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SERVOACTUATOR STUDY FOR SPACE SHUTTLE APPLICATIONS

GENERAL DYNAMICS
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SERVOACTUATOR STUDY FOR SPACE SHUTTLE APPLICATIONS

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ABSTRACT

The purpose of this study was to provide designs and data to aid in the selection of aerodynamic surface control and thrust vector control servoactuators during the final definition phase of the space shuttle. The study consisted of four basic activities: establishment of requirements, industry survey, conceptual designs, and trade-off evaluation. Point design configurations were established for the flight control applications that had the greatest impact on vehicle interfaces. Parametric weight data was developed to provide trend information for those applications not specifically configured. The design configurations were to employ any type of power and control that represent current state-of-the-art technology. This included hydraulic, mechanical, electromechanical, and gas servoactuator development. Additionally, the trade off included a digital configuration for comparison to the conventional analog type. The parameters used in the comparison evaluation were — weight, reliability, maintainability, checkout capability and cost. Weight was developed in quantitative form. The remainder of the parameters were in qualitative form to provide relative comparisons of the point designs.

ACKNOWLEDGEMENTS

The author wishes to express appreciation to all the companies and individuals who responded to the industry survey and especially to the following organizations, which provided technical information specifically for space shuttle applications.

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Valencia, California

Aerospace Hydraulics Division
Parker Hannifin Corporation
Los Angeles, California

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SUMMARY

The objectives of this study were to survey the state of the art of flight control servoactuators, produce conceptual designs for space shuttle applications, and provide basic information to assist in servoactuator selection during the final definition phase of the space shuttle program.

The study was divided into four basic activities: (1) establishment of requirements and ground rules for the flight control subsystems, (2) industry survey of current state of the art, (3) conceptual design and point design configurations for various flight control applications, and (4) trade-off evaluation.

The principal subject of the study was redundancy mechanization. Failure criteria rather than reliability goals were established as a requirement. The basic failure criteria requirement was fail operate, fail safe for aerodynamic flight controls and thrust vector controls. With respect to each subsystem this criteria established redundancy as follows:

a. Aerodynamic surface controls — fail operate; fail operate, degraded performance (booster and orbiter).

b. Thrust vector controls (TVC)/engine (orbiter) — fail operate, fail to null.

c. Thrust vector controls/engine (booster — fail to null.

The industry survey included review and analysis of hydraulic, mechanical, electromechanical, gas, and digital development. Nearly all the effort and advances in the development of fly-by-wire multiple channel/system servoactuators has been on analog hydraulic servoactuators. Development work has been conducted in the other disciplines but not of sufficient magnitude to consider them equal to the state of the art of hydraulic servoactuators. An annotated bibliography report GDC-DC70-013 containing 110 references was submitted to NASA-MSC at the conclusion of the survey.

Three point designs were established for each of the following applications:

a. Booster elevator (largest load application on booster).

b. Orbiter elevator (largest load application on orbiter).

c. Orbiter aileron (smallest load application).

d. Orbiter TVC.

e. Booster TVC.

All configurations except one were either analog electrohydraulic or analog electromechanical upper servo stages controlling hydraulic power to linear hydraulic actuators. One configuration for the orbiter aileron was an analog electromechanical using spring clutch servos to engage electrical power to a ballscrew output. In addition, one digital electrohydraulic configuration was created for comparison to the aerodynamic surface control analog configurations.

Some of the prominent trends and results are as follows:

a. The electromechanical configuration was heavier and contained less output redundancy than the hydraulic configurations for the orbiter aileron. It would have to be in a stronger competitive position to be considered further because commonality would dictate that it be used on other applications.

b. The digital configuration compared favorably with respect to weight and reliability on the small load application, but became heavier for large load applications due to the penalties involved in providing large digitizer pistons. The digital concept was new and did not represent anything existing that has had the benefit of some development. As such there remains some doubt concerning its functional and performance capabilities.

c. On the largest load application (booster elevator), a four-channel, four-power actuator configuration/side weighs approximately 20% less (over 2000 lb total vehicle weight impact) than a four-channel, three-power actuator configuration. This weight difference is the result of failure criteria requirements. A three-power circuit/actuator arrangement can collectively produce 300% of required hinge moment (100% after two failures) whereas a four-power circuit/actuator arrangement can produce only 200% of required hinge moment normally to guarantee 100% capability after two failures. The increased weight totals for the three-power circuits are due to increased weights of actuators, transmission, power source, fuel, and hydraulic power generation over that of four power circuits.

d. Considering only actuators and hydraulic transmission for intermediate load applications (booster rudder and aileron, orbiter rudder), four actuators weigh less than three for loads down to approximately 10,000 ft-lb. The weight differences become small, however, and future cost effective analysis must determine the weight versus complexity cross over point.

e. In a four-power circuit/actuator configuration, for the large load applications, reducing the actuators by two and inserting switching valves obviously reduces the overall weight. However, to go one step further and reduce the power circuits by two gains little or nothing because the actuators must then double in hinge moment capability. This is based on a groundrule of no castrastrophic actuator failures for either case and only one power circuit failure for the latter case.

f. Self-monitoring mechanization of servo channels whether they be force summing or active/standby appears to result in higher reliability. The self-monitoring detection and switching for each channel is in parallel with all other channels whereas cross monitoring, which depends on cross connections of some form, is not. The reliability models used in this study were sensitive to the parallel versus series arrangement of detection and switching elements.

g. Weight and ease of installation can be improved for TVC servoactuators by reducing redundancy of power actuators and applying the failure criteria to only the servo control portion. This is reasonable due to the short operating time per flight.

h. Reducing the power actuator redundancy as mentioned in g, a servoactuator common to orbiter and booster and operating off vehicle APU power has considerable appeal. The lowered redundancy at the output makes the booster configuration (22 servoactuators) more palatable and separate engine-driven hydraulic circuits can be eliminated.

i. TVC servoactuators should employ mechanical feedback so that automatic centering (fail to null) is achieved after failure.

j. For aerodynamic surface controls, force summing mechanization of control channels using self-monitoring techniques is easily adaptable to manual detection and correction by the flight crew.

k. An electromechanical, four-control channel mechanization is versatile in that one basic unit can be used to control any combination of hydraulic circuits and output actuators.

SECTION 1

INTRODUCTION

Studies on reusable vehicles for space applications have been conducted for a decade. Parametric studies over this period have derived a concept called the space shuttle. This concept can be classified as a vehicle that combines the technology of missiles and aircraft. Aircraft flight control systems during this era have also been evolving towards "fly-by-wire," where control mechanization is electrical rather than mechanical. Larger and higher performance aircraft, survivability, and better tactical weapons delivery are primary reasons for the need to advance the state of the art of flight controls. However, until that time when the confidence and reliability of electrical control can match mechanical control, fly-by-wire systems will employ multiple channels (systems) to a more redundant level. A major part of the fly-by-wire development has dealt with redundancy mechanization of servoactuators.

The space shuttle application adds a new burden to the development of fly-by-wire servoactuators: space and re-entry environment.

The objectives of this study are to produce conceptual designs of servoactuators suitable for use on the space shuttle with supporting data that will provide the basic information for servoactuator selection during the final definition phase of the space shuttle.

The approach used was to establish point designs based on two criteria: vehicle requirements as known at the start of this study and current state-of-the-art technology. The technology includes the use of electrical, mechanical, and gas power as well as hydraulics. Additionally, the study includes evaluation of a digital servoactuator versus the conventional analog servoactuator.

A primary objective of any design is to achieve the desired reliability with the minimum number of parts. As such, redundancy is a means whereby the desired reliability is achieved. Ideally, a servoactuator would consist of a single functional path of 100% reliability. Since this is not possible, the next logical step would be to add redundancy only as required until reliability goals are met. This procedure assumes that reliability goals are known, can be properly apportioned, and failure rate data is accurate. Unfortunately, failure rate data for nonelectronic components is subject to error.

The principal subject of this study is the application of redundancy mechanization techniques in servoactuators applicable to the space shuttle. To accomplish this purpose and also to eliminate dependency on absolute reliability numbers, the requirements are stated in terms of failures permitted for mission success and/or mission safety (e.g., fail operate, fail safe). This approach is conservative and generally defines the limit of complexity required.

This introduction is Section 1 of this report. Section 2 covers the industry survey. separate annotated bibliography report, GDC-DCB70-013, includes references used herein and a glossary of terms commonly used in redundancy mechanization discussions. Section 3 includes a vehicle baseline description, servoactuator requirements, and ground rules and assumptions needed to establish point designs. Section 4 is a discussion on various techniques used in servoactuator development and lists the candidate point designs and rationale for selection. Section 5 displays the failure modes and effects analysis of each design. The FMEA is a backcheck on adequacy of design to meet failure criteria requirements and an expose' of weak elements within a design. Section 6 establishes the basic parameters and data used for tradeoff evaluation. Section 7 covers the tradeoff evaluation. Section 8 includes the conclusions and recommendations.

SECTION 2

INDUSTRY SURVEY

2.1 GENERAL

As a part of the industry survey, approximately 45 letters were sent to 36 companies. Follow-up telephone conversations and letters were made primarily to those companies that responded to the original survey letters. Activity in this regard included many personal contacts with representatives of the responding companies. In addition the Defense Documentation Center, Alexandria, Va., and the NASA linear tape were contacted for data retrieval. The time period covered was mainly 1960 to 1970.

The survey yielded approximately 150 documents of which 110 are included in the annotated bibliography, report number GDC-DCB70-013. The documents may be broken down as follows:

Government-sponsored reports	22
Technical papers and articles	29
Industry-sponsored reports	38
Miscellaneous	21

Information gathered is not exclusively on "multiple fault correcting" servoactuator development. It also includes general design data that helped form parametric data used in the evaluation.

2.2 STATE OF THE ART

2.2.1 <u>GENERAL</u>. Hydraulics dominate the scene not only in power control but also in application of redundancy mechanization techniques. Of the 110 documents described in Section 2.1, approximately 75% were concerned with the application of hydraulics. Development of redundancy mechanization is almost solely a hydraulic venture. The reason for hydraulic technology dominance is rather obvious: much of the recent development efforts have been based on conversion of existing aircraft flight control systems to fly-by-wire, where a configuration was already in existence employing hydraulic power.

"Redundancy mechanization" is a term used herein that describes special techniques used in implementing fly-by-wire control. For many reasons (survivability, better weapons delivery, increased manual control complexity, increased structure life, etc.) flight control systems are evolving towards fly-by-wire. A larger burden is placed on the servoactuator to accomplish more functions. As electrical command replaces mechanical command, redundancy is increased to offset the loss of inherent reliability of a manual control system. In a pure fly-by-wire system it is logical to combine the electrical command conversion devices with the power control actuator. This total package is termed the servoactuator. Therefore, within the development of fly-by-wire control, the servoactuator must accept multiple electrical command signals in addition to multiple power systems and produce a proper output. Due to anticipated failures and fast reaction time required, the servoactuator usually must also incorporate logic to accomplish automatic fault correction. The techniques used to accomplish this staggering task is therefore called redundancy mechanization. The following paragraphs briefly outline some of the development effort that has been accomplished or is in progress.

2.2.2 HYDRAULIC APPLICATIONS. Many programs have been sponsored by the Air Force Flight Dynamics Laboratory (AFFDL). A triple redundant actuator was developed by Weston Hydraulics for a program conducted by Sperry.[1,2*] The unit used "force summing" of the three active command inputs with an electronic model to give two fail operate capability. Hydraulic Research developed a quad redundant servoactuator that was installed and flown in a B-47, accumulating approximately 18 flight hours in fly-by-wire mode.[3,4] The actuator was an active/standby arrangement using hydraulic logic for fault detection and had two fail operate capability.

Hydraulic Research also built a servoactuator for the TWEAD program.[4] This actuator is installed on an F4C. The control package is fail operate, fail safe and performs as a series servo in conjunction with manual control. Bertea built a fail operate actuator that employs a hydro-mechanical force summing technique called mid value logic for Douglas as a part of a program to evaluate a redundant fly-by-wire system.[5] The 680J program in quest of survivability as well as fly-by-wire has been responsible for continuing development. LTV electrosystems built a duplex integrated package that incorporated two hydraulic power supplies with the servoactuator. The four control channels in this unit use electromechanical rather than the conventional electrohydraulic signal conversion techniques.[6] The four electromechanical actuators are force summed. LTV is now developing an electromechanical signal conversion control that uses servo motors and differentials in a velocity summing arrangement. In each case the outputs from the control portion operate conventional hydraulic power spools and actuators. In addition to the many development programs that resulted in hardware either for ground (non flight rated) or flight test, AFFDL has sponsored several studies that did not carry through to the hardware stage but are definite contributions to the fly-by-wire development.

The SST is another activity that has contributed to servoactuator development. Bertea is now working on a quadruple system for the horizontal tail.[7] The configuration has four hydraulic power systems and four electrical command inputs that sum with mechanical inputs from the pilot. The configuration is not purely fly-by-wire but electrical command redundancy is quadruple because electrical control is considered mandatory for safe flight.

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*Numbers represent references at the end of the report.

Moog developed a triple redundant servoactuator for the Saturn S-IVB stage.[8] The redundancy in this case is limited to the servo channels. Detection-correction is not used in the unit. Fault correction consists of a failure in a control channel being overridden by the two good channels. The failed channel is not de-activated. The actuator provides fail operate capability and can tolerate many combinations of dual faults. Moog refers to this design as a majority voting servoactuator.

The programs and hardware listed above are by no means the total effort that has been pursued in the development of hydraulic servoactuators, but do represent some of the more prominent efforts.

Electronic models have been used as a substitute for an electrohydraulic servo channel to provide intelligence for comparison of signals. This was quite often necessary in an aircraft equipped with two hydraulic systems but requiring fail operate capability in the servo control portion. Fail operate normally requires a minimum of three signals for comparison. There seems to be some disagreement as to the effectiveness of using electronic models. Electrohydraulic servos have been modeled; however, there has been difficulty in simulating wear, temperature effects, and fluctuating load of an electrohydraulic servo valve.

2.2.3 MECHANICAL/ELECTROMECHANICAL APPLICATIONS. Mechanical and electromechanical are grouped together because of their similarity. The servoactuator or actuating device is essentially the same in either system. The major difference is in power transmission. A pure mechanical system uses mechanical shafting from the power source to the servoactuator. An electromechanical system uses electrical power transmission and electric motors at the servo.

The development of electromechanical servoactuators for primary flight controls and specifically fly-by-wire flight controls is not as advanced as its hydraulic counterpart. Development has been progressing but practically no hardware testing has been accomplished on multiple system servoactuators for large load applications. The primary advantages an electromechanical system has over hydraulics is long term storage at cold temperature, elemination of fluids and seals, and adaptability to environmental extremes.

Curtiss Wright designed and built an all-mechanical aileron control system for the North American F-100 simulator.[9] Mechanical shafting delivered power from the power source to the servo clutch. A torroidal clutch was used to clutch in power on command to power hinge actuators located at the aileron. This system was "flown" on the simulator for evaluation. The conclusions were that the mechanical system servo performance capabilities compared favorably with those of a hydraulic servoactuator.[10]

There have been many electromechanical servo applications primarily for small power applications. Examples of these applications are:

Marc Condor missile steering:	Spring clutch servo
Mark 37 torpedo fin control:	Magnetic clutch servo
QH-50C helicopter:	Magnetic clutch servo
Sergeant missile:	Magnetic clutch servo
KC-135 — horizontal stab. manual trim:	Magnetic clutch servo
Apollo service module — engine gimballing:	Magnetic clutch servo

None of these applications would classify as automatic-fault-correcting servoactuators although the Apollo system has a redundant power supply, command, and servo clutch arrangement that can be switched in manually.

2.2.3.1 Servo Clutch. The heart of the electromechanical servoactuator is the servo clutch. Figure 2-1 is a simplified schematic of an electromechanical arrangement. The servo clutch is analogous to the electrohydraulic servo and power spool in a hydraulic system. The three basic types that have been used in servo applications are the torroidal clutch, spring clutch, and magnetic clutch.

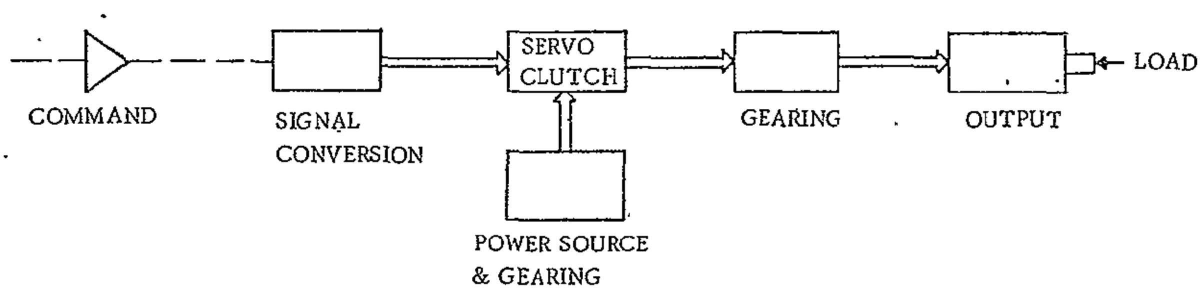


Figure 2-1. Electromechanical System

The torroidal clutch has a continuous rotating input shaft, variable rollers, center torroid, and differential rollers attached to the output shaft. See Figure 2-2. In the position shown there is no rotation of the output shaft. A command input changes the angle of the variable rollers which in turn causes the center torroid to change speed with respect to the input shaft. This causes rotation of the differential rollers and

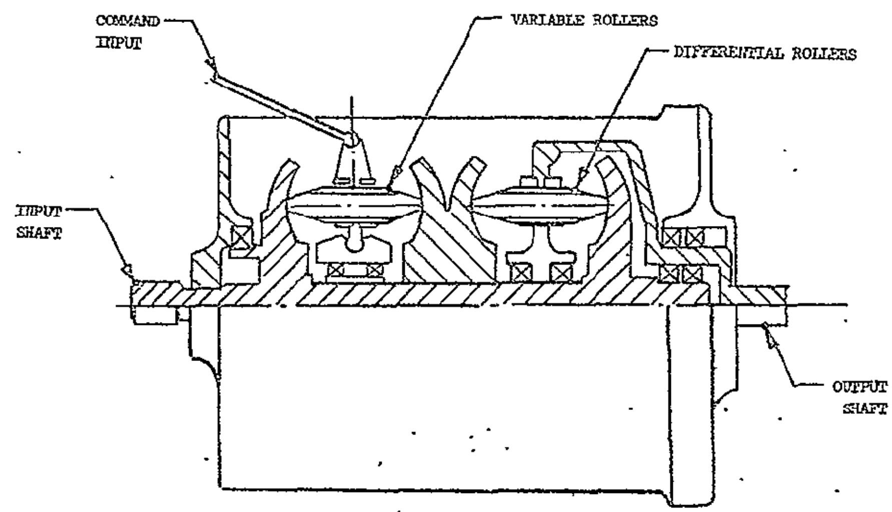


Figure 2-2. Torroidal Clutch

output shaft. The unit is bi-directional and has good resolution. Disadvantages of this device are the requirement for a lubrication sump, high heat generation, and high force inputs for large step inputs due to the gyro effect.

The spring clutch consists of an input shaft, spring energizer, two springs to give bi-directional output, an output shaft, and brake as shown in Figure 2-3 (only one spring assembly is shown). The spring clutch is by nature an on-off device. When on, the clutch transmits power at full rate. Problem areas for this device include lowered response as size increases (spring inertia), dead zone (spring engagement motion), and high threshold (there must be positive clutch engagement). It is attractive in that there is no appreciable heat generation. To eliminate the first two problems the unit can be designed such that the spring rotates with the input shaft and clearances reduced so that spring engagement travel is minimized. This, however, eliminates the brake from the design which in effect disconnects the output from ground when there is no error signal.

An example of the magnetic clutch is shown in Figure 2-4. When the clutch coil is energized, the magnetic field is produced, crosses the powder gap, and causes radial alignment of the iron particles in the gap forming chains across the gap. These chains couple input to the output (drive disc) where the output torque is proportional to input current. This type clutch has been applied successfully, as mentioned before, for

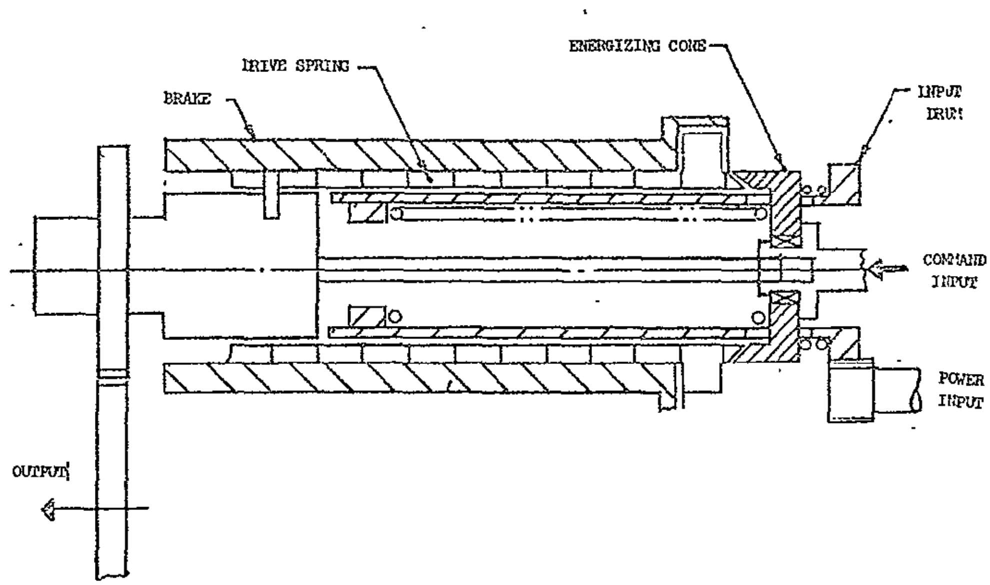


Figure 2-3. Spring Clutch

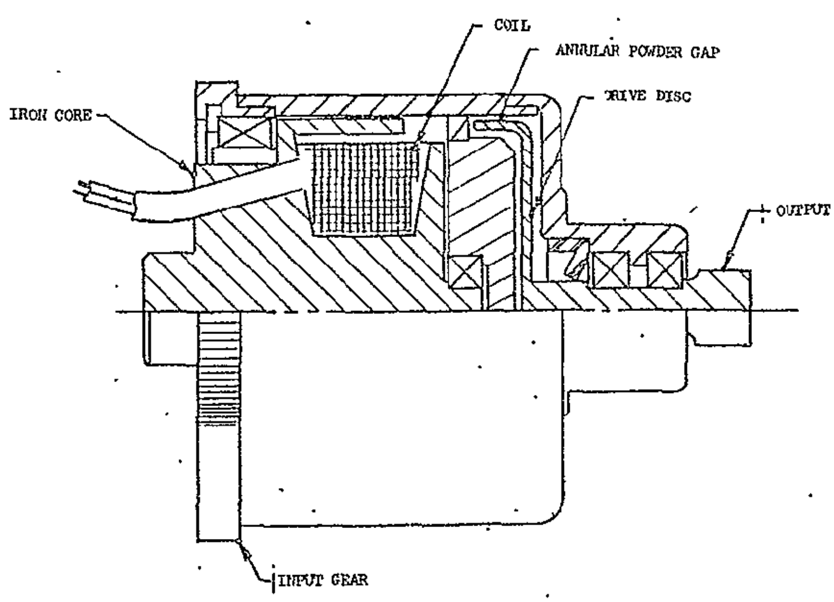


Figure 2-4. Magnetic Particle Clutch

small power applications. However as size increases response lowers as seen by observing Figure 2-5. Also there are problems in repeatability where experience has shown that due to shearing action the magnetic particles start to lose their effectiveness and the output torque versus input current relationship changes after repeated cycling. The clutch would suffer severe heating problems unless a brake is used when applied to control of aerodynamic surfaces because the clutch absorbs power when holding against an external load.

