emfinger; Sperry Pheonix Company; Report No. AFFDL-TR-67-53; July 1967.

5. "Design and Evaluation of a Single Axis Redundant Fly-By-Wire System"; Vernon Sethre, Richard V. Hupp, and Gerald A. Rayburn; McDonnell Douglas Corporation; July 1968.

APPENDIX I

CONVENTIONAL SYSTEM DEFINITION

A. GENERAL

The primary flight control system of the A-7 aircraft was analyzed with respect to cost, weight, volume, and vulnerable area. The primary flight control system is defined as all mechanical and hydraulic components required to deflect a control surface.

References utilized in compiling this data are:

Vought Aeronautics Division
A-7 Weight & Balance
Report No. 2-59330/8R-5350

Vought Aeronautics Division
Weight Control Date Book

Vought Aeronautics Division
A-7 Flight Controls System Design
Report No. 2-51724/5R-5131

B. ASSUMPTIONS

The following criteria and assumptions were used as a basis for the data derived in this study:

1. Hydraulic System

   - The pressure and return trunk lines for PC-1 and PC-2 systems are assumed to average 3/4 inch tubing size.
   - Length of pressure and return trunk lines are assumed to be equal.
   - PC-1 and PC-2 trunk lines, pumps, reservoirs, valves, filters, and fluid are assumed identical.
   - PC-1 and PC-2 trunk to actuator lines are assumed equal in length and to average 3/8 inch tubing size.
   - Vulnerable area was considered as that area of the system exposed in a horizontal plane only.

2. Flight Control System

   - Only moving linkage was considered; bracketry or supporting structure is not included.
   - Costs of components were based upon the following:

     Ten components were selected from the A-7 controls as representative of components making up the complete controls mechanism. The cost of these ten components was divided by their weights to derive a cost per pound factor. This factor was applied to similar components to arrive at an estimated cost.

* Vulnerable area was based upon that portion of the system exposed in a horizontal plane and not shielded by other components of the control system.

C. HYDRAULIC SYSTEM

Item	Weight Lbs.	Projected Area in^2	Volume in^3	Cost
PC-1 SYSTEM	(2310")			
Plumbing (Trunk Lines)	47.0	1730.0	1025.0	5500.00
Pump	13.66	60.0	150.0	835.00
Reservoir	8.23	42.0	265.0	450.00
Accummulator	2.99	10.0	26.2	700.00
Filters	6.78	48.0	93.0	185.00
Valves (Check, Relief, Etc)	4.12	61.2	53.5	80.00
Fluid (MIL-H-5606)	13.21			4.00
PC-2 SYSTEM	(ASSUME SAME AS PC-1 SYSTEM ABOVE)			
Pitch Control (UHT)	(1140")			
Plumbing (Trunk Line to Actuator)	6.74	428	127	1690.00
Fluid	7.32			2.20
Rudder	(326")			
Plumbing (Trunk Line to Actuator)	1.98	126	37.3	483.00
Fluid	0.75			0.23
Aileron	(3550")			
Plumbing (Trunk Line to Actuator)	21.03	1330	394	5250.00
Fluid	2.31			0.69
Spoiler	(1405")			
Plumbing (Trunk Line to Actuator)	8.28	528	156	2080.00
Fluid	1.04			0.31
AFCS Actuators	(627")			
Plumbing (Trunk Line to Actuator)	3.70	235	69.7	930.00
Fluid	0.22			0.07
TOTAL	245.35	6549.4	4009.4	25944.50

D. LINKAGE SYSTEM

1. Longitudinal

	WEIGHT LBS	PROJ AREA IN^2	VOLUME IN^3	COST
215-28025				
215-21085-1 Stick Grip	1.25	Hidden by Housing	16.90	50.00
215-21129-1 Sensor Assy	5.00	Hidden by Housing	32.90	500.00
215-28145-2 Housing Assy	3.51	43.03	141.70	173.80
215-28146-2 Arm Assy	1.76	19.94	25.90	194.30
CV15-401293-4 Cyl Assy	3.05	24.75	44.90	125.00
215-28344-2 Bob Wgt Assy	12.35	7.84	19.10	145.00
215-78055				
215-38303-2 Bellcrank Assy	0.28	4.20	4.30	29.85
CVC559C5130H77 Conn Link	0.98	76.90	90.80	70.40
215-38304-2 Arm Assy	0.22	0.51	3.38	24.30
CVC554C4903H84 Conn Link	0.77	73.50	86.75	55.30
215-78303-1 Arm Assy	0.14	1.06	2.15	15.47
CVC564C3983H180 Conn Link	0.72	54.75	59.25	51.60
CV15-408535-2 Idler Assy	0.14	1.19	2.15	15.47
215-38322-1 Conn Link Assy	0.77	55.85	60.30	55.30
215-48070				
215-48305-2 Bellcrank Assy	0.67	4.74	10.31	72.90
215-48318-1 Link Assy	0.95	68.70	80.90	68.20
215-68030				
CV21-658554-1 Fitting Assy	0.10	Hidden by AFCS Act	0.33	7.18
215-68418-1(2R) Spring	0.34	18.30	25.25	3.96
215-68310-3 Link Assy	0.25	Hidden by AFCS Act	3.46	17.95
215-68102-1 Feel Strut Assy	2.95	Partly Hidden 7.02	31.10	525.00
215-68080-1 Trim Actuator	1.20	Hidden by AFCS Act	20.59	412.00
215-68115-2 Arm Assy	0.32	6.51	4.92	35.35
215-68107-2 Bellcrank Assy	0.75	Hidden by AFCS Act	11.53	82.90
215-68109-1 Bellcrank Assy	0.37	6.62	5.69	40.85
215-68110-2 Link Assy	0.32	5.60	4.93	35.35
215-68305-2 Bob-Wgt Assy	11.00	5.40	15.21	122.00

215-68304-2 Conn. Link	0.40	Hidden by AFCS Act	3.68	28.75
215-68113-2 Arm Assy	0.49	Hidden by AFCS Act	7.54	54.20
215-68111-2(2R) Link Assy	0.66	2.31	10.17	72.95
215-68310-1 Link Assy	0.25	4.12	2.03	17.95
215-68103-2 Bellcrank Assy	0.70	13.80	11.67	77.40
CVC511-785H90 Conn Link	0.24	0.44	3.30	50.25
210-3223-10 AFCS Actuator	17.00	46.40	102.50	4011.00
CV15-601060-7 Cyl Assy	4.13	Hidden by AFCS Act	35.80	125.00
215-68108-2 Arm Assy	1.20	Hidden by AFCS&Feel Strut	18.47	132.50
215-28332-2 Link Assy	0.46	18.63	18.29	33.10
215-28154-2 Support Instl	0.84	5.04	12.92	92.90
215-38323-3 Link Assy	1.02	3.60	86.80	73.30
215-28308-2 Arm Assy	0.47	7.84	7.24	51.90
215-48025 (Split)				
215-48303-2(2R) Bellcrank	1.08	8.60	15.88	119.20
CV15-608027-3(2R) Strut Assy	5.10	17.55	22.38	907.50
215-48304-2(2R) B'crk Assy	1.04	8.60	15.88	115.00
CVC542-2321-90(2R) Rod Assy	0.76	0.44	8.06	54.60
CV15-608517-2(2R) B'crk Assy	0.52	7.36	8.00	57.50
CVC511-2760-180(2R) Rod Assy	0.80	21.00	12.32	57.40
215-48038				
CV15-160059(2R) U.H.T. Arm	25.35	30.40	42.30	635.00
CV15-601051-17&-18 Cyl Assy	65.16	114.80	404.40	7390.00
CV15-608056-5(2R) Rod Assy	1.70	44.10	25.90	122.10
215-48100-2 Guide Assy	1.60	15.60	19.80	343.10
CV15-608548-3&-4 Arm Assy	1.14	10.25	17.50	126.00
CV15-608549-2 (2R) Arm Assy	0.64	7.46	10.10	70.70
CV15-608514-4(2R) Support Assy	1.02	14.39	15.70	112.80
CV15-60835-12(4R) Rod Assy	2.36	20.40	8.48	169.30
TOTAL	186.29	909.54	1751.61	18,030.83

2. Lateral

214-28025	WEIGHT LB	PROJ AREA IN^2	VOLUME IN^3	COST
215-28143-3 Link Assy	0.20	Inside Stick Column	---	14.37
215-28158-2 B'crk Assy	0.26	Inside Stick Column	---	28.70
215-28155-1 Link Assy	0.51	9.37	5.52	90.75
215-28144-4 Arm Assy	0.15	Inside Stick Column	---	16.58
215-28021				
215-28310-2 B'crk Assy	0.42	11.23	6.67	46.40
215-28332-1 Link Assy	0.38	25.6	19.43	27.30
215-28109-1 Link Assy	1.64	25.09	22.04	292.00
215-28313-2 B'crk Assy	0.36	2.1	5.7	39.80
CVC512-3721H180 Conn Link	0.70	37.21	29.2	50.30
215-28154 Support Instl				
215-28153-2 Arm	0.22	2.52	3.50	24.30
215-28152-3 Arm	0.33	2.52	5.25	36.45
215-38021 Cont. Instl				
215-38314 B'crk Assy	0.49	6.75	7.78	54.20
215-38474 Conn Link	0.60	42.0	30.0	43.10
215-38305 Arm Assy	0.25	2.5	3.97	27.65
215-38305 Arm Assy	0.25	2.5	3.97	27.65
CVC564B4568H180 Conn Link	0.60	35.9	45.68	43.10
CVC512-4970H90 Conn Link	0.81	39.0	49.70	58.10
215-38105-2 B'crk Assy	0.36	8.22	5.72	25.85
215-38020 Cont Instl				
215-38306 Link Assy	0.45	Shielded by Stab Act.	2.06	32.30
210-32277	6.51	Shielded by Stab Act.	31.58	738.00
215-38315 Link Assy	0.27	Shielded by Stab Act.	2.7	19.38
215-38107 Arm Assy	0.32	Shielded by Stab Act.	5.08	35.30
215-38109 Conn Link	0.14	Shielded by Stab Act.	2.22	10.93
215-38316 Arm Assy	0.24	Shielded by Stab Act.	3.81	26.50
215-38106	0.39	Shielded by Stab Act.	6.2	43.10
215-38108 Arm Assy	0.35	Shielded by Stab Act.	5.56	38.65

<u>215-88030 (Aileron)</u>				
215-88102-1(2R) Conn Link	0.84	10.64	6.54	149.70
215-88109-3 & -4 Arm Assy	1.28	16.66	18.82	141.50
CVC541-1342H180(2R) Conn Link	0.51	20.24	5.96	36.60
215-88108-3 & -4 Arm Assy	0.94	13.12	14.47	103.90
215-88105-1(2R) Conn Link	0.90	4.73	1.63	64.60
<u>215-88020 (Aileron)</u>				
215-88021-3(2R) Bellcrank Assy	8.08	31.84	26.90	202.00
215-88024-2(2R) Adapter Assy	0.86	6.66	2.87	22.10
215-82031-1(2R) Ail Actuator	23.50	146.94	173.10	2663.00
215-88021-2(2R) Bellcrank Assy	7.06	31.84	26.55	177.00
215-88022-2(4R) Conn Link	5.02	30.32	17.33	124.50
<u>215-78030 (Spoiler)</u>				
215-72031-1 & -2 Spoiler Act	27.46	146.94	173.10	2545.00
215-78031-2 (4R) B'crk Assy	6.12	38.64	94.20	676.00
215-78032-2(2R) Conn Link	1.00	Shielded by 215-78031-2	15.39	110.40
215-78032-3(2R) Conn Link	0.90	Shielded by 215-78031-2	13.85	99.40
215-78301-1(2R) Turnbuckle	0.88	4.45	11.18	63.20
215-78301-2(2R) Turnbuckle	0.84	4.45	11.18	60.30
<u>215-38020 (Cont'd)</u>				
215-38400-1 Spring 215-38400-2 Spring Nested	0.125	Shielded by Stab. Act.	1.97	1.43
215-38302-2 Link Assy (2 reqd)	0.48	Shielded by Stab. Act.	7.62	53.10
215-38102-2 B'crank Assy	0.57	Shielded by Stab. Act.	9.05	63.00
210-32230-9 AFSC Act	15.73	27.0	21.6	4011.00
215-38318 Link Assy	0.48	0.44	8.65	53.10
<u>215-38035 Control Inst</u>				
215-38327 B'crank	0.35	5.5	5.57	38.70
CVC512-1531H90 Conn Link	0.382	15.31	12.1	27.40
215-38308-2 B'crank	0.98	14.18	15.5	100.82
215-38310 Arm Assy	0.71	12.6	11.3	78.40
215-38309 Arm Assy	0.58	8.4	9.2	64.00
215-38030 Trim Actuator	0.90	7.1	13.0	308.50

215-78021

215-78102-1(2R) Conn Link	7.06	54.88	62.00	1258.00
CVC513-2918H160(2R) Conn Link	1.28	56.98	47.00	91.90
215-78103-2(2R) Arm Assy	0.50	Partly hid by 215-78105 8.10	15.38	55.20
215-78105-2(2R) Arm Assy	0.48	11.76	14.76	53.00
CVC516-2358H180(2R) Conn Link	0.84	Hidden by 215-78101	28.20	60.30
215-78101-1(2R) Conn Link	5.42	41.86	56.80	964.00
215-78104-3&-4 Bellcrank Assy	0.98	Partly hid by 215-78106 7.06	15.08	108.20
215-78106-2(2R) Arm Assy	0.044	11.36	6.77	48.60
CVC544-5000H173(2R) Conn Link	1.92	108.95	94.00	138.00
215-88101-2(2R) Conn Link	1.08	32.14	15.90	77.60
215-78107-3&-4 Arm Assy	0.90	15.00	13.85	99.40
CVC515-3326H180(2R) Conn Link	1.06	54.84	32.90	74.80
TOTAL	141.82	1,244.07	1,424.61	16,958.41

3. Directional System

	WEIGHT LBS	PROJ AREA IN^2	VOLUME IN^3	COST
215-28026				
CV15-408514-1 Tube	0.68	12.03	35.99	48.90
CV15-408036-3&-4 Arm Assy	1.64	10.28	31.98	81.20
CV15-408039-11&-12 Pedal Assy	2.06	34.86	67.16	102.00
CV15-408042-10(2R) Rod Assy	0.40	1.58	2.61	28.75
215-28301-3&-4 Bellc'k Assy	1.04	12.92	13.19	113.80
CV15-408080-7(2R) Cable Assy	1.54	10.63	3.92	238.50
MS20220-2(2R) Pulley	0.18	13.64	4.61	
215-28022				
CV15-408521-1(2R) Link Assy	0.10	1.40	1.15	
NAS305-24-2935 Cable Assy	1.11	40.40	3.82	338.50
NAS305-24-2714 Cable Assy	1.20	37.30	4.28	326.00
MS20220A1(2R) Pulley	0.13	1.26	1.44	
$MS20220A_2$(2R) Pulley	0.35	2.43	5.38	
215-78055				
MS20220A2(2R) Pulley	0.35	2.43	5.38	
MS20220A1(2R) Pulley	0.13	1.26	1.44	
MS21251-52(2R) Turnbuckle	0.18	3.42	0.58	
MS21256-2(4R) Cup	0.003	Negligible	Negligible	
215-48070				
MS20220A2(2R) Pulley	0.35	2.43	5.38	
NAS305-35-1010 Cable Assy	0.49	13.89	1.75	130.40
NAS305-35-1212 Cable Assy	0.57	16.68	2.03	141.50
215-48105-11 Cable Assy	0.38	25.00	1.35	52.75
MS20219A2 Pulley	0.066	0.44	0.61	
215-38021				
MS20219A2 Pulley	0.066	0.44	0.61	
MS21251-35 Turnbuckle	0.028	1.70	0.08	
215-68025				
CV15-158110-4 Linkage Assy	5.01	14.50	29.05	554.00
CV511-806H180 Push Rod	0.24	5.60	3.07	17.23
215-68300-1 Arm Assy	0.28	3.78	4.45	29.85
CV15-158036-2 Arm Assy	0.39	4.70	5.74	43.10
215-68131-1 Transducer	0.80	12.07	4.80	

CVC541-1072H180 Push Rod	0.23	5.12	2.51	16.50
CV15-158039 Idler Assy	0.24	4.05	3.81	26.50
CV15-158037 Arm Assy	0.36	2.0	5.29	39.80
RE4F7 Rod End (2R)	0.36	2.72	2.40	4.00
NAS27-7-32	0.16	0.90	0.53	2.00
210-32230-10 AFCS Actuator	17.00	48.10	102.50	4011.00
215-68105-2 Arm Assy	0.36	Hidden by AFCS	5.30	39.80
215-68106-2(2R) Link Assy	0.50	Hidden by AFCS	7.36	55.25
215-68104-3&-4 B'crk Assy	0.74	Hidden by AFCS	10.89	81.80
CVC542-1098H180 Push Rod	0.26	Hidden by AFCS	2.97	18.70
215-68101-2 B'crk Assy	0.49	3.95	7.21	54.20
CV15-158121-1 Strut Assy	1.23	23.54	20.53	219.00
215-68100-2 B'crk Assy	0.29	4.38	4.26	32.10
215-68020				
CV15-158785-1 Rod Assy	0.45	3.88	4.51	41.00
CV15-158137-3 Rod Assy	1.09	5.88	2.31	78.15
CV15-158597-2 B'crk Assy	3.53	13.69	51.90	390.00
CV15-158761-2 Link Assy	3.10	23.30	47.70	74.20
215-68051-1 Linkage Assy	1.72	11.40	26.50	191.00
215-62100-1 Valve Assy	0.93	6.00	7.06	105.25
CV15-151039-1 Cyl. Assy	6.03	20.39	33.66	
TOTAL	58.83	466.37	601.05	8,411.73

E. SAMPLE CALCULATIONS

1. Determine average length of trunk line with total PC-1 trunk line weight known.

   PC-1 trunk line wt. = 47#

   Assumes trunk lines average 3/4" tubing size

   Pressure line material = Cres 304 Steel; 0.042" wall

   Return line material = 6061-T6 Alum.; 0.065" wall

   Wt/In (Steel) = 0.0265#/In.

   Wt/In (Al) = 0.0139#/In.

   Assume length of press. line = length of ret. line.

   Average (Press. & Ret.) Wt/In = $\frac{0.0265 + 0.0139}{2}$ = 0.0202#/In.

   PC-1 = $\frac{47\#}{0.0202\#/\text{In.}}$ = 2310 in. length

   Area (External) = 2310 in. (0.75) = 1730 in^2

   Vol. (External) = 2310 $(0.75)^2$ = 1025 in^3

2. Determine cost of plumbing:

		3/4"	3/8"
Tubing cost:	3/4" Steel =	\$0.85/Ft.	\$0.54
	3/4" Alum =	\$0.27/Ft.	0.12
	2/	\$1.12/Ft.	2/1 0.66
Average Cost/Ft.	=	\$0.56/Ft.	\$0.33

   Assume 1 bend per 2 Ft.

   Assume 10 min. per bend (including assy. of coupling)

   Assume average length line = 5 Ft.

   Cost per Bend/Ft. = 0.17 (1.5) (\$7.7)/2 = \$0.98

   Cost per Ms 21902-6 Union = \$0.45

   Cost per Ms 21902-12 Union = \$2.25

   Cost per Ms 21921-6 Coupling = \$0.20

   Cost per Ms 21921-12 Coupling = \$0.40

Cost of 5 Ft., 3/4" tubing Assy - 5(0.56) + 1(2.25) + 2(0.4) + 5(0.98) = \$10.75

Cost/Ft. of 3/4" tubing Assy = $\frac{\$10.75}{5 \text{ Ft.}}$ = \$2.38/Ft.

Cost/5 Ft., 3/8" tubing Assy = 5(0.33) + 1(0.45) + 2(0.20) + 5(0.98) = \$7.40

Cost/Ft. of 3/8" tubing Assy = $\frac{\$7.40}{5}$ = \$1.84/Ft.

APPENDIX II

IAP SYSTEM DEFINITION

A. AERODYNAMIC REQUIREMENTS

Aerodynamic information was taken from "Hydraulic System Design", LTV Aerospace Report No. 2-51725/5R-5126. The aerodynamic requirements of static hinge moment and no-load surface rate for each surface have been established by the appropriate aerodynamics group as the input for the hydraulic requirements.

For the Aileron, the requirements were a surface rate of 100°/sec. which corresponds to 3.14 gpm flow. The maximum hinge moment required was 19,200 in-lbs. The corresponding available moment was 15,090 in-lbs. for single system operating condition. The system was marginal for that condition and was used for the IAP as designed for the conventional system.

For the spoiler/deflector, the surface rate requirements was 200°/sec. which corresponded to a flow of 3.90 gpm. The maximum required hinge moment was 8820 in-lbs. compared with an available hinge moment of 11,780. The size of the IAP system was reduced to meet the required hinge moment.

For the UHT, the surface rate required was 25°/sec. which corresponded to 7.87 gpm flow. The maximum required hinge moment was 189,500 in-lbs. as compared with an available hinge moment of 387,500 in-lbs. Again, the size of the IAP system was determined by reducing the conventional system by the hinge moment ratio.

For the rudder, the surface rate was 75°/sec. which corresponded to 2.11 gpm flow. The maximum hinge moment required was 24,200 in-lbs. compared with an available hinge moment of 34,600 in-lbs. This reduction ratio was applied to the conventional system to size the IAP system.

B. ACTUATOR SIZING

The ratio of the available hinge moment to the required hinge moment was used to determine the actuator sizes. This reducing ratio optimized the size of the actuators on a hinge moment basis. The reducing factors calculated and applied are as follows: Aileron 1.0, Spoiler .75, UHG 0.50, Rudder 0.67.

The reducing factors were applied directly to the conventional system power, weight, and volume to determine the corresponding IAP values. However, to determine the projected area, the factors were taken to the 2/3 power and applied to the conventional system. There would be very little reduction in cost due to a reduction in actuator size alone, so the same values for the conventional system were used for the IAP. A tabulation of the reducing factor is shown in Table XX. Gross horsepower is defined as the power resulting from the application of full pressure at full rated flow, as would be exerted in the conventional constant pressure system.