2.2.3.2 Harmonic Drives. A harmonic drive is basically a gear reduction device. However, there is some development effort to adapt this technique to servo control. In addition to describing 20 operational applications of harmonic drives, Reference 11 describes a potential electromagnetic servo solution. Figure 2-6 is a schematized version of the device. "A series of electromagnets are positioned about the flexspline, and are progressively excited to produce a rotating field vector which serves to radially deflect the low inertia flexspline. Mechanical rotation occurs only at the low speed circular spline and output shaft. This type of structure appears capable of very high acceleration rates well in excess of those obtainable with state-of-the-art electromechanical drives and servoactuators."[11]

2.2.4 GAS APPLICATIONS. Some development work has been accomplished on pneumatic servoactuators but very little hardware is in existence. Bendix conducted a study for the Air Force Flight Dynamics laboratory evaluating the feasibility of

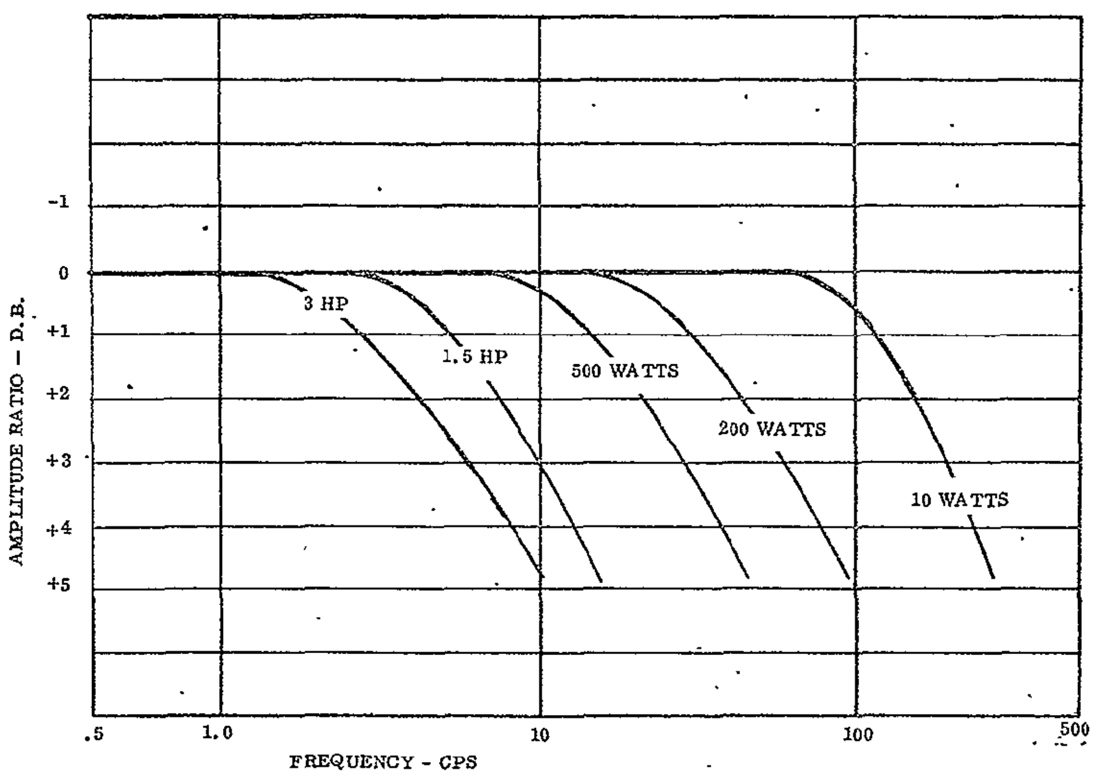


Figure 2-5. Typical Frequency Response Curves — Magnetic Clutch

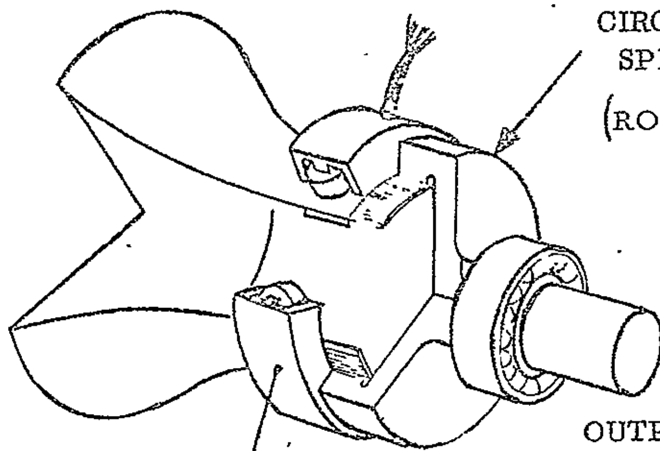


Figure 2-6. Electromagnetic Harmonic Drive

applying the Bendix Dynavector* actuator (operating on 50 psig compressor bleed air) to actuate flight control surfaces.[12] A pneumatic servoactuator was built and tested by Parker-Hannifin under contract to NASA-MSFC.[13] The unit was designed to use -250°F hydrogen gas. Weston built and tested a pneumatic digital servoactuator under contract to NASA-MSFC. This unit was also designed to use -250°F hydrogen gas.

The work on pneumatic development has been essentailly to advance the state-of-the-art of pneumatic servos to the level that hydraulics reached nearly 20 years ago. As such there is no specific correlation between pneumatic servo development and fly-by-wire development.

2.2.5 DIGITAL APPLICATIONS. Most of the development effort on digital servo actuators has been in the field of hydraulics. The industry survey did not disclose any effort conducted on multiple channel servoactuators.

*Trademark of the Bendix Corporation

There is very little operational hardware in existence. The Convair 990 commercial jet uses a digital open loop arrangement for low rate trim of the horizontal stabilizer. There have been feasibility studies and breadboard models tested of single channel digital servos.[14,15,16]

Digital servoactuators fall into two broad categories and are classified in relation to the input signal characteristics.

a. Incremental — The input signal is in the form of a series of pulses sometimes referred to as a pulse train. In this case a hydraulic control unit might consist of a single valve, digital pump, and associated sequencing and polarity valves as shown in block form, Figure 2-7.

b. Parallel-binary — The input signal is in the form of parallel or simultaneous signals to multiple coils of a valve or to multiple valves. An example of a multiple coil valve is shown in Figure 2-8. Each coil produces twice the output of the adjacent coil. The smallest coil is sized to produce the smallest incremental change required. Thus, the valve can receive one or more pulses in the same time interval and produce an output proportioned to digital input. The digital/analog interface is at the first stage of the valve in this case. Another example of parallel mechanization is shown in Figure 2-9. Here the unit is completely in digital form to the output actuator. The actuator in this case is arranged in binary parallel form.

An incremental type digital servoactuator has less severe failure modes than its analog equivalent in that a hardover signal is only one pulse or one incremental change at the output. However, this type is usually severely rate limited because of the limiting valve cycling frequency and the small incremental change per cycle required to achieve positional accuracy.

The parallel types begin to resemble analog with respect to failure modes. The worst hardover in this case is equivalent to a half hardover in a conventional analog servoactuator.

The digital strut appears to be impractical when applied to large load applications because of the size and complexity of the output actuator.

2.3 SUMMARY

Table 2-1 summarizes in general the state of the art for the various disciplines. A conspicious trend is the absence of multiple systems (redundancy mechanization) development except in hydraulic analog servoactuators.

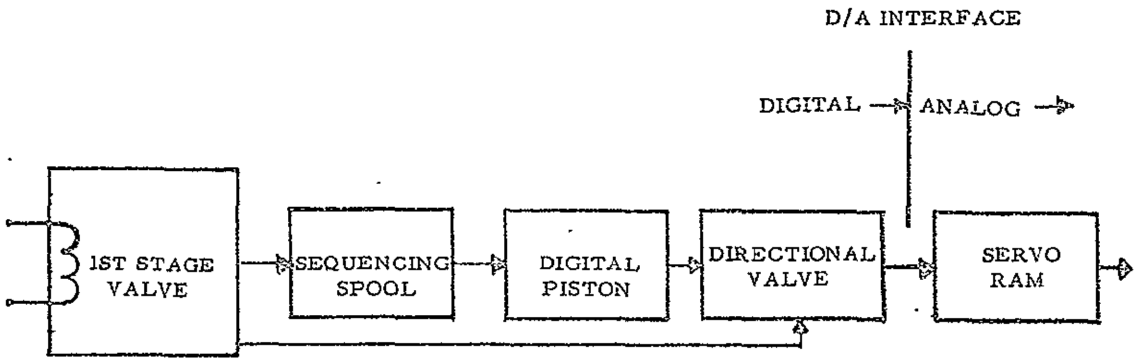


Figure 2-7. Incremental Digital Servo

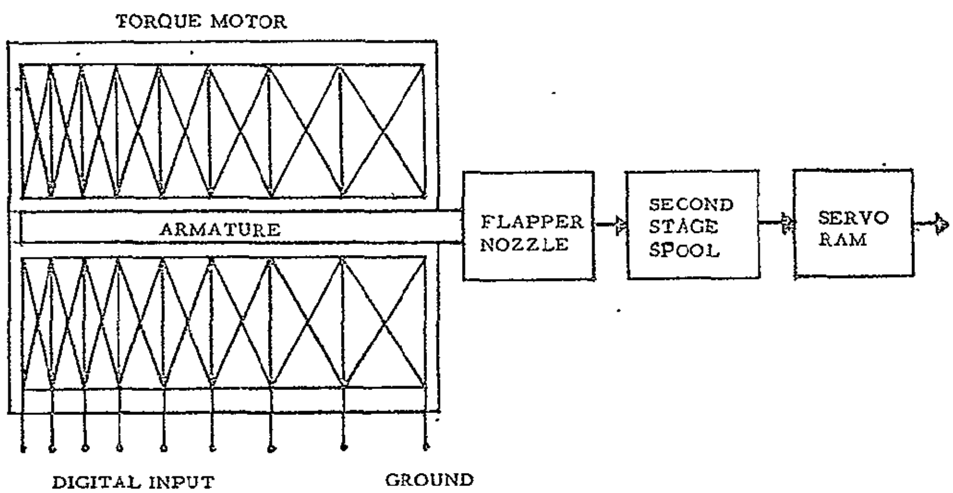


Figure 2-8. Parallel Digital Servo

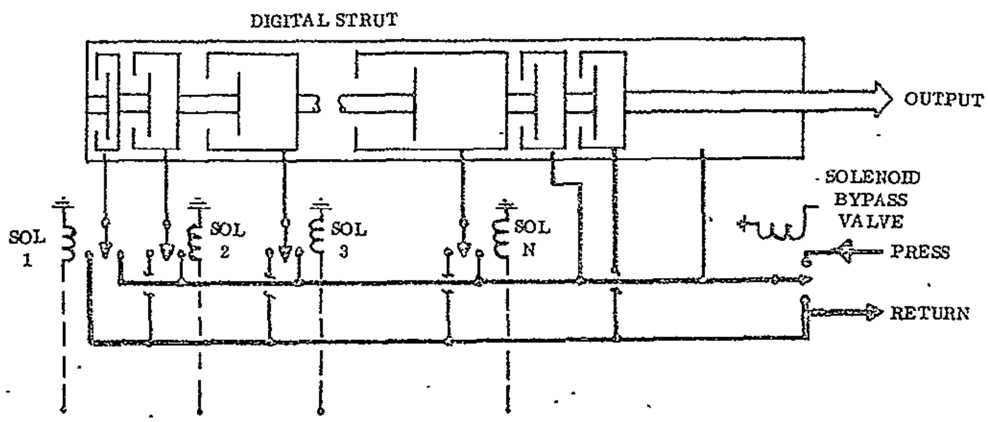


Figure 2-9. Digital Strut Actuator

Table 2-1. State of the Art Summary

	MECHANICAL	ELECT/MECH	GAS	HYDRAULIC
POWER ACTUATION OPERATIONAL	YES	YES	YES	YES
FLIGHT CONTROL SERVOS OPERATIONAL	YES (SMALL <2 HP)	YES (SMALL <2 HP)	NO	YES
DEVELOPMENT TESTING			YES	
MULTIPLE CHANNEL & POWER SERVOS WITH AUTOMATIC FAULT CORRECTING				
OPERATIONAL	NO	NO	NO	YES
DEVELOPMENT TESTING	NO	NO	NO	
STUDIES	NO	NO	NO	

2.4 RECOMMENDATIONS

Based on the survey, the following types of servoactuators were considered further:

a. Hydraulic analog for all applications.

b. Electromechanical analog for orbiter aileron only.

An electromechanical arrangement was considered to be more adaptable to the orbiter aileron application because of the low load requirements and severe environment. (Refer to Section 3 for requirements.)

One digital configuration was studied to provide a comparison with analog servoactuators.

Neither the electromechanical nor digital configuration as shown in Section 4 is taken from previous studies per se. They are configured to meet automatic fault correcting criteria as established in this report and as such represent new concepts employing redundancy mechanization.

Gas servo applications are not considered further. They appear to be the least desirable of all the disciplines in terms of development effort required to achieve operational status within the short term future slated for space shuttle development. The scope of this study is to configure point designs that have the best chance of succeeding using the technology of today (1970).

SECTION 3

REQUIREMENTS

3.1 GENERAL

For purpose of establishing point design configurations the following vehicle baseline definition, servoactuator requirements, ground rules, and assumptions apply.

3.2 VEHICLE BASELINE DESCRIPTION

3.2.1 GENERAL. The space shuttle concept consists of a two-stage, fully reusable system mounted in a piggyback fashion at liftoff. The lower vehicle acts as the booster, the upper vehicle completes the orbital phase of the mission.

During the boost phase, only the booster vehicle's rocket engines are operating and propellant is supplied from this stage. At staging, the orbital stage's rocket engines ignite and continue thrusting to orbit. The booster element enters from the staging altitude and decelerates to subsonic velocity where it starts its cruise engines and flies back to the launch site unmanned under automatic control. Following preflight checkout, both elements are ready for payload integration, vertical assembly, refueling, and relaunch.

The vehicles are both equipped with a fixed wing and a horizontal and vertical tail. Roll, pitch, and yaw control is achieved by using aileron, elevator, and rudder control surfaces. The surfaces are not split.

3.2.2 VEHICLE STATISTICS

a. Orbiter

Approximate gross landing weight	200,000 lb
Number of rocket engines	2
Thrust rating/engine	400,000 lb (S.L.)
Both engines gimballed	

b. Booster

Approximate gross landing weight	500,000 lb
Number of rocket engines	11
Thrust rating/engine	400,000 lb (S.L.)
All engines gimballed	

3.2.3 <u>ALLOWABLE FAILURES - ROCKET ENGINES AND TVC</u>

a. Orbiter — Critical Mission Segment

Lose one engine and/or TVC* — safe abort

b. Booster

Lose one engine and/or TVC* — complete mission

Lose two engines and/or TVC* — safe abort

3.2.4 <u>ALLOWABLE FAILURES - AERODYNAMIC SURFACE CONTROLS.</u> None. Elevators, rudder, and ailerons must function.

3.3 SERVOACTUATOR REQUIREMENTS

3.3.1 <u>PERFORMANCE.</u> For all functions the following characteristics apply:

Positional accuracy	0.2% of total travel
Frequency response (no load, closed loop)	
Frequency	
Amplitude ratio at 10% of valve saturation	±3 db
Phase lag input amplitude	45 deg

Refer to Table 3-1 for other performance criteria.

3.3.2 <u>ENVIRONMENT.</u> See Figures 3-1 and 3-2 for temperature-time histories for the orbiter and booster, respectively.

3.4 GROUND RULES AND ASSUMPTIONS

3.4.1 <u>INTERFACES</u>

3.4.1.1 <u>Command Channels.</u> There are four electrical command channels available for use. It is assumed that all four signals are identical and proper to the servoactuator interface. The servoamplifier and summing point are included in the servoactuator.

*Fail to null position - hardover failures not allowed, (e.g., no engine shutdowns permitted due to thrust vector control (TVC) failure).

Table 3-1. Performance Requirements

Function	Stage	Max Static Hinge Moment Ft-Lb	Max Rate No Load Deg/Sec	Total Travel Deg.	Redundancy Required	Degraded Performance (after 1 failure)	Degraded Performance (after 2 failures)
Aileron	Orbiter	4,800/side	40	±30	Fail Operate, Fail Operate Degraded Perf.	Negligible	67% Max H.M. 75% Max Rate
Aileron	Booster	19,200/side	40	±30	Same As Above.	Same As Above.	Same As Above.
Rudder	Orbiter	27,000	30	±30	Same As Above.	Same As Above.	Same As Above.
TVC each axis	Booster	65,000	*10	± 7	Fail to Null	-	-
TVC each axis	Orbiter	65,000	*10	± 7	Fail Operate, Fail To Null	Negligible	-
Rudder	Booster	135,000	30	±30	Fail Operate, Fail Operate Degraded Perf.	Same As Above.	67% Max H.M. 75% Max Rate
Elevator (ea of 2 Units)	Orbiter	66,500	30	±40	Same As Above.	Same As Above.	100% Max H.M. 75% Rate
Elevator (ea of 2 Units)	Booster	325,000	30	±40	Same As Above.	Same As Above.	100% Max H.M. 75% Rate

*TVC Actuators Are Always Loaded. At 10°/Sec, Hinge Moment Is 44,000 Ft-Lbs.

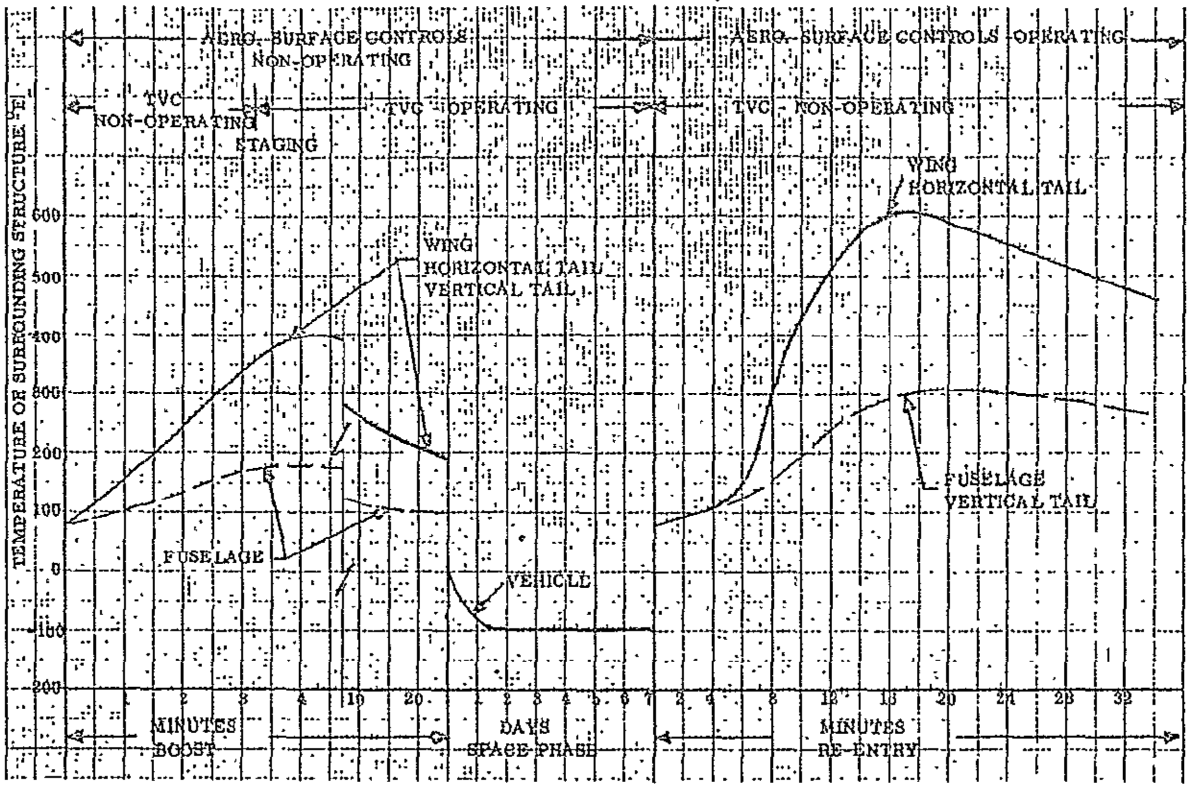


Figure 3-1. Orbiter — Internal Structural Temperature vs. Time

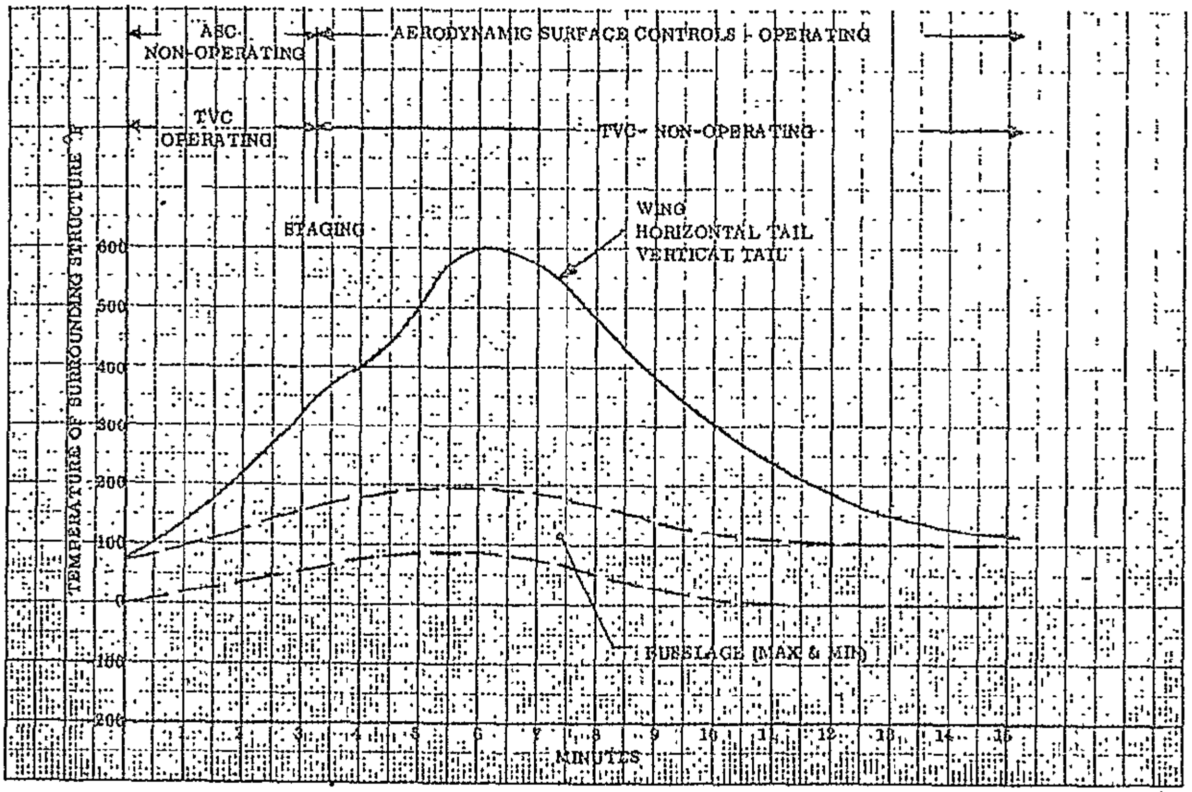


Figure 3-2. Booster — Internal Structural Temperature vs. Time

3.4.1.2 Power Sources

a. Electrical

Number of systems available	4
115/200 Vac, 3 phase, 400 Hz	

b. Hydraulic

Number of systems available	
Vehicle APU	4
Rocket engine accessory pad	As required
Pressure	
No flow	4000 psi
Full flow	2500 psi
Temperature	
Maximum — Normal operating	350°F
Minimum — Non-operating	-20°F

An operating pressure of 4000 psig maximum is chosen as a ground rule based on trends of present day large aircraft. The B-70 used 4000 psig and the SST has gone to 4000 psig after many studies on the merits of 4000 psig versus the standard 3000 psig. Boeing showed a net 7% weight saving in favor of 4000 psig but noted that the weight savings may be lost or reversed if the flight control actuators' sizes were determined by stiffness requirements.[17] Subsequent to that study, Boeing selected 4000 psig. Results of an actuator weight study by Moog showed that optimum pressure is from 3000 to 4000 psi for usual size loads (1000 to 10,000 ft-lb torque) but becomes higher for larger loads. However, the magnitude of weight improvement in the actuator is small.[18] Perhaps that particular tradeoff of optimum pressure should be conducted for each new aerospace application; however, that tradeoff is not within the scope of this study. Regardless of the eventual selection, 4000 psig used here will still yield valid comparison evaluations.