TABLE XX

IAP SYSTEM ACTUATOR SIZING

ACTUATOR REQUIREMENT:

$$= \text{Flow} \times \frac{1}{A} \times \text{Force}$$

$$= \text{gpm}\ (231\ \text{in}^3/\text{gal})\ \left(\frac{1}{60}\ \text{sec/min}\right)\left(\frac{1}{12}\ \text{in/ft}\right) \times \frac{1}{A} \times PA \times \frac{1}{550}\ \frac{\text{ft-lb}}{\text{sec}}$$

$$= \text{gpm}\ (231)\left(\frac{1}{60}\right)\left(\frac{1}{12}\right)(2960)\ \frac{1}{550}$$

$$|HP| = (1.727)(\text{gpm})$$

Aileron - Single System

REDUCING FACTOR		FLOW (gpm)	AREA (in)	STROKE (in)	GROSS POWER/SYST. (H.P.)
1.0					
1.0	up	3.14	1.21	5.0	5.41
	down	2.50	0.96	5.0	4.31

Spoiler - Single System

REDUCING FACTOR		FLOW (gpm)	AREA (in)	STROKE (in)	GROSS POWER/SYST. (H.P.)
8820/1178 0.75		2.92	.907	3.72	5.04

UHT - Single System

REDUCING FACTOR		FLOW (gpm)	AREA (in)	STROKE (in)	GROSS POWER/SYST. (H.P.)
189,500/387,500	Ext.	3.98	3.53	5.71	6.96
	Ret.	3.32	2.98	5.71	5.72

Rudder - Both Systems

REDUCING FACTOR		FLOW (gpm)	AREA (in)	STROKE (in)	GROSS POWER/SYST. (H.P.)
24,200/34,600 0.67		1,415	2.37	2.94	2.44
				Total	20.38

4-55888-21(R1)

C. AFCS-TRIM ACTUATOR RESIZING

The AFCS trim actuators also were resized. For the purpose of determining the power requirements, reduction factors were determined by the ratio of the force required to the force available in each case. The factors were 0.0096 for yaw, 0.0224 for roll, and 0.0257 for pitch. These factors were then used to calculate the power required for the AFCS units.

For the purpose of determining weight and volume, the above described factors become unrealistic because the size reduction is limited. Consequently, an estimated size reduction of 50% was used. Projected area was estimated by taking the volume to the 2/3 power. A summary of the actuator parameters is:

	Actuator		
	Yaw	Roll	Pitch
Maximum Required Output Force (lbs.)	15	35	40
Current Available Output Force (lbs.)	1560	1560	1560
Current Available Flow Rate (gpm)	0.425	0.687	0.429

D. HYDRAULIC POWER REQUIREMENTS

The power required to be delivered to the actuators has been described above. The method of delivering this power will now be discussed. To deliver full flow at maximum pressure would require an excessive amount of wasted power. To take advantage of a reduced pressure requirement at higher flow rates, a pump was defined for which the pressure developed droops as the flow rate increases (Figure 8). Maximum hydraulic horsepower of this type of system is only about 33% of that for a conventional constant pressure unit. Based on duty cycle requirements described in Section III, the duty cycle horsepower requirements are established in Table XXI.

4-55888-21(R1)

TABLE XXI

IAP SYSTEM PUMP HP REQUIREMENTS

SYSTEM	Surface Total Gross HP	Peak Pump HP (1/3 gross HP) Required	Duty Cycle HP Required At Surface
Aileron	5.417	1.806	0.813
Spoiler/Deflector	5.04	1.68	0.756
UHT	7.004	2.334	2.03
Rudder	2.48	0.827	0.533

5888-21(R1)

E. MOTOR SIZING

For sizing the individual drive motors, 95% hydraulic pumping efficiency was assumed. Valve pressure drop losses are included in the hydraulic power requirements; however, these are quite small when operating under load due to the large orifices associated with the soft cut-off type system.

The motor size and weight are extrapolated from the ABEX P/N PQ8573A-2 unit as a reference.

The following is a tabulation of the motor sizing details of each of the subsystems:

	Pump Output Horsepower Cont. Duty	Motor Output Horsepower Cont. Duty	Motor Wt-Lbs	Dia. (in)	Length (in)	Vol. (in^3)	Area (in^2)	Cost ($)
Aileron	0.813	0.856	7.8	5.0	5.4	106	27	540
Spoiler	0.756	0.796	7.5	5.0	5.2	100	26	520
UHT	2.03	2.14	15.0	5.0	6.0	118	30	600
Rudder	0.533	0.561	6.1	5.0	4.1	80	21	410

The motor for the UHT was sized directly from the ABEX unit and the remainder of the motors were sized in accordance with the following assumed relationships.

Weight, W, horsepower

Length, L, $(\text{horsepower})^{2/3}$

Diameter, D, = constant 5.0 in.

Area = DXL

Cost $\sim$ $(\text{horsepower})^{2/3}$

F. HYDRAULIC PUMP SIZING

The hydraulic pump considered for the integrated system is a standard 3000 psi aircraft type variable displacement unit with a soft cut-off characteristic. The soft cut-off feature can be achieved simply by using a stiff spring restraint on the standard pump compensator mechanism. Although several types of variable displacement pumps can be used satisfactorily, the sizing was based on a Vicker pump selection guide.

4-55888-21(R1)

The pumps were sized in accordance with the following assumed relationships:

Weight $\sim$ (flow) from Vickers' selection guide

Diameter = constant 3.5" in desired range

Length $\sim$ weight

Cost $\sim$ weight

The following is a tabulation of the pump sizing details:

	Max. Flow (gpm)	Wt. (#)	Dia. (in)	Length (in)	Volume $(in)^3$	Area $(in)^2$	Cost ($)
Aileron	3.14	3.8	3.5	3.8	36	14.0	570
Spoiler	2.92	3.5	3.5	3.5	34	12.0	525
UHT	3.98	4.0	3.5	4.0	38	14.0	600
Rudder	1.42	3.0	3.5	3.0	29	10.5	450

G. INTEGRATED HYDRAULIC CIRCUIT

The integrated hydraulic system, although actually packaged together with pump, motor, and actuator-valve package, was considered separately to avoid interface complications which would require considerable design detail. The system portion of ABEX P/N PQ8573A-2 integrated power supply was used for basic sizing. This unit contains a 25 in^3 reservoir with fill, bleed and drain valves, relief valve, filter with by-pass valve, and mechanical reset indicator. This package is considered satisfactory for all A7 systems except the UHT, which will require approximately 50 in^3 reservoir size. The following is a tabulation of the sizing details for the hydraulic system portion of the integrated package:

	Wt. (#)	Dia. (in)	Length (in)	Volume (in^3)	Area (in^2)	Cost ($)
Aileron	6	6	5	140	30	470
Spoiler	6	6	5	140	30	470
UHT	7	6	6	168	36	470
Rudder	6	6	5	140	30	470

4-55886-21(R1)

H. CONSTANT SPEED DRIVE & ELECTRICAL POWER GENERATION SYSTEM

Continuous duty power requirements for each generator system is determined as follows:

$$P = \frac{\text{(hydraulic power required)}}{\text{(pump eff.)(motor eff.)(transmission eff.)}}$$

Hydraulic power required for each channel is:

Aileron X2	=	1.626 h.p.
Spoiler X2	=	1.512
UHT X2	=	4.060
Rudder	=	.533
Total		7.731 h.p.

$$P = \frac{7.731}{(.95)(.85)(.98)}$$

P = 9.8 h.p. = 7.3 KVA

Sizing of the CSD unit was based on the A-7 weight data handbook and Sundstrand product catalog for a 17 h.p. 10KVA unit. The basic sizing data for the CSD is as follows:

Weight = 45 lbs.
Length = 13.7 in.
Dia. = 10.25 in.
Volume = 1130 in^3
Area = 141 in^2
Cost = $9300

Cost was evaluated by weight ratio from the present A-7 counterpart.

The 10 KVA generator was sized from the A-7 weight data handbook with physical dimensions and cost evaluated by weight ratio from the present A-7 counterpart. The sizing data is as follows:

Weight = 35 lbs.
Length = 7.5 in.
Dia. = 8 in.
Volume = 380 in^3
Area = 60 in^2
Cost = $875

I. TRUNK LINE CIRCUIT COMPONENTS

The various circuit components were sized directly from present A-7 counterparts. The various components are discussed below:

1. Line Contractors

This electro-magnetic device is required to bring each generator on the main line bus. It is controlled by the generator control circuit box. Sizing is as follows:

Envelope - 4" dia. X 2" thickness
Est. Weight = 5 lbs.
Est. Cost = $25

2. Generator Control Box

The control box contains circuitry to control the generator power thru functions of voltage and frequency to permit or deny access of generator power to the main bus through the main line contractor. Sizing is as follows:

Envelope - 3" X 4" X 6" package
Est. Weight = 5 lbs.
Est. Cost = $350

3. Differential Current Transformers

There are six transformers required for each generator circuit, on each per phase for both power and ground legs. These transformers function as generator differential current monitors which supply information to the generator control circuit. These units are sized as follows from the A-7 system.

Envelope - 2" X 2" x 1/2" package
Est. Weight = 1/4 lb.
Est. Cost = $5

J. ELECTRICAL DISTRIBUTION SYSTEM

The wire length was established to each unit from the control panel. The length between stations was determined and the vertical and horizontal distance estimated to each unit. The same method was used to determine the line length from the generator to the control panel.

One volt drop from the control panel to each unit was assumed. Also, a one volt drop was assumed from the generator to the control panel. Having established the power required and selected an electrical system of 115V 3 phase 400 cps, the current in each line was calculated. From the current and the assumed one volt drop on each line, the resistance of each line was established. A line size to each IAP unit and from the generator to the control panel was selected. Wire weight, area, and volume were calculated. Cost of the wiring was calculated at $3 per pound.

4-55888-21(R1)

1. Wire Lengths

Electrical Line Length from Control Panel to Units

	To Side Of Fuselage	Station Length Along Fuselage	Vertical Along Fuselage	To Unit From Fuselage	Total	Total plus 20% for Routing
Aileron	24	478-330 = 148	60	180	412	494
Spoiler	24	478-330 = 148	60	75	307	368
UHT	24	640-330 = 310	25	0	359	430
Rudder	24	653-330 = 323	50	40	437	524

Electrical Line Length from Generator to Control Panel

Station Length	Plus 20% for Routing
552-330 = 222	266

2. Wire Sizing

Aileron

Current per phase $= (1/3)\ \dfrac{1.27\ \text{KVA}\ (1000)}{115\ \text{V}} = 3.68$ Amps.

Allowable resistance per ft. $= \dfrac{10}{3.68\ (494/12)} = \dfrac{1}{151}\ \dfrac{\text{ohm}}{\text{ft}}$

Ft/Ohm = 151 Use 156.6 or size A.W.G. 18

Wt. = 203.4 ft/lb. D = .040 in.

Spoiler

Current per phase $= (1/3)\ \dfrac{(1.18)(1000)}{115\text{V}} = 3.42$ amps.

Allowable resistance per ft. $= \dfrac{1.0}{3.42\ (368/12)} = \dfrac{1}{105}\ \dfrac{\text{ohm}}{\text{ft.}}$

Ft/Ohm = 105 Use 98.5 or size A.W.G. 20

Wt. = 323.4 ft/lb. D = .032 in.

4-55686-21(R1)

UHT

Current per phase = $(1/3)\ \dfrac{3.18\ (1000)}{115V}$ = 9.2 amps

Allowable resistance per ft. = $\dfrac{1.0}{9.2\ (430/12)} = \dfrac{1}{330}\ \dfrac{\text{ohm}}{\text{ft.}}$

Ft/ohm = 330 Use 314.0 or size A.W.G. 15

Wt. = 63.8 ft/lb D = .072 in.

Rudder

Current per phase = $(1/3)\ \dfrac{(.79)(1000)}{115V}$ = 2.29 amps

Allowable resistance per ft. = $\dfrac{1.0}{(2.29)(524/12)} = \dfrac{1}{100}\ \dfrac{\text{ohm}}{\text{ft}}$

Ft/Ohm = 100 Use 98.5 or size A.W.G. 20

Wt. = 323.4 ft/lb. D = .032 in.

Generator to Control Panel

Current per phase = $(1/3)\ \dfrac{10(1000)}{115}$ = 28.9 amps

Allowable resistance per ft. = $\dfrac{1.0}{28.9\ (266/12)} = \dfrac{1}{641}\ \dfrac{\text{ohm}}{\text{ft.}}$

Ft/Ohm = 641 Use 629.6 or size A.W.G. 12

Wt. = 50.6 ft/lb D = .081 in.

3. Total Wire Length/System

	Distance	3 Phase	Total
Aileron	494	1482	2964 (2 sides)
Spoiler	368	1104	2208 (2 sides)
UHT	430	1290	2580 (2 sides)
Rudder	524	1572	1572 (one unit)
Generator	266	798	798 (one unit)

4-55888-21(R1)

4. Wire Weight

		25% Insulation	Total
Aileron	2964 in $(\frac{1}{12}$ ft/in$)(\frac{\#}{203.4\ ft})$ = 1.22#	0 .31	1.53 lbs.
Spoiler	2208 in $(\frac{1}{12}$ ft/in$)(\frac{\#}{323.4\ ft.})$ = .57#	0 .14	0.71
UHT	2580 in $(\frac{1}{12}$ ft/in$)(\frac{\#}{63.8\ ft.})$ = 3.36#	0 .84	4.20
Rudder	1572 in $(\frac{1}{12}$ ft/in$)(\frac{\#}{323.4\ ft.})$ = .41#	0 .10	.51
Generator	798 in $(\frac{1}{12}$ ft/in$)(\frac{\#}{50.6\ ft.})$ = 1.32#	0 .44	1.76
		Total	8.71 lbs/sys.

5. Wire Area

		50% for Insulation	Dia.	Area (Dia. XL)
Aileron	D = 0.04	0 .02	0 .06	178 in^2
Spoiler	D = 0.032	0 .016	0 .048	106 in^2
UHT	D = 0.072	0 .036	0 .108	279 in^2
Rudder	D = 0.032	0 .016	0 .048	75
Generator	D = 0.081	0 .040	0.121	97
			Total	735 in^2/sys

6. Wire Volume

			Volume (A X L)
Aileron	DIA = 0.06	A = 0.00283	8.4 in^3
Spoiler	DIA = 0.048	A = 0.00181	4.0
UHT	DIA = 0.108	A = 0.00916	23.6
Rudder	DIA = 0.048	A = 0.00181	2.8
Generator	DIA = 0.121	A = 0.0115	9.2
		Total	48.0 in^3/sys.

4-55888-21(R1)

K. SIGNAL LINKAGE

1. General

The control linkage from the pilot to each surface is essentially the same as the conventional counterpart with the exception of the transfer of all AFCS electro-hydraulic actuators from just aft of the feel spring to the IAP's. One other major exception is elimination of the FIA (Feel Isolation Actuator) of the conventional aileron - spoiler linkage in favor of a "dwell" linkage for each spoiler, preventing motion and force feedback from the "down" spoiler as the other is actuated. The aileron trim function was also considered transferred from the conventional electro-mechanical servo to the aileron IAP. The spoiler IAP contains no trim or stabilization function. In the case of the longitudinal (UHT) system, the E/M trim function was not altered as it is a parallel trim system (stick motion with trim) which actuates the linkage feel spring ground point and must remain at the feel spring location.

2. Aileron Spoiler Signal Linkage

The net result in linkage re-arrangement is tabulated below:

	Wt. (#)	Vol. (in^3)	Area (in^2)	Cost ($)
Add dwell linkage (2 required)	6.0	60	80	250
Add pushrod to replace AFCS Supporting Linkage	0.1	10	12	1
Subtract E/M Trim Act & Linkage	-2.6	-30	-34	-488
Net for IAP Linkage	+3.5	+40	+58	-$237

3. UHT Signal Linkage

The linkage removed from the conventional system AFCS package was considered equivalent to the added linkage required to sum pilot and AFCS commands in the IAP, therefore, no significant change affecting evaluating parameters was considered for the IAP system.

4. Rudder Signal Linkage

The AFCS linkage removed was considered equivalent to that added in the IAP, as in the UHT system, resulting in no significant change in parameters.

L. PARAMETER SUMMATION

The previous paragraphs have delineated the addition and deletion of components based on the conventional system and established the parameters for the additional components added. The parameters of the components added are defined in Appendix I. Table XXII lists the components added and their parameters. Table XXIII lists the components subtracted.

4-55588-21(R1)

TABLE XXII

COMPONENTS ADDED TO THE CONVENTIONAL SYSTEM

UNIT	ITEM	WEIGHT	AREA	VOLUME	COST
Aileron (2 sets)	Circuit Breaker	3.00	24.0	24.0	$ 60.00
	Wiring	5.04	356.0	16.8	16.00
	Motors	31.20	108.0	424.0	2160.00
		39.24	338.0	514.8	2236.00
Spoiler (2 sets)	Circuit Breaker	3.00	24.0	24.0	60.00
	Wiring	1.44	212.0	8.0	6.00
	Motors	30.00	104.0	400.0	2080.00
		34.44	340.0	432.0	2146.00
UHT (2 sets)	Circuit Breaker	3.00	24.0	24.0	60.00
	Wiring	8.40	560.0	47.2	26.00
	Motors	60.00	120.0	472.0	2400.00
		71.4	714.0	553.2	2486.00
Rudder	Circuit Breaker	1.56	12.0	12.0	30.00
	Wiring	0.52	76.0	2.8	2.00
	Motors	12.20	42.0	160.0	820.00
		14.28	130	174.8	852.00
Power System	Generator	70.0	120.0	760.0	1750.00
	CSD	90.0	282.0	2260.0	18600.00
	Line Contactor	10.0	16.0	50.0	50.00
	Gen. Control Panel	10.0	48.0	240.0	700.00
	Current Transformers	3.0	36.0	24.0	60.00
	Trunk Line Wiring	1.76	94.0	9.2	5.00
	Master Gen. Switch	1.0	4.0	4.0	10.00
	Master Gen. Indicator	1.0	2.0	2.0	10.00
		186.8	602.0	3349.0	21185.00
Aileron IAP (2 sets)	Pumps	15.2	56.0	144.0	2280.00
	Hydraulic Pack	24.0	120.0	560.0	1880.00
	Shut Off Valve	2.0	8.0	16.0	300.00

4-55888-21(R1)

TABLE XXII

COMPONENTS ADDED TO THE CONVENTIONAL SYSTEM (CONTINUED)

UNIT		ITEM	WEIGHT	AREA	VOLUME	COST
Aileron IAP	(2)	AFCS Pack	16.0	34.0	22.0	$ 8022.00
		Actuator Pack	23.5	146.94	173.1	2662.00
			80.7	365.0	915.0	15144.00
Spoiler IAP	(2)	Pumps	14.0	48.0	136.0	2190.00
		Hydraulic Pack	24.0	120.0	560.0	1880.00
		Actuator Pack	16.8	120.0	130.0	2544.00
			54.8	288.0	826.0	6524.00
UHT IAP	(2)	Pumps	16.0	56.0	152.0	2400.00
		Hydraulic Pack	28.0	144.0	672.0	1880.00
		Shut Off Valve	2.0	8.0	16.0	300.00
		AFCS Pack	16.0	34.0	22.0	8022.00
		Actuator Pack	32.8	72.0	202.0	7390.00
			94.8	157.0	1064.0	19992.00
Rudder IAP		Pump	6.0	21.0	58.0	900.00
		Hydraulic Pack	12.0	60.0	280.0	940.00
		Shut Off Valve	1.0	4.0	8.0	150.00
		AFCS Pack	8.0	17.0	11.0	4011.00
		Actuator Pack	6.0	20.3	27.3	685.00
			33.0	72.0	384.3	6686.00
Linkage		1" Dia. Push Rod	0.10	12.0	10.0	1.00
		2 Dwell Packs	6.00	80.0	60.0	250.00
			6.1	92.0	70.0	251.00
TOTAL			615.54	3216.3	8223.1	$ 77502.00

4-55858-21(R1)

TABLE XXIII

COMPONENTS SUBTRACTED FROM CONVENTIONAL SYSTEM

UNIT	ITEM	WEIGHT	AREA	VOLUME	COST
Hydraulic	PC_1 & PC_2	191.98	3902.0	3225.0	$ 15508.00
	Plumbing	53.37	2647.0	784.0	10436.00
	FIA Cyl. Assy.	6.51	Shielded	32.0	738.00
	Aileron Act. (2)	23.50	147.0	173.0	2663.00
	Spoiler Act. (2)	22.46	147.0	173.0	2545.00
	Roll AFCS Act.	15.73	27.0	22.0	4011.00
	Yaw AFCS Act.	17.00	48.0	102.0	4011.00
	Pitch AFCS Act.	17.00	46.0	102.0	4011.00
	Rudder Valve Assy.	0.93	6.0	17.0	105.00
	Rudder Actuator	6.03	20.0	34.0	685.00
	UHT Cyl. Assy. (2)	65.16	115.0	404.0	7390.00
		419.87	7105.0	5068.0	52103.00
Linkage	Roll Trim Act.	2.6	34.0	30.0	488.00
TOTAL		422.47	7139.0	5098.0	$ 52581.00

4-55888-21(R1)

13. AIRCRAFT C.G. SHIFT

The weight and center of gravity of the airplane were used as the starting point for determining the C.G. shift. The airplane weight of 16,122 lbs., moment of 7,454,600 in-lbs., and C.G. at station of 462.38 were taken from "Actual Weight and Balance" Report No. 2-59330/8R-5350. In describing the IAP system, a number of components were added which had certain weights and station locations in the airplane. These components are tabulated in Table XXII. A number of the conventional system components had to be removed. These components are tabulated in Table XXIII. After subtracting the weight of the removed components from the weight of the added components, the result was added to the total weight of the airplane. The same procedure was used to evaluate the increased moment on the airplane. The tabulation below indicates a 1.64 in. shift in C.G. in the aft direction.