The pressure droop characteristic (2500 psi, maximum flow) is another ground rule established in anticipation of the eventual design requirements. That ground rule is discussed further in Section 3.4.2.

The 350°F maximum operating fluid temperature is chosen as a reasonable compromise between two design considerations that tug in opposite directions — stiffness and thermal conditioning. Bulk-modulus (e.g., actuator physical stiffness) lowers as fluid temperature increases. The weight penalty for transferring heat from a hydraulic system naturally goes up as maximum fluid operating temperature is lowered.

3.4.2 OUTPUT POWER. Preliminary data on surface hinge moment and rate requirements usually define the end limits; that is, maximum stall hinge moment and maximum no load rate. The output of a hydraulic system will give a parabolic curve (A) as shown in Figure 3-3, and this curve usually defines the requirements of hinge moment versus rate. However, the intermediate points are not defined and it is questionable whether they represent real requirements. If a constant output horsepower curve (B) can be defined as the real limit of hinge moment versus rate required for the intermediate points, then a reduction in delivered horsepower is possible. By allowing hydraulic pressure to droop with increased flow, the output will resemble curve C. Shown another way the required developed power in the hydraulic system is significantly reduced as shown in Figure 3-4. This affects APU size, APU fuel, transmission line size, and valve size. The APU and fuel weights reduce and the hydraulic transmission line size and valve weight increase (e.g., lower allowable pressure drop). However, for the space shuttle application, only warm oil need be considered where tubing friction losses are low. The resulting increase in line weights is more than offset by savings in APU and fuel weight. There are some undesirable features in allowing the pressure to droop. One is that the servovalve flow gain becomes non-linear as shown in Figure 3-5. The flow gain is normally defined as output flow rate versus valve input at a constant pressure drop. The slope (flow gain) varies from a maximum for small valve inputs to a

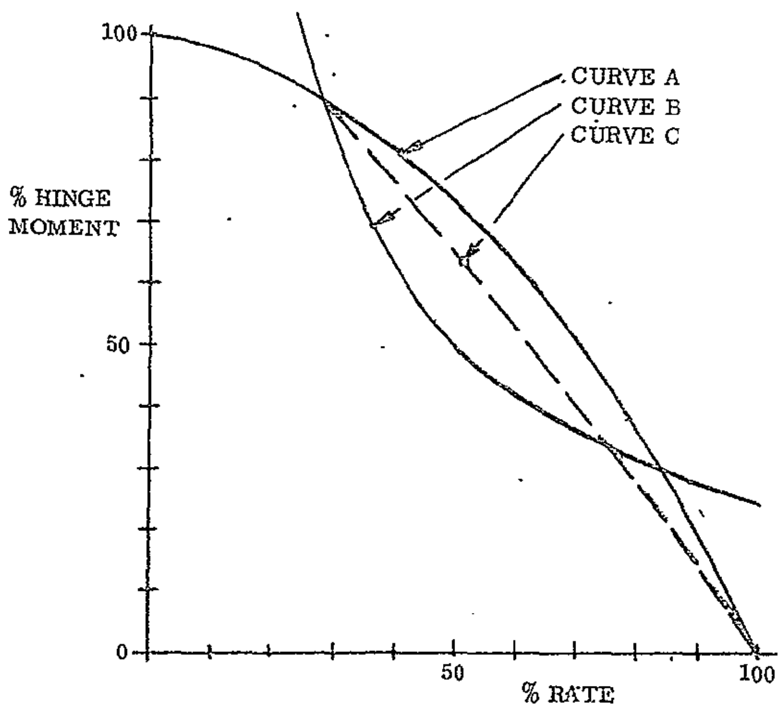


Figure 3-3. Hinge Moment vs. Rate

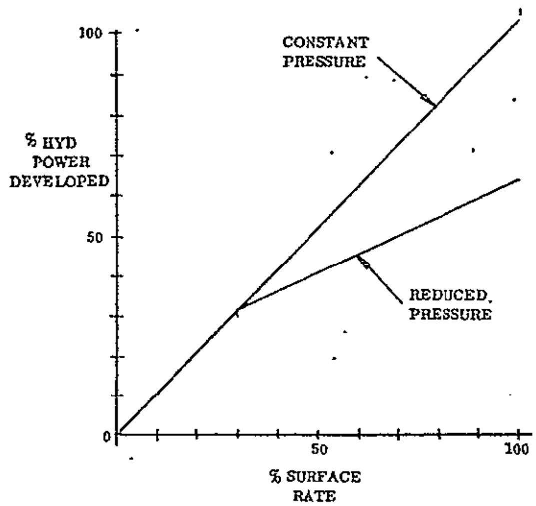


Figure 3-4. Hydraulic Power vs. Surface Rate

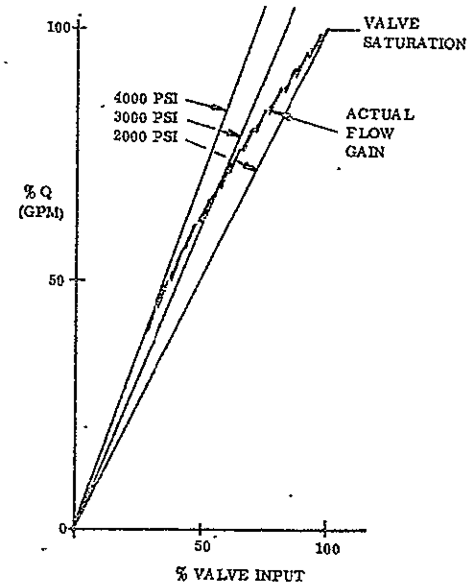


Figure 3-5. Servo Valve Flow Gain

minimum for full valve input due to the reduction in inlet pressure. This variance may be approximately 25% but is not considered to be a problem. The valve can be designed to compensate for this non-linearity.

Allowing pressure to droop is not valid if maximum rate at high hinge moment (typical of TVC) is defined as a requirement.

3.4.3 ALLOWABLE CORRECTION (SWITCHING) TRANSIENTS. The MSC space shuttle straight wing orbiter was examined to determine the vehicle response to control surface failure transients. Failure transients were simulated as triangular pulses having a specified failure surface rate and a specified area. The vehicle was examined at a cruise altitude of 20,000 ft and a velocity of 250 knots and at surface rates of 5 to 30 deg/sec and accumulated errors of 0.2 and 0.5 deg-sec. Maximum error for the case of 30 deg/sec surface rate and 0.5 deg-sec accumulated error for an elevator trailing edge down failure is +0.048g and -0.044 g. These accelerations are within the failure transient specification contained in MIL-F-8785B, "Flying Qualities of Piloted Airplanes," paragraph 3.5.5, which requires transients to be less than ±0.05g for a first failure.

The 0.5 deg-sec transient is well within the capability of an automatic fault correcting actuator. For example, in a position summing arrangement, assuming a control channel (secondary actuator) time constant of 0.02 sec, an output rate due to one channel hardover of 30 deg/sec, a failure not sensed until the channel is hardover, and a 30 millisecond switching time, the output transient would be approximately 0.1 deg-sec.

The 0.5 deg-sec transient may also be applied to the orbiter TVC as a reasonable allowable transient. The booster TVC criteria is not based on degree-seconds since one engine could go hardover and control could be maintained. The limit is simply physical restraint and the allowable transient is more liberal than for all other applications.

The allowable transients listed above are much more liberal than can normally be permitted on high performance aircraft (which established correction transient allowables for most servoactuator designs). As such, correction transients become diminished in importance as evaluation criteria for this study.

3.4.4 DUTY CYCLE. Duty cycles are defined so that power source fuel weight can be determined. Duty cycles are established for aerodynamic surface controls only. Thrust vector controls operate for only short durations (approximately three minutes compared to approximately two hours for aerodynamic surface controls on the booster) and exert a minor influence on fuel weight. The duty cycle is given in terms applicable to hydraulic systems for the largest surface or demand on each vehicle. As such fuel weight will only be a factor in the largest surface servoactuator tradeoff comparison. See Table 3-2 for mission segments and duration.

Table 3-2. Mission Segments and Duration

	TIME ~ SECONDS	
MISSION SEGMENT	ORBITER	BOOSTER
ENTRY	2000	920
TRANSITION	100	100
CRUISE	600	4700
LANDING	300	480
MISSION DURATION	3000	6200

The duty cycles are

a. Entry, Transition, Landing: 15% of maximum elevator rate — continuous; 100% of maximum elevator rate — 10% of the time

b. Cruise: 5% of maximum elevator rate — continuous; 100% of maximum elevator rate — 1% of the time

From the above data an average power in % of maximum delivered power can be derived:

Maximum Delivered Power = 100% Elevator Rate x 2500 psi

Constant HP Loss (Pump & System Leakage = 10% of a 4000 psi System

$$= 10\% \left(\frac{4000}{2500}\right) \times \text{Max Del power}$$

$$= 16\% \text{ of Max Del power}$$

$$\text{Continuous Cycling @ } 15\% = 15\% \left(\frac{4000}{2500}\right) = 24\% \text{ of Max Del power}$$

$$@\ 5\% = 5\% \left(\frac{4000}{2500}\right) = 8\% \text{ of Max Del power}$$

[at lower surface rates pump discharge pressure is approximate 4000 psi]

Steady State Power = Constant Losses + Continuous Cycling

$$HP_{AVE} = HP_{S.S.} + K(HP_d - HP_{S.S.})$$

HP_{AVE} = Average Power

$HP_{S.S.}$ = Steady State Power

K = % of Time @ Max Rate

HP_d = Delivered Power = 100%

3.4.4.1 <u>For Entry, Transition, and Landing</u>

$$HP_{AVE} = (16\% + 24\%) + 10\% [100\% - (16 + 24)]$$

$$HP_{AVE} = 46\% \text{ of Max Delivered Power}$$

Adjusting to APU shaft power where delivered power $\cong$ 0.9 shaft power,

HP_{AVE} = 46% x 0.9 = <u>41.5% of APU Shaft Output Power</u>

3.4.4.2 <u>For Cruise</u>

$$HP_{AVE} = (16\% + 8\%) + 1\% [100 - 24]$$

$$HP_{AVE} = 25\% \text{ of Max Delivered Power}$$

or 25 x 0.9 = <u>23.5% of APU Shaft Output Power</u>

3.4.4.3 <u>Average Power Over Total Mission</u>

a. Orbiter:

$$HP_{AVE} = \frac{41.5\%\ (\text{Entry} + \text{Transition} + \text{Landing Time}) + 23.5\%\ (\text{Cruise Time})}{\text{Total Mission Duration}}$$

$$HP_{AVE} = \frac{41.5\%\ (2000 + 100 + 300) + 23.5\%\ (600)}{3000}$$

$$HP_{AVE} = \underline{38\% \text{ of APU Rated Shaft Output Power}}$$

b. Booster:

$$HP_{AVE} \doteq \frac{41.5\%\ (920 + 100 + 480) + 23.5\ (4700)}{6200}$$

$$HP_{AVE} = \underline{30\% \text{ of APU Rated Shaft Output Power}}$$

The average power shown above converted to hp-hrs and used with specific fuel consumption data shown in Section 6, Figure 6-10, determines fuel weight for each elevator configuration; see Section 7.

SECTION 4

CONCEPTUAL DESIGN

4.1 POWER CIRCUITS

All aerodynamic surface controls are required to be fail operate, fail safe. In this case fail safe means fail operate, degraded mode. Allowable degradation in terms of hinge moment is 67% for ailerons and rudders and no degradation for elevators. To show convenient comparisons of the different applications, their requirements are converted to output power. This is done to provide a convenient base regardless of the type of power used (e.g., hydraulic, electrical, mechanical). The power listings below do not represent actual power developed but represent the minimum theoretical output required as shown on the constant power curve, Figure 3-3, Section 3. The constant power curve passes through the half hinge moment, half rate point which is used for this comparison. Table 4-1 shows the output power required when applying multiple circuits per the failure criteria. For example, each of four circuits need only supply half the power (hinge moment) that is required of each of three circuits. This indicates that a four-power circuit arrangement is a candidate for large load applications because of the possible weight reduction over three-power circuits. The chart also identifies the type of power to be used based on the recommendations of the industry survey.

Table 4-1. Output Power Required

Application	Min Req't (hp)	Multiple Systems (hp/sys) 3 Systems	4 Systems
Aileron - orbiter	3	2	1
*Rudder - orbiter	6.4	4.3	2.15
*Aileron - booster	12.2	8.1	4
*Elevator - orbiter	31.5	31.5	15.75
*Rudder - booster	32	21.5	10.25
*Elevator - booster	155	155	77.5
*TVC	19.7 Based on 70.7% (Pitch + Yaw H.M. @ 10 deg/sec)		

*Applicable to hydraulic power and control only. Only hydraulics is considered for these applications because it is the only technology that has combined large power requirements, multiple system output, and automatic failure correcting capability.

4.2 DESIGN APPLICATIONS

Aerodynamic surface controls range from 9600 ft-lb hinge moment for the orbiter aileron to 650,000 ft-lb for the booster elevator and suggest no common solution for all. To do a separate tradeoff for each of six surfaces would dilute the overall effort. It is reasonable then to break the applications down as follows:

a. Booster critical application (largest hinge moment)

b. Orbiter critical application (largest hinge moment)

c. Smallest size

The elevator for each vehicle is the critical application. They have the greatest effect on vehicle weight (subsystem and interfaces). The orbiter aileron has the least effect on vehicle weight, is the smallest size, and is a candidate for something other than hydraulic power. The intermediate applications are not configured, but the data generated is applicable. For example, qualitative comparisons for the elevator servoactuators will apply to the intermediate size servoactuators due to similarity of the control portions. Weight is the major difference that might exist between the orbiter elevator and rudder.

Three configurations are established for each application to provide basis for tradeoff evaluation. These configurations are all of the analog type. One digital configuration is established for comparison to the analog versions.

Three analog configurations are established for each of the booster and orbiter TVCs. They are studied separately because of the different levels of redundancy required. The number of power systems to be used for TVC is based solely on redundancy requirements and not weight.

4.3 DESIGN CONCEPTS

As stated before, the majority of work has been accomplished in hydraulics. There are many types of redundancy mechanizations possible. Figure 4-1 is a flow chart showing many combinations that have been conceived to give automatic fault correcting capability. Reviewing the space shuttle requirements, the aerodynamic surface controls must be fail operate, fail operate-degraded mode. This requires a minimum of three power circuits, three output actuators, and four control channels. Four channels are required to provide a two out of three agreement after one failure. Electronic models are not used in this study.

The flow chart is broken down to show major classifications of techniques used to achieve automatic fault correcting capability. Some of these terms are unique to this report and may not agree with all published data to date. The following discussions describe these classifications.

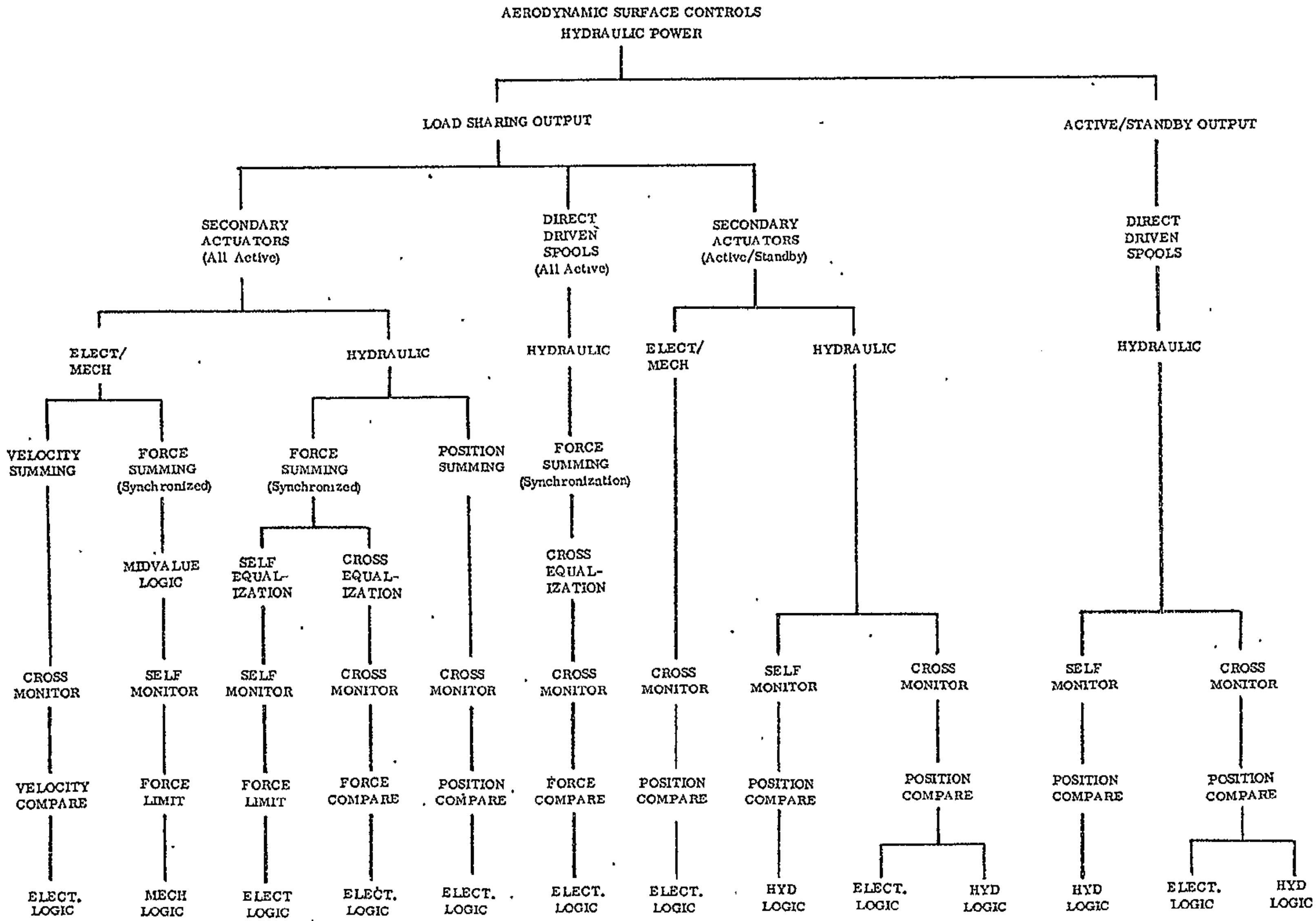


Figure 4-1. Flow Diagram — Redundancy Mechanization, 2 Fail Operate

4.3.1 LOAD SHARING VS. ACTIVE/STANDBY. (This section is concerned only with the power output portion of a servoactuator. Sections 4.3.2 through 4.3.11 are discussions on the control portions only.)

Load sharing has all output actuators active where they share the load equally. Active/standby has only one output actuator active at one time. All other output actuators are bypassed.

4.3.2 SECONDARY ACTUATORS VS. DIRECT DRIVEN SPOOLS. A secondary actuator (sometimes referred to in other literature as servo ram or mod piston) is an intermediate actuator that converts an upper stage servochannel command to a mechanical output to drive a power spool.

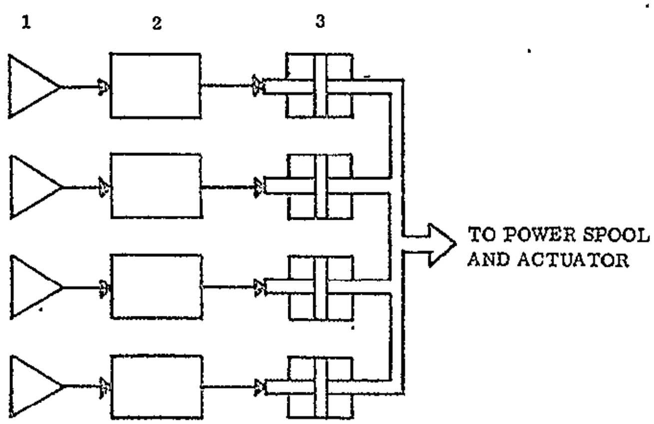
Secondary actuators came into being as a means of providing a mechanical equivalent of an electrical signal so that convenient summation with a pilot's manual input to a servoactuator was possible. In a pure fly-by-wire arrangement, the need to produce a mechanical output from an electrohydraulic servo to sum with a pilot's mechanical command is unnecessary. A secondary actuator may be used, but for different reasons: remote location of a control package from a power valve and actuator, or the need to provide a convenient output of each command/servo channel for monitoring.

"Direct driven spool" as used herein describes a power spool driven hydraulically by an upper stage (either a spool or first stage amplifier). In this case, secondary actuators are not used.

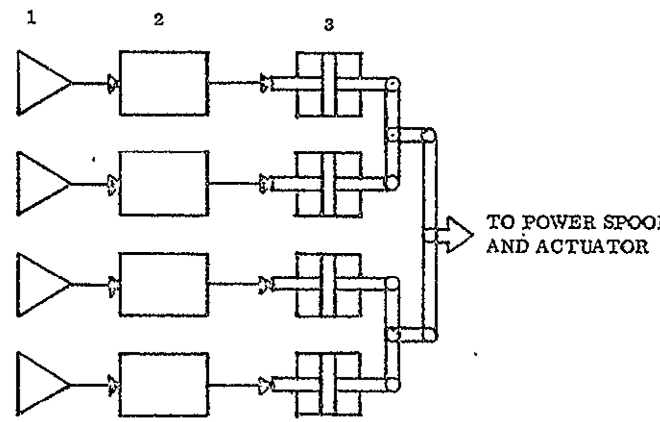
4.3.3 FORCE SUMMING. All channel outputs are active and in parallel. Where secondary actuators are used (see Figure 4-2A) the output positions are common, and any force unbalance is due primarily to channel mismatch. Force summing arrangements must be synchronized to maintain static stiffness around null, prevent dead band, and reduce power drain. The power spools must also be synchronized either by some force balancing means or by close tolerance fabrication in the case of tandem spools. If secondary actuators are not used and direct driven spools cannot be synchronized by fabrication, force signals from the power output actuators can be used to synchronize the channels. Maintaining stability becomes a problem when inserting a pressure feedback loop within a position feedback loop where the control or synchronization loop gain is nearly equal to the controlled or position loop gain.