Airplane Weight		16,122.00#
Airplane C.G.		462.38 in.
Airplane Moment		7,454,660.00 in#
Weight Added		615.54
Weight Removed		422.77
	Net Added	192.77
Moment Added		342,010.6
Moment Removed		226,253.
	Added	115,757.6

New Weight $16,122 + 192.77 = 16,314.77$

New Moment $7,454,660 + 115,757.6 = 7,570,417.6$

New C.G. $7,570,417.6/16,314.77 = 464.02$

4-55888-21(R1)

APPENDIX III

SURVIVABILITY ANALYSIS

A. INTRODUCTION

The object of this analysis is to determine the relative survivabilities of the conventional and the integrated actuator package hydraulic systems and their associated linkage systems. Survivability analysis for each system requires as input:

1. Number of hits in system

2. The probability of hitting and killing each subsystem

3. System kill criteria

B. NUMBER OF HITS IN SYSTEM

For comparison purposes, the number of hits on each system for a given number of hits on the aircraft can be computed as follows:

$$\text{Number of hits on system} = (\text{number of hits on aircraft}) \cdot \frac{\text{vulnerable area of system}}{\text{total aircraft presented area}}$$

The vulnerable areas of each of the systems in the horizontal plane are tabulated as follows:

System	Area
Total aircraft area	82000 in^2
Conventional hydraulic system	7106.97 in^2
IAP Power System	3078.0 in^2
Linkage - Conventional	2194.24 in^2
Linkage - IAP	2252.36 in^2

4-55888-21(R1)

To determine the minimum number of hits on the aircraft required to kill the system with the smallest vulnerable area, the assumption is made that the hits received by the aircraft are uniformly distributed over its presented area in the horizontal plane. The required number of hits is then determined by solving the above equation, given the number of hits in the system. The system with the smallest vulnerable area is the Linkage - Conventional, and its kill criteria specifies that only one hit, if properly located, is required for a kill. Therefore:

$$1 \text{ system hit} = (\text{"R" hits on aircraft}) \; \frac{2194.24 \text{ in}^2}{82000 \text{ in}^2}$$

$$\text{"R"} = 37.5 \text{ hits on aircraft (minimum)}$$

This value is rounded to 40 hits.

Then, for comparison purposes, the ratio of hits on the four system for a given number of hits on the aircraft is:

System	No. of Hits
Conventional	3.5
IAP	1.5
Linkage - Conventional	1
Linkage - IAP	1

For 80 rounds hitting the aircraft, seven would hit the conventional hydraulic system, three would hit the IAP, and two rounds would hit both the Conventional and IAP linkages. Since 80 rounds result in integer hits being received by each system, this number is used in comparisons.

C. PROBABILITY OF SUBSYSTEM KILL

The probability of killing a subsystem is a combination of the probability of hitting the subsystem and the probability of killing the subsystem after having hit it. The probability of hit in each subsystem is obtained by taking area to the total system vulnerable area. Since the number of hits in the system is based on the system vulnerable area, this method is valid. In equation form:

$$P_{HIT} = \text{Probability of subsystem hit} = \frac{\text{subsystem vul. area}}{\text{total system vul. area}}$$

4-55888-21(R1)

The probability of subsystem kill given a subsystem hit (conditional probability) is assigned and takes into account the fact that while many subsystems may be damaged by a hit, they would not necessarily be made inoperable. Worst case analysis would set the conditional probability equal to 1.

$$P_{KH} = \text{Probability of subsystem kill given a hit} = \text{assigned}$$

The two probabilities for each subsystem are combined using the multiplication law of probability and result in the subsystem probability of kill.

$$P = P_{HIT} * P_{KH}$$

D. SYSTEM KILL CRITERIA

Kill criteria reflects the necessary combinations of subsystems needed to kill a system and thus, the aircraft. Each of the systems evaluated are composed of groups of subsystems arranged in series and/or parallel combinations. Series combinations require that all units must be operating in order that the total system functions. In parallel combinations, at least one of the units must be operating.

E. COMPUTATION OF PROBABILITY OF SURVIVAL

For a kill criteria specifying that only one subsystem of a system made up of N subsystems need be killed to kill the aircraft, the probability of survival is calculated directly using the binomial expansion. For the i TH subsystem, the probability of subsystem kill is;

$$P_K\ (i) = \frac{R!}{(R-1)!}\ P\ (i)\ \left[1 - P\ (i)\right]^{R-1}$$

and the probability of survival is;

$$P_S\ (i) = 1 - P_K\ (i)$$

where;

R = number of rounds hitting system

$P\ (i)$ = probability of hitting and killing the i TH subsystem

For $R = 1$, the computation reduces to:

$$P_K\ (i) = P\ (i)$$

Calculating probability of survival for a kill criteria which requires that two or more subsystems of N subsystems must be killed simultaneously in order that the system be killed is accomplished by using an extension of the binomial expansion. As an example of the computational procedure, consider the case:

N = number of subsystems = 4

R = number of rounds on system = 3

kill criteria = kill systems 1 and 2 simultaneously

A list of possible combinations of three rounds hitting four subsystems is shown in the following table. Combinations of rounds which result in aircraft kill are noted.

	SUBSYSTEM NUMBER				
	1	2	3	4	
Number of Hits	3	0	0	0	
	2	1	0	0	system kill (K_1)
	2	0	1	0	
	2	0	0	1	
	0	3	0	0	
	0	2	1	0	
	0	2	0	1	
	1	2	0	0	system kill (K_2)
	0	0	3	0	
	0	0	2	1	
	1	0	2	0	
	0	1	2	0	
	0	0	0	3	
	1	0	0	2	
	0	1	0	2	
	0	0	1	2	
	1	1	1	0	system kill (K_3)
	1	0	1	1	
	0	1	1	1	
	1	1	0	1	system kill (K_4)

4-55888-21(R1)

For the first kill combination, system kill results if at least one of two rounds hits subsystem 1, and at least one of one round hits subsystem 2, or

$$K_1 = P\,(X_1 \geqq 1)\ P\,(X_2 \geqq 1)$$

Using the binomial expansion and combining via the multiplication law

$$K_1 = \left\{ \sum_{i=1}^{2} \binom{2}{i} P_1^{i} \left[1-P_1\right]^{2-i} \right\} \cdot \left\{ \sum_{j=1}^{1} \binom{1}{j} P_2 \left[1-P_2\right]^{1-j} \right\}$$

and reducing

$$K_1 = \left\{ 2P_1^{\ 1} \left[1-P_1\right]^{1} + P_1^{\ 2} \left[1-P_1\right]^{0} \right\} \cdot \left\{ P_2^{\ 1} \left[1-P_2\right]^{0} \right\}$$

All cases which result in aircraft kill are analyzed in the same manner.

Making the assumption that each of the round assignment combinations is equally likely to occur, the probability of killing the system for this particular kill criteria becomes a weighted average of K_1, K_2, K_3, and K_4. Thus;

$$P_K \text{ (subsystems 1 and 2)} = \frac{1}{NC} \Big[K_1\ (2,\ 1) + K_2\ (1,\ 2) + K_3\ (1,\ 1) + K_4\ (1,\ 1) \Big]$$

where K (M, n) denotes the kill of (m) rounds hitting subsystem 1 and (n) rounds hitting subsystem 2. NC is the total number of round combinations possible,

$$NC = \binom{R + N - 1}{R} = \frac{(R + N - 1)!}{R!\ (N - 1)!} = 20.$$

The expression for P_K (1 and 2) can be further manipulated to ease computations.

$$P_K \text{ (subsystems 1 and 2)} = \frac{1}{NC} \sum_{I_1 = 1}^{R - 1} \sum_{I_2 = 1}^{R - I_1} \binom{I_X + S-1}{S - 1} K\,(I_1,\ I_2)$$

$$= \frac{1}{\binom{R + N-1}{R}} \sum_{I_1 = 1}^{R - 1} \sum_{I_2 = 1}^{R - I_1} \binom{I_X + S-1}{S-1} \left[\sum_{=1}^{I_1} \binom{I_1}{i} \cdot P_1 \cdot Q_1^{I_1 - i} \right] \left[\sum_{=1}^{I_2} \binom{I_2}{j} \cdot P_2 \cdot Q_2^{I_2 - j} \right]$$

4—55868—21 (R1)

$$P_K = \frac{1}{\binom{R+N-1}{R}} \sum_{I_1=1}^{R-1} \sum_{I_2=1}^{R-I_1} \binom{I_X + S-1}{S-1} \cdot \left[1 - Q_1^{I_1}\right] \cdot \left[1 - Q_2^{I_2}\right]$$

where

$Q = (1 - P)$

$I_X = R - I_1 - I_2$

S = remaining number of components on which $(R - I_1 - I_2)$ rounds are spent

The probability of survival for kill criteria consisting of more than two subsystems can be calculated by expanding the preceeding equations. For the probability of survival of N out of T subsystems, the kill probability expression is

P_K (subsystems 1, 2, 3 ---N)

$$= \frac{1}{\binom{R+N-1}{R}} \sum_{I_1=1}^{I_X} \sum_{I_2=1}^{I_X} \sum_{I_3=1}^{I_X} \cdots \sum_{I_N=1}^{I_X} \binom{I_X + S-1}{S-1} \left[1 - Q_1^{I_1}\right] \left[1 - Q_2^{I_2}\right] \cdots \left[1 - Q_N^{I_N}\right]$$

where

$I_X = R - I_1 - I_2 - \cdots - I_N$

The probability of survival is then

$P_S = 1 - P_K$ (subsystems 1, 2, 3 --- N)

After each probability of kill has been computed for every kill criteria, the total system probability of survival is obtained through application of the probability law of addition for events which are not mutually exclusive. For a system with three kill criteria, for example, the total probability of kill is

$$\text{Probability}\left[A \text{ and/or } B \text{ and/or } C\right] = P(A) + P(B) + P(C) - P(AB) - P(AC) - P(BC) + P(ABC)$$

4-55888-21(R1)

A simplier method of obtaining the same result is;

$$\text{Probability}\left[\text{A and/or B and/or C}\right]$$
$$= 1 - \left[1-P(A)\right]\cdot\left[1 - P(B)\right]\cdot\left[1 - P(C)\right]$$

The probability of system survival with three kill criteria is then

$$P_S = 1 - \text{Probability}\left[\text{A and/or B and/or C}\right]$$
$$= 1 - \left\{1 - \left[1 - P(A)\right]\cdot\left[1 - P(B)\right]\cdot\left[1 - P(C)\right]\right\}$$
$$= \left[1 - P(A)\right]\cdot\left[1 - P(B)\right]\cdot\left[1 - P(C)\right]$$
$$P_S = \left[P_S(A)\right]\cdot\left[P_S(B)\right]\cdot\left[P_S(C)\right]$$

Therefore, total aircraft survivability is determined by calculating the separate probabilities of system survival for each given kill criteria and multiplying all together.

F. SURVIVABILITY SENSITIVITY ANALYSIS

A sensitivity analysis was performed to determine the effect variations in the number of hits on the aircraft would have on probability of survival for each of the four systems considered. The number of rounds hitting the aircraft was varied from 0 to 200 and the probability of survival for each system determined. Results are shown in Figure 19. All systems are sensitive to the number of rounds hitting the aircraft; however, the same relative order of survivability was maintained for the four systems.

G. DIGITAL COMPUTER PROGRAM DESCRIPTION

A digital program was written to assist in the computation of system survivabilities. The program, written in Basic Fortran IV, was run on an IBM 360/40 Computer through the System/360 Remote Access Computing System. It requires 9456 bytes of core location and determines the probability of survival of a system for a given number of hits in approximately one second of computer time.

After reading in required data, the program first calculates factorial values from 1 through 50 for later use. It then determines the probabilities of kill and survival for each kill criteria specified and makes a progressive computation of total system survivability. The program is able to handle kill criteria of from order one to order eight. The number of system hits can be varied from 1 to 16.

4-65888-21(R1)

Input required by the program in the correct order is as follows:

TITLE - descriptive heading

K - number of vulnerable subsystems

P(K) - kill probability for each subsystem

N - total number of subsystems

R - number of hits on the system

Repeated for each order of kill criteria	KC	-	order of kill criteria
	NCNT	-	number of kill criteria for each order
	S, T, U	---	subsystems involved in kill criteria

Output, all in printed form, consists of:

descriptive page heading

number of subsystems considered

number of rounds hitting system

the specified kill criteria

the system probability of kill for that particular kill criteria

the system probability of survival for that particular kill criteria

total progressive system probability of survival(the final value is the system probability of survival for the given number of rounds)

Samples of input and output and a listing of the entire program can be provided by FDCL, Air Force Flight Dynamics Laboratory.

4-55888-21(R1)

APPENDIX IV

SIMPLEX RELIABILITY ANALYSIS

A. INTRODUCTION

The reliability study described in this report was made to establish quantitative reliability predictions for the Simplex Actuator Package designed and manufactured as a part of the AFCS Integrated Actuator R & D Program. The reliability analysis was conducted to establish Mean Time Between Failure (MTBF) and Probability of Mission Success (R) for all operating modes.

B. SUMMARY

Table XXIV presents the results of reliability analyses performed on the predicted failure rates (λ), mean time between failures (MTBF), and reliability (R, probability of successful mission) as shown for each mode of operation, as defined in paragraph 3.0. Mission time (t) is defined as two hours.

C. DESCRIPTION OF EQUIPMENT

The reliability analysis contained herein pertains to the hydraulic power servo actuator package (Simplex Actuator, P/N 401-13850) capable of the following four modes of operation:

(1) Manual operation similar to that of conventional hydraulic power control units.

(2) Operation equivalent to that of a normal series mode servo plus power cylinder.

(3) Operation equivalent to that of a normal parallel mode servo plus power cylinder.

(4) Manual operation with an integrated emergency power supply providing the required hydraulic power.

1. Manual Operation (Mode 1)

A tandem servo valve, positioned directly by pilot input through a mechanical linkage, directs hydraulic supply pressure to a tandem power cylinder for extend and retract operations. Hydraulic fluid, provided by three (3) independent power suppliers (P_1, P_2, P_E), is ported through the unit for single or dual system operation. Hydraulic power supplies P_1 and P_2 are provided by the "airplane" hydraulic systems. Hydraulic power supply P_E is provided by an emergency system integrated into the simplex actuator package.

2. Series Operation (Mode 2)

Hydraulic power from P_1 supply is routed through a control solenoid valve to an electrohydraulic servo valve. The electrohydraulic servo valve, controlled by SAS system input signals, supplies hydraulic pressure to position an auxiliary ram. The auxiliary ram, through a mechanical linkage, positions the tandem servo valve directing hydraulic power to the power cylinder for extend and retract operations. Pilot inputs and auxiliary ram outputs are summed through the mechanical linkage for tandem servo valve control. In this mode hydraulic pressure is routed through the parallel mode solenoid control valve to the authority control pistons of the auxiliary ram to limit auxiliary ram authority over the tandem servo valve. A sequence valve piloted by P_1 pressure from the series mode control valve ports the outputs of the electrohydraulic servo valve to the auxiliary ram. Loss of P_1 pressure or de-energization of the control solenoid shuttles the spring loaded sequence valve which ports both sides of the auxiliary ram piston to return. This action allows auxiliary ram to be quickly returned to neutral.

3. Parallel Operation (Mode 3)

Parallel mode of operation is accomplished in the same manner as series mode of operation, except the parallel control solenoid is also energized. This action results in directing hydraulic supply pressure to the authority control piston on the pilot input linkage and ports the auxiliary ram authority control pistons to return. This allows the auxiliary ram to have full authority over the tandem servo valve. Pilot input is limited to override authority.

4. Manual Operation, Emergency Power (Mode 4)

Control of the tandem servo valve and power cylinder are accomplished in the same manner as normal manual operation except hydraulic power is provided by the integrated emergency system. Pressure from P_1 supply is routed through a control solenoid to a switching valve. Loss of P_1 pressure or energization of the solenoid valve causes the spring load, P_E pressure assisted, switching valve to shuttle. This action blocks the P_1 supply and allows pressure from P_E supply to be routed to the tandem servo valve for power cylinder operations.

The emergency power system is an integrated motor/pump and reservoir with associated filters, pressure relief valves, check valves and fill provisions necessary for a self-contained power supply.

D. RELIABILITY DESIGN ANALYSIS

1. Reliability Series Block Diagrams

Presented in Figures 75,76,77 and 78 are the series block diagrams for each mode of operation. Total failure rate (λ) in failures per million operating hours is shown for each item along with the part name, number and quantity of each item required for each mode of operation. A mission item, (t), of 2.0 hours is used in computing the reliability from the block diagram equation. The total equipment failure rate (λ) for each mode is indicated below each diagram.

2. Math Models

a. Basic Reliability Model

To determine the reliability (R) of the Simplex Actuator Package, the components are assumed to be regularly maintained and operated during their useful life. The basic reliability equation is

$$R = e^{-\lambda t}$$

where

e = Base of Natural Logarithms

λ = Failures per Flight Hour

t = Mission Time in Hours

b. Series System Model

The reliability of components in series can be calculated using the "Product Law of Reliabilities;", i.e.,

$$R = R_1 \cdot R_2 \cdot R_3 \cdot \cdots \cdot R_N$$

or $$R = e^{-\lambda_1 t} \cdot e^{-\lambda_2 t} \cdot \cdots \cdot e^{-\lambda_N t};$$

therefore, $$R = e^{-\sum_{i=1}^{N} \lambda_i (t)}$$

and $$\lambda = \sum_{i=1}^{N} \lambda_i$$

This analysis considered all parts in series; therefore, the above equation applies to each mode of operation.

c. Mean Time Between Failure (MTBF) Equation

The calculated MTBF values were derived by the equation

$$MTBF = \frac{1}{\text{Failure Rate}}$$

which considers only chance failures. By this assumption, it is implied that "early" and "wearout" failures are eliminated by good debugging and maintenance practices.

3. Failure Rate Assessment

Piece parts of the Simplex Actuator Package have been listed in Table XXV which shows part number, name, and failure rate assigned. The failure rate data was derated based on equipment usage and failure experience gained from similar products produced by LTV-E. The products include:

(a) Titan III Servoinjectors (1,264 units)

(b) Minuteman Servo Actuators (22,000 units)

(c) Boeing 727 Feel Systems Computers and Actuators (583 units)

(d) F-8 and A-7 APC Throttle Servo Actuator (1,000 units)

(e) F-8 and A-7 Aircraft Stabilization System Actuators (1,000 units)

(f) Boeing 737 and 747 Feel Computers

(g) Boeing 747 Stabilization Actuator

(h) Boeing 747 Rudder Hydraulic Brake

a. Failure Rate Sources

The following failure rate data sources were used for failure rate assignment:

- FARADA, Bureau of Naval Weapons Failure Rate Data Handbook
- RADC, Rome Air Development Center Failure Rate Data
- MIL-HDBK-217A, Reliability Stress and Failure Rate Data for Electronic Equipment
- LTV-E Predictions - This covers predictions by LTV-E Reliability Engineers based on experience and general analyses. It is often used in conjunction with other available data to upgrade and degrade failure rate information on similar parts.
- Vendor Predictions - These include data obtained from vendors or subcontractors on identical or similar components.

b. Parts Derating

The derating factor indicated in Table XXV is applied to selected source failure rates to adjust the assessed rates depending on the stress and environments of the source data and that anticipated for each part of the subject design. This factor is further adjusted based on failure experience gained from similar products produced by LTV-E.

4. Failure Modes and Effects Analysis

The Reliability Failure Mode and Effect Analyses are presented in Table XXVI. A listing of all major components is given for the Simplex Actuator Package breakdown.

As a result of previous experience on similar parts plus additional general failure analysis, the probable modes of failure and effects on the Actuator package performance are tabulated. Only operational modes have been considered. Storage and nonoperation of the actuator package may result in failure modes of the O-rings while no other portion of the package is affected. Wearout failures are considered as being prevented by good maintenance practice in this analysis.

5. Sample Calculations

(a) MODE 1 - Manual Operation

$\lambda = 84.2285$*

*Failure Rate is expressed in failures per million operating hours.

$$MTBF = \frac{1}{\lambda} = \frac{1 \times 10^6}{84.2285}$$

MTBF = 11,875 hours

$R = e^{-\lambda t}$ where (t) = 2.0 hrs.

$$R = e^{-(84.2285 \times 2.0)}$$

R = .999832

(b) MODE 2 - Manual Operation Plus Augmentation

$\lambda = 279.7245$*

$$MTBF = \frac{1}{\lambda} = \frac{1 \times 10^6}{279.7245}$$

MTBF = 3570 hours

$$R = e^{-\lambda t}$$

$$R = e^{-(279.7245 \times 2.0)}$$

R = .999440

(c) MODE 3 - Autopilot Operation

$= 356.3905*$

$$MTBF = \frac{1}{\lambda} = \frac{1 \times 10^6}{356.3905}$$

MTBF = 2810 hours

$$R = e^{-\lambda t}$$

$$R = e^{-(356.3905 \times 2.0)}$$

$$R = .999288$$

(d) MODE 4 - Emergency Power Operation

$= 372.5675*$

$$MTBF = \frac{1}{\lambda} = \frac{1 \times 10^6}{372.5675}$$

MTBF = 2685 hours

$$R = e^{-\lambda t}$$

$$R = e^{-(372.5675 \times 2.0)}$$

$$R = .999256$$

Emergency Power Supply System Only

$$\lambda \text{ MODE 4} - \lambda \text{ MODE 1} = \lambda_E$$

$$\lambda = 288.3390$$

$$MTBF = \frac{1}{\lambda} = \frac{1 \times 10^6}{288.3390}$$

MTBF = 3460 hours

$$R = e^{-\lambda t}$$

$$R = e^{-(288.3390 \times 2.0)}$$

$$R = .999425$$

(a) $R = e^{-\lambda t}$

$$R = e^{-(84)(2) \times 10^{-6}} = e^{-168 \times 10^{-6}} = 1 - t$$

$$R = 1 - (168 \times 10^{-6})$$

$$R = .999832$$

(b) $R = e^{-(280)(2) \times 10^{-6}} = 1 - 5 = 1 - (560 \times 10^{-6})$

$$R = .999440$$

(c) $R = .999440$

(d) $R = e^{-(372)(2) \times 10^{-6}} = 1 - 5 = 1 - (744 \times 10^{-6})$

$$R = .999256$$

$$R = e^{-(288)(2) \times 10^{-6}} = 1 - t = 1 - (576 \times 10^{-6})$$

$$R = .999425$$

TABLE XXIV

Reliability Analysis Summary

	Failure Rate (λ) Failures/ 10^6 Hrs.	Mean Time Between Failure (MTBF) Hrs.	Reliability (R)* (Probability of Success)
MODE 1 - Manual Operation	84.2285	11,875	.999832
MODE 2 - Manual Operation Plus Augmentation	279.7245	3,570	.999440
MODE 3 - Autopilot Operation	356.3905	2,810	.999288
MODE 4 - Emergency System Operation	372.5675	2,685	.999256
Emergency Power Supply Only	288.3390	3,460	.999425

*Reliability (R) Computed using Time (t) = 2.0 Hrs.