The output force of each channel being unique to that channel is used for self monitoring or comparison with the other channels in fault detection and correction logic. In a hydraulic arrangement, signals proportional to ΔP developed at each channel output (secondary actuator) are usually used for this purpose.

A change in output does not occur after a failure thereby allowing a long switching time. There is negligible loss in performance after failure. Complexity is high because of the synchronization required.

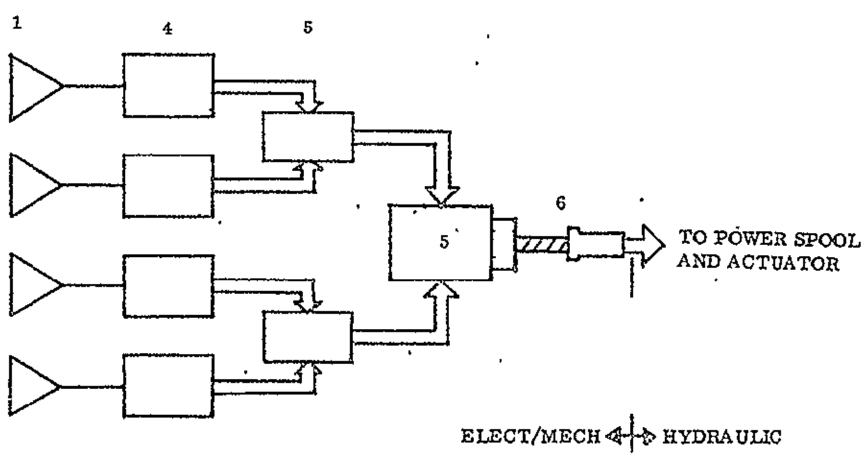


A. FORCE SUMMING - SECONDARY ACTUATORS, ALL ACTIVE (SYNCHRONIZATION REQ'D)



B. POSITION SUMMING - SECONDARY ACTUATORS, ALL ACTIVE

1. SERVO AMPLIFIER
2. E/H SERVOVALVE
3. SECONDARY ACTUATOR
4. SERVO MOTOR
5. DIFFERENTIAL
6. BALL SCREW



C. VELOCITY SUMMING - SECONDARY BALL SCREW ACTUATOR

Figure 4-2. Control Channel Mechanization

Another method of force summing is mid value logic. In a four-channel, secondary actuator arrangement, all actuators are in parallel. Detents are used with unequal breakout forces in opposite directions. Channel mismatch causes the secondary actuators to break out of their detents in the ensuing force fight. The unequal breakout forces are to ensure that one channel stays in detent and provides the output. There is a considerable problem in preventing all actuators from breaking out and/or suffering numerous channel disengagements (nuisance tripping) via fault detection and correction logic with this arrangement.

4.3.4 POSITION SUMMING. All channel outputs are active and in series. Position summing, Figure 4-2B, does not require synchronization (no force fight between channels). Since an individual channel is not resisted by adjacent channels, a failure such as a hardover will cause output motion of the power actuator. In fact an output motion must occur for failure detection because position is unique to each channel and provides the intelligence for comparison to other channels to detect and switch out faults. When comparing force summing to position summing the methods used to deactivate a faulty channel are opposite. A secondary actuator in a force summing arrangement must be bypassed since all actuators are in parallel. A secondary actuator in a position summing arrangement is centered and locked. Therefore, failures in a position summing arrangement reduce position authority of the common output which in turn reduces loop gain and maximum rate capability of the output actuators unless some means of gain changing is used.

4.3.5 VELOCITY SUMMING. A redundancy mechanization technique employing electromechanical servo channels developed by LTV electrosystems uses velocity summing; see Figure 4-2C. The outputs of servo motors are summed through differentials to provide an input to a hydraulic power spool. As channels are de-activated, velocity output of the ball screw to the hydraulic power spool is reduced. Velocity summing is similar to position summing in that all outputs are in series, but failures in velocity summing reduce the velocity of the power spool rather than position which affects acceleration of the output actuator and not rate.

4.3.6 ACTIVE/STANDBY-CONTROL CHANNELS. Active/standby is self explanatory in that there is usually only one active output at one time. Engagement of standby channels are by predetermined sequence. All channels receive commands and fault detection and switching is achieved by position comparison of an element within each channel. There is no degradation in performance after a failure.

4.3.7 SYNCHRONIZATION. In a force summing arrangement, synchronization usually consists of an equalization (or compensation) signal feedback to force all channels to seek a common null. Self equalization is contained within the channel; see Figure 4-3A. Load pressure is allowed to build up in a secondary actuator to a limit that satisfies synchronization (approximately 1000 psi in a 4000 psi system). As load pressure exceeds this limit, a proportionally larger signal is fed back to bias the servovalve in the direction to reduce the actuator ΔP. The correction signal stays

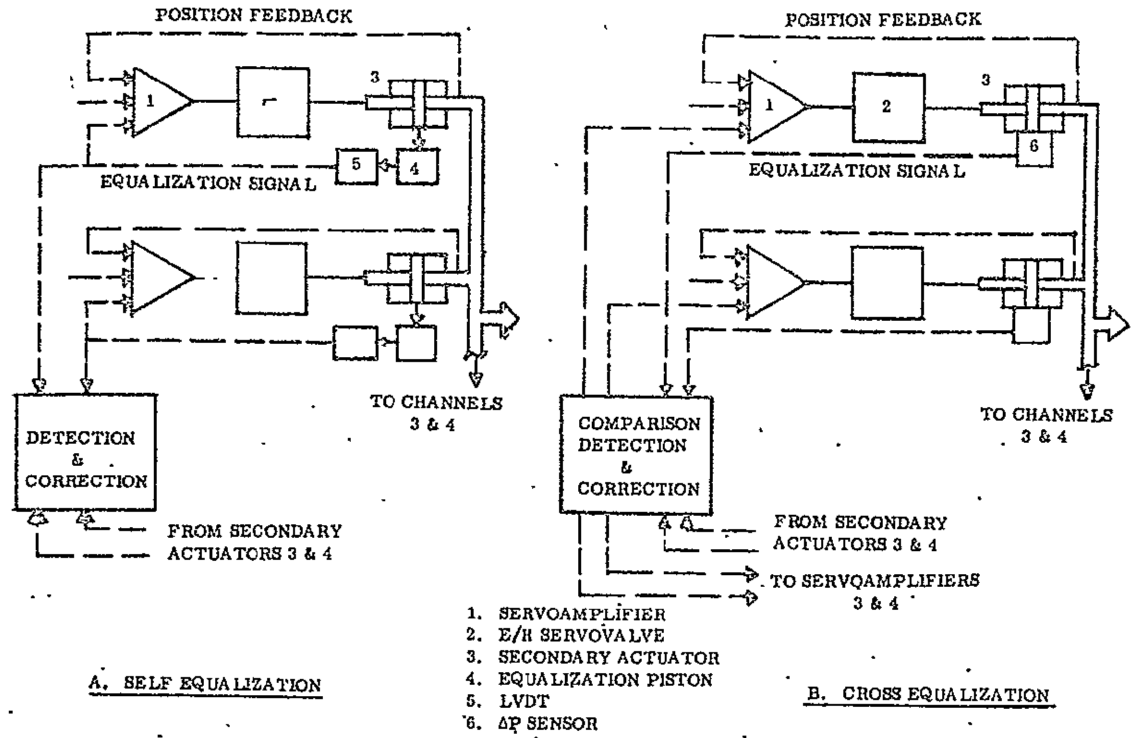


Figure 4-3. Control Channel Synchronization

nearly constant until the ΔP reaches a minimum value (approximately 200 psi) then will reduce to zero. To accomplish this hysteresis effect, the equalization network consists of a preloaded piston and a LVDT. The equalization signal is also directed to a detection and switching function for de-activating the channel at a "failure threshold" limit.

Force signals are averaged between two adjacent channels before being fed back as a biasing signal in a cross equalization scheme; see Figure 4-3B. Detection and correction in this case consist of comparing channel outputs (force) two-by-two and logically determining the faulty channel.

Self equalization and cross equalization are synchronizing techniques that were devised primarily to meet the broader classification of self monitoring and cross monitoring, respectively.

4.3.8 CROSS MONITORING. Cross monitoring or inter channel monitoring refers to any comparison arrangement requiring interchannel connections to implement fault detection logic. Figure 4-3B is an example of cross monitoring where some form of "majority vote logic" is used to detect and switch out a failure.

4.3.9 SELF MONITORING. Self monitoring or intra channel monitoring does not use interchannel connections. The self equalization technique of Section 4.3.7 and its fault indication method is an example of self monitoring. In Figure 4-4, a monitoring channel is added for comparison to each channel that provides an output. There are no

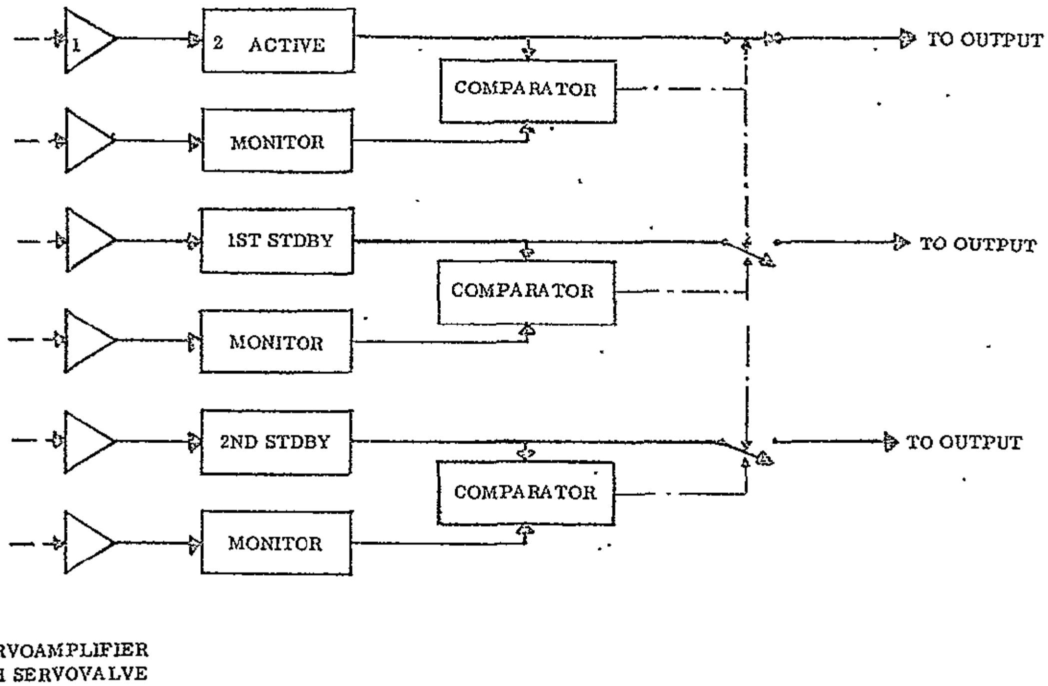
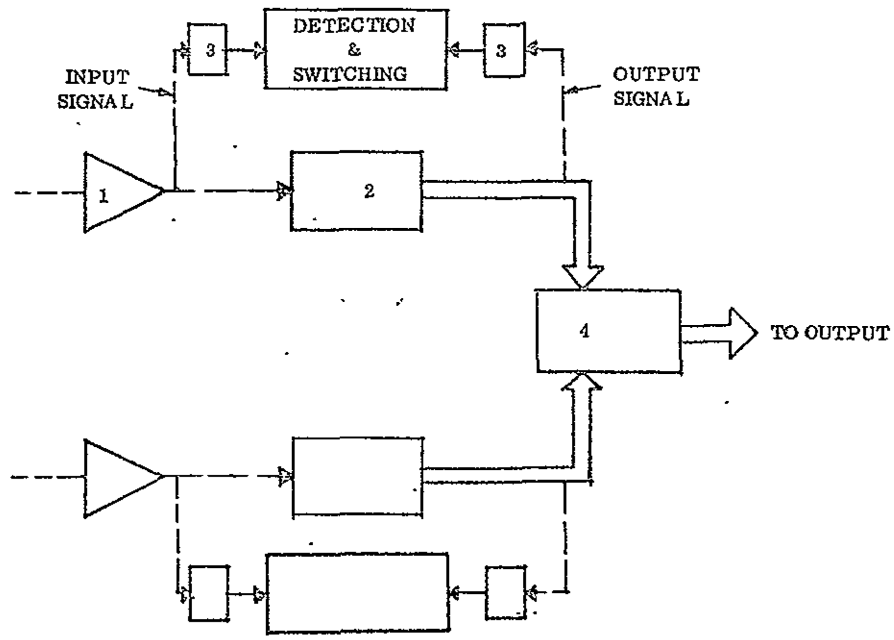


Figure 4-4. Self Monitoring – Active/Standby

interconnections between the channels that provide outputs other than a device to engage standby channels. This approach requires more servo channels but fewer power supplies when compared to cross monitoring. For example, using electro hydraulic servo channels, cross monitoring requires four servo channels and four hydraulic systems whereas self monitoring requires six servo channels and three hydraulic systems for two fail operate capability.

In Figure 4-5, the fault detection method is simply the comparison of output to input within one channel. This method is simple but has some disadvantages. For example, if the output meets stall load and the stall lasts the duration of a time delay, all channels may be de-activated at once.

4.3.10 ELECTRONIC LOGIC VS. HYDRAULIC LOGIC. Two fault detection and correction methods widely used are electronic logic and hydraulic logic. Electronic logic has all comparison or monitoring information converted to electrical signals and utilizes electronic devices for failure detection and switching. It is versatile in that it can be packaged remote from the servoactuator, or its function can be removed entirely from the servoactuator subsystem and assigned to the vehicle computer. Hydraulic logic is more of a specialty design integrated with the type of redundancy mechanization employed by the servoactuator. For example, the hydraulic logic developed by Hydraulic Research for active/standby switching bears little resemblance to the hydromechanical logic used in the F-111 damper servoactuator.[19] Hydraulic logic by necessity is packaged within the servoactuator.



1. SERVOAMPLIFIER
2. SERVO-MOTOR OR CLUTCH
3. TIME DELAY
4. DIFFERENTIAL

Figure 4-5. Self Monitoring – Input/Output

4.3.11 MONITOR VS. MONITORLESS. Figure 4-1 with all of its classifications of techniques for redundancy mechanizations describes monitored systems only. A monitored servoactuator is synonomous with detection-correction in this text. In a monitorless servoactuator, a fault is not switched out but is simply overpowered by the remaining good channels. An example of a monitorless servoactuator is the one developed for the Saturn S-IVB by Moog, commonly called a majority voting servoactuator. In general, to achieve the same level of failure capability, a monitorless design requires more redundancy than a monitored design; however, it is not encumbered with detection and switching elements.

4.4 CONFIGURATIONS

4.4.1 SELECTION CRITERIA. As stated before in Section 4.2, three configurations for each of five applications were established and, per the recommendations based on the industry survey, hydraulic analog concepts are configured for all applications and one electromechanical configured for the smallest load application (orbiter aileron). In addition, one digital concept was established to meet the surface controls failure capability. It should be noted here that a concept using electromechanical servos to control hydraulic power stages is classified as a hydraulic concept.

Hydraulic concepts were given preliminary evaluation based on redundancy performance, complexity, and development status. Redundancy performance includes criteria such as nuisance tripping and effects after failure.

The electromechanical concept (compared to other electromechanical) was evaluated on normal performance as well as the criteria above.

4.4.2 AERODYNAMIC SURFACE CONTROLS. Refer to Figure 4-6 for descriptions of the configurations and where they are applied. These were selected to represent the best candidates and also to provide meaningful trends over wide ranges of loads and sizes. Electromechanical control (velocity summing) is used with three hydraulic power circuits to delete the need of a fourth small hydraulic circuit just for control channel power. Electromechanical control could also be used with four hydraulic power circuits. A weight comparison is made in Section 8 between electromechanical and hydraulic servo control portions only. The electromechanical control is a candidate for all three applications so that this comparison can be made.

Although a four-power system appears to weigh less than a three-power system for large load applications, a three-power system is a candidate for the elevator applications to show the weight trends of four versus three through this range of large loads.

The booster and orbiter elevators have the same configurations because they have much in common. Although the booster loads are much larger, both surfaces must be classified as being big. They are the driving functions for both vehicles (e.g., they determine interface sizes and quantities).

One principal difference between the elevator configurations 1 and 2 (see Figures 4-7 and 4-8, respectively) is the physical installation. Configuration 1 has four separate control packages, each mounted integrally with an actuator. Configuration 2 has one large control package and could be installed remote from the actuator such that only fluid and electrical lines interconnect the two. The control package location must not be so remote that fluid line compliance would seriously degrade hydraulic stiffness.

4.4.2.1 Booster and Orbiter Elevator – Configuration 1 (Figure 4-7). The unit has an all active-force summing control mechanization. Four hydraulic power systems and four separate valve/actuators are used. The control sections are force summed through a mechanical shaft. The secondary actuators are synchronized by a self-equalization loop fed back within each channel to keep all secondary actuators within predetermined synchronous limits. The equalization signal (force signal) is also used for fault detection and correction. When a channel output reaches a threshold of failure, power is removed from the continuously energized shutoff valve. With the shutoff valve off, the pressure operated bypass valve shifts to bypass the secondary actuator. Hydraulic pressure to a power actuator is not affected by a channel failure that de-activates a secondary actuator. Each control channel loop is closed by electrical position feedback from secondary actuator to servo amplifier.

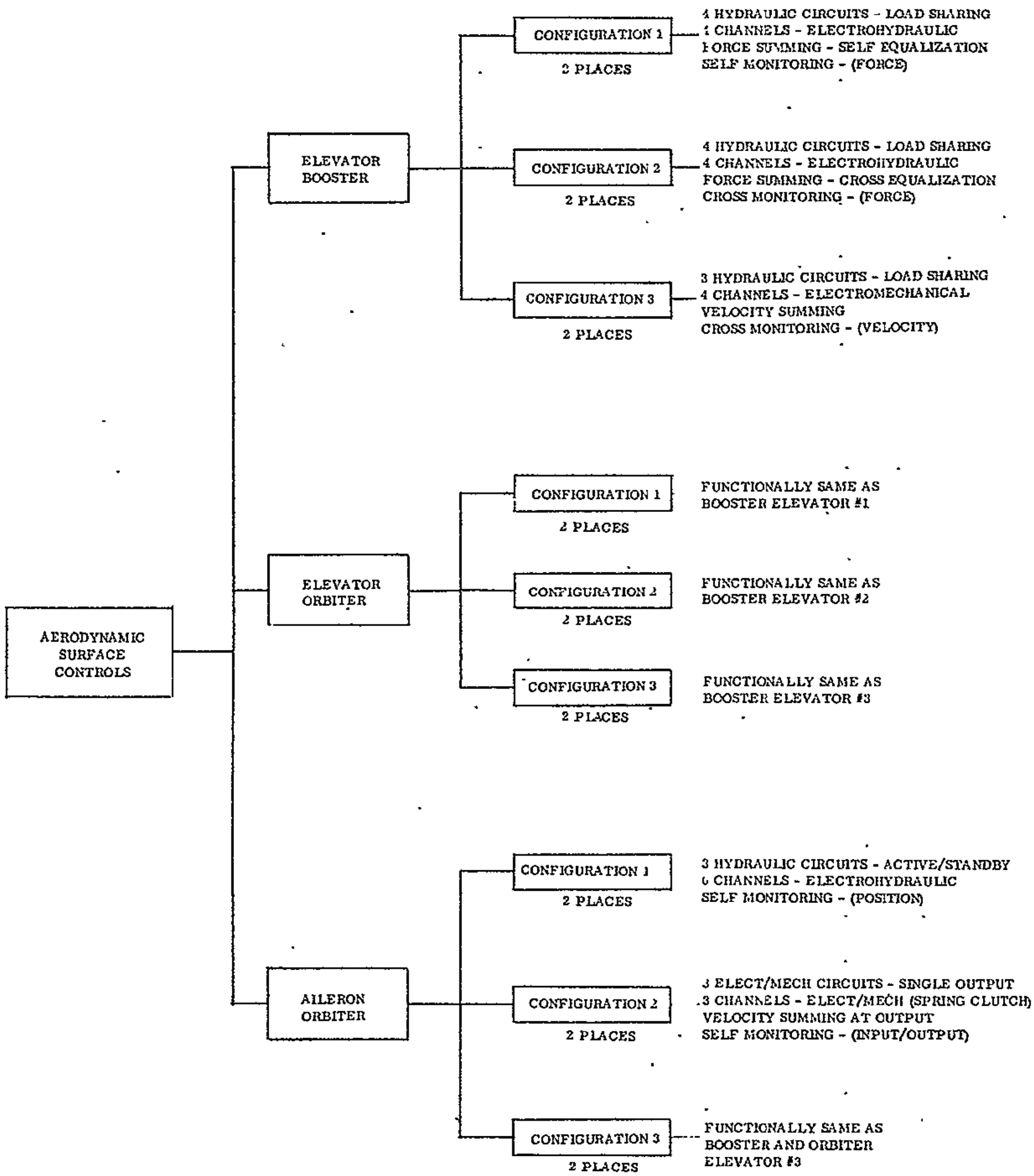


Figure 4-6. Aerodynamic Surface Control Applications

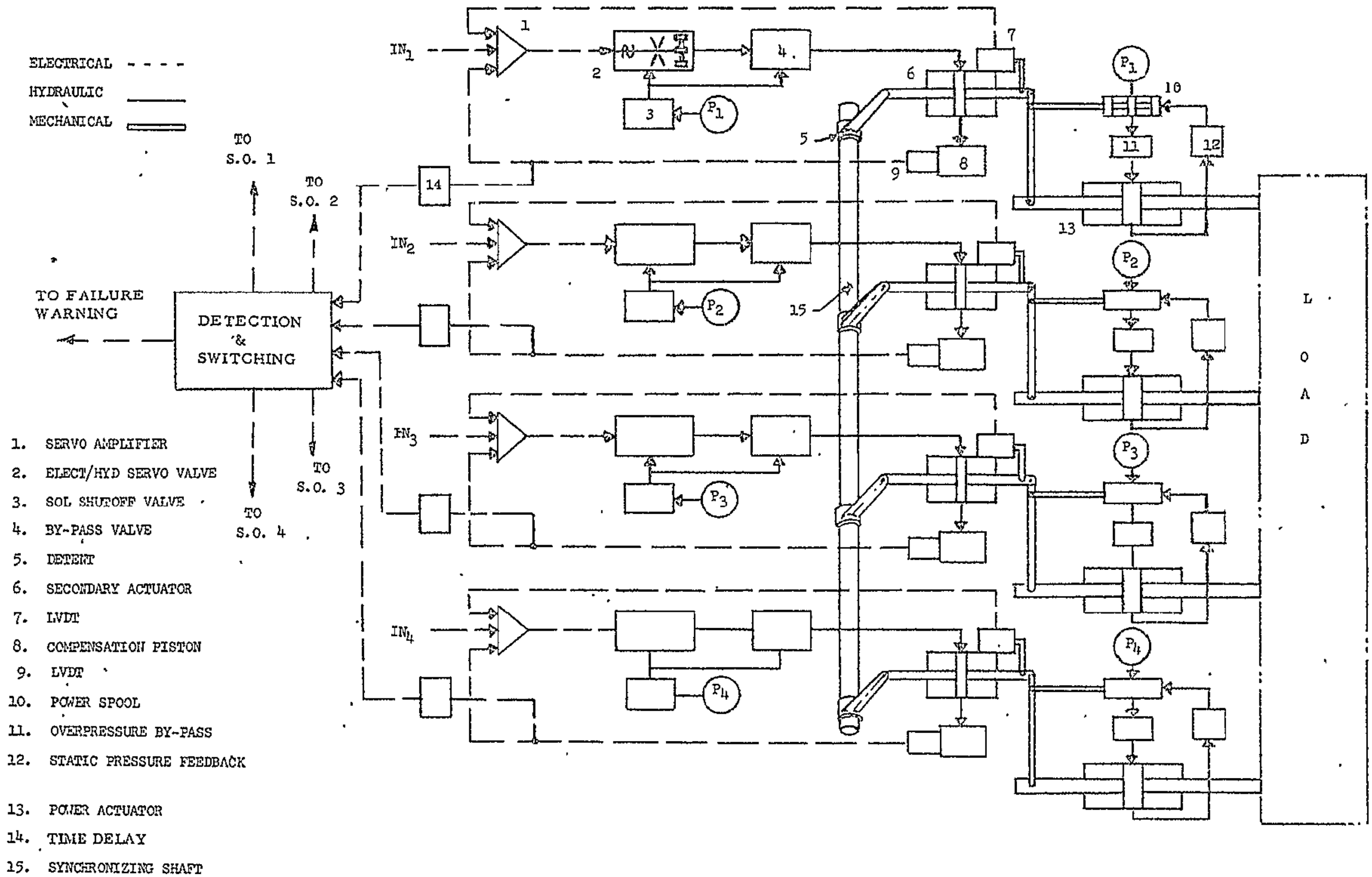


Figure 4-7. Booster and Orbiter Elevator – Configuration 1

Each power actuator is sized for one-half the maximum hinge moment so that 100% of the required hinge moment is available after two hydraulic failures. Null synchronization of the power spools is achieved by use of static pressure feedback. This in effect softens the pressure gain of each power spool at null, thus preventing a force fight between hydraulic circuits. A jammed secondary actuator or power spool can be driven out of its detent by the three good channels. The detent breakout force is set below the output of any two secondary actuators operating within their synchronous limits. This is to prevent disengagement of channels when they are driving against this load. Over pressure bypass on the output is provided to allow the three good outputs to follow commands. Mechanical feedback closes the outer loop from the actuator to the power spool.