TABLE XXV
TOTAL COMPONENT PART
RELIABILITY PREDICTION WORKSHEET

DATE 25 September 1969
PAGE NUMBER 1 of 7

PROJECT AFFDL EQUIPMENT Simplex Actuator ENGINEER Hickerson

PART CLASS	GENERIC FAILURE RATE FAILURES PER 10^6 HOURS (GFR)	APPLICATION FACTOR (K_A)	ADJUSTED FAILURE RATE FAILURES PER 10^6 HOURS (GFR x K_A)	QUANTITY (N)	PART CLASS FAILURE RATE FAILURES PER 10^6 HOURS (GFR x K_A x N)	REMARKS
Arm - Input Assembly	.012	1.0	.012	11	.132	401-13861
Retainer, Bearing	.20	.6	.12	2	.24	401-13853-01
Bearing	.875	.25	.29	2	.58	MKP6A
Housing Assembly	.4	1.0	.4	1	.40	401-13800-01
Cap, Linkage	.02	1.0	.02	1	.02	401-13854-01
Crank	.012	1.0	.012	1	.012	401-13855-01
Link Assembly	.012	1.0	.012	1	.012	401-13856-01
Shim	.001	1.0	.001	1	.0001	401-13870-01
Piston, Locking	1.0	1.0	1.0	2	2.0	401-13862
Sleeve, Locking	2.5	1.0	2.5	1	2.5	401-13863
Slider, Locking	2.5	1.0	2.5	1	2.5	401-13864
Retainer, Locking	.20	.6	.12	1	.12	401-13865
Valve, Sol. - 3-Way	76.6	1.0	76.6	2	153.20	401-13830

TOTAL DEVICE FAILURE RATE (FR) =

OPERATION FACTOR (K_O) = x( )
TOTAL DEVICE IN-USE FAILURE RATE =

TABLE XXV

TOTAL COMPONENT PART RELIABILITY PREDICTION WORKSHEET

DATE 25 September 1969
PAGE NUMBER 2 of 7

PROJECT ______ EQUIPMENT ______ ENGINEER ______

PART CLASS	GENERIC FAILURE RATE FAILURES PER 10^6 HOURS (GFR)	APPLICATION FACTOR (K_A)	ADJUSTED FAILURE RATE FAILURES PER 10^6 HOURS (GFR x K_A)	QUANTITY (N)	PART CLASS FAILURE RATE FAILURES PER 10^6 HOURS (GFR x K_A x N)	REMARKS
Valve, Sol., 4-Way	76.6	1.0	76.6	1	76.6	401-13831
Spring, Aux. Ram	30.0	.05	1.5	2	3.0	401-13819-04
Nut, Jam, Aux. Ram	.001	1.0	.001	1	.001	401-13825-01
Sleeve, Outer, Aux. Ram	2.50	1.0	2.50	1	2.50	401-13828-01
Piston, Aux. Ram	2.50	1.0	2.50	2	5.00	401-13829-01
Retainer, Sleeve, Aux. Ram	.20	.6	.12	1	.12	401-13837-01
Sleeve, Intmd., Aux. Ram	2.50	1.0	2.50	2	5.00	401-13826-01
Piston, Aux. Ram	2.50	1.0	2.50	1	2.50	401-13827-01
Retainer, Sleeve, Aux. Ram	.20	.6	.12	1	.12	401-13872-01
Linear XDCR, Aux. Ram	10.0	1.0	10.0	1	10.0	401-13806-01
Retainer, XDCR, Aux. Ram	.20	.6	.12	1	.12	401-13838-01
Nut, Jam, XDCR, Aux. Ram	.001	1.0	.001	1	.001	401-13839-01
Sequence Valve Assy.	5.00	1.0	5.00	1	5.00	401-13813-01

TOTAL DEVICE FAILURE RATE (FR) = ______

OPERATION FACTOR (K_O) = x(______)
TOTAL DEVICE IN-USE FAILURE RATE = ______

TABLE XXV
TOTAL COMPONENT PART
RELIABILITY PREDICTION WORKSHEET

DATE 25 September 1969

PROJECT ______ EQUIPMENT ______ ENGINEER ______

PART CLASS	GENERIC FAILURE RATE FAILURES PER 10^6 HOURS (GFR)	APPLICATION FACTOR (K_A)	ADJUSTED FAILURE RATE FAILURES PER 10^6 HOURS ($GFR \times K_A$)	QUANTITY (N)	PART CLASS FAILURE RATE FAILURES PER 10^6 HOURS ($GFR \times K_A \times N$)	REMARKS
Retainer, Sequence Valve	.20	.6	.12	1	.12	401-13817-01
Spring, Sequence Valve	30.0	.05	1.50	1	1.50	401-13819-03
Slider	2.50	1.0	2.50	1	2.50	401-13815-01
Sleeve	2.50	1.0	2.50	1	2.50	401-13814-01
Servo Valve Assy.	76.0	1.0	76.0	1	76.0	401-13809-01
Servo Valve Assy.				1		401-13802-01
Slider	2.50	1.0	2.50	1	2.50	401-13804-01
Sleeve	2.50	1.0	2.50	1	2.50	401-13803-01
Retainer (Servo Valve)	.2	.6	.12	1	.12	401-13835-01
Switching Valve Assy.				1		401-13820-01
Slider	2.50	1.0	2.50	1	2.50	401-13822-01
Sleeve	2.50	1.0	2.50	1	2.50	401-13821-01
Retainer, Switching Valve	.20	.6	.12	1	.12	401-13823-01

TOTAL DEVICE FAILURE RATE (FR) = ______

OPERATION FACTOR (K_O) = x( )
TOTAL DEVICE IN-USE FAILURE RATE = ______

TABLE XXV
TOTAL COMPONENT PART
RELIABILITY PREDICTION WORKSHEET

DATE 25 September 1969
PAGE NUMBER 4 of 7

PROJECT ______ EQUIPMENT ______ ENGINEER ______

PART CLASS	GENERIC FAILURE RATE FAILURES PER 10^6 HOURS (GFR)	APPLICATION FACTOR (K_A)	ADJUSTED FAILURE RATE FAILURES PER 10^6 HOURS (GFR x K_A)	QUANTITY (N)	PART CLASS FAILURE RATE FAILURES PER 10^6 HOURS (GFR x K_A x N)	REMARKS
Spring, Outer	30.0	.05	1.5	1	1.5	401-13819-01
Spring, Inner	30.0	.05	1.5	1	1.5	401-13819-02
Barrel Assy.				1		401-13840-01
Barrel	1.1	1.0	1.1	1	1.1	401-13840-02
Barrel	1.1	1.0	1.1	1	1.1	401-13840-03
Piston & Rod Assy.	2.2	1.0	2.2	1	2.2	401-13842-01
Seal Plate (Center Dam)	.2	1.0	.2	1	.2	401-13841-01
Nut (Bushing & Seal)	.001	1.0	.001	1	.001	401-13846-01
Bushing	.046	1.0	.046	1	.046	401-13847-01
Piston	12.2	.1	1.22	1	1.22	401-13884-01
Washer (Piston)	.0001	1.0	.0001	1	.0001	401-13834-01
Nut (Piston)	.001	1.0	.001	1	.001	401-13833-01
Linear XDCR	10.0	1.0	10.0	1	10.0	401-13808-01

TOTAL DEVICE FAILURE RATE (FR) = ______

OPERATION FACTOR (K_O) = ______ x(______)
TOTAL DEVICE IN-USE FAILURE RATE = ______

TABLE XXV
TOTAL COMPONENT PART
RELIABILITY PREDICTION WORKSHEET

DATE 25 September 1969
PAGE NUMBER 5 of 7

PROJECT _____ EQUIPMENT _____ ENGINEER _____

PART CLASS	GENERIC FAILURE RATE FAILURES PER 10^6 HOURS (GFR)	APPLICATION FACTOR (K_A)	ADJUSTED FAILURE RATE FAILURES PER 10^6 HOURS (GFR x K_A)	QUANTITY (N)	PART CLASS FAILURE RATE FAILURES PER 10^6 HOURS (GFR x K_A x N)	REMARKS
Retainer, (XDCR)	.2	.6	.12	1	.12	401-13845-01
Washer, (XDCR)	.0001	1.0	.0001	2	.0002	401-13834-02
Fitting (XDCR Probe)	.001	1.0	.001	1	.001	401-13843-01
Rod End Assy.	2.9	1.0	2.90	1	2.90	40.-13812-01
Bushing	.046	1.0	.046	1	.046	401-13847-01
Bayonet	1.0	.5	.5	2	1.0	401-13848-01
Bayonet	1.0	.5	.5	2	1.0	401-13848-02
Relief Valve Assy.	3.35	1.0	3.35	1	3.35	401-13852-01
Relief Valve Assy.	3.35	1.0	3.35	1	3.35	401-13852-02
Check Valve	1.67	1.0	1.67	2	3.34	401-13807-01
Check Valve	1.67	1.0	1.67	3	5.01	401-13807-02
Retainer, Check Valve	.2	.6	.12	2	.24	401-13859-01
Retainer, Check Valve	.2	.6	.12	2	.24	401-13860-01

TOTAL DEVICE FAILURE RATE (FR) =

OPERATION FACTOR (K_O) = x(   )
TOTAL DEVICE IN-USE FAILURE RATE =

TABLE XXV

TOTAL COMPONENT PART

RELIABILITY PREDICTION WORKSHEET

DATE 25 September 1969

PAGE NUMBER 6 of 7

PROJECT ______ EQUIPMENT ______ ENGINEER ______

PART CLASS	GENERIC FAILURE RATE FAILURES PER 10^6 HOURS (GFR)	APPLICATION FACTOR (K_A)	ADJUSTED FAILURE RATE FAILURES PER 10^6 HOURS (GFR X K_A)	QUANTITY (N)	PART CLASS FAILURE RATE FAILURES PER 10^6 HOURS (GFR X K_A X N)	REMARKS
Spacer, Check Valve	.0001	1.0	.0001	1	.0001	401-13869-01
Junction Box	.001	1.0	.001	1	.001	401-13874-01
Press Switch	11.9	.1	11.9	3	35.70	401-13811-01
Filter (Screen)	4.15	1.0	4.15	2	8.30	401-13818-01
Filter Element	4.15	1.0	4.15	1	4.15	401-13832-01
Retainer, Filter	.2	.6	.12	1	.12	401-13849-01
Motor/Pump	M=67.0 P=71.0	1.3	88.0 94.0	1	182.0	401-13871-01
Spring, Rsvr. (Outer)	.5	1.0	.5	2	1.0	401-13819-05
Spring, Rsvr. (Inner)	.5	1.0	.5	2	1.0	401-13819-06
Piston, Rsvr.	1.0	.2	.2	1	.2	401-13866-01
Guide, Spring, Rsvr.	.001	1.0	.001	1	.001	401-13867-01
Retainer, Spreing, Rsvr.	.2	.6	.12	1	.12	401-13868-01
Tube Assy.	.22	1.0	.22	1	.22	401-13850-03

TOTAL DEVICE FAILURE RATE (FR) =

OPERATION FACTOR (K_O) = x( )

TOTAL DEVICE IN-USE FAILURE RATE =

TABLE XXV
TOTAL COMPONENT PART
RELIABILITY PREDICTION WORKSHEET

DATE 25 September 1969
PAGE NUMBER 7 of 7

PROJECT ______ EQUIPMENT ______ ENGINEER ______

PART CLASS	GENERIC FAILURE RATE — FAILURES PER 10^6 HOURS (GFR)	APPLICATION FACTOR (K_A)	ADJUSTED FAILURE RATE — FAILURES PER 10^6 HOURS (GFR x K_A)	QUANTITY (N)	PART CLASS FAILURE RATE — FAILURES PER 10^6 HOURS (GFR x K_A x N)	REMARKS
Tube Assy.	.22	1.0	.22	1	.22	401-13850-04
Connector	.011	6.0	.066	4	.264	MS3106R10SL-3S
Connector	.044	6.0	.264	1	.264	MS3102R28-11P
Connector	.058	6.0	.348	1	.348	MS3102R22-14P
Connector	.011	6.0	.066	2	.132	MS3106E-10SL-4S
Connector	.022	6.0	.132	1	.132	MS24266R-14-T-7S

TOTAL DEVICE FAILURE RATE (FR) = ______

OPERATION FACTOR (K_O) = ______ x(______)
TOTAL DEVICE IN-USE FAILURE RATE =

TABLE XXVI

Failure Mode and Effects Analysis

Item Description/ Part Number	Function	Failure Mode	Failure Effects	Remarks
Arm - Input Assy. 401-13861	Positions locking piston and main servo valve input arm.	Structural	Loose manual control.	Low probability of occurrence; design safety factor 2.
Housing Assy. 401-13800-01	Structural mounting for components, fluid flow and pressure vessel.	Structural	Actuator fails hard over. Worst case.	Low probability of occurrence. Design safety factor 2.
Piston, Locking (2) 401-13862	Contains slider and has matched piston areas to lock slider.	Structural	Would bind slider. Loose manual control.	
		Leakage between pis-on and housing.	Cause unbalanced position of locking piston.	
Slider, Locking 401-13864	Position main servo valve input arm.	Structural	Would bind. Loose manual control.	Low probability of occurrence. Design factor 2.
Sleeve, Locking 401-13863	Contains slider.	Structural	Would bind slider. Loose manual control.	Low probability of occurrence. Design factor of safety 2.
Solenoid Valve, 3-Way, 401-13830	Supplies hydraulic pressure to actuator. Switching Valve	Leakage	(P_1) leaks to (R_1) return, leaks external.	When P_1 becomes 1000 psi, emergency pump pressure over comes force on switching valve to shuttle valve.
		Electrical short.	Fails to actuate. (Fails closed.)	
Solenoid Valve 3-Way, 401-13830	Supplies hydraulic pressure to E/H, Servo Valve, Aux. Ram, and locking piston.	Leakage	P_1 leaks to return - leaks external.	
		Electrical short	Fails to actuate auto-pilot and augmentation modes.	Pilot flys on manual mode.

TABLE XXVI
(Continued)

Item Description/ Part Number	Function	Failure Mode	Failure Effects	Remarks
Piston, Aux. Ram (2) 401-13829-01	Positions main servo valve input arm.	Structural	Would bind. Loose autopilot and augmentation modes.	Pilot flys on manual mode.
Piston, Intmd., Aux. Ram (1) 401-13827-01		Leakage	Performance tolerances affected.	
Sleeve, Outer, Aux. Ram, (1) 401-13828-01	Contains aux. ram pistons	Structural	Would bind piston loose autopilot and augmentation modes.	Pilot flys on manual mode.
Sleeve, Intmd., Aux. Ram, (2) 401-13826-01		Leakage	Performance tolerances affected.	
Spring, Aux. Ram (2) 401-13819-04	Balances aux. ram pistons.	Structural	Aux. ram fails hardover. Worst case.	Pilot corrects input and flys manually.
Linear XDCR, Aux. Ram (1) 401-13806-01	Provides aux. ram position	Structural	Slug breaks - loose autopilot feedback.	Pilot flys on manual mode.
	Feedback to autopilot controls	Electrical low resistance.	False feedback signal to autopilot controls.	
		Electrocal open	Loose autopilot and augmentation modes.	
Sequence Valve Assy. 401-13813-01	Directs flow between C_1 and C_2 ports on aux. ram.	Structural	Bind would cause loss of control of aux. ram.	Pilot flys on manual mode.
		Leakage	Performance tolerance affected.	
Servo-Valve Assy. 401-13809-01	Provides control flow to the sequence valve.	Structural	Aux. ram fails hardover.	Pilot flys on manual mode.
		Electrical low resistance.	False flow control.	
		Open	Loose autopilot and augmentation modes.	

TABLE XXVI
(Continued)

Item Description/ Part Number	Function	Failure Mode	Failure Effects	Remarks
Main Servo Valve	Meters P_1 and P_2 pressure and R_1 and R_2 pressures to cylinder A_1, A_2, and A_3, A_4 areas for ram operation.	Structural	Uncontrolled flow of cylinder ports.	Low probability of occurrence.
		Metering edge scored by contamination.	Performance tolerance affected.	Low probability of occurrence. System fluid filtered to 10 microns.
		Leakage	Irregular flow from valve.	Will be observed during assembly and corrected.
Switching Valve	Switches hydraulic pressure from A/P primary (P_1) system to power cylinder integrated motor/pump system.	Structural	Spool could fail to shuttle due to bind. Emergency fluid could be pumped into P_1 system.	Could cause loss of control of actuator.
		Leakage	If P_1 emergency pressure leakage could deplete fluid quantity in emergency system.	Could cause limited use of emergency system.
		Contamination	Contamination could cause binding as indicated above.	Could cause loss of control of actuator.
Barrel Assembly (2) 401-13840-01	Contains Piston rod and fluid pressure.	Structural	Loss of actuator - bind piston rod.	Design factor of safety 2.
		Leakage (Areas A_1 and A_2.)	Deplete fluid in P_1 and emergency system.	Fly on P_2 system.
		Leakage (Areas A_3 and A_4.)	Deplete fluid in P_2 system.	Fly on P_1 or emergency system.
Linear XDCR, Piston Rod (1) 401-13808-01	Provides actuator piston rod position Feedback to autopilot controls.	Structural	Slug Breaks - loose autopilot feedback.	Pilot flys on manual mode.
		Electrical, low resistance.	False feedback signal to autopilot controls.	
		Electrical open.	Loose autopilot and augmentation modes.	

TABLE XXVI
(Continued)

Item Description/ Part Number	Function	Failure Mode	Failure Effects	Remarks
Piston and Rod Assy. 401-13842-01	Provides linear output for servo package.	Structural	Bind or jam barrel.	Could cause loss of control of actuator.
Relief Valve Assy. 401-13852-01	Relieves excessive emergency return pressure to return of R_1	Structural	Failure of valve to relieve could cause over pressure and possible rupture of emergency return system.	Fly on P_1 or P_2 systems.
		Leakage (External)	Deplete emergency and P_1 fluid supplies.	Fly on P_2 system.
Relief Valve Assy. 401-13852-02	Relieves excessive discharge pressure at the emergency pump to return.	Structural	Over pressure supply system.	Possible loss of emergency system components.
		Leakage (External)	Deplete emergency fluid supply.	Fly on P_1 or P_2 systems.
Check Valve (7 Total) 401-13807	Check P_1 reverse flow at inlet. (1)	Structural	Back-flush inlet filter to A/C system - contamination.	Cause possible contamination of upstream components.
	Checks P_2 reverse flow at inlet. (1)		(Same).	
	Checks emergency return flow to R_1 at relief valve. (1)		Deplete emergency fluid supply.	Fly on P_2 system.
	Doublechecks P_1 flow to emergency pressure. (2)		Over pressure emergency components, possible rupture.	Loose emergency and P_1 system; fly on P_2.
	Check emergency supply flow to return (1)		(same)	
	Checks emergency reservoir flow to pump case drain. (1)		(Same.)	
		Leakage	(Same over a longer period of time.)	(Same.)

TABLE XXVI
(Continued)

Item Description/ Part Number	Function	Failure Mode	Failure Effects	Remarks
Pressure Switch (3) Total 401-13811-01	Indicates P_1, P_2 and emergency system supply pressure.	Structural	Deplete system fluid.	Fly on other system.
		Leakage	Deplete system fluid.	
		Electrical short.		
Filter (screen) (2) total 401-13818-01	Stops contamination from entering servo package from A/C fluid systems.	Structural	Contamination enters package - cause possible valve jam and wear.	Fly on other system.
Filter element	Stops contamination from entering servo package fluid system from emergency pump.	Structural	(Same above).	(Same above).
Motor (Part of Motor/Pump Assy.) (1) 401-13871-01	Power source for emergency pump.	Structural	Bearings burn out causing lock rotor. Loose motor.	Loose emergency supply system.
		Electrical open or shorted rotor or stator.		
Pump (Part of Motor/Pump Assy.) (1) 401-13871-01	Emergency hydraulic power supply.	Structural (Internal)	Fail to operate.	Loose emergency supply system.
		External	Deplete emergency fluid supply.	
		Contamination	Cause excessive wear - Degrade performance.	
Reservoir (1)	Fluid storage (Boot-strap type.)	Structural	Deplete fluid supply.	Loss of system.
		Leakage (IN-ternal and external.)	Loss of fluid and pump cavitation.	

TABLE XXVI
(Continued)

Item Description/ Part Number	Function	Failure Mode	Failure Effects	Remarks
Connectors, Electrical (9 total)	Junction for electrical circuit.	Electrical short or open corrosion.	Loss of circuit. False signal.	Loss of system.