4.4.2.2 Booster and Orbiter Elevator – Configuration 2 (Figure 4-8). This unit is all active-force summing. Equalization (force) signals from secondary actuators are averaged between channels before fedback to the servoamplifiers to synchronize channels. Electronic fault detection and correction logic is used. When one channel fails, the logic compares channels on a two-by-two basis, detects the faulty channel and bypasses the secondary actuator by sending a signal to de-energize the normally energized shutoff valve which in turn cycles the bypass valve to bypass. The four channel control and the power spools are integrated into one package and can be remote from the output actuators.

Each of the output actuators is sized for one-half hinge moment so that 100% of the required hinge moment is available after two hydraulic failures. The power spools are synchronized by fabrication, where all circuits null within approximately 1000 psi ΔP of each other. There are no provisions for power spool jams other than the overpowering force of the secondary actuators. Electrical position feedback is used to close all control loops.

4.4.2.3 Booster and Orbiter Elevator – Configuration 3, (Figure 4-9); Orbiter Aileron – Configuration 3. This unit is all electromechanical in the control stages. The ac servomotor and differentials arranged in an all active, velocity summing arrangement provide a ballscrew output to drive hydraulic power spools. The servo motors incorporate a fixed phase and control phase winding and normally electrically energized brake (brake off). The control phase accepts a variable voltage input to control the output. Each channel incorporates velocity (tachometer) feedback which is also used for channel comparison. When a channel reaches threshold of failure (disagreement with adjacent channels) the electronic detection and switching logic signals the faulty channel to shut down. Power is removed from the control phase winding and brake, locking the output. The output velocity to the ballscrew is reduced 25% after each channel failure. A back-up differential and ballscrew is provided. Should the primary ballscrew/differential jam, the back-up ballscrew is driven out of its detent to drive the power spools.

Each output actuator is sized for full hinge moment to provide 100% capability after two failures. The triple tandem spools are synchronized by close tolerance fabrication.

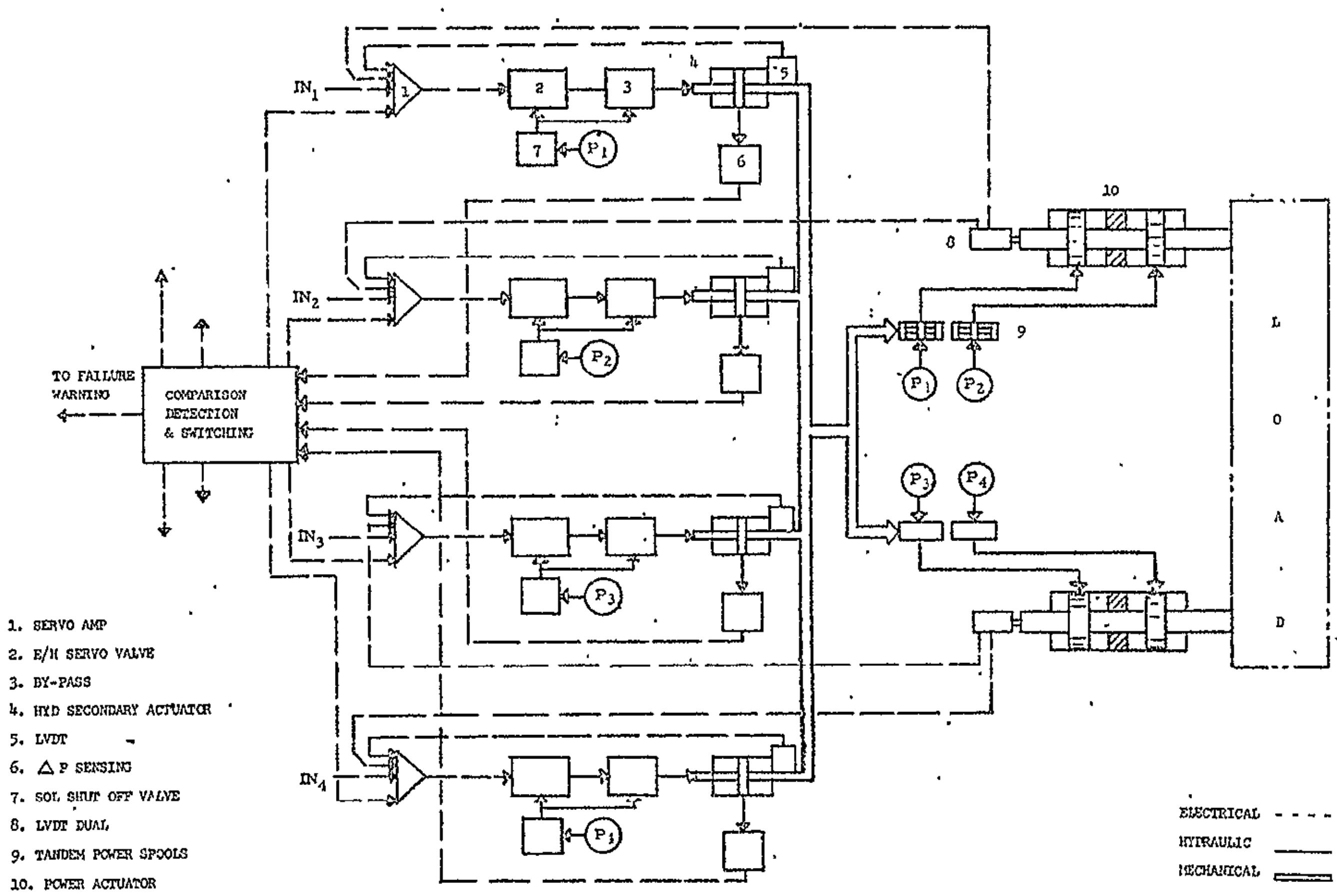
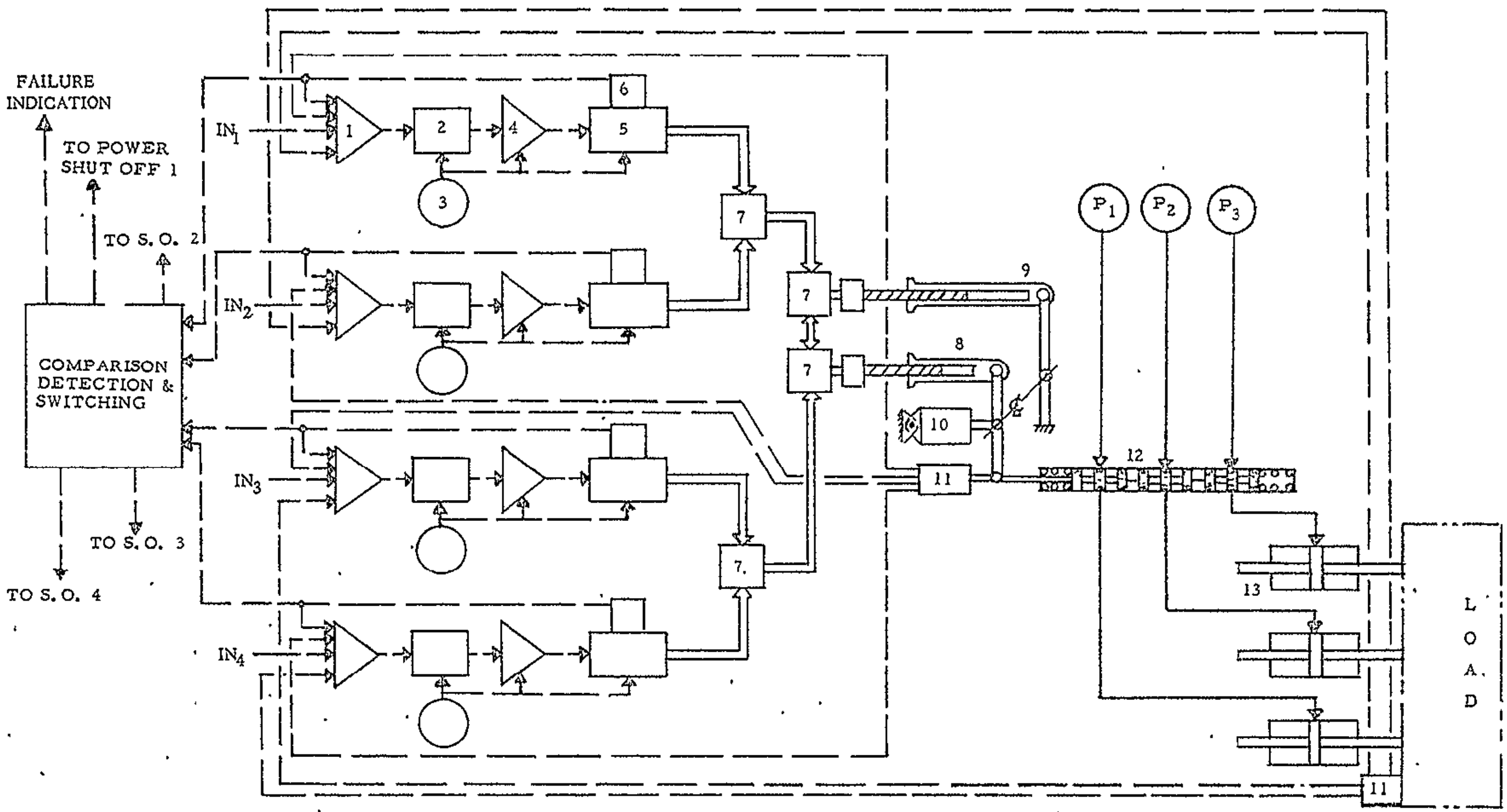


Figure 4-8. Booster and Orbiter Elevator – Configuration 2



1. SUMMING AMPLIFIER
2. ELECTRONICS
3. ELECT. POWER
4. POWER AMPLIFIER
5. A.C. SERVO MOTOR & BRAKE
6. TACHOMETER
7. DIFFERENTIAL
8. BALL SCREW
9. STANDBY BALL SCREW
10. FUNK STRUT
11. LVDT
12. POWER SPOOL
13. POWER ACTUATOR

ELECTRICAL
HYDRUALIC
MECHANICAL

Figure 4-9. Booster and Orbiter Elevator – Configuration 3; Orbiter Aileron – Configuration 3

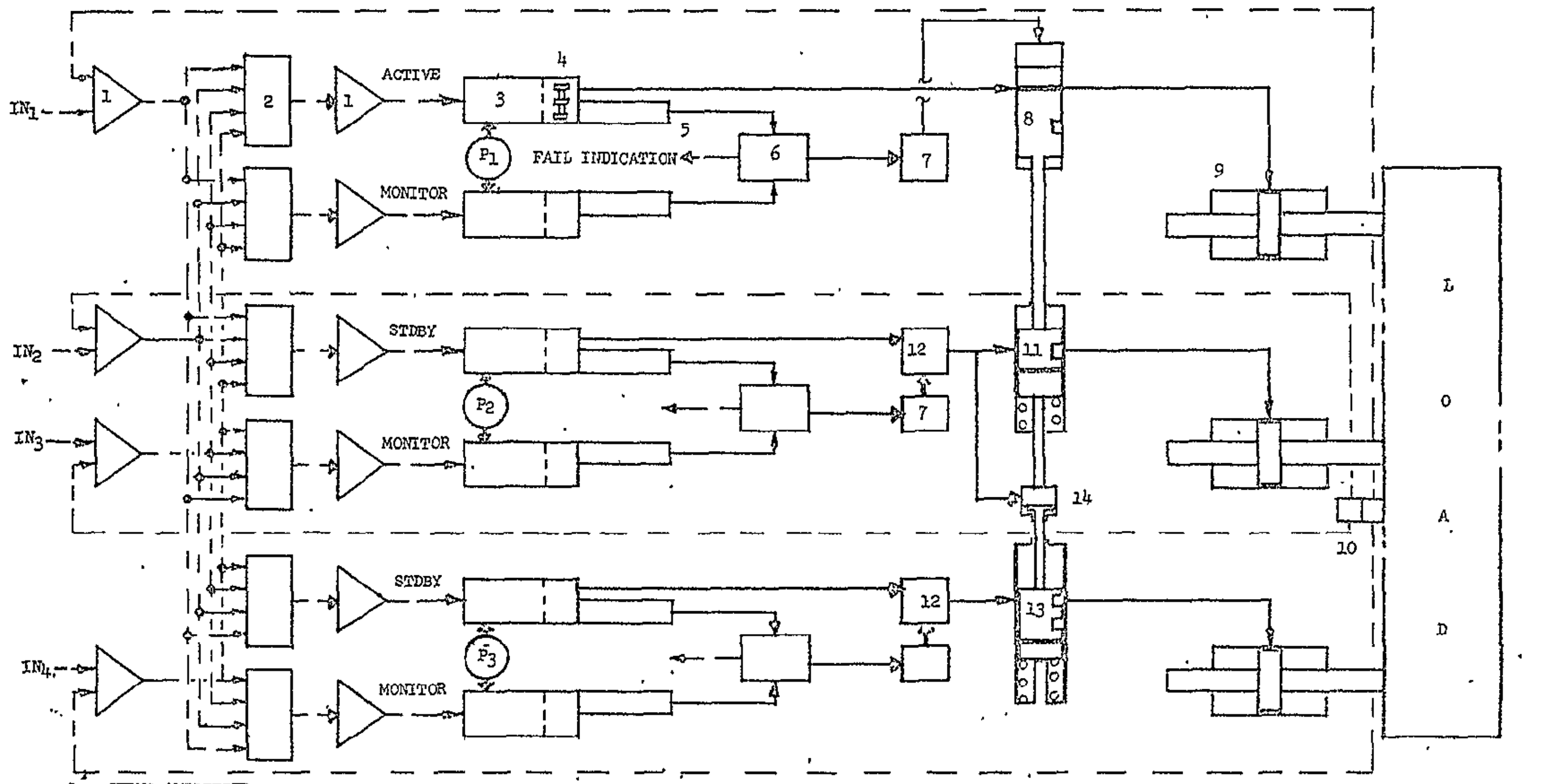
4.4.2.4 Orbiter Aileron – Configuration 1 (Figure 4-10). The unit is active/standby employing hydraulic logic. Secondary actuators are not used. The active and two standby circuits each have two servo channels, one being a monitor with no output. The system is self monitoring with no cross comparison between channels that have outputs. The lockout solenoid valves are energized momentarily to cycle the unit to the starting position shown. Fault correction logic consists of hydraulic flapper/nozzle assemblies that port pressure proportional to electrohydraulic valve spool position to a spring loaded slide valve called a comparator. If spool positions disagree between the active channel and its monitor by a predetermined amount, the ΔP on the comparator causes it to shift, which in turn causes the engage valve to shift, bypassing the active output and engaging the first standby. The lockout valve (de-energized) cycles upon collapse of control pressure to prevent re-engagement of the channel. If either standby channel should fail first, separate shutoff valves cycle to prevent engagement. A hydraulic piston pressurized by the first standby circuit provides an additional engage valve position should the first standby fail first. Pressure switches in the comparator valve provide intelligence for failure indication.

Because the mechanization requires six servo channels, voters are placed between the four channel inputs and the six driving servo amplifiers. This allows up to two erroneous command inputs to be voted out before reaching the servos. Electrical feedback is used to close the control loop. Each actuator is sized to 100% of maximum required hinge moment.

4.4.2.5 Orbiter Aileron – Configuration 2 (Figure 4-11). This unit is all electromechanical employing spring clutches. Three electrical power systems are clutched in upon electrical command and summed through a triple differential to provide output to a single ballscrew actuator. Two clutches are required per circuit to give bi-directional motion. The input signals from four channels are conditioned to permit only signals of sufficient level to energize the solenoids (or coil energizers) to prevent clutch slipping. The clutches transmit power from a continuous rotating input shaft at constant rate. Fault detection is self monitoring, comparing input to output. If there is no output with an input signal present, power is removed from that channel and the output is locked by means of a brake incorporated within each clutch assembly. The signal conditioning electronics also includes failure detection and switching logic to remove an erroneous command signal so that comparison can be made after one failure.

Due to the triple differential summing arrangement, failures cause unequal effects. Should either channel 1 or 2 fail, for example, the output ballscrew rate is reduced 25%. If channel 3 fails, the output rate is reduced 50%. There is no degradation in hinge moment after failure but there is a degradation in surface rate.

4.4.3 THRUST VECTOR CONTROLS. See Figure 4-12 for descriptions of the configurations and where they are applied.



1. SERVO AMPLIFIER
2. VOTER
3. E/H SERVO VALVE
4. POWER SPOOL
5. HYD MONITOR
6. HYD COMPARATOR
7. SOL. LOCKOUT VALVE
8. SHUTOFF & BYPASS VALVE
9. POWER ACTUATOR
10. LVDT (QUADRUPLE)
11. BYPASS & ENGAGE VALVE 2
12. SHUTOFF & BYPASS VALVE
13. BYPASS & ENGAGE VALVE 3
14. HYD. COUPLING

ELECTRICAL
HYDRAULIC
MECHANICAL

Figure 4-10. Orbiter Aileron – Configuration 1

1. SERVO AMPLIFIER
2. SIGNAL AVERAGER & THRESHOLD DETECTOR
3. EXTEND SOLENOID
4. RETRACT SOLENOID
5. EXTEND SPRING CLUTCH
6. RETRACT SPRING CLUTCH
7. TIME DELAY
8. SELF MONITORING & SWITCHING
9. TIME DELAY
10. LVDT
11. TRIPLE DIFFERENTIAL
12. BALLSCREW OR POWER HINGE
13. LVDT
14. POWER SOURCE & GEARING

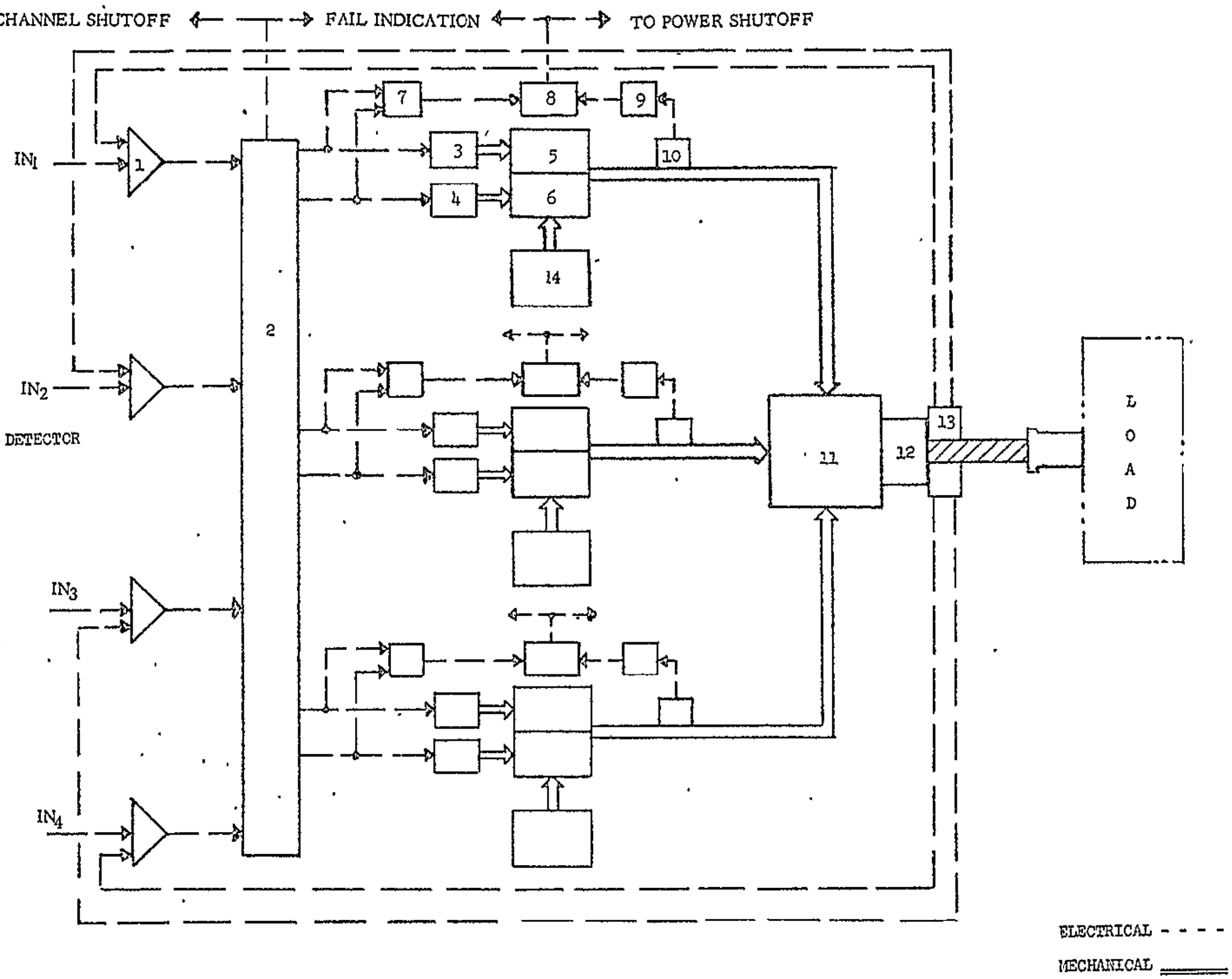


Figure 4-11. Orbiter Aileron – Configuration 2

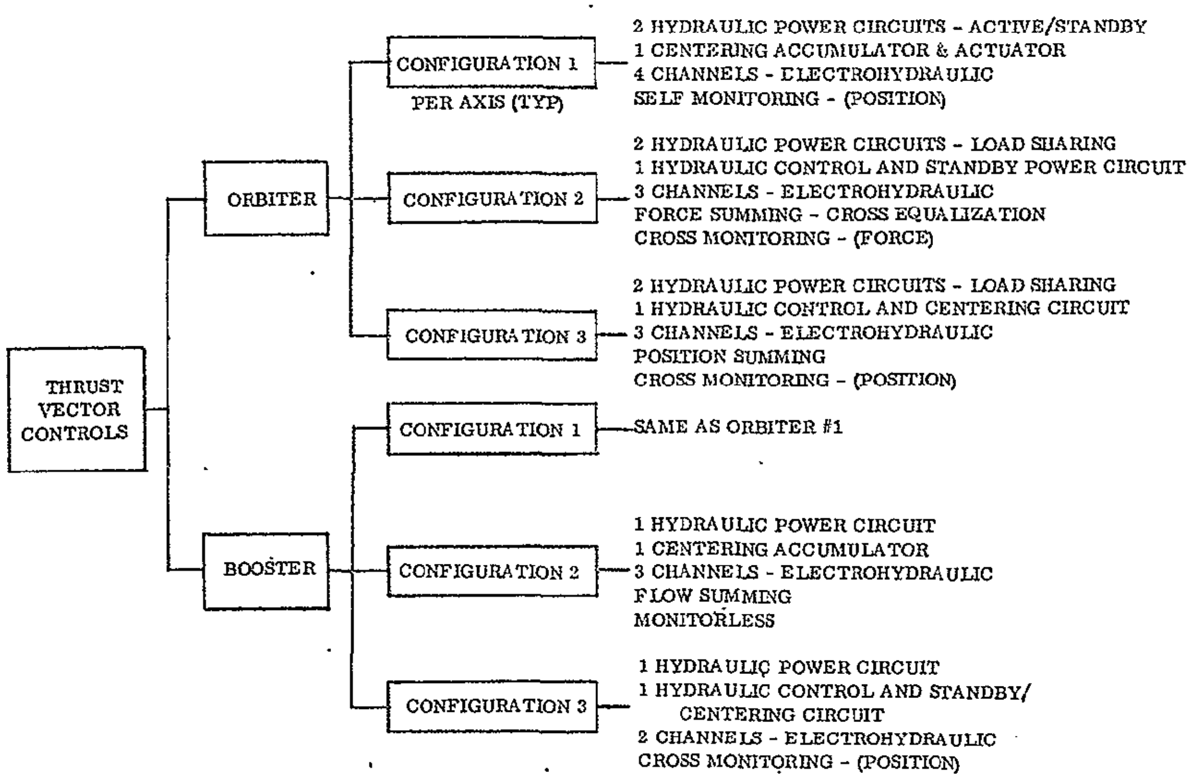


Figure 4-12. TVC Applications

The orbiter and booster TVC configuration 1 are common and utilize the vehicle central APU driven hydraulic circuits. The centering power for this case is an accumulator checked off from an active hydraulic circuit.

To see how the APU systems are arranged, refer to Figure 4-13. The dual blocks represent dual tandem actuators and the numbers within the blocks represent the power circuits. One can see that one hydraulic circuit loss does not shut down any TVC. A second hydraulic failure shuts down two TVC systems on the booster and one TVC system on the orbiter. The configuration is more redundant than is normally required (fail to null) for the booster, but it eliminates the need for 11 separate hydraulic circuits, one per rocket engine. The addition of power distribution lines from the APU hydraulics is the only power penalty applied to this configuration since the power supplies are sized by the aerodynamic surface controls.

Configurations 2 and 3 for the orbiter require three active hydraulic systems per TVC due to continuous channel control power consumption.

The booster configurations represent three levels of redundancy. Configuration 1 has the highest level to be able to use only 4 hydraulic power circuits for all 11 engines. Configuration 3 has the least redundancy in the control channels, sufficient to prevent hardover on the first failure. This configuration meets only the fail to null requirement. Configuration 2 has three channels because it is monitorless in the control channels. The only detection and switching occurs when the actuator is centered after loss of

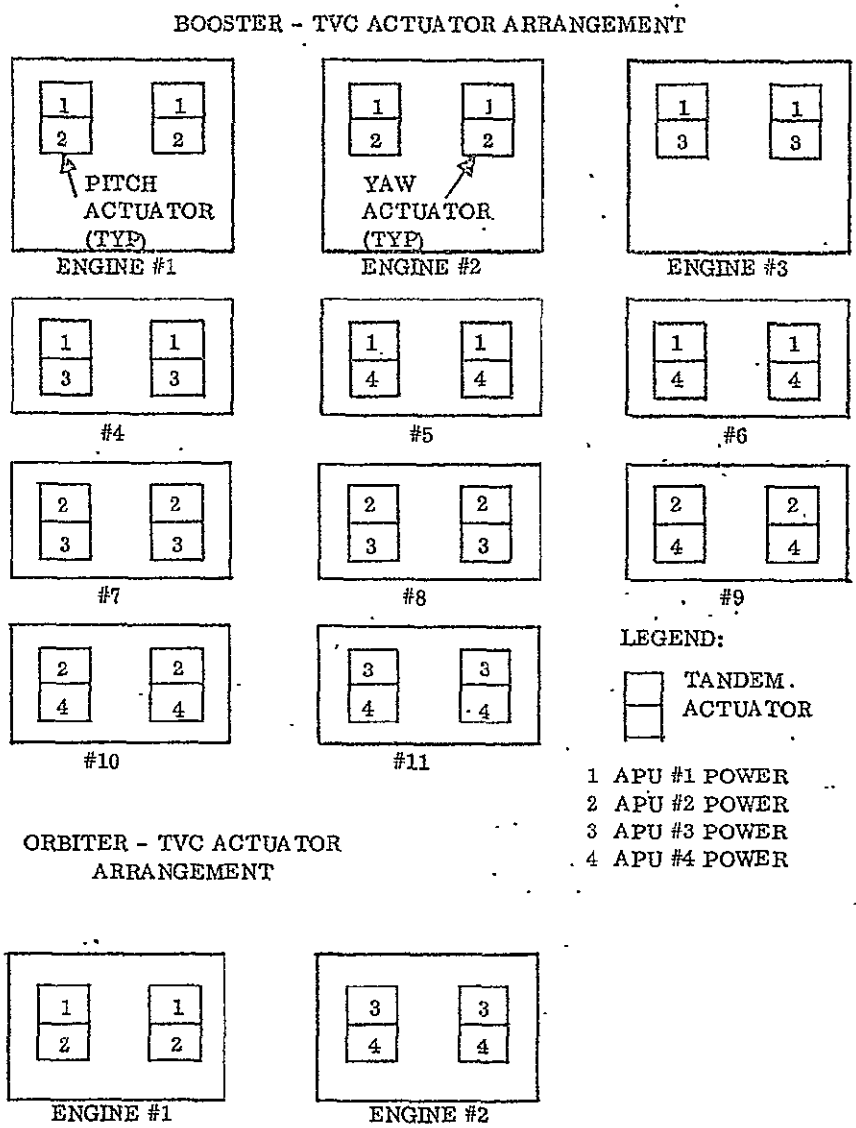
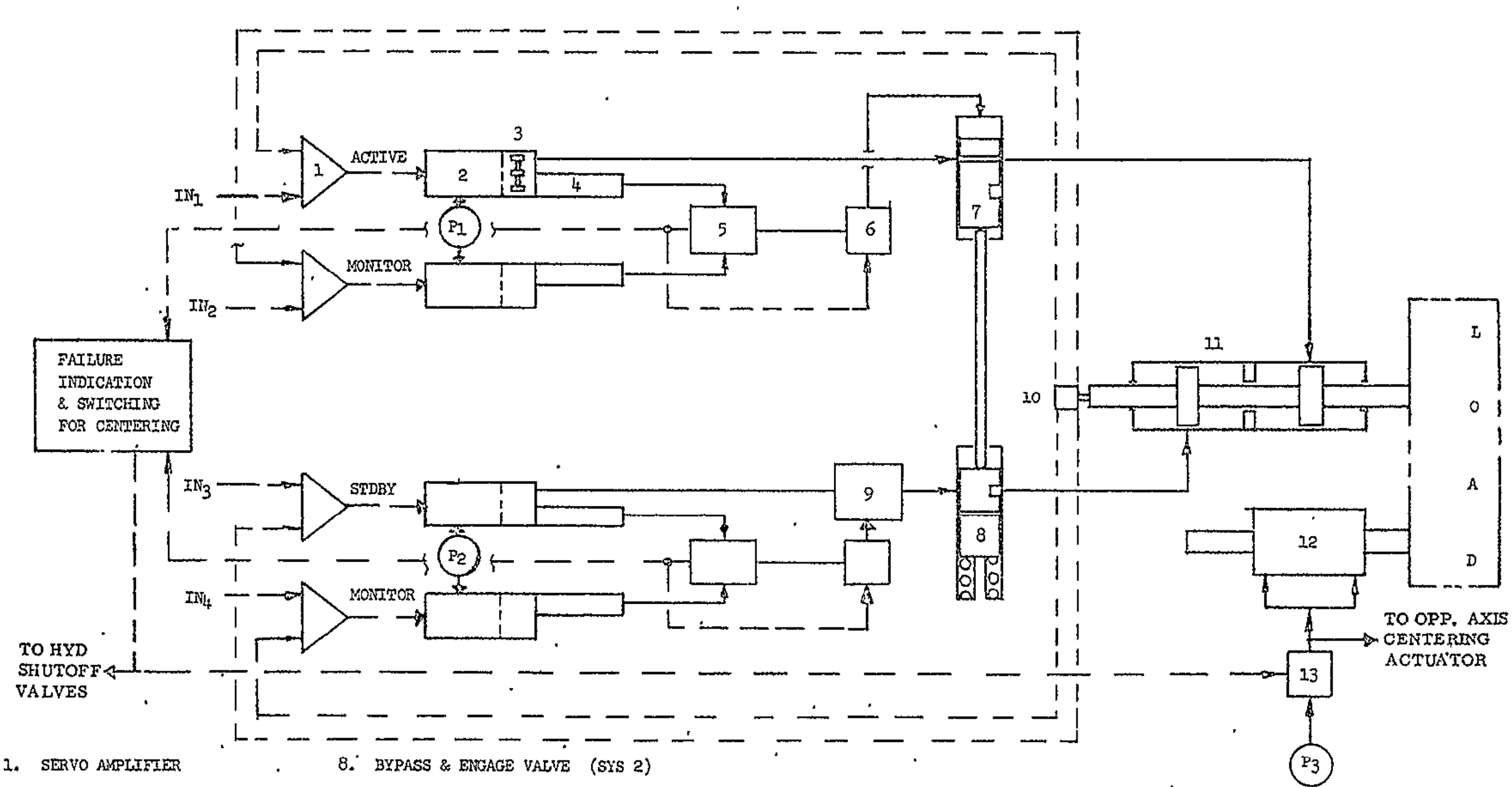


Figure 4-13. APU Driven Hydraulic Circuits for TVC – Configuration 1

hydraulic control power. Configuration 2 is fail operate degraded if the failure is confined to the upper servo portions where redundancy exists. It can survive many combinations of failures providing there are no two like failures.

4.4.3.1 Orbiter and Booster TVC – Configuration 1 (Figure 4-14). The unit is active/standby employing hydraulic logic. The active and standby circuits each have two servo channels, one being a monitor with no output. The unit is self monitoring with no cross monitoring between P_1 and P_2 outputs. The lockout solenoid valves (6) are energized momentarily to cycle the unit to the starting position shown. Fault correction logic consists of hydraulic flapper/nozzle assemblies (4) that port pressure proportional to electrohydraulic valve spool position to a spring-loaded slide valve (5) called a comparator. If spool positions disagree between active channel and its monitor



1. SERVO AMPLIFIER
2. E/H SERVO VALVE
3. POWER SPOOL
4. HYD. MONITOR
5. HYD. COMPARATOR
6. SOL LOCKOUT VALVE
7. SHUTOFF & BYPASS VALVE (SYS 1)
8. BYPASS & ENGAGE VALVE (SYS 2)
9. SHUTOFF & BYPASS VALVE (SYS 2)
10. LVDT (QUADRUPLE)
11. POWER ACTUATOR
12. CENTERING ACTUATOR
13. SOL. CENTERING VALVE

ELECTRICAL - - - -
HYDRAULIC ______
MECHANICAL ══════

Figure 4-14. Orbiter and Booster TVC – Configuration 1 by a predetermined amount, the ΔP on the comparator causes it to shift, which in turn causes the engage to shift bypassing the active output and engaging the standby. The lockout valve (de-energized) cycles upon collapse of control pressure to prevent re-engagement of the channel. If the standby should fail first, its shutoff and bypass valve (9) prevents engagement. Pressure switches in the comparator provide intelligence for failure indication and actuator centering. After two failures, P_1 and P_2 hydraulic pressure is shutoff and the centering valve energized, porting P_3 (accumulator) to the centering and locking actuator.

Each half of the tandem actuator is sized for full hinge moment. Electrical position feedback is used to close the loop around the servoactuator.

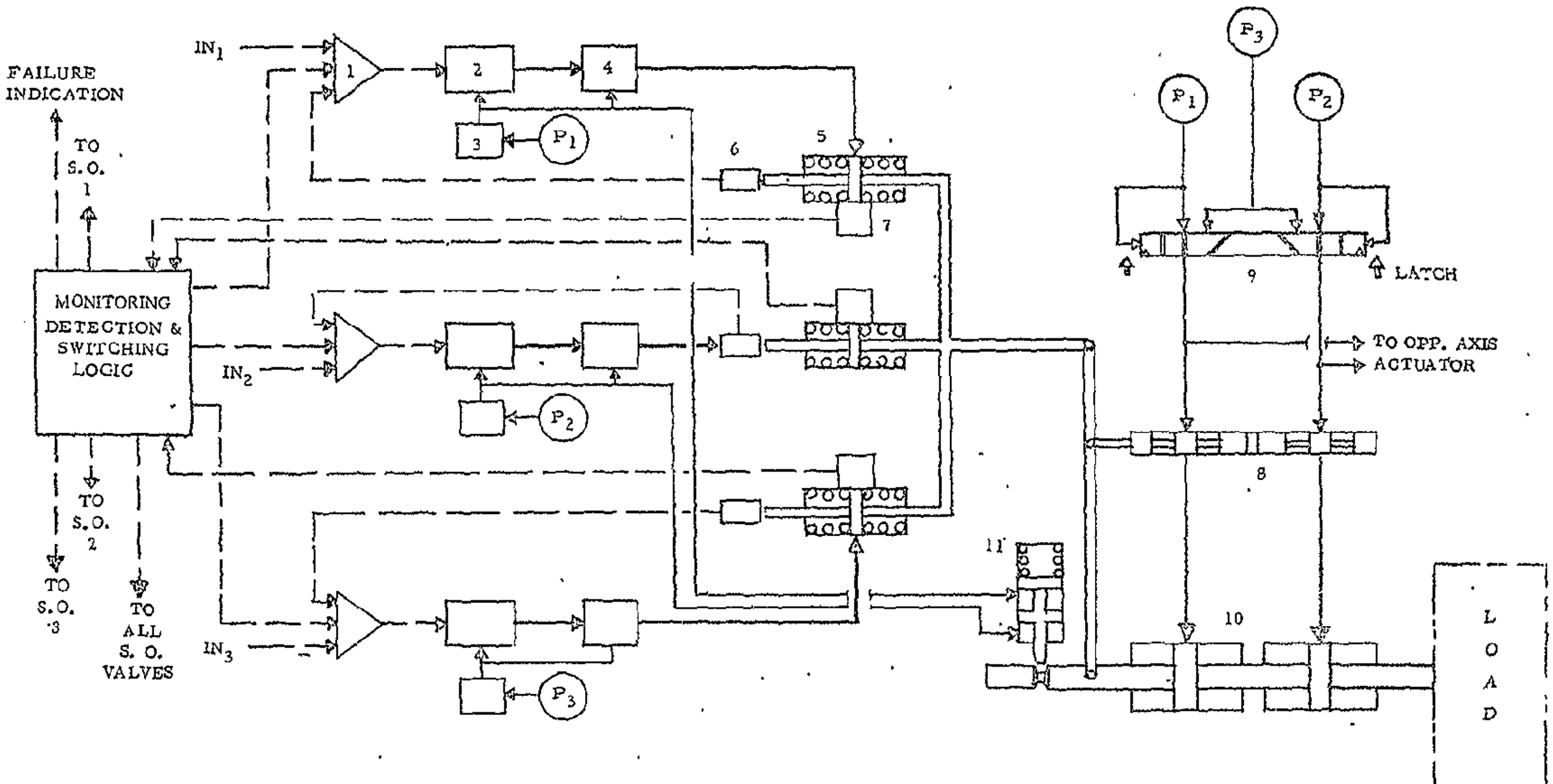
4.4.3.2 Orbiter TVC – Configuration 2 (Figure 4-15). Three active channels powered by three hydraulic circuits provide a common output that controls a tandem output actuator. P_3 is used as a standby and switched into the power actuator in event of loss of either P_1 or P_2. The servo channels use secondary actuators mechanized to provide a force summed output. To minimize the force unbalance the three channels are synchronized by force signals (ΔP) returned to the electronic comparison, detection and switching logic where an equalization signal is fed back to each channel forcing them to a common null. If one channel (e.g., secondary actuator ΔP) reaches a predetermined threshold of disagreement with the other channels, it is shut off via the detection and switching logic, and the secondary actuator is bypassed.

Electrical position feedback closes the control loop from the secondary actuator to the servo amplifier. After a second failure all channels are shut down and all secondary actuators center. Center position of the secondary actuators is coincident with geometric center of the power actuators due to the mechanical feedback arrangement.

Each half of the tandem power actuator is sized to provide full hinge moment.

4.4.3.3 Orbiter TVC – Configuration 3 (Figure 4-16). Two hydraulic circuits (P_1 and P_2) and corresponding servo channels provide active outputs to a tandem power actuator, whereas the servo channel powered by P_3 serves only as a model. All channels use secondary actuators mechanized to provide position summing of the outputs of channels 1 and 2. If one secondary actuator position reaches a predetermined threshold of disagreement with the other secondary actuators, the electronic detection and switching logic shuts off that channel. As hydraulic pressure collapses the affected secondary actuator centers and locks. If an active actuator is centered, the output stroke to the power spool is halved and gain and output rate are reduced. A second failure that will cause sufficient disagreement between the two remaining channels will shut down all channels via the detection and switching logic. All secondary actuators center and lock. In addition the power control actuator is by-passed by action of the power by-pass valves, allowing the centering actuator to center and lock the TVC. The centering valve is energized by the same intelligence that de-activates all channels.

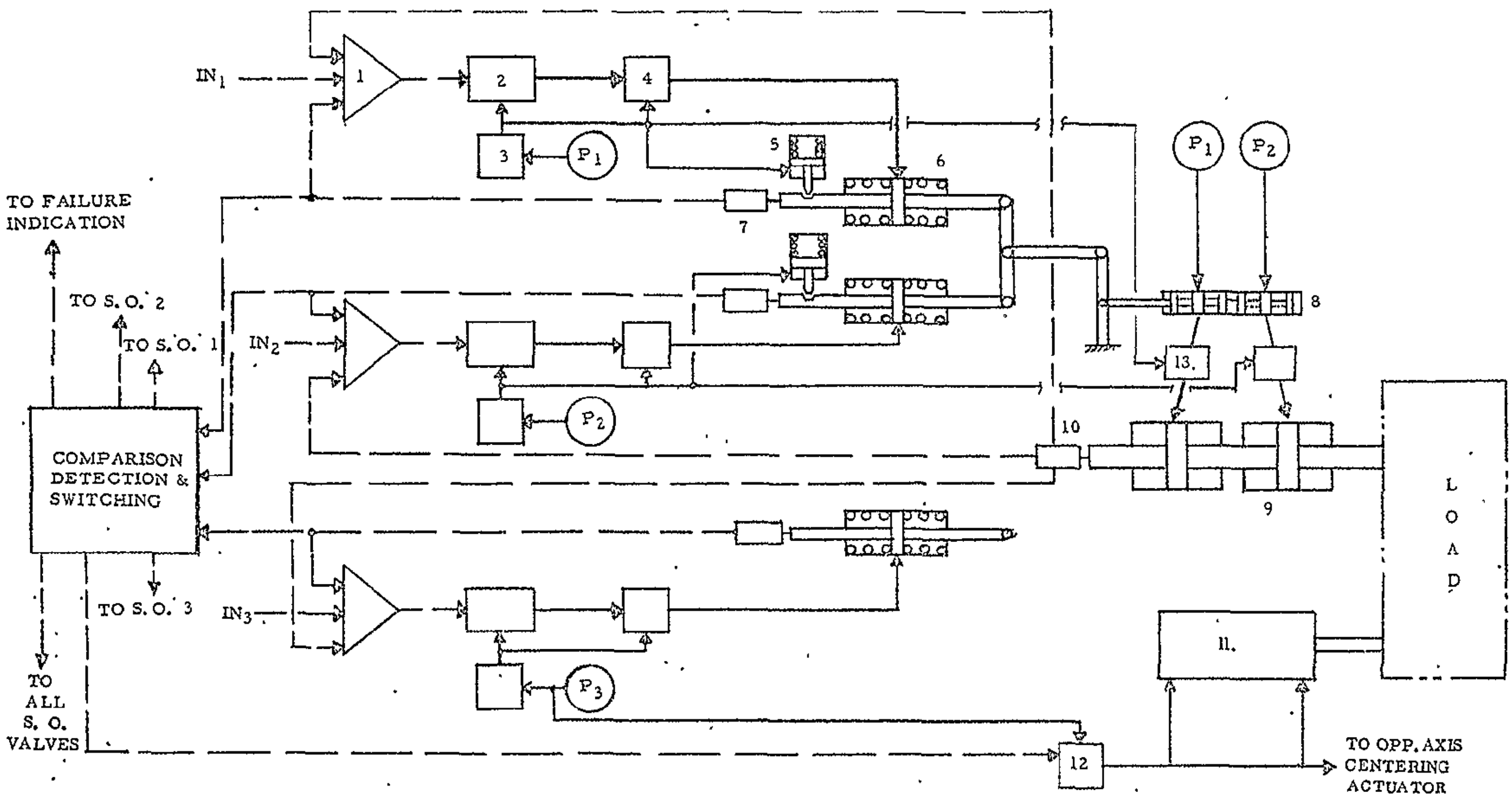
Each half of the tandem power actuator is sized to provide full hinge moment.