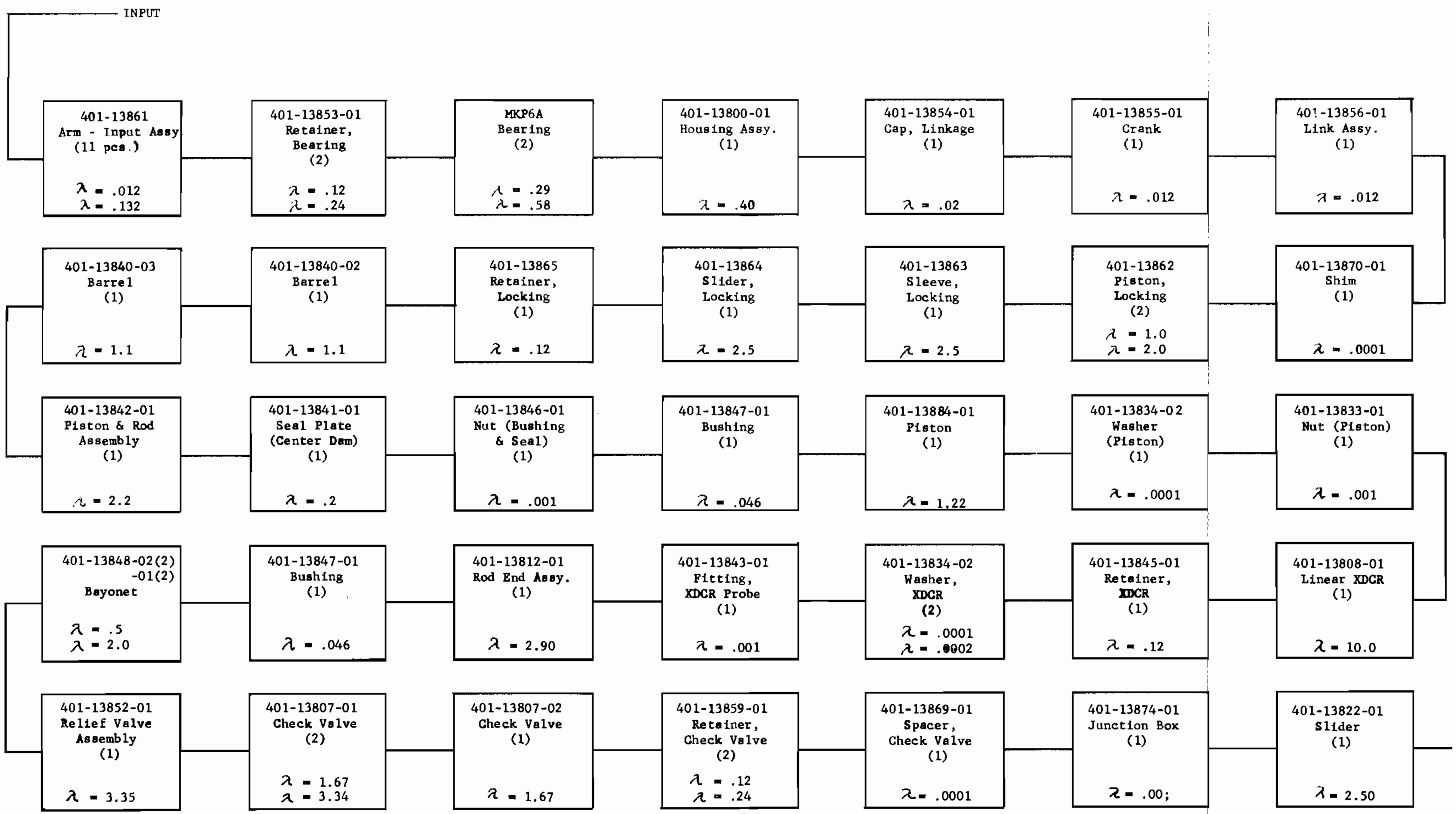
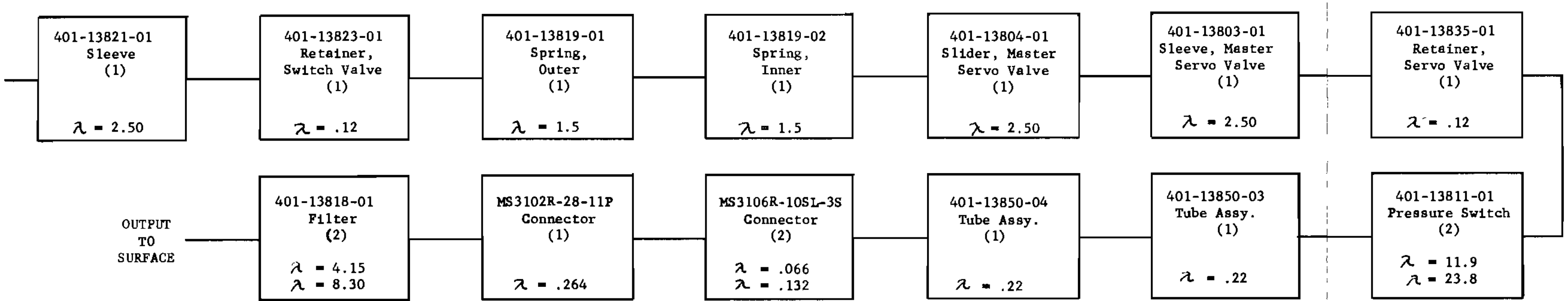
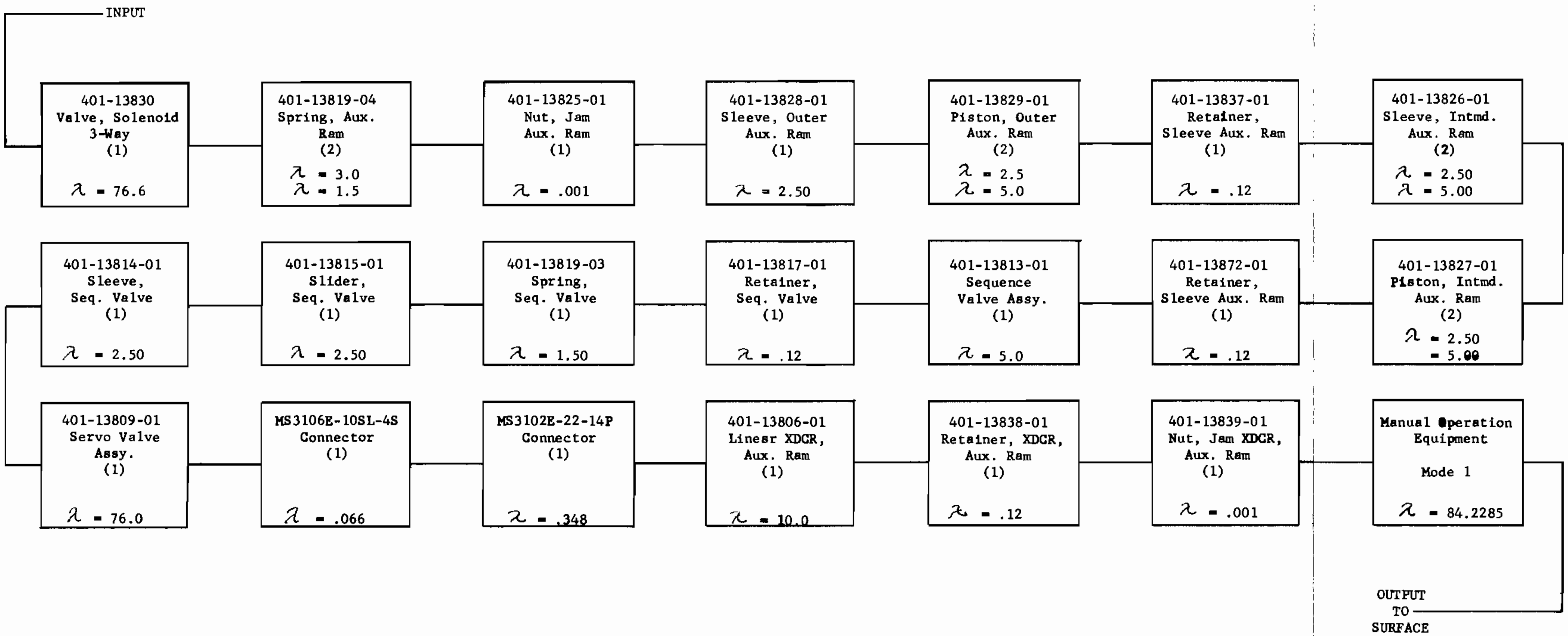


FIGURE 75. MANUAL OPERATION – MODE 1 (CONVENTIONAL HYDRAULIC POWER CONTROL UNIT) (SHEET 1 OF 2)



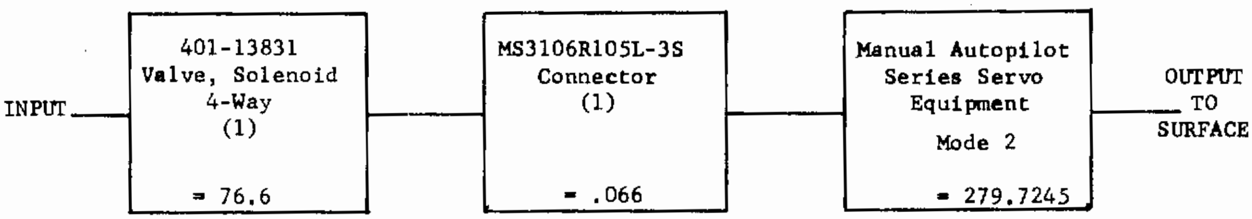
$$\lambda = \sum_{i=1}^{n} \lambda_i = 84.2285 \frac{\text{FAILURES}}{10^6 \text{ Operating Hrs.}} \quad \text{(MODE 1)}$$

FIGURE 75. MANUAL OPERATION – MODE 1 (CONVENTIONAL HYDRAULIC POWER CONTROL UNIT)
(SHEET 2 OF 2)



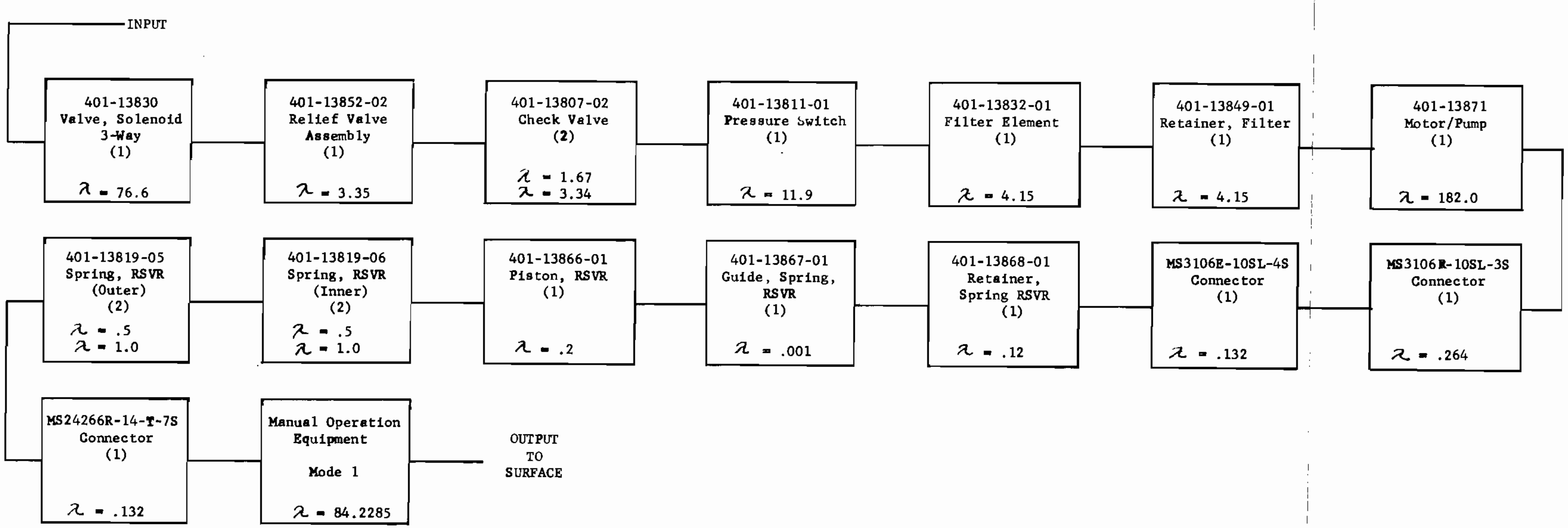
$$\lambda = \sum_{i=1}^{n} \lambda_i = 279.7245 \ \frac{\text{FAILURES}}{10^6 \text{ Operating Hrs.}} \quad \text{(MODE 2)}$$

FIGURE 76. MANUAL OPERATION PLUS AUGMENTATION – MODE 2
(NORMAL AUTOPILOT SERIES SERVO PLUS POWER CYLINDER)



$$\lambda = \sum_{i=1}^{n} \lambda_i \quad = 356.3905 \ \frac{\text{FAILURES}}{10^6 \text{ Operating Hrs.}} \quad \text{(MODE 3)}$$

FIGURE 77. AUTOPILOT – MODE 3 (NORMAL AUTOPILOT PARALLEL SERVO PLUS POWER CYLINDER)



$$\lambda = \sum_{i=1}^{n} \lambda_i = 372.5675 \frac{\text{FAILURES}}{10^6 \text{ Operating Hrs.}} \quad \text{(MODE 4)}$$

FIGURE 78. EMERGENCY POWER OPERATION – MODE 4
(MANUAL OPERATION WITH INTEGRATED EMERGENCY POWER SUPPLY)

APPENDIX V

SUMMARY OF SIMPLEX ACCEPTANCE TESTING

A. INTRODUCTION

The Simplex actuator was tested in accordance with LTVE acceptance test procedure (ATP) 406-10571. The S/N 2 and S/N 3 met all performance requirements. The S/N 1 unit met all performance requirements except the emergency system velocity. S/N 1 was a development and qualification actuator which did not have latest improvements incorporated on the S/N 2 and 3 units such as stronger reservoir springs to provide higher emergency system return pressure. A brief description and results of the more significant tests are given in the remainder of this section. The referenced paragraphs refer to the ATP.

B. PISTON VELOCITY FOR SMALL DISPLACEMENTS (Para 5.4)

This test consists of measuring piston velocity at displacements of the manual input lever of ± .015 inch from null. The average velocities obtained were:

	Average Velocity In/Sec
S/N 1	.078
S/N 2	.114
S/N 3	.079

The specified limit is .07 to .20 in/sec.

C. OVERPOWER FORCE (Para 5.7)

The force required to break out the parallel mode locking pistons with the auxiliary ram hardover was evaluated. The test was performed in both directions. Results were as follows:

	Overpower Forces	
	Retract lbs.	Extend lbs.
S/N 1	20.5	19.5
S/N 2	21.0	20.0
S/N 3	20.0	20.0

The specifications requirement is 17 to 21 pounds.

D. CAPABILITY OF PILOT TO REVERSE DIRECTION (Para 5.8)

This test measures the main ram velocity that can be obtained when the pilot overcomes a hardover auxiliary ram in the parallel mode of operation.

	Piston Velocity	
	Retract in/sec	Extend in/sec
S/N 1	7.0	7.1
S/N 2	8.48	5.05
S/N 3	7.72	7.50

The specified limit is 4.63 in/sec minimum.

E. VALVE TRAVEL AND FLOW (Para 5.9)

This test determines the valve flow through each orifice separately as a function of manual input lever displacement. The flows obtained at maximum valve displacement of .75 inch and at 500 psi ΔP across each orifice are as follows:

S/N 1

System 1		System 2	
Orifice	Flow gpm	Orifice	Flow gpm
P1 to A1	8.86	A1 to R1	10.05
P1 to A2	9.10	A2 to R1	10.2
P2 to A3	11.4	A3 to R2	11.6
P2 to A4	11.6	A4 to R2	13.2

S/N 2

System 1		System 2	
Orifice	Flow gpm	Orifice	Flow gpm
P1 to A1	10.0	A1 to R1	10.0
P1 to A2	10.7	A2 to R1	10.6
P2 to A3	11.9	A3 to R2	11.6
P2 to A4	12.0	A4 to R2	12.7

S/N 3

System 1 Orifice	Flow gpm	System 2 Orifice	Flow gpm
P1 to A1	9.6	A1 to R1	10.3
P1 to A2	9.6	A2 to R1	10.4
P2 to A3	12.7	A3 to R2	12.4
P2 to A4	13.0	A4 to R2	13.6

The flow limits for this condition is as follows:

System 1 Orifice	Flow gpm	System 2 Orifice	Flow gpm
P1 to P1	8.6 to 10.7	A1 to R1	8.6 to 10.7
P1 to A2	9.7 to 11.9	A2 to R1	9.7 to 11.9
P2 to A3	11.3 to 14.3	A3 to R2	11.3 to 14.3
P2 to A4	11.3 to 14.1	A4 to R2	11.3 to 14.1

F. EMERGENCY SYSTEM TESTS

1. Static Pump Output Pressure (Para 5.12.5.1)

This test consists of measuring the static pressure output of the emergency pump when operating at rated speed and zero piston velocity. Results obtained are as follows:

	Output Pressure (psi)
S/N 1	1610
S/N 2	1560
S/N 3	1580

Spec limit is: 1500 ± 100 psi.

2. Velocity (Para 5.12.5.2)

This test consists of measuring the no-load piston velocity of the actuator being powered by the emergency motor pump. The purpose of this test is to ensure that sufficient surface rate is obtained when operating in emergency mode. Results are:

	Retract Velocity in/sec	Extend Velocity in/sec
S/N 1	4.00	3.16*
S/N 2	3.98	3.62
S/N 3	4.04	3.81

The specification limit is 3.5 in/sec minimum.
*S/N 1 did not have the stronger reservoir springs which increased emergency return pressure.

3. Switching Times (Para 5.12.6)

This test consists of measuring the elapsed times required for the emergency system to switch from the armed condition to the on-line, on-line to armed condition and unarmed condition to on-line. Tests results were as follows:

	Armed Condition to On-Line sec	On-Line to Armed Condition sec	Unarmed to Armed Condition sec
S/N 1	.05	.05	.4
S/N 2	.11	.05	.4
S/N 3	.2	.4	.25
Maximum time allowed	.5	.5	1.5

4. Frequency Response (Para 5.17)

The phase lag and the amplitude ratio of the input signal to the output signal while the unit was being powered by the emergency motor pump were evaluated. The test was performed with simulated spring and mass load. The test results were as follows:

	Frequency at 45° Phase Lag
S/N 1	.9
S/N 2	1.45
S/N 3	1.5

The specification requirement was that the phase lag of 45° between the input and output signal should not occur before .8 cps.

G. INTERNAL LEAKAGE (Para 5.13)

The leakage of actuator (P1 and P2 systems) both at neutral and bottomed out condition was measured. The test results were as follows:

	Neutral Leakage cc/min	Bottomed Out Extended Leakage cc/min	Bottomed Out Retracted Leakage cc/min
S/N 1			
P1 section	234	26	11
P2 section	180	11	34
S/N 2			
P1 section	207	118	138
P2 section	158	18	14
S/N 3			
P1 section	182	40	47
P2 section	194	22	20

The maximum allowable neutral leakage is 328 cc/min and the maximum allowable bottomed out (or loaded) leakage is 820 cc/min.

H. VALVE FORCE

1. Static Valve Force (Para 5.14.1)

The forces required to move the manual input lever with the actuator in the static condition were as follows:

	Retract Force oz.	Extend Force oz.
S/N 1	4.2	4.15
S/N 2	1.8	1.2
S/N 3	1.5	2.75

The maximum allowed force is 4.8 ounces.

2. Dynamic (Para 5.14.2)

Force input to the manual input lever was measured as the unit was cycled from .1 to 5 cps at a displacement of 1.57 inches peak-to-peak. The following maximum forces were measured:

	Forces (lbs)
S/N 1	± 3.0*
S/N 2	± .6 @ 3.5
S/N 3	± 1.5 @ 4 cps

*A tare force (due to mass and friction) was not subtracted from this figure. The maximum allowable force was ± 1.75 lbs.

I. SAS MODE TEST

1. Force Feedback in SAS Mode (Para 5.15)

The force feedback which will be felt by the pilot when operating in the SAS mode was measured at a frequency of .5 and 1.0 cps. The test results were as follows:

	Force Feedback lbs.
S/N 1	± .3
S/N 2	± .3
S/N 3	± .2

The maximum feedback force is ± 1.75 pounds.

2. Authority Travel (Para 5.16)

The authority of the SAS mode was measured. The results were as follows:

	Authority Travel inches
S/N 1	.348
S/N 2	.354
S/N 3	.342

The specific authority travel in the SAS mode is .352 ± .010 inches.

J. AUTOPILOT TRANSIENT VELOCITY

This test consists of measuring the drift rate of the actuator when the parallel mode is engaged. There is no signal to the servo amplifier (grounded input), and the outer servo loop is opened by removing the excitation to the main ram's LVDT. If the drift exceeds the .2 in/sec specification limit, the lock pistons are shimmed to achieve an acceptable drift rate. The purpose of shimming the lock pistons is to adjust the main servo valve to its null position with the parallel mode engaged with no electrical inputs to the auxiliary ram. After shimming the following test results were obtained:

	Autopilot Transient Velocity in/sec
S/N 1	.054
S/N 2	.071
S/N 3	.076

APPENDIX VI

SUMMARY OF QUALIFICATION TESTS

A. General

The qualification testing of the Simplex Actuator was performed at the Arlington and Garland facilities of LTV Electrosystems, Inc., Garland Division. The life and endurance, proof pressure, compatibility, full stroke cycles, impulse cycling, high and low temperature environmental test, ultimate loads, and burst pressure tests were performed at the Arlington facility. The shock, vibration, humidity, acceleration and EMI tests were performed at the Garland facility.

B. Description of Tests

1. Life and Endurance

a. Test with P_1 and P_2 Supply Pressure Applied

The life and endurance test consisted of a preliminary low temperature operation test in which the actuator oil and environment was reduced to -65°F and the actuator was cycled for ten strokes. After some preliminary functional tests, the endurance cycling tests were begun. The test spectrum was as follows:

Cycles No.	Stroke %	Mode	Load(C) Pounds	Load(T) Pounds	Frequency Hz
40,000	2.5	Series	None	None	2.8
20,000	10	Parallel	None	None	1.5
2,000	25	Parallel	3,800	3,000	.7
1,800	50	Manual	7,400	5,900	.4
200	95	Manual	14,000	11,300	.2

The spectrum was repeated ten times. The first two series were run at a inlet fluid temperature of 275 $^{+0}_{-25}$ °F and the next eight series were run at 200 $^{+25}_{-0}$ °F. The doors of the environmental test box were open and ambient air (approx. 80°F) was allowed to circulate through the environmental box. See Figure 79 for photograph of actuator in test fixture.

b. Backup Systems

The following endurance cycle spectrum was performed upon the actuator using the backup (emergency) system pump only to supply the hydraulic power:

No. of Cycles	Stroke %	Frequency Cps	Load(C) Pounds	Load(T) Pounds
10,000	10	1.5	None	None
1,500	25	.3	3,800	3,000
800	50	.1	7,400	5,900
100	95	.1	None	None

The test was also run with the door removed from the environmental test chamber and approximately 80°F ambient air was allowed to circulate around the test actuator. Thermocouples mounted on the pump housing, system 1 barrel, emergency reservoir of valve housing, in ambient air adjacent to actuator, and pump case drain line were used to monitor the temperatures during this test.

2. Proof Pressure

The test actuator was proofed at 4500 psi with both the hydraulic fluid and ambient air temperature maintained at 275 $^{+0}_{-25}$ °F. The actuator was proofed for five minutes with the piston bottomed out in both extend and retract directions.

The test was done first with the proof pressure applied to both P_1 and P_2 sections, and then repeated with the proof pressure applied to each section individually. The return passages were proofed to 2500 psi.