1. SERVO AMPLIFIER
2. 2 STAGE E/H SERVO VALVE
3. SOL SHUT OFF VALVE
4. BY-PASS VALVE
5. SECONDARY ACTUATOR
6. LVDT
7. ΔP SENSING
8. POWER SPOOL
9. SWITCHING VALVE
10. POWER ACTUATOR
11. ACTUATOR LOCK

ELECTRICAL - - -
HYDRAULIC ——
MECHANICAL

Figure 4-15. Orbiter TVC – Configuration 2



1. SERVO AMPLIFIER
2. 2 STAGE E/H SERVO VALVE
3. SOL. SHUT OFF VALVE
4. BYPASS
5. ACTUATOR CENTER LOCK
6. SECONDARY ACTUATOR
7. LVDT
8. POWER SPOOL
9. POWER ACTUATOR
10. LVDT
11. CENTERING & LOCKING ACTUATOR
12. SOL. CENTERING VALVE
13. BYPASS

ELECTRICAL ------
HYDRAULIC ——
MECHANICAL ═══

Figure 4-16. Orbiter TVC – Configuration 3

4.4.3.4 Booster TVC – Configuration 2 (Figure 4-17). This unit is monitorless (no detection or correction in the servo channels). It has one active hydraulic circuit that powers three servo channels. The first stage electrohydraulic valve outputs are flow summed to drive a common power spool, which in turn controls the power actuator. A bad output from any channel will attempt to drive the power spool. Mechanical feedback from the spool to each first stage will direct the two good channels to oppose the discrepant signal. Single faults in the servo upper stages, therefore, are overpowered and no switching is required. There is some degradation (unsymmetrical response, slight output position change, loss of control sensitivity about-null) after a failure depending on the nature of failure. The unit can survive many combinations of dual failures but cannot tolerate two like failures (e.g., two channel hardovers in the same direction).

Centering and locking provisions are incorporated into the power actuator. A second hydraulic circuit, P_2 (accumulator), is switched in to center the actuator should the active circuit, P_1, fail.

4.4.3.5 Booster TVC – Configuration 3 (Figure 4-18). This unit has one active hydraulic circuit, P_1. A second hydraulic circuit is used to power a monitoring channel and provide centering. The two servo channels are required to prevent a hardover output. Hydraulic logic is used (e.g., the monitors, comparator; and shutoff and bypass are the same as described in Section 4.3.3.1). A secondary actuator is used on the active channel. When there is sufficient disagreement between channels, pressure is removed from the secondary actuator and it centers. Center position is coincident with geometric center of the power actuator due to the mechanical feedback arrangement. The centering valve is triggered by either loss of P_1 hydraulic pressure or servo channel disagreement, to port centering power to the tandem power actuator. Should P_2 fail first, P_1 is available to center the power actuator.

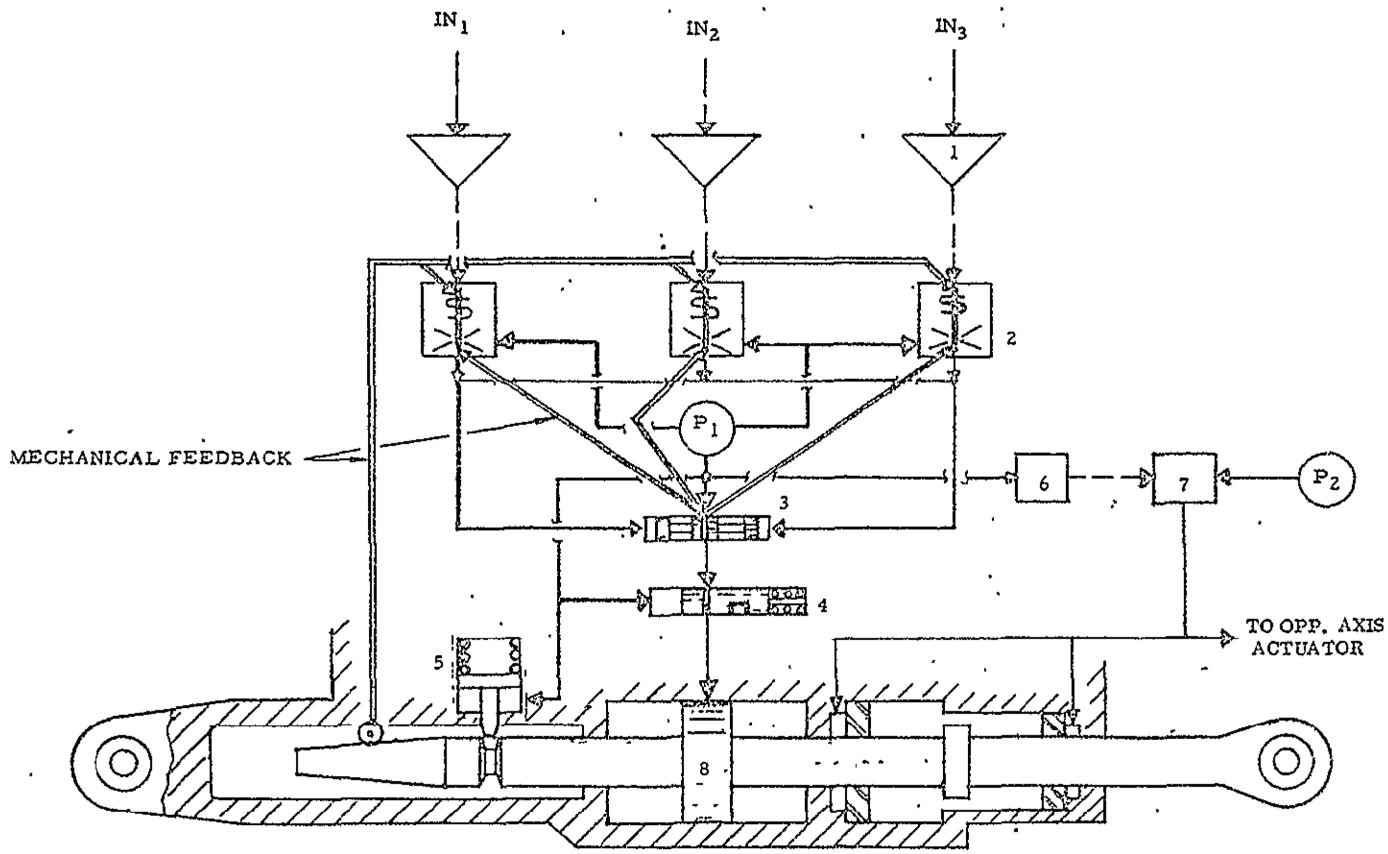
4.4.4 DIGITAL CONFIGURATION. One digital design is configured to satisfy the two fail operate requirements for aerodynamic surface controls. This configuration, Figure 4-19, was selected because it is all digital except for the output actuator. To extend digital design to include the output actuator (digital strut) would result in a very large and complex actuator. Torque motors rather than solenoid valves are used in the upper stages because of the severe cycling requirements (number and frequency).

Figure 4-20 is a functional schematic that shows detail operation of one channel. Command signals are received in incremental form. For each signal pulse, the torque motor drives the pilot stage spool. Pressure is boot strapped to drive the pilot stage spool after initial movement to reduce the signal level and pulse duration required on the torque motor. The pilot stage spool directs pressure to the main power spool, positioning it to open pressure to one cylinder port. The flow from the opposite cylinder port drives the digitizer spool against a stop. The displacement of the digitizer spool represents the incremental displacement of the actuator for one pulse. When the signal is removed from the torque motor, the pilot stage and power spool center, blocking the actuator ports and recycling the digitizer spool.

Two digitizers are used. The smaller gives 0.2% positional accuracy per requirements of Section 3, but has limited rate capability. Rate capability is based on the assumption that the maximum practical cycling rate attainable is 30 Hz. The larger digitizer gives maximum rate required but has a larger displacement per cycle.

The large digitizer would normally receive signals for large surface position changes. The small digitizer acts as a vernier for accurate positioning. The digitizers operate in sequence, not in parallel.

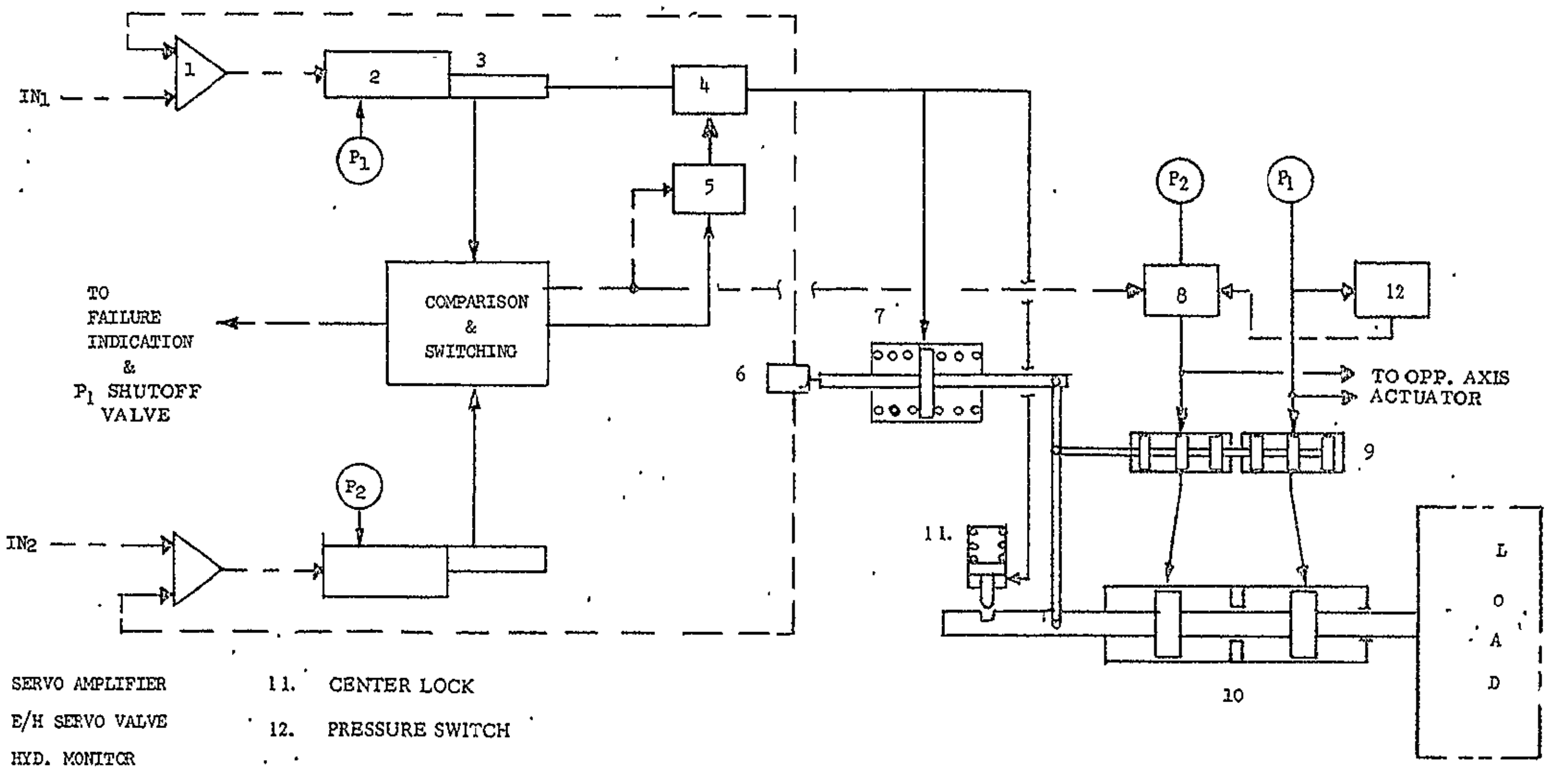
The fault correction consists only of overpressure sensing on the output actuators. If one actuator is not in agreement and is driven by the other three, it will be switched off when sufficient disagreement is reached. This "force voting" on the output requires four systems to enable correction after a second failure.



1. SERVO AMPLIFIER
2. SINGLE STAGE E/H SERVO VALVE
3. POWER SPOOL
4. BY-PASS
5. CYLINDER LOCK
6. PRESSURE SWITCH
7. SOLENOID OPERATED CENTERING VALVE
8. POWER PISTON

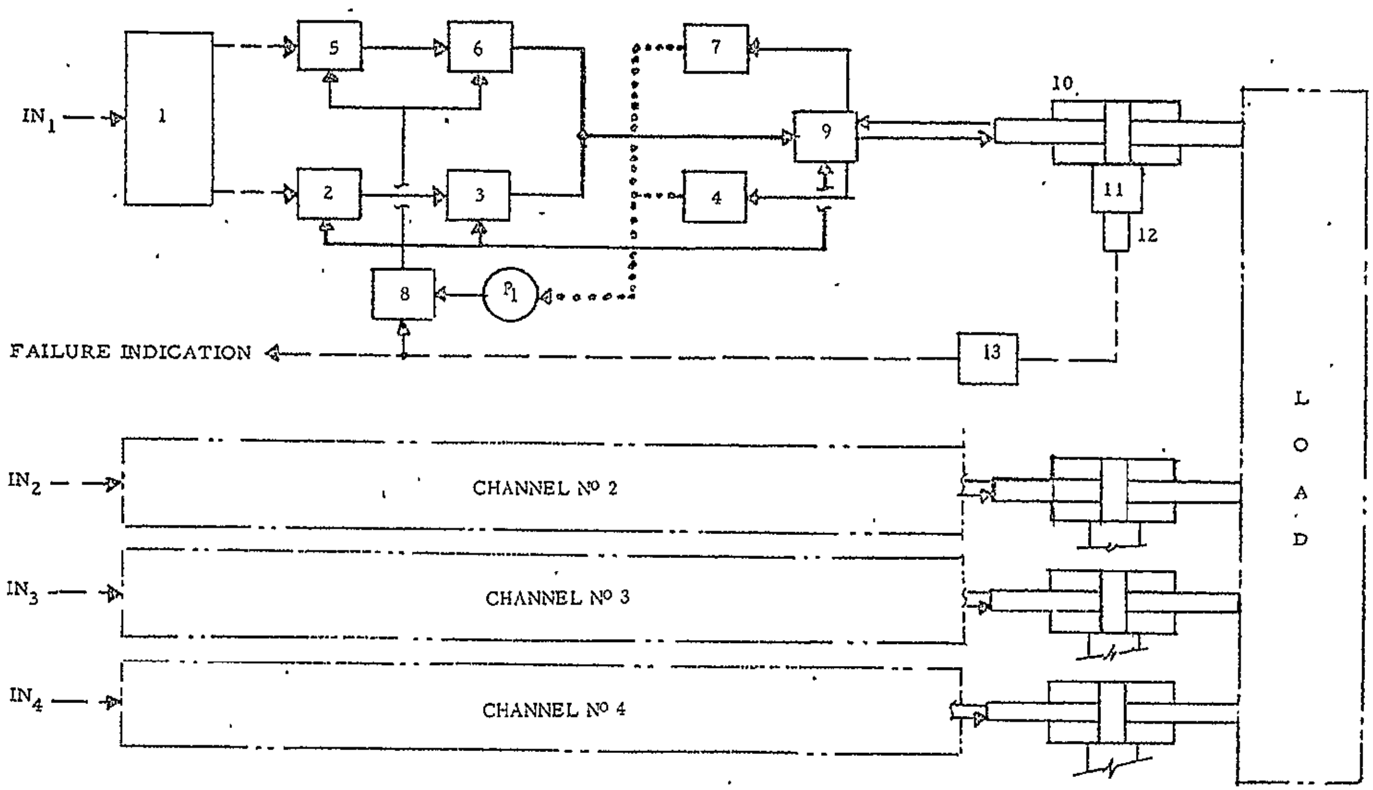
ELECTRICAL - - -
HYDRAULIC
MECHANICAL

Figure 4-17. Booster TVC – Configuration 2



1. SERVO AMPLIFIER
2. E/H SERVO VALVE
3. HYD. MONITOR
4. SHUTOFF & BYPASS VALVE
5. SOL LOCKOUT VALVE
6. LVDT
7. SECONDARY ACTUATOR (CENTERS)
8. CENTERING VALVE
9. POWER SPOOL
10. POWER & CENTERING ACTUATOR
11. CENTER LOCK
12. PRESSURE SWITCH

Figure 4-18. Booster TVC – Configuration 3



1. VALVE CONTROLLER
2. PILOT VALVE (LOW RATE)
3. POWER SPOOL (LOW RATE)
4. RETURN DIGITIZER SPOOL (LOW RATE)
5. PILOT VALVE (HIGH RATE)
6. POWER SPOOL (HIGH RATE)
7. RETURN DIGITIZER SPOOL (HIGH RATE)
8. SHUTOFF VALVE
9. PRESS OPERATED BYPASS
10. POWER ACTUATOR
11. OVERPRESSURE RELIEF
12. OVERPRESSURE SENSOR
13. TIME DELAY

ELECTRICAL - - - -
HYDRAULIC ———
HYD RETURN • • • •

Figure 4-19. Digital Configuration

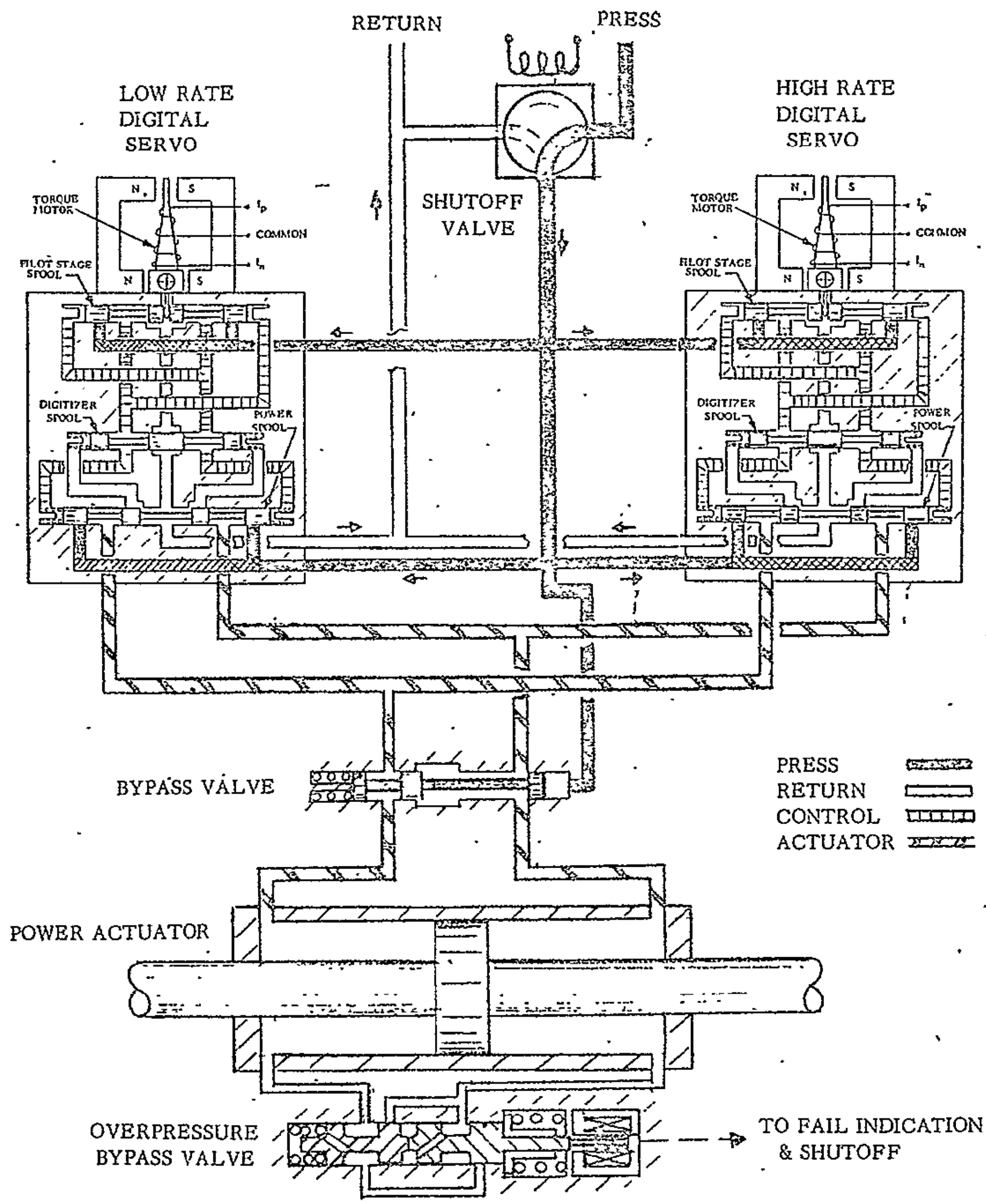


Figure 4-20. Digital Servo Schematic

SECTION 5

FAILURE MODES AND EFFECTS ANALYSIS

5.1 DEFINITIONS

Refer to pages 5-2 through 5-23 for the failure modes and effects analysis of the eleven candidate configurations.

Jamming failures of output linear hydraulic actuators are not considered in this analysis. This is in keeping with past experience that this type of failure mode is virtually non-existent.

The last column, entitled Failure Category, carries the following definitions:

Category I - Single failure or failure level that could cause loss of personnel or vehicle.

Category II - Single failure or failure level whereby the next associated failure could cause loss of personnel or vehicle.

Category III - Single failure or failure level that can be sustained without loss of primary mission objectives; or the single failure or failure level whereby the next associated failure could cause loss of primary mission objectives.