3. Compatibility Tests

The emergency reservoir was completely filled (as measured after one retract stroke) and the unit was cycled at maximum rate with P_2 section operative and P_1 section inoperative. The actuator was cycled for 25 cycles with the emergency system operative and 25 cycles with the emergency system inoperative. The extension of the reservoirs indicator was measured at the completion of the test in order to determine the amount of reservoir fluid lost.

4. Full Stroke Cycle Test

The actuator was next cycled to 15,000 full stroke no load cycles. The bottoming of the actuator was adjusted until a maximum pressure peak of 3300 $\pm$ 100 psi was obtained. Visual and leakage tests were conducted at completion of the test. The hydraulic fluid temperature was maintained at 90 $\pm$ 20^{o}F and the air temperature was 80 $\pm$ 10^{o}F.

5. Impulse Cycling

The tests were run with the unit bottomed out in both extend and retract positions. The 25,000 impulse cycles were applied in each position. The pressure pulse was a 12-cps square wave which varied from 500 to 3750 psi. Fluid and air temperatures were maintained at 200 $\pm$ 10^{o}F.

6. Environmental Tests

a. Low Temperature Operations & Rapid Warmup

The actuator was soaked for 24 hours at a fluid and ambient temperature of -65 $\pm$ 10^{o}F. The actuator was cycled 5 strokes at a minimum pressure and 5 cycles at 3000 psi. The fluid and ambient temperature was increased to $-20^{\circ}F$ and the unit was cycled 5 strokes. The unit was then rapidly warmed up to $275^{\circ}F$. Th: actuator was cycled manually at 6 cpm during this warmup. The input force as a function of fluid temperature was monitored during the rapid warmup.

b. High Temperature Operation with P_1 and P_2 Pressure Applied

The ambient air and hydraulic fluid temperatures were increased to $275 {}^{+0}_{-25} {}^{\circ}F$ and allowed to stabilize for 30 minutes. The ambient air was then increased to $330 {}^{+0}_{-10} {}^{\circ}F$ in approximately 1 minute and then maintained for 4 1/2 minutes. After the ambient air temperature had attained $330 {}^{+0}_{-10} {}^{\circ}F$, the actuator was stroked 5 complete strokes in the manual mode. This 30 minutes at $275^{\circ}F$ and 4 1/2 minute cycle at $330^{\circ}F$ was repeated a total of 50 times.

c. High Temperature Emergency System Operation - The actuator backup system reservoir was filled with hydraulic fluid and the actuator was soaked at $200 \pm 10^{\circ}F$ ambient air temperature for five hours. The ambient air temperature was then increased to $330 {}^{+0}_{-10} {}^{\circ}F$ and held for 1 minute. The ambient air temperature was then reduced to $200 \pm 10^{\circ}F$ and the test run was begun. The motor pump was run at rated speed and maximum flow at a pressure of 350 psi for 30 minutes. The next test consisted of running with the pump cycling between maximum pressure (no flow) and maximum flow at a cycle rate of 6 cpm for 30 minutes. The next test condition consisted of running at maximum pressure (no flow) for 10 minutes and maximum flow for 1 minute. This cycling schedule was continued for 60 minutes. The test was performed by connecting a bypass with a needle valve between A_1 and A_2 test ports. The valve was opened to obtain maximum flow at a pressure of 350 psig and closed to obtain maximum pressure.

7. Ambient Functional Test

Upon completion of the above tests, the actuator was operated manually and electrically in the parallel (autopilot) mode, the static pump output pressure, the emergency velocity and the internal leakage of the actuator were measured.

8. Vibration

The actuator was vibrated in all three major axes (Figure 80) at 2g sinusoidal from 51 to 500 cps in order to determine the resonance frequencies. In the Z axis (vertical) resonances at 51 and 214 cps were observed. The test procedure was to dwell at each of these resonances for 10 minutes and then cycle the frequency of the vibration from 5 to 500 to 5 cps at 5g sinusoidal (at frequencies below 52 cps the vibration was controlled to prescribed frequency-amplitude schedule) for the remainder of the hour. The test procedure was the same for the lateral and longitudinal axis. Resonance at 99 cps was found in the lateral axis. A brief test which consisted of manual and parallel (autopilot) operation, measurements of valve current, auxiliary ram frequency response, threshold, emergency pump no-flow pressure output, emergency system velocity, and internal leakage of both systems were performed.

9. Shock

The actuator was subjected to shocks of 10g level along the lateral, longitudinal, and vertical axes and in each direction along these axes. The shock pulse duration was 11 milliseconds with peak acceleration occurring at approximately 5.5 milliseconds as obtained with a Barry 150 shock machine. At the conclusion of same series of functional tests which had been performed after the vibration tests were performed.

10. Humidity

The test article was placed in the humidity chamber in its approximate installation attitude. A 24-hour cam actuator programmer controlled the chamber environment to parameters as follows:

Temperature (°F)	Humidity (%)	Time (Hrs)
Gradual increase to +160	95	2
+160	95	6
Gradual decrease to +80	85+	16

The above schedule of test conditions was repeated for a total of five continuous 24-hour cycles. Upon completion the actuator was subjected to the same functional tests performed after vibration.

11. Acceleration

The test actuator was placed in the centrifuge and an acceleration of 5.8g was applied for one minute in the +X direction and 13g was applied for one minute in the +Z direction (See Figure 80) for definition of these directions). This was the 115% no yield level condition. For the next test condition, ultimate load, an acceleration of 17.7g for one minute in the

+Z direction and an acceleration of 7.6g for one minute in the +X direction was imposed upon the actuator. Upon completion of this test, the test article was subjected to the same functional tests which were performed after the vibration tests.

12. EMI Tests

Electromagnetic interference tests were performed on one F-4 Stabilator Control Simplex Actuator in accordance with the requirements of MIL-I-6181D, but specifically adapted to the unit under test by LTV EMI Test Procedure Report No. 4-24300W/9P-53.

The results of the tests performed are presented below.

EMI Test Performed	Frequency Range	Compliance With MIL-I-6181D
RF Conducted Interference	.15 MHz to 25 MHz	
Steady State		Yes
Transient		Slightly below
RF Radiated Interference	.15 MHz to 1.0 GHz	
Steady State		Yes
Transient		Slightly below
RF Conducted Susceptibility	.15 MHz to 10 GHz	Yes
RF Radiated Susceptibility	.15 MHz to 10 GHz	Yes
AF Conducted Susceptibility	50 Hz to 15 kHz	Yes

The results of the tests indicate that the Simplex Actuator is somewhat marginal with respect to the requirements of specification MIL-I-6181D. The actuator complies with the steady state and susceptibility requirements of MIL-I-6181D but does not comply with the transient requirements. It should be recognized, however, that the transient interference occurred only while the actuator was being driven to the extremes of its travel, a situation that would not likely occur during normal flight conditions.

13. Ultimate Loads

The prescribed utlimate loads were imposed upon the test article by blocking the actuator in the test fixture and applying the required pressure to the test actuator. At the same time the prescribed loads were applied to the manual input lever.

14. Burst Pressure

The ambient air and hydraulic fluid temperatures were increased to $275\ ^{+0}_{-25}\ ^{o}F$ and a pressure of 7500 ± 300 psig was applied to the actuator for one minute with the piston rod both fully retracted and fully extended. The actuator was operated in the parallel mode for this test. The pressure was reduced to 3000 psig when the piston rod was moved from the retracted to the extend positions.

C. Test Results

The actuator performed satisfactorily during most of the tests except for a number of seal failures, failures of check valves and inlet filters. The failures do not invalidate test results as sufficient corrective action was taken.

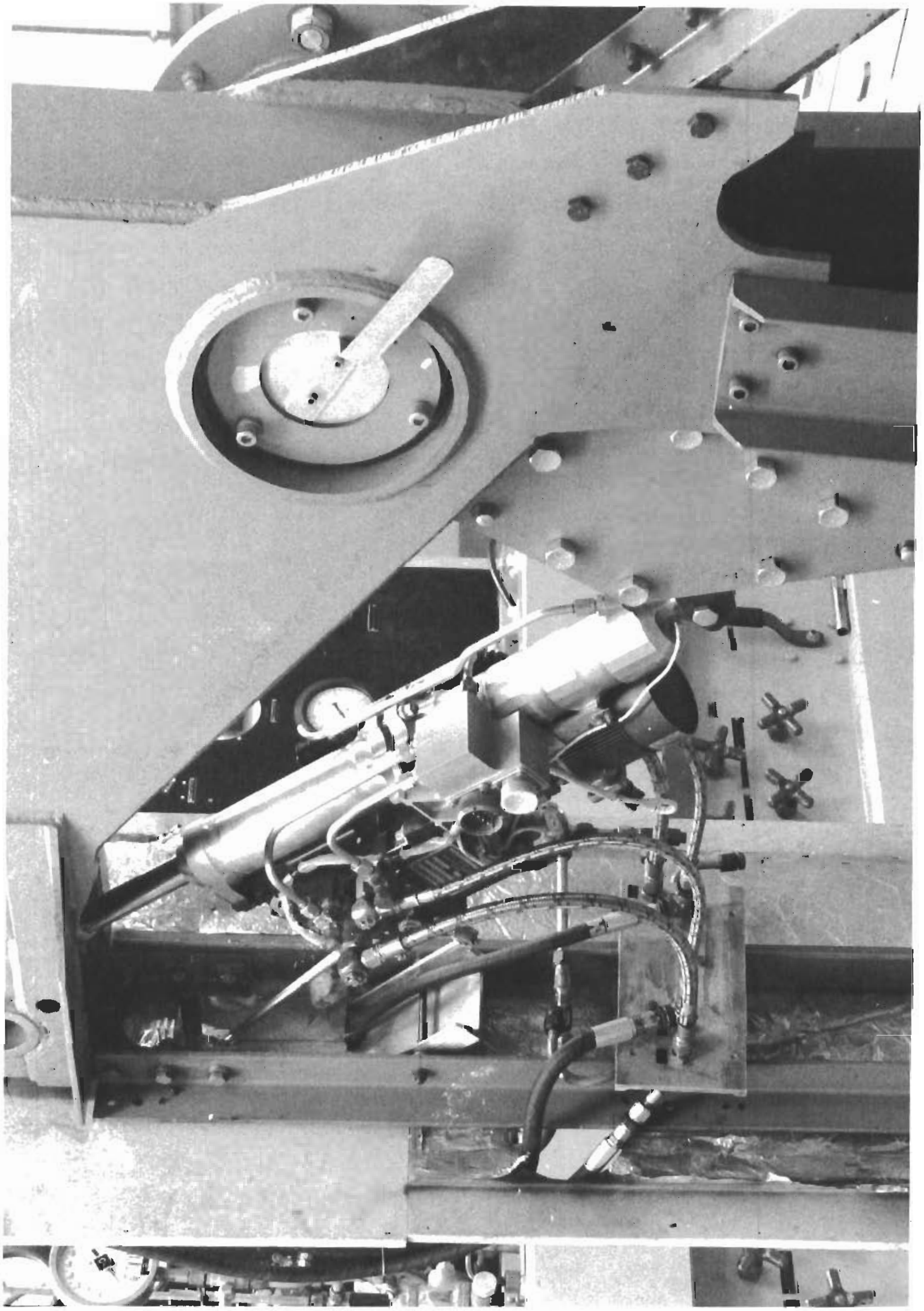


FIGURE 79. SIMPLEX ACTUATOR IN TEST FIXTURE

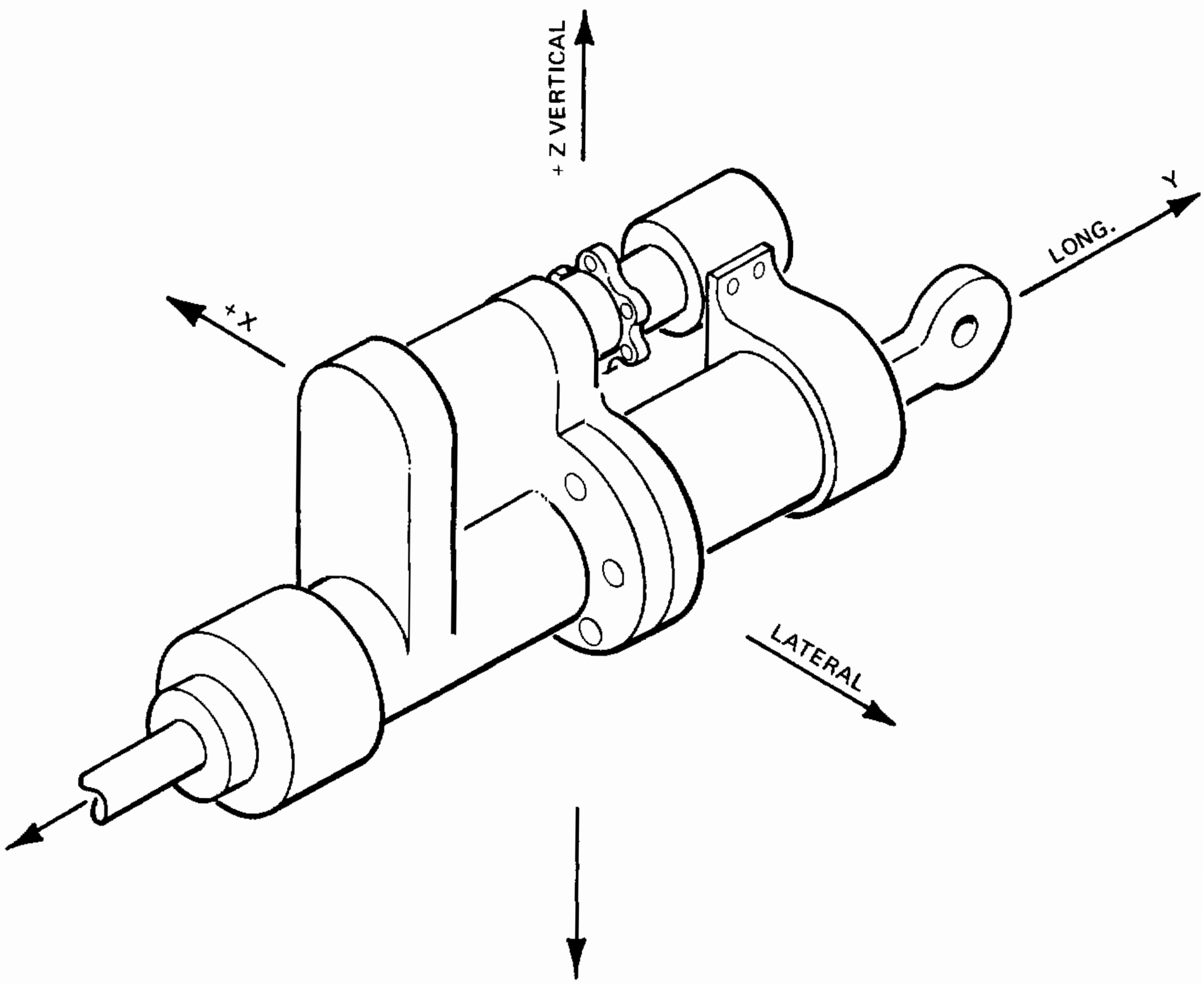


FIGURE 80. DEFINITION OF AXES

APPENDIX VII

DUPLEX FUNCTIONAL TESTS

A. GENERAL

The objective of these tests was to determine the basic characteristics of the Duplex Integrated Actuator Package. The Duplex IAP is a fly-by-wire unit having two-fail-operate capability. Therefore, operating characteristics with various failures and combinations of failures was evaluated. Specific tests which were conducted are:

1. Weight
2. Proof pressure
3. Frequency response
4. Failure transients
5. Slew rate
6. Threshold
7. Load temperature
8. Input power

B. WEIGHT

The total dry weight of the Duplex Integrated Actuator Package is 166.8. This does not include case drain heat exchangers which were added for testing purposes.

C. PROOF PRESSURE

The high pressure sections of the package were proofed by driving the input to cause the actuator to bottom out in both extend and retract directions. This constitutes maximum expected pressure.

The return sections were proofed by removing the return pressure relief valves and supplying 500 psi by an external source.

D. FREQUENCY RESPONSE

1. Amplitude Effects

To determine the effect of amplitude on frequency response, a single channel of the inner loop was examined. An input signal equivalent to a percentage of maximum output was applied to the channel. A Metrolog 204A servo analyzer was used to evaluate the amplitude ratio and phase shift. The results are as follows:

Signal Amplitude (% of max. output)	Frequency at 45° Phase Shift (H_Z)
5	40
10	30
20	30

2. Coupling Effects

The affect of various failure conditions on the frequency response of the four channel force summing unit was evaluated for a 5% signal. The results show that worst response is obtained when two channels are operating and two channels are passive and are being back driven. This is due to the inertia of the back driven motors. The conditions when one channel is hard over does not seriously affect the response since the load as seen by the operating channels is a spring load rather than inertial.

Condition	Break Frequency (H_Z)
4 channels operating	45.0
3 channels operating & 1 passive	15.0
2 channels operating & 2 passive	4.0
3 channels operating & 1 hardover	37.0
2 channels operating, 1 passive & 1 hardover	11.0

3. Total Package

The frequency response of the total package is, of course, a function of the loop gain and signal amplitude. The unit was tested at a loop gain of 11 1/sec, an output mass of 1200 in.#sec^2 to simulate a control surface, and input signal corresponding to ± .16 in. of piston output at .5 H_Z

Condition	Break Frequency (H_Z)
Both systems	1.8
PC_1 only	1.75
PC_2 only	1.75

E. FAILURE TRANSIENTS

The effects of various types of failures of the four channel fly-by-wire unit was measured by plotting the output of each of the four units together with the main servo valve and piston output. Figures 81 through 86 show the resultant main servo valve transient as a result of various failures. Figures 87 through 89 show the resultant main piston transient as a result of various failures.

F. SLEW RATE

The actuator slew rates were measured and recorded as follows:

Condition	Slew Rate (in/sec)
Both Systems	
Retract	7.65
Extend	7.03
PC_1 only	
Retract	6.90
Extend	7.28
PC_2 only	
Retract	7.66
Extend	7.15

G. THRESHOLD

The following threshold characteristics were measured with both systems operating:

Condition	Threshold (% of input)
All channels operating	.0448
3 channels operating & 1 passive	.0448
2 channels operating & 2 passive	.071
2 channels operating & 1 passive & 1 hardover	.0448

H. LOAD-TEMPERATURE TESTS

These tests were conducted to determine the steady state and/or transient package temperatures under various loading conditions.

1. Static No-Load

Figure 90 shows the package temperatures in a no-load static condition. The limited temperature is because the thermal energy is generated in the pump and is removed by the case drain heat exchanger.

2. No-Load Cycling

Figure 91 shows the package temperature resulting from cycling the unit at .5 H_Z and a 1.2 in. peak-to-peak with a mass load of 1200 in.#sec^2 and no spring load.

3. Static Load

Figure 92 shows package temperatures with the actuator holding a static load of 4000#.

4. Load Cycling

Figure 93 shows the package temperatures resulting from load cycling the actuator at .5 H_Z and an output amplitude of $\pm$ 1.6 in. and output load of $\pm$4000 lbs.

I. INPUT POWER

The standby or steady state current to the 3 Ø 400 H_Z pump motors at no external load is approximately 22 amps per phase at a power factor resulting in approximately 3.8 KW input power to each motor.

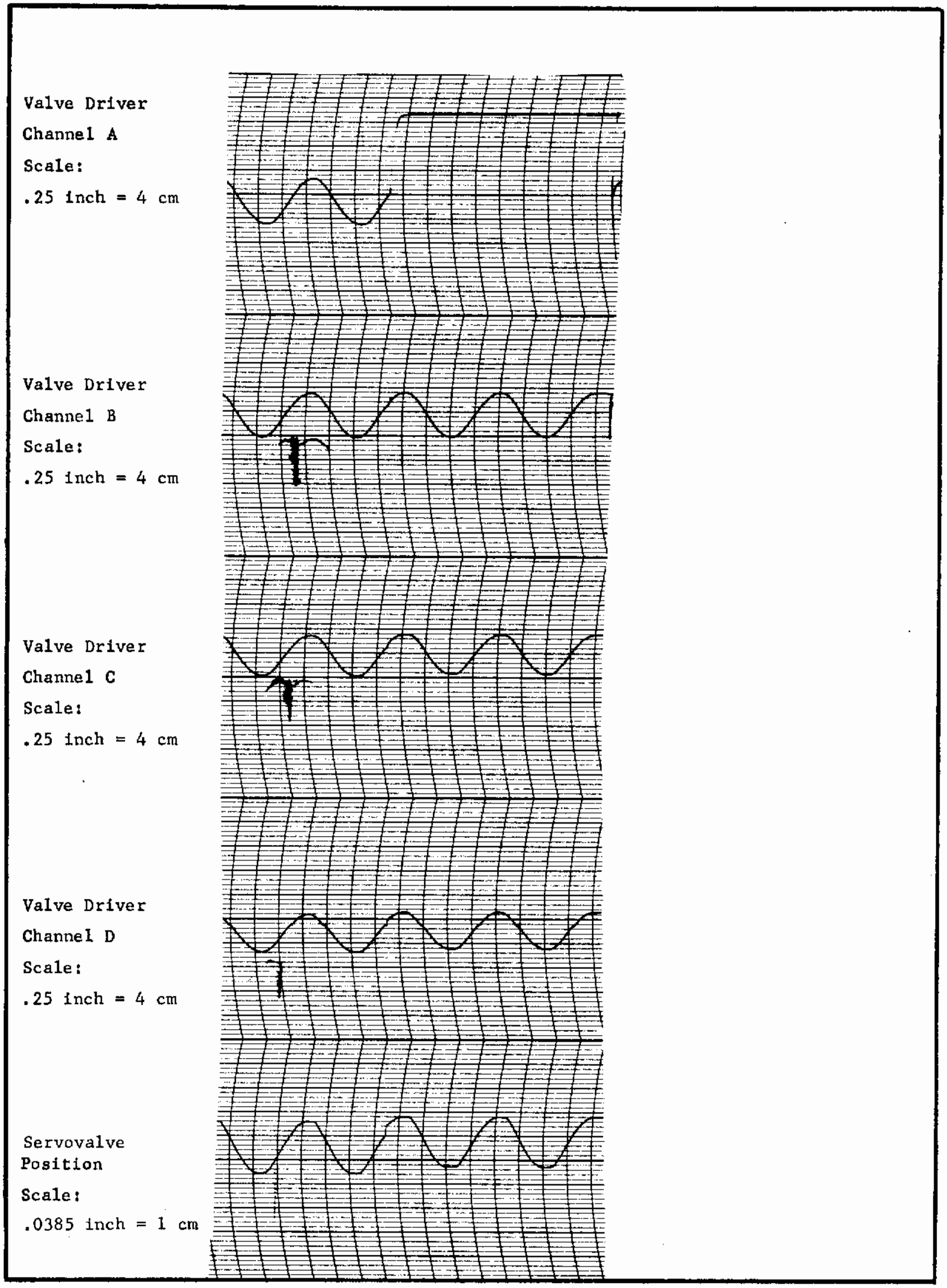


FIGURE 81. SYSTEM TRANSIENT – CHANNEL A HARD OVER BUT NOT DISCONNECTED

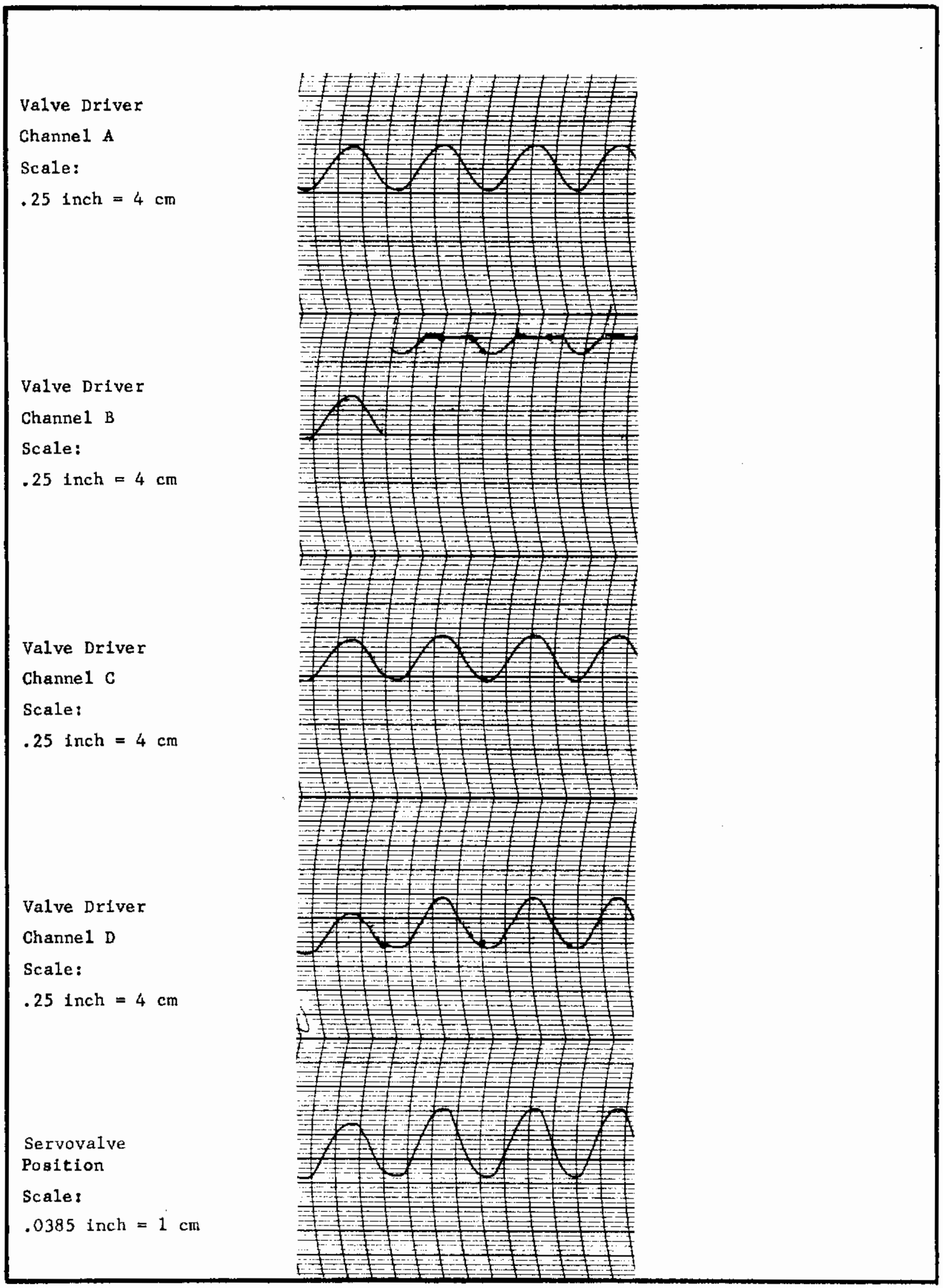


FIGURE 82. SYSTEM TRANSIENT – CHANNEL B HARD OVER AND CHANNEL D DISCONNECTED SIMULTANEOUSLY

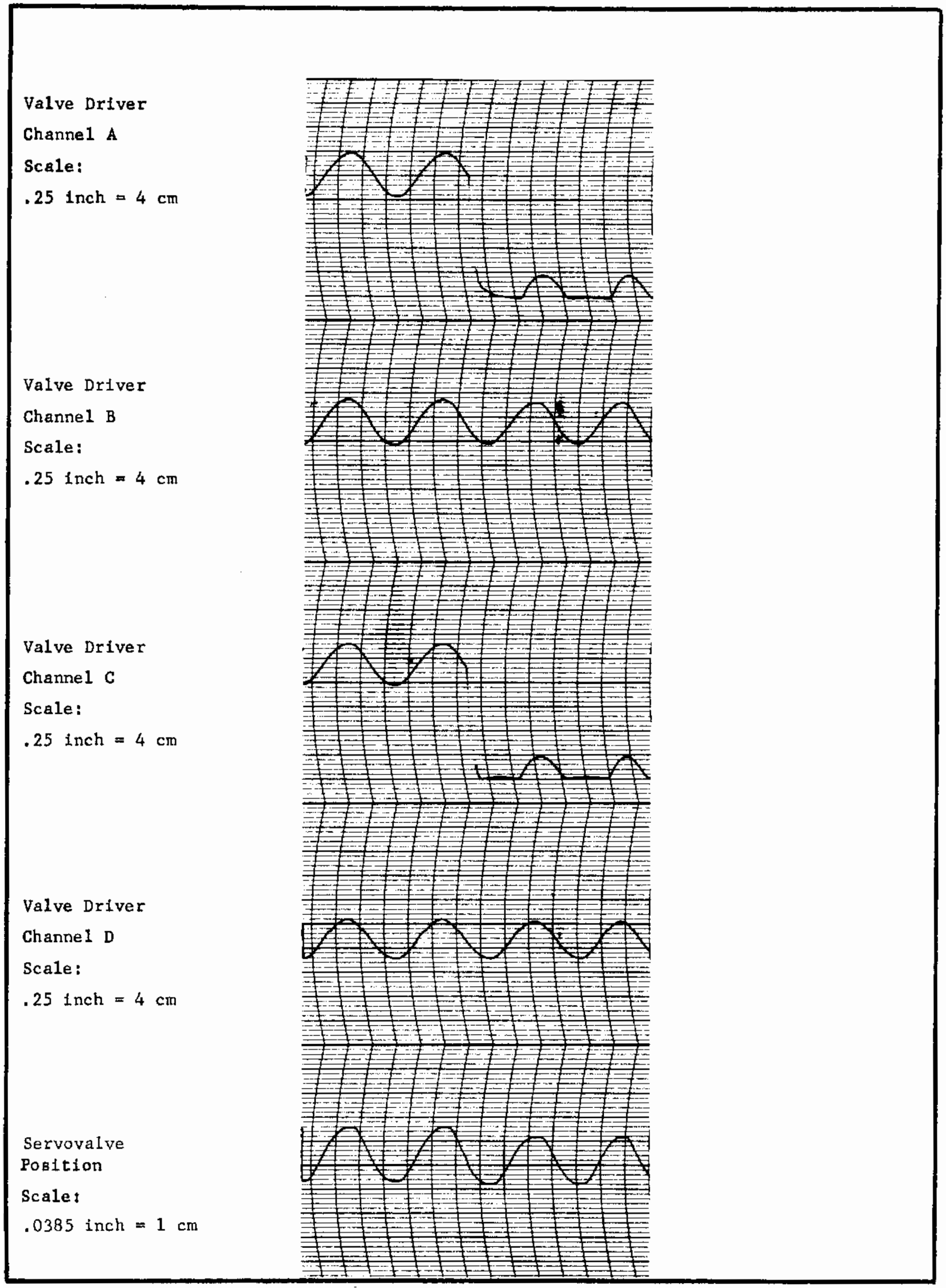


FIGURE 83. SYSTEM TRANSIENT – CHANNELS A & C HARD OVER BUT NOT DISCONNECTED

Valve Driver
Channel A
Scale:
.25 inch = 4 cm

Valve Driver
Channel B
Scale:
.25 inch = 4 cm

Valve Driver
Channel C
Scale:
.25 inch = 4 cm

Valve Driver
Channel D
Scale:
.25 inch = 4 cm

Servovalve
Position
Scale:
.0385 inch = 1 cm

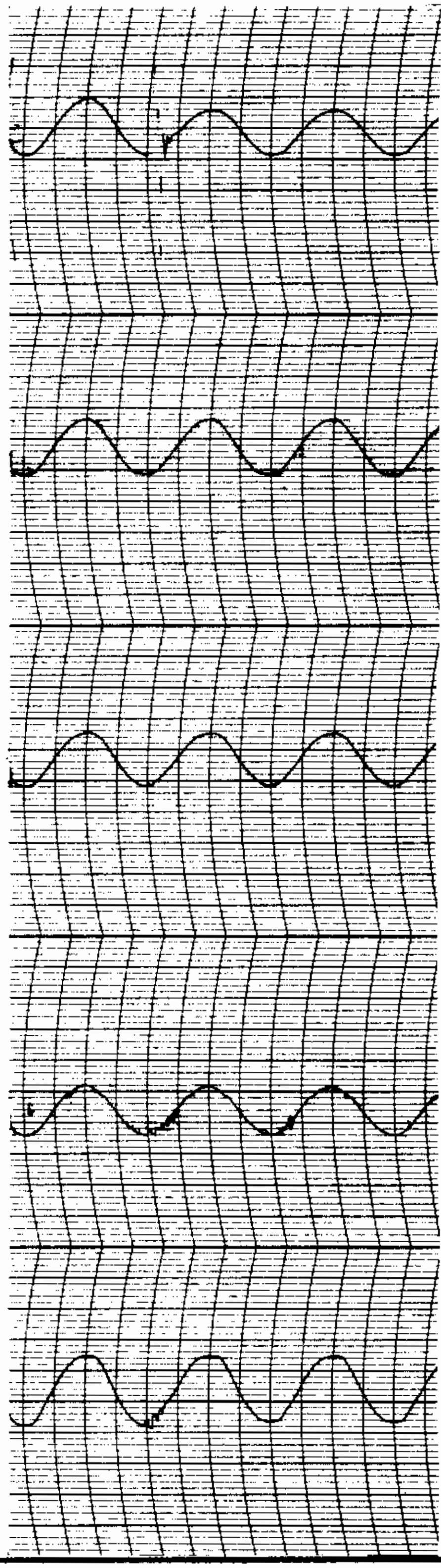


FIGURE 84. SYSTEM TRANSIENT – CHANNEL A HARD OVER TO DISCONNECT

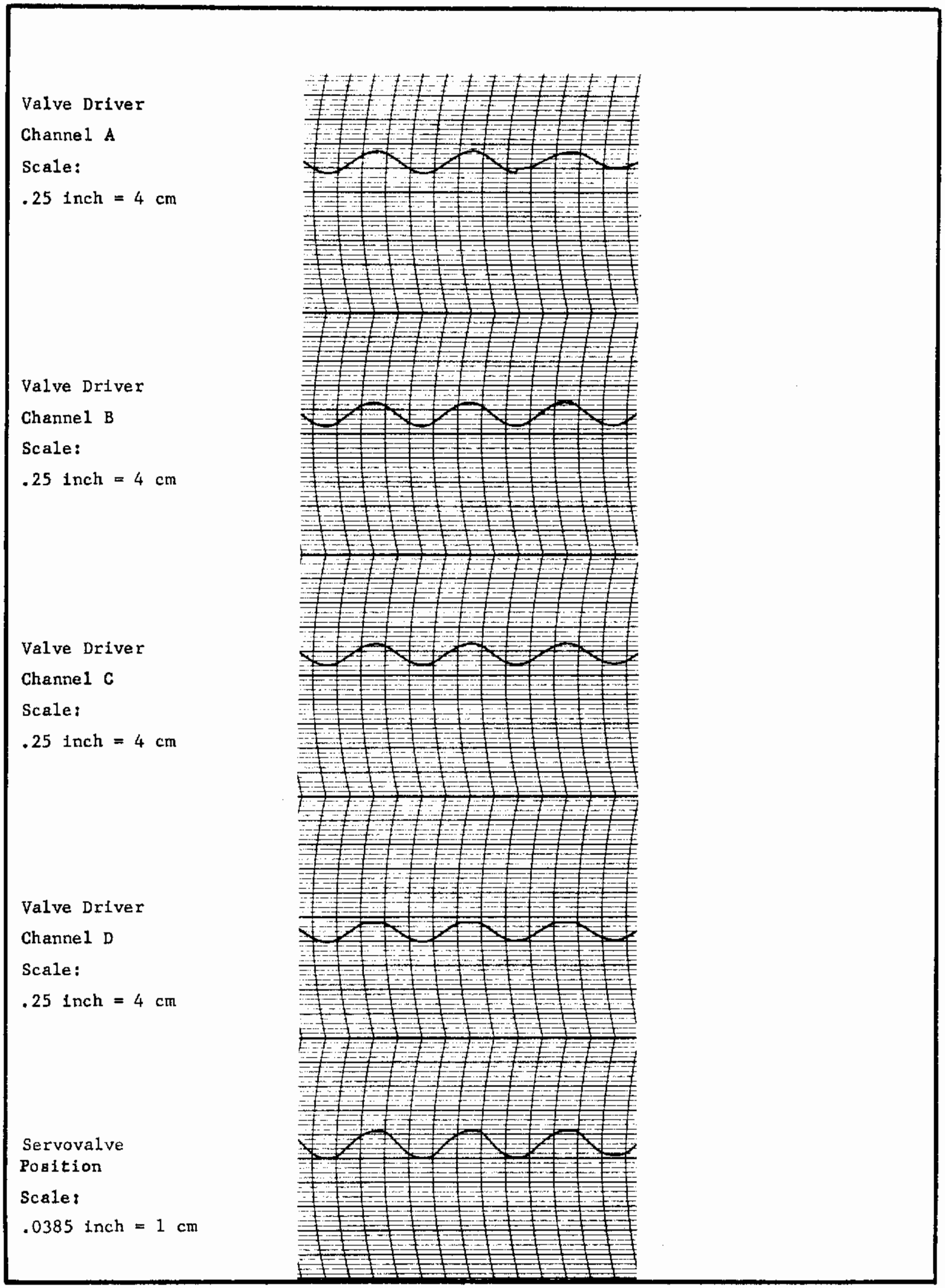


FIGURE 85. SYSTEM TRANSIENT — CHANNEL A DISCONNECTED

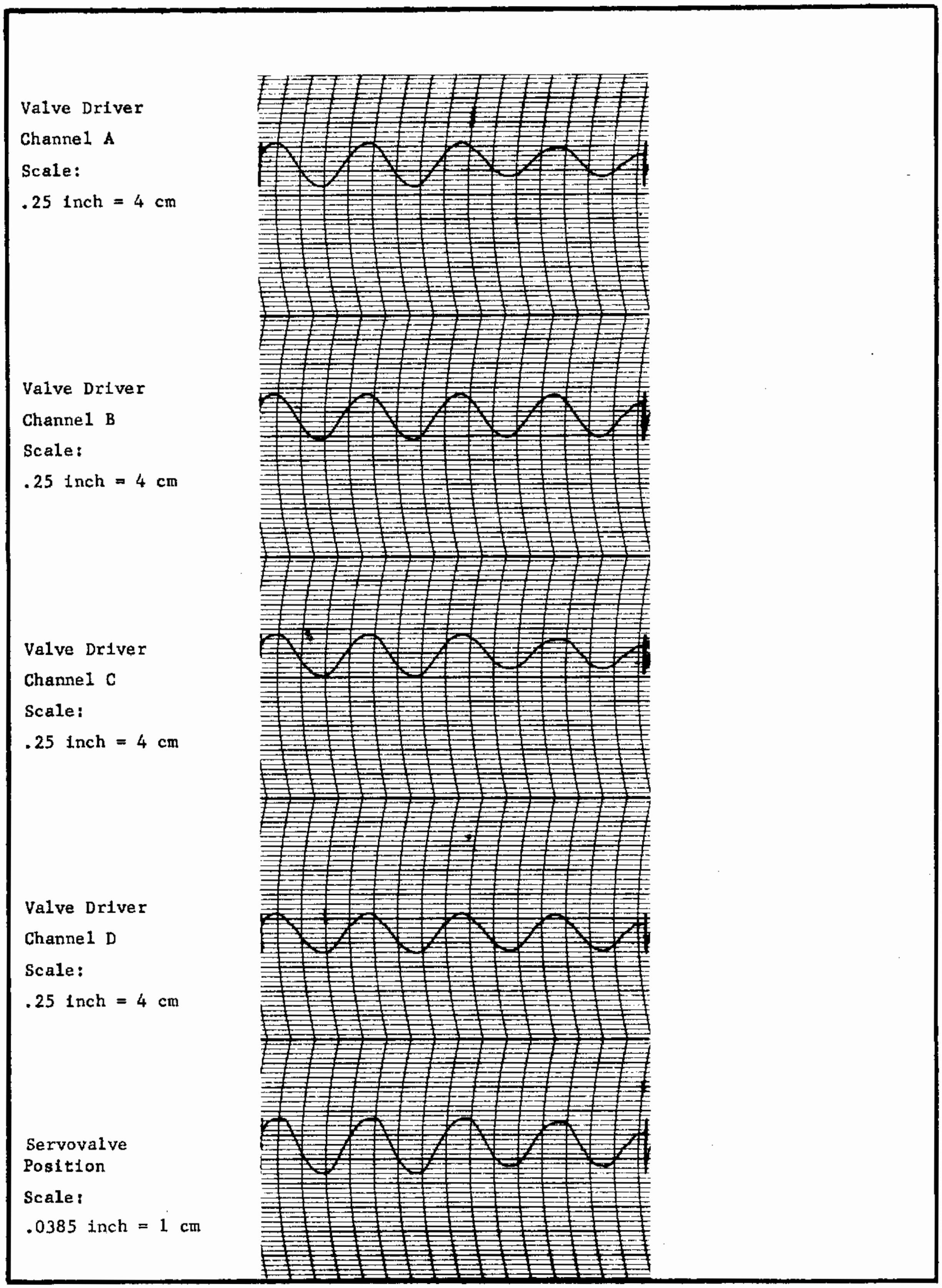


FIGURE 86. SYSTEM TRANSIENT – CHANNELS A & B DISCONNECTED SIMULTANEOUSLY

Valve Driver
Channel A
Scale:
.25 inch = 4 cm

Valve Driver
Channel B
Scale:
.25 inch = 4 cm

Valve Driver
Channel C
Scale:
.25 inch = 4 cm

Valve Driver
Channel D
Scale:
.25 inch = 4 cm

Main Ram
Position
Scale:
.062 inch = 1 cm

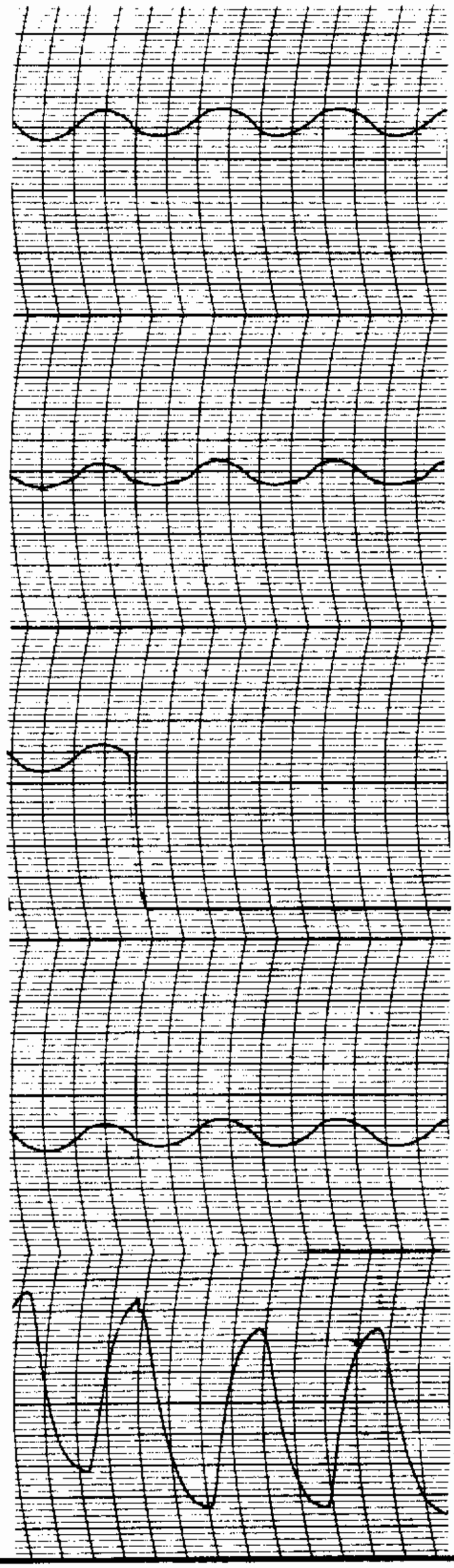
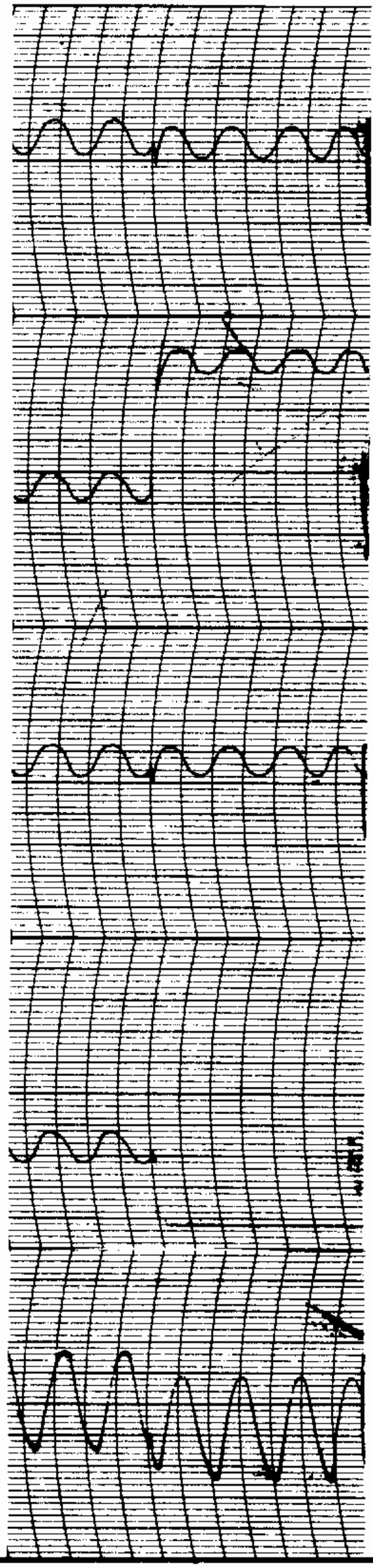


FIGURE 87. SYSTEM TRANSIENT – CHANNEL C HARD OVER BUT NOT DISCONNECTED

Valve Driver
Channel A
Scale:
.25 inch = 4 cm

FIGURE 88



Valve Driver
Channel B
Scale:
.25 inch = 4 cm

Valve Driver
Channel C
Scale:
.25 inch = 4 cm

Valve Driver
Channel D
Scale:
.25 inch = 4 cm

Main Ram
Position
Scale:
.2 inch = 1 cm

FIGURE 88. SYSTEM TRANSIENT – CHANNELS B & D HARD OVER BUT NOT DISCONNECTED

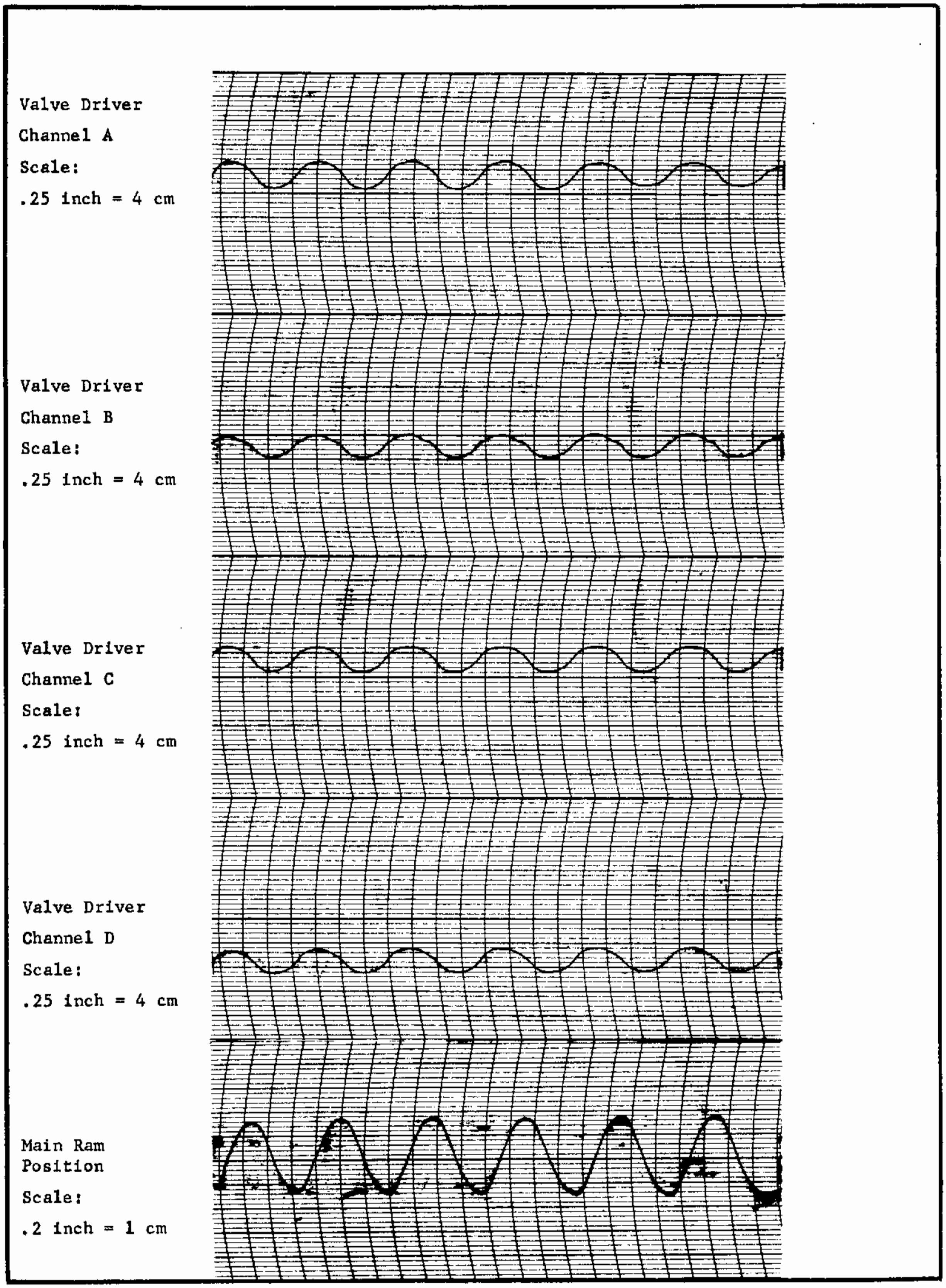


FIGURE 89. SYSTEM TRANSIENT CHANNEL D DISCONNECTED ONLY

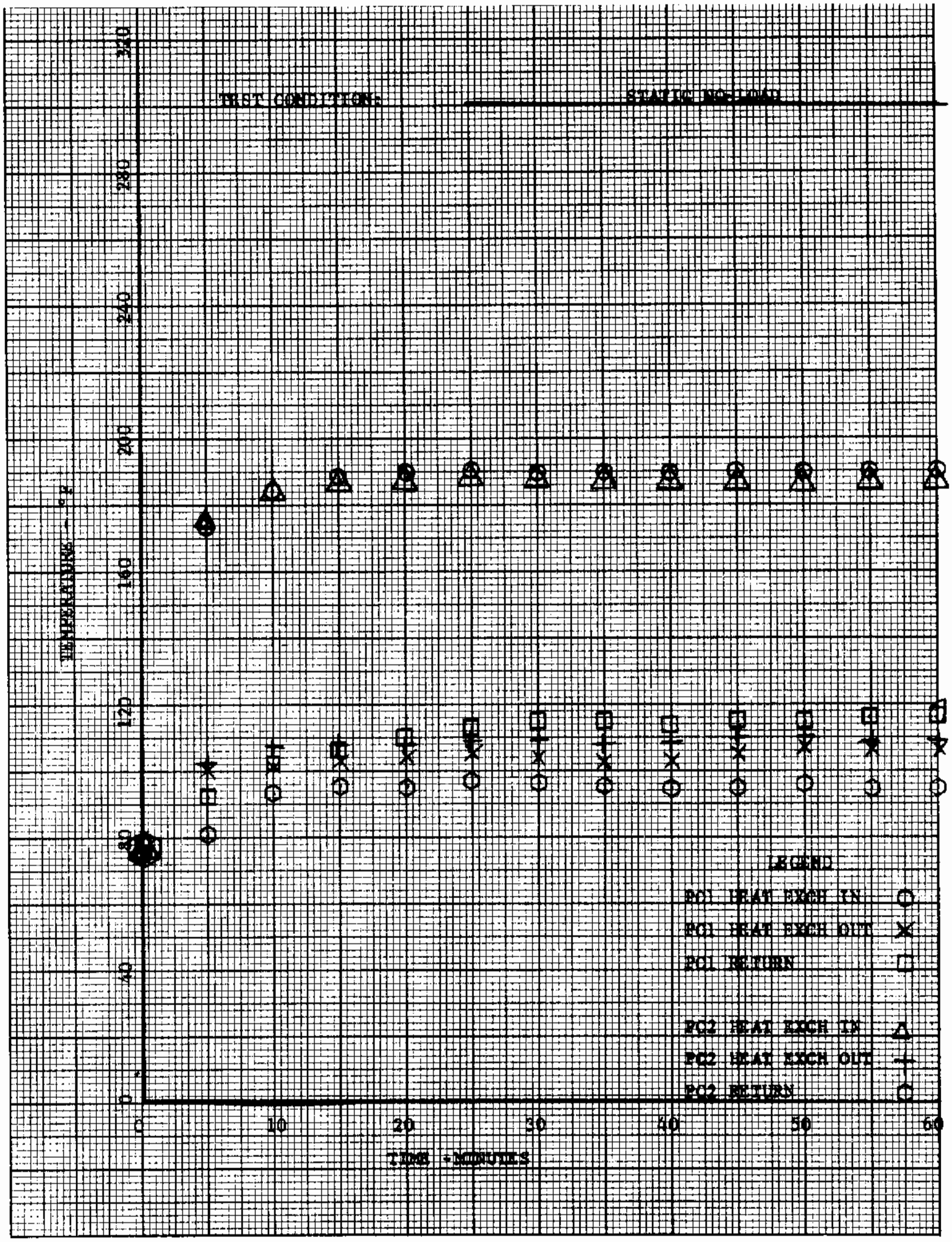


FIGURE 90. DUPLEX ACTUATOR P/N 401–15200 LOAD TEMPERATURE TEST

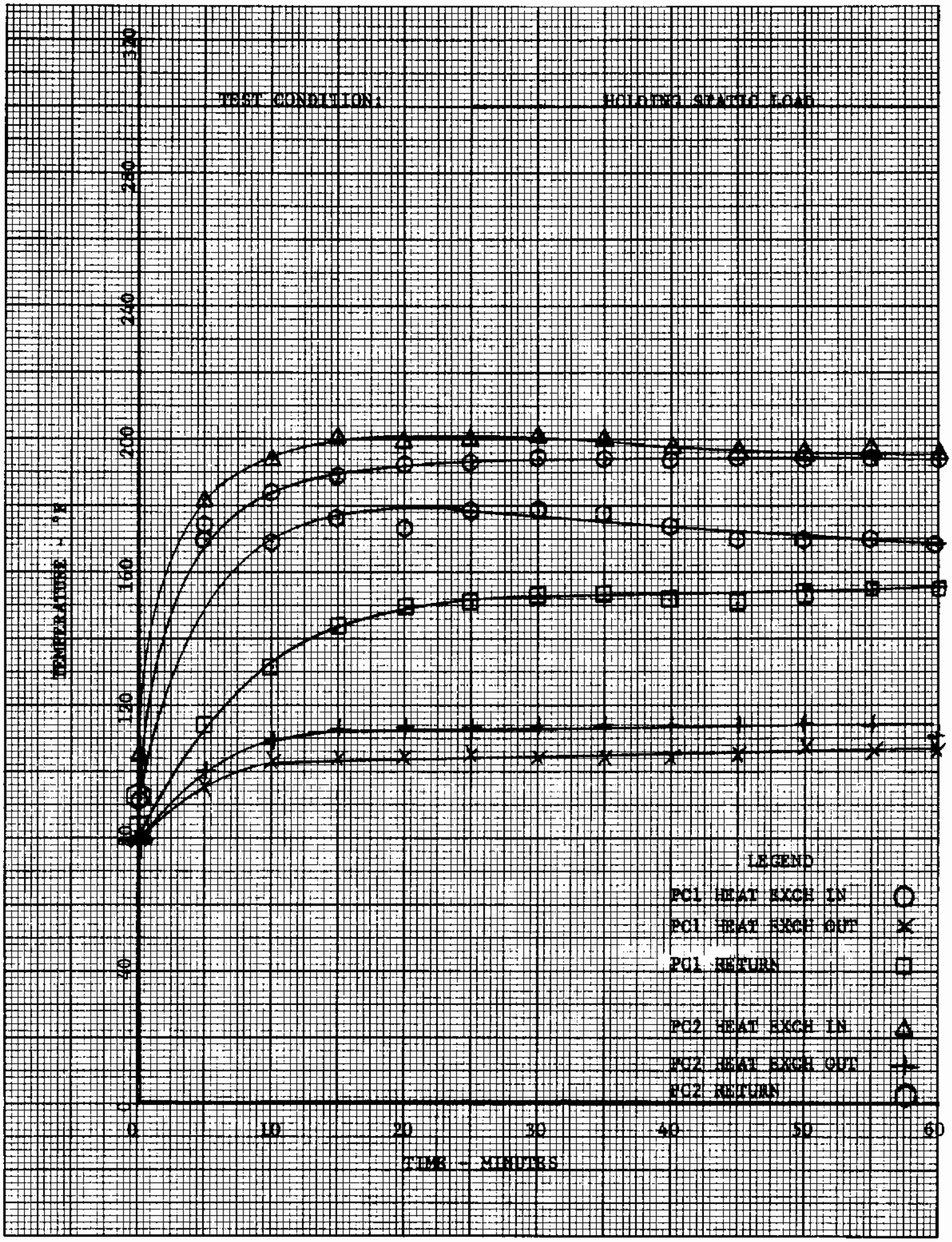


FIGURE 91. DUPLEX ACTUATOR P/N 401–15200 LOAD TEMPERATURE TEST

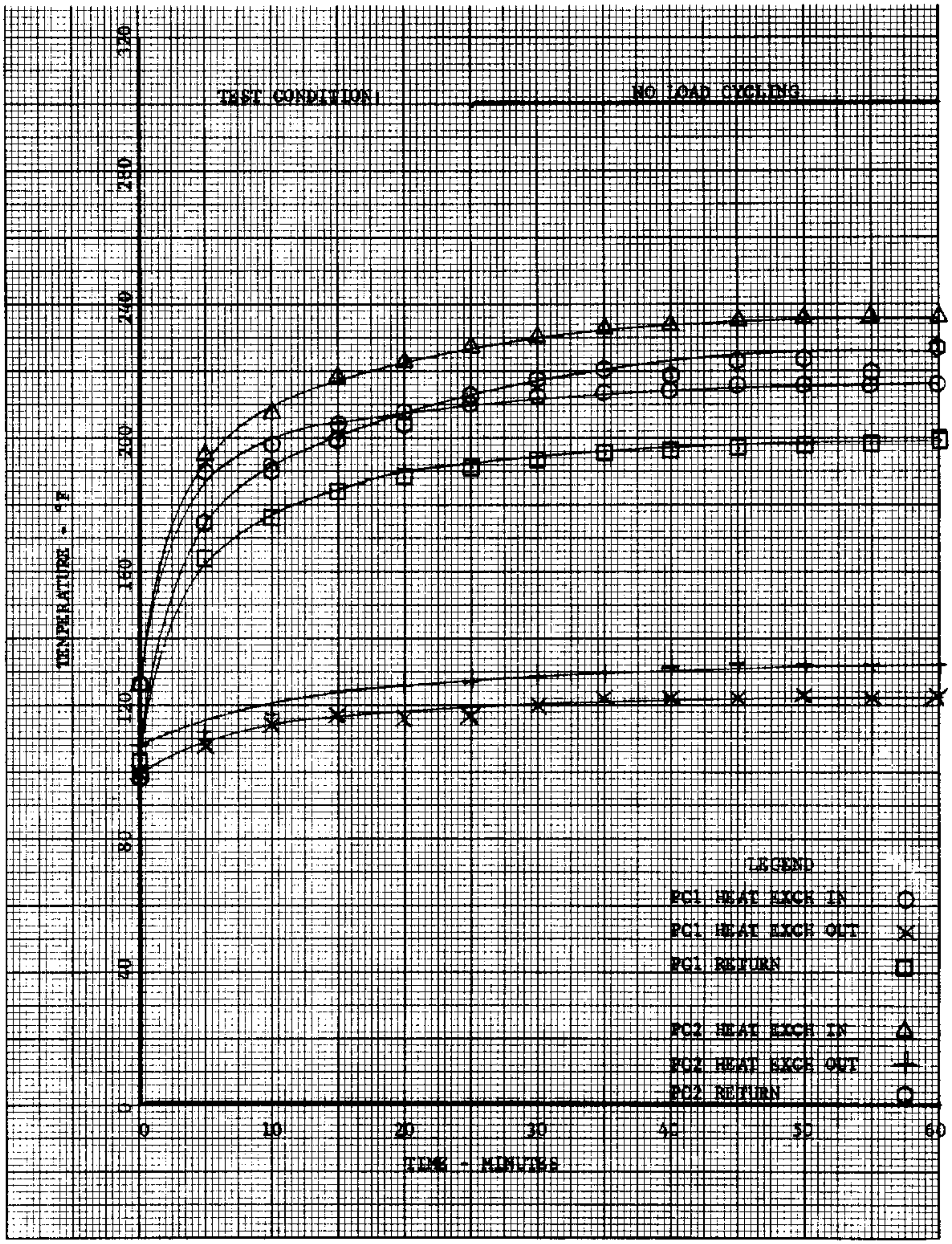


FIGURE 92. DUPLEX ACTUATOR P/N 401–15200 LOAD TEMPERATURE TEST

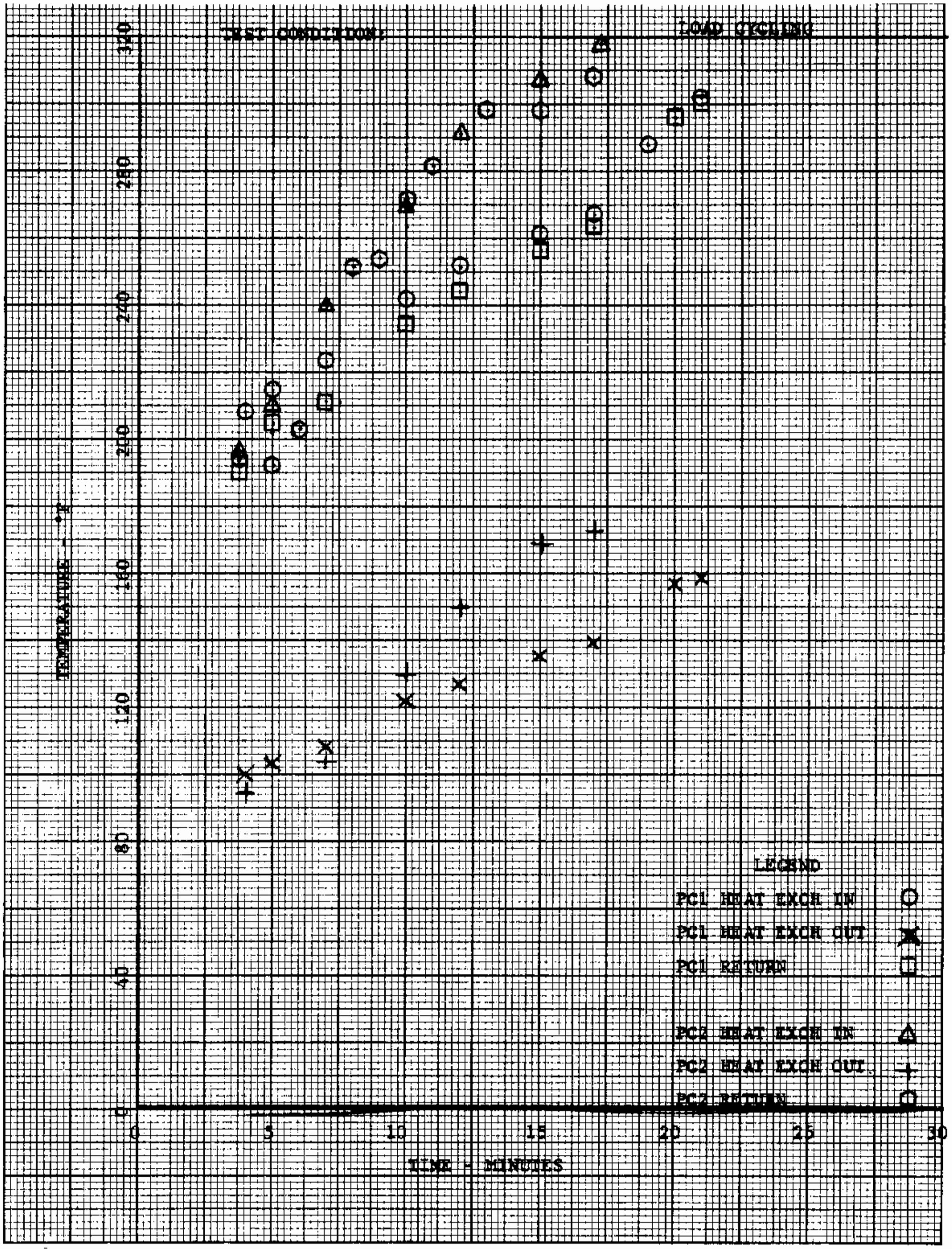


FIGURE 93. DUPLEX ACTUATOR P/N 401-15200 LOAD TEMPERATURE TEST

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13. ABSTRACT

A study was conducted to determine the feasibility of implementing present day fighter and attack aircraft with integrated actuator packages in the primary flight control sytem. The integrated package concept results in a control system having a considerably lower vulnerability to small arms fire than the conventional control system. A Simplex actuator, containing a single self-contained hydraulic supply was designed to meet the performance and structural interface requirements of the F-4 stabilator actuator. Three units were fabricated and functionally tested. One of the Simplex units was subjected to a comprehensive qualification test. The other two units were delivered to McDonnell Aircraft Company for flight testing. A Duplex integrated actuator, containing dual self-contained hydraulic systems and quadruplex electrical input channels, was designed, built, and tested. Test results demonstrate the basic feasibility of electrically signalled integrated actuator packages. Specific recommendations for further development of the integrated package concept are included.

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14. KEY WORDS	LINK A ROLE	LINK A WT	LINK B ROLE	LINK B WT	LINK C ROLE	LINK C WT
Integrated Actuator						
Power-By-Wire (PBW)						
Fly-By-Wire (FBW)						
Survivable Flight Control System						
Flight Control Actuation						
Vulnerability						

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