ASC		CONFIGURATION 1, ELEVATOR - BOOSTER AND ORBITER			USAGE: 2 PLACES		SHEET 1
ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
1	LOSS OF ONE HYD. SYSTEM	• FLUID LOSS • PUMP FAILURE	LOW PRESS WARNING	SECONDARY ACTUATOR BYPASSED	NO OUTPUT FROM ONE CIRCUIT. PASSIVE SEC. ACTUATOR DRIVEN BY 3 GOOD CHANNELS	NO DEGRADATION	III
*2	LOSS OF TWO HYD. SYSTEMS	SAME AS 1	SAME AS 1	TWO SECONDARY ACTUATORS BYPASSED	NO OUTPUT FROM TWO CIRCUITS. PASSIVE SEC. ACTUATORS DRIVEN BY 2 GOOD CHANNELS	DEGRADATION IN SURFACE RESPONSE GAIN & STIFFNESS	II
3	HARDOVER SIGNAL	• LOSS OF SIGNAL • ELECT. HARDOVER • OPEN FEED BACK • PLUGGED NOZZLE HYD. AMPLIFIER	FAULT INDICATION -DETECTION AND SWITCHING LOGIC	FORCE (ΔP) SIGNAL FROM SEC. ACTUATOR DE-ENERGIZES SHUTOFF. SEC. ACTUATOR BYPASSED.	OUTPUT FROM ONE CONTROL CHANNEL DE-ACTIVATED. ALL POWER CIRCUITS OPERATIVE	SAME AS 1	III
4	HARDOVER SIGNAL-WITH ONE CHANNEL DE-ACTIVATED	SAME AS 3	SAME AS 3	SAME AS 3	2 CONTROL CHANNELS DE-ACTIVATED. ALL POWER CIRCUITS OPERATIVE	SLIGHT DEGRADATION IN SURFACE RESPONSE	II
5	JAMMED BYPASS SPOOL - (WONT BYPASS)	• CONTAMINATION • BROKEN SPRING	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	HARDOVER SIGNAL-(SECOND FAILURE)		SAME AS 3	FORCE SIGNAL DE-ENERGIZES SHUTOFF. SEC. ACTUATOR WONT BYPASS	SECONDARY ACTUATOR BREAKS OUT DETENT AND DRIVES POWER SPOOL. GOOD CIRCUITS "BUCK" OUT HARDOVER ATTEMPT. SMALL SURFACE POSITION CHANGE	SAME AS 2	II
6	JAMMED SEC. ACTUATOR	• CONTAMINATION • STRUCTURAL FAILURE	SAME AS 3	FORCE SIGNAL DE-ENERGIZES SHUTOFF	3 GOOD CHANNELS MAINTAIN POSITION CONTROL AND DRIVE JAMMED ACTUATOR OUT OF DETENT. OVER-PRESSURE BYPASS OPENED ON AFFECTED POWER ACTUATOR	SAME AS 2	II
7	JAMMED MAIN POWER SPOOL	SAME AS 6	SAME AS 3	SAME AS 6	SAME AS 6	SAME AS 2	II

ASC	CONFIGURATION 1, ELEVATOR - BOOSTER AND ORBITER				USAGE: 2 PLACES		SHEET 2
ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
8	DETECTION AND SWITCHING LOGIC-ONE OPEN	BROKEN WIRE OR OPEN CONNECTION	FAULT INDICATION-DETECTION AND SWITCHING LOGIC	SHUTOFF VALVE DE-ENERGIZED SEC. ACTUATOR BY-PASSED	OUTPUT FROM ONE CONTROL CHANNEL DE-ACTIVATED. ALL POWER CIRCUITS OPERATIVE	NO DEGRADATION	III
9	SOL SHUTOFF VALVE - OPEN	SAME AS 8	NOT DETECTED	SAME AS 8	SAME AS 8	SAME AS 8	III
10	SOL SHUTOFF VALVE-STUCK IN ENERGIZED POSITION	CONTAMINATION	NOT DETECTED	NONE	NONE	NO EFFECT	III
	HARDOVER SIGNAL (2ND FAILURE)	-	SAME AS 8	SHUTOFF VALVE DE-ENERGIZED. VALVE WONT CYCLE	SAME AS 5	SAME AS 2	II
11	CONTROL MISMATCH, PRESS, VOLTAGE, GAIN, RESPONSE	-	NOT DETECTED UNLESS MISMATCHES REACH FAILURE DETECTION THRESHOLD. (NUISANCE TRIP)	NONE REQUIRED UNTIL FAILURE THRESHOLD IS REACHED, THEN RESULT IS SAME AS 3	EQUALIZING CIRCUITS ON EACH CHANNEL PROVIDE BIAS SIGNAL TO SERVO AMP TO FORCE ALL CHANNELS TO COMMON NULL.	NO EFFECT	III
12	SYNCHRONIZING SHAFT-BROKEN	STRUCTURAL FAILURE	NOT DETECTED	NONE	LARGE DEAD ZONE AND LOSS OF STIFFNESS	SERIOUS DEGRADATION IN PERFORMANCE & POSSIBLE LOSS OF PITCH CONTROL	I
13	INTERNAL LEAKAGE HIGH RATE	• FAILED SEAL ACTUATOR PISTONS • EROSION/WEAR LAPPED SPOOLS NOZZLE	NOT DETECTED UNLESS CHANNEL PERFORMANCE DEGRADES TO FAILURE THRESHOLD	NONE UNTIL FAILURE THRESHOLD IS REACHED, THEN RESULT IS SAME AS 3	• FLUID HEATING • LOWER SERVO GAIN • LOWER LOAD CAPABILITY ON ONE CIRCUIT	SAME AS 1	III
14	EXTERNAL LEAKAGE HIGH RATE	• ROD DYNAMIC SEAL • STATIC SEAL TO AMBIENT	LOW PRESS WARNING WHEN CIRCUIT FLUID IS DEPLETED	SECONDARY ACTUATOR BY-PASSED WHEN CIRCUIT IS DEPLETED.	SAME AS 1. POTENTIAL SAFETY HAZARD EXISTS WITH OIL SPILLAGE	SAME AS 1	III

ASC — CONFIGURATION 2, ELEVATOR - BOOSTER AND ORBITER — USAGE: 2 PLACES — SHEET 1

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
1	LOSS OF ONE HYD. SYSTEM	• FLUID LOSS • PUMP FAILURE	• LOW PRESS WARNING • FAULT INDICATION - DETECTION AND SWITCHING LOGIC	PASSIVE FAILURE DETECTED BY CROSS MONITORING. SHUT OFF VALVE DE-ENERGIZED. SEC ACTUATOR BYPASSED	NO OUTPUT FROM ONE CIRCUIT. PASSIVE SEC. ACTUATOR DRIVEN BY 3 GOOD CHANNELS	NO DEGRADATION	III
2	LOSS OF TWO HYD. SYSTEMS	SAME AS 1	SAME AS 1	TWO SECONDARY ACTUATORS BYPASSED	NO OUTPUT FROM TWO CIRCUITS. PASSIVE SEC. ACTUATORS DRIVEN BY 2 GOOD CHANNELS	DEGRADATION IN SURFACE RESPONSE GAIN & STIFFNESS	II
3	HARDOVER SIGNAL	• LOSS OF SIGNAL • ELECT. HARDOVER • OPEN FEEDBACK • PLUGGED NOZZLE HYD. AMPLIFIER	FAULT INDICATION - DETECTION & SWITCHING LOGIC	ACTIVE FAILURE DETECTED BY CROSS MONITORING. SHUT-OFF VALVE DE-ENERGIZED. SEC. ACTUATOR BYPASSED.	OUTPUT FROM ONE CONTROL CHANNEL DE-ACTIVATED. ALL POWER CIRCUITS OPERATIVE	SAME AS 1	III
4	HARDOVER SIGNAL - WITH ONE CHANNEL DE-ACTIVATED	SAME AS 3	SAME AS 3	SAME AS 3	2 CONTROL CHANNELS DE-ACTIVATED. ALL POWER CIRCUITS OPERATIVE	SLIGHT DEGRADATION IN SURFACE RESPONSE	II
5	JAMMED BYPASS SPOOL-(WONT BY-PASS)	• CONTAMINATION • BROKEN SPRING	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	HARDOVER SIGNAL (2ND FAILURE)	-	SAME AS 3	DETECTION SIGNAL DE-ENERGIZES SHUTOFF. SEC. ACTUATOR WONT BYPASS	3 GOOD CHANNELS MAINTAIN CONTROL. LOSS IN SERVO GAIN & RESPONSE	SAME AS 2	II
6	JAMMED SEC. ACTUATOR	• CONTAMINATION	NOT DETECTED	• EACH SECONDARY ACTUATOR HAS 800 LBS NET FORCE. 4 CHANNELS HAVE 3200 LBS AVAILABLE, APPROX 10 TIMES FORCE NEEDED TO SHEAR CHIP	• NO EFFECT IF JAM IS CLEARED	NO EFFECT	III
		• STRUCTURAL FAILURE		• JAM WOULD NOT CLEAR ONLY IF MASSIVE STRUCTURAL FAILURE OCCURRED	• SERVO ACTUATOR CAN'T FOLLOW COMMANDS. 2ND SERVOACTUATOR STALLED	• LOSS OF CONTROL. SURFACE POSITION FIXED	I
7	JAMMED MAIN POWER SPOOL	SAME AS 6	SAME AS 6	SAME AS 6	SAME AS 6	SAME AS 6	I

ASC	CONFIGURATION 2, ELEVATOR - BOOSTER AND ORBITER				USAGE: 2 PLACES		SHEET 2
ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
8	DETECTION AND SWITCHING LOGIC-ONE OPEN	BROKEN WIRE OR OPEN CONNECTION	SAME AS 3	SAME AS 3	SAME AS 3	SAME AS 1	III
9	SOL. SHUTOFF VALVE - OPEN	SAME AS 8	SAME AS 3	SAME AS 3	SAME AS 3	SAME AS 1	III
10	SOL. SHUTOFF VALVE - STUCK IN ENERGIZED POSITION	CONTAMINATION	NOT DETECTED	NONE	NONE	NO EFFECT	III
	HARDOVER SIGNAL (2ND FAILURE)		SAME AS 3	DETECTION SIGNAL TO SHUT-OFF VALVE. VALVE CAN'T SHUTOFF. SECONDARY ACTUATOR WON'T BY-PASS	3 GOOD CHANNELS MAINTAIN CONTROL. LOSS IN SERVO GAIN & RESPONSE	SAME AS 2	II
11	CONTROL MISMATCH - PRESS, VOLTAGE, GAIN, RESPONSE	-	NOT DETECTED UNLESS MISMATCHES REACH FAILURE DETECTION THRESHOLD (NUISANCE TRIP)	NONE REQUIRED UNTIL FAILURE THRESHOLD IS REACHED, THEN RESULT IS SAME AS 3	EQUALIZING CIRCUITS BETWEEN CHANNELS PROVIDE AVERAGING BIAS SIGNALS TO EACH SERVO AMP TO FORCE ALL CHANNELS TO A COMMON NULL	NO EFFECT	III
12	INTERNAL LEAKAGE - HIGH RATE	• FAILED SEAL ACTUATOR PISTONS • EROSION/WEAR LAPPED SPOOLS NOZZLE	NOT DETECTED UNLESS CHANNEL PERFORMANCE DEGRADED TO FAILURE THRESHOLD	NONE UNTIL FAILURE THRESHOLD IS REACHED, THEN RESULT IS SAME AS 3	• FLUID HEATING • LOWER SERVO GAIN • LOWER LOAD CAPABILITY ON ONE CIRCUIT	SAME AS 1	III
13	EXTERNAL LEAKAGE - HIGH RATE	• ROD DYNAMIC SEAL • STATIC SEAL TO AMBIENT	LOW PRESS WARNING WHEN CIRCUIT FLUID IS DEPLETED	SECONDARY ACTUATOR BY-PASSED WHEN CIRCUIT IS DEPLETED	SAME AS 1. POTENTIAL SAFETY HAZARD EXISTS WITH OIL SPILLAGE	SAME AS 1	III

ASC — CONFIGURATION 3, ELEVATOR - BOOSTER AND ORBITER CONFIGURATION 3, AILERON - ORBITER USAGE: 2 PLACES SHEET 1

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
1	LOSS OF ONE HYD. SYSTEM	•FLUID LOSS •PUMP FAILURE	LOW PRESS WARNING	NONE	NO OUTPUT FROM ONE CIRCUIT	NO DEGRADATION	III
2	LOSS OF TWO HYD. SYSTEMS	SAME AS 1	SAME AS 1	NONE	NO OUTPUT FROM 2 CIRCUITS	DEGRADATION IN SURFACE RESPONSE, GAIN & STIFFNESS	II
3	HARDOVER SIGNAL	•LOSS OF SIGNAL •ELECT HARDOVER •OPEN FEEDBACK	FAULT INDICATION-DETECTION AND SWITCHING LOGIC	FAILURE DETECTED BY CROSS MONITORING. POWER REMOVED FROM SERVO MOTOR AND BRAKE.	OUTPUT FROM ONE CHANNEL LOCKED. VELOCITY OUTPUT OF BALL SCREW (SECONDARY ACTUATOR) REDUCED 25%	SLIGHT DEGRATION IN SURFACE RESPONSE	III
4	HARDOVER SIGNAL-ONE CHANNEL OFF AND LOCKED	SAME AS 3	SAME AS 3	SAME AS 3	SAME AS 3, EXCEPT WITH 2 CHANNELS OFF, VELOCITY OF OUTPUT REDUCED 50%	SAME AS 3	II
5	LOSS OF TACH. FEEDBACK	•OPEN COIL •BROKEN WIRE	SAME AS 3	AFFECTED CHANNEL OUTPUT VELOCITY BECOMES NON LINEAR WITH RESPECT TO ERROR SIGNAL. IF FAILURE THRESHOLD REACHED, POWER REMOVED FROM SERVO MOTOR AND BRAKE	MOTOR ROTATION AT NULL. 3 GOOD CHANNELS COUNTER ROTATE TO PREVENT OUTPUT POSITION CHANGE. INCREASED POWER CONSUMPTION AT NULL. IF DETECTION REMOVES POWER FROM MOTOR, RESULT IS SAME AS 3.	SAME AS 3	III
6	LOSS OF 2 TACH. FEEDBACKS	SAME AS 5	SAME AS 3	SAME AS 5	SAME AS 4	SAME AS 3	II
7	JAMMED DIFFERENTIAL OUTPUT - SUMMING OUTPUT OF 2 MOTORS	•CONTAMINATION •BEARING FAILURE •STRUCTURAL FAILURE	NOT DETECTED	NONE. FAILURE EQUIVALENT TO 2 SIMULTANEOUS CHANNEL FAILURES. LOGIC DISABLED & ALL CHANNELS REMAIN ON.	SAME AS 4	SAME AS 3	II
	HARDOVER SIGNAL (2ND FAILURE)	–	SAME AS 3	WITH THIS ADDITIONAL FAILURE LOGIC WILL SHUT OFF THE NON JAMMED SYSTEMS	•ONE SERVOACTUATOR FIXED. 2ND SERVOACTUATOR STALLED (ELEVATOR) •ONE SERVOACTUATOR FIXED. OPP SIDE ACTUATOR STILL OPERABLE (AILERON)	•ELEVATOR FIXED. LOSS OF CONTROL •IF FIXED POSITION IS NEAR EXTREME POSITION - LOSS OF CONTROL	I I

ASC CONFIGURATION 3 ELEVATOR - BOOSTER AND ORBITER CONFIGURATION 3, AILERON - ORBITER SHEET 2

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
8	JAMMED STANDBY OUTPUT - BALL SCREW OR DIFFERENTIAL	SAME AS 7	NOT DETECTED	NONE- STANDBY NORMALLY GROUNDED AGAINST FUNK STRUT	NO EFFECT	NO EFFECT	III
9	JAMMED ACTIVE OUTPUT - BALL SCREW OR DIFFERENTIAL	SAME AS 7	SAME AS 8	WHEN FORCE LEVEL EXCEEDS FUNK STRUT BREAKOUT, STANDBY BALL SCREW CONTROLS OUTPUT FROM ALL CHANNELS	NO EFFECT	NO EFFECT	III
10	JAMMED HYDRAULIC POWER SPOOL	•CONTAMINATION	SAME AS 8	•BALL SCREW OUTPUT IS 800 LBS, SUFFICIENT TO SHEAR CONTAMINATION	•NO EFFECT IF JAM IS CLEARED	•NO EFFECT	III
		•STRUCTURAL FAILURE		•JAM WOULD NOT CLEAR ONLY IF MASSIVE STRUCTURAL FAILURE OCCURRED	•ELEVATOR SERVOACTUATORS STALLED	•LOSS OF CONTROL	I
					•ONE AILERON SERVOACTUATOR OPERABLE & ONE STALLED.	•IF FIXED ACTUATOR IS NEAR EXTREME POSITION - LOSS OF CONTROL	II
11	FAILED MOTOR BRAKE - (LOCKED MOTOR)	•OPEN COIL	SAME AS 3	ONE MOTOR LOCKED. FAILURE DETECTED BY CROSS MONITORING. POWER REMOVED FROM SERVO MOTOR AND BRAKE.	SAME AS 3	SAME AS 3	III
12	FAILED MOTOR BRAKE (WONT LOCK)	•JAMMED MECHANISM	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	HARDOVER SIGNAL (2ND FAILURE)	-	SAME AS 3	SAME AS 3	REMAINING CHANNELS BACK DRIVE THRU OPEN. NO OUTPUT UNLESS HYD POWER SPOOLS' FLOW FORCES & CENTERING SPRING FORCE LESS THAN MOTOR FIXED PHASE REACTING TORQUE.	SERIOUS DEGRADATION. LOSS OF CONTROL SENSITIVITY.	II
13	INTERNAL HYDRAULIC LEAKAGE - HIGH FLOW	•PISTON SEAL •EROSION/WEAR POWER SPOOL	NOT DETECTED	NONE	•FLUID HEATING •LOWER LOAD CAPABILITY ON ONE CIRCUIT	•SAME AS 1	III
14	EXTERNAL HYDRAULIC LEAKAGE - HIGH FLOW	•ROD DYNAMIC SEAL •STATIC SEAL TO AMBIENT	LOW PRESS WARNING WHEN FLUID CIRCUIT IS DEPLETED	NONE	SAME AS 1	SAME AS 1	III

ASC CONFIGURATION 1, AILERON - ORBITER USAGE: 2 PLACES SHEET 1

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
1	LOSS OF ACTIVE HYD. SYSTEM	•FLUID LOSS •PUMP FAILURE	LOW PRESS WARNING	ENGAGE VALVE SHIFTS AT LOW PRESS. LOCKOUT VALVE SHIFTS TO PREVENT RE-ENGAGEMENT	ACTIVE CIRCUIT BYPASSED. FIRST STANDBY ENGAGED NO PERFORMANCE DEGRADATION	NO DEGRADATION	III
2	LOSS OF STANDBY HYD. SYSTEM	SAME AS 1	SAME AS 1	SHUTOFF & BYPASS VALVE IN AFFECTED CIRCUIT SHIFTS TO BYPASS. LOCKOUT, SAME AS 1.	AFFECTED STANDBY CIRCUIT CANNOT BE ENGAGED	NO EFFECT	III
3	LOSS OF TWO HYD SYSTEMS - ONE ACTIVE	SAME AS 1	SAME AS 1	SAME AS 2. ENGAGE VALVE SHIFTS TO NEW POSITION.	REMAINING CIRCUIT ENGAGED. NO PERFORMANCE DEGRADATION	SAME AS 1	II
4	HARDOVER SIGNAL-ACTIVE OR MONITOR CHANNEL	•ELECT HARDOVER •PLUGGED NOZZLE HYD. AMPLIFIER	FAIL INDICATION - COMPARATOR SPOOL	COMPARATOR SPOOL SHIFTS, DUMPING PRESS. ENGAGE VALVE SHIFTS AT LOW PRESS. LOCKOUT, SAME AS 1	SAME AS 1.	SAME AS 1	III
5	HARDOVER SIGNAL-STANDBY OR MONITOR CHANNEL	SAME AS 4	SAME AS 4	COMPARATOR SPOOL SHIFTS, CYCLING SHUTOFF & BYPASS VALVE. LOCKOUT, SAME AS 1	FAILED CIRCUIT CANNOT BE ENGAGED. NO CHANGE TO ACTIVE CONTROL	SAME AS 1	III
6	HARDOVER SIGNAL-ON FIRST STAND-BY - (ACTIVE CHANNEL FAILED)	SAME AS 4	SAME AS 4	COMPARATOR SPOOL SHIFTS, DUMPING PRESS. TO HYD CPLING. ENGAGE VALVE SHIFTS AT LOW PRESS LOCKOUT, SAME AS 1	SAME AS 3	SAME AS 1	II
7	SUMMING AMPLIFIER - BAD OUTPUT	•ELECT OPEN •ELECT SHORT •OPEN FEEDBACK	NOT DETECTED	NONE	NONE. BAD SIGNAL BLOCKED BY VOTERS & DISENGAGED FROM VOTER LOGIC	NO EFFECT	III
8	SUMMING AMPLIFIER - 2 FAILED	SAME AS 7	NOT DETECTED	NONE	NONE. BAD SIGNAL BLOCKED BY VOTERS	NO EFFECT	II
9	JAMMED POWER SPOOL	CONTAMINATION	SAME AS 4	•SAME AS 4 - IF ACTIVE •SAME AS 5 - IF STANDBY	SAME AS 4 - IF ACTIVE SAME AS 5 - IF STANDBY	SAME AS 1	III
10	FAILED HYDRAULIC MONITOR	•BROKEN •BLOCKED RESTRICTOR	SAME AS 4	SAME AS 9	SAME AS 9	SAME AS 1	III

ASC CONFIGURATION 1, AILERON - ORBITER SHEET 2

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
11	JAMMED COMPARATOR (INNEUTRAL)	• CONTAMINATION • BROKEN SPRING	• NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	HARDOVER SIGNAL (2ND FAILURE)		• NOT DETECTED	NONE. IF ON ACTIVE CHANNEL CREW MUST SWITCH OFF ACTIVE SYSTEM HYD. PRESSURE	OUTPUT WILL FOLLOW HARDOVER. OPPOSITE AILERON WILL MOVE IN SAME DIRECTION	LARGE ROLL TRANSIENT. SAME AS 1 AFTER SWITCHING	I
12	JAMMED COMPARATOR SPOOL-END POSITION	CONTAMINATION	SAME AS 4	NONE	THIS FAILURE IS RESULT OF PREVIOUS CHANNEL FAILURE. HAS NO EFFECT ON SUBSEQUENT FAILURES	NO EFFECT	III
13	SOL. LOCKOUT VALVE-STUCK IN ENERGIZED POSITION	SAME AS 12	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	ERRATIC CHANNEL OUTPUT (2ND FAILURE)	-	SAME AS 4	COMPARATOR SPOOL CYCLES BETWEEN NEUTRAL & END POSITION, SHIFTING ENGAGE VALVE	FAILURE MODE OF LOCKOUT VALVE CANNOT PREVENT ERRATIC CHANNEL FROM RE-ENGAGING	INTERMITTENT SWITCHING BETWEEN CHANNELS CAUSE UNDESIRABLE BUT NOT CATASTROPIC TRANSIENTS	III
14	SOL LOCKOUT VALVE - OPEN	• BROKEN WIRE • OPEN COIL	NOT DETECTED	NONE IN FLIGHT	NONE IN FLIGHT. AFFECTED CHANNEL CANNOT BE ENGAGED DURING GRD CHECKOUT	NO EFFECT	III
15	JAMMED ENGAGE VALVE	• CONTAMINATION	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	HARDOVER-ACTIVE CHANNEL (2ND FAILURE)	-	SAME AS 4	NONE	DEFECTIVE CIRCUIT CANNOT BE DISENGAGED. OUTPUT GOES HARDOVER. OPP. AILERON MOVES IN SAME DIRECTION	LOSS OF CONTROL	I
16	FAILED VOTER-BAD OUTPUT	• ELECT OPEN • ELECT SHORT	SAME AS 4	SAME AS 4 - IF ACTIVE SAME AS 5 - IF STANDBY	SAME AS 4 - IF ACTIVE SAME AS 5 - IF STANDBY	SAME AS 1	III
17	INTERNAL LEAKAGE - HIGH RATE	• PISTON SEAL • EROSION/WEAR LAPPED SPOOLS	NOT DETECTED	NONE, UNLESS CREW SWITCH TO STANDBY	• FLUID HEATING • LOWER SERVO GAIN • LOWER LOAD CAPABILITY	POSSIBLE REDUCTION IN VEHICLE ROLL RESPONSE UNTIL CREW SWITCHES TO STANDBY	III
18	EXTERNAL LEAKAGE-HIGH RATE	• ROD DYNAMIC SEAL • STATIC SEAL TO AMBIENT	SAME AS 1	SAME AS 1	SAME AS 1. POTENTIAL SAFETY HAZARD WITH OIL SPILLAGE	SAME AS 1	III

ASC CONFIGURATION 2, AILERON - ORBITER USAGE: 2 PLACES SHEET 1

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
	NOTE: SPEED REDUCTION AFTER FAILURE DEPENDS ON CHANNEL FAILURE DUE TO SUMMING ACTION OF DIFFERENTIAL. FOR THIS ANALYSIS, THE FOLLOWING DEFINITIONS APPLY FOR ONE AILERON:						
				CHANNEL OUTPUT LOCKED	MAX. RATE CAPABILITY		
				NO. 1	75%		
				NO. 2	75%		
				NO. 3	50%		
				NO. 1 & 2	50%		
				NO. 1 & 3	25%		
				No. 2 & 3	25%		
1	LOSS OF NO. 1 POWER INPUT - ONE AILERON	•ELECT OPEN-MOTOR •JAM OR OPEN GEAR BOX	INPUT TO OUTPUT DETECTION	DETECTION SWITCHES OFF INPUT POWER TO AFFECTED CLUTCH	OUTPUT OF CHANNEL NO. 1 LOCKED. SERVOACTUATOR OPERATES AT 75% MAX. RATE	VEHICLE ROLL RESPONSE 88% OF NORMAL MAX.	III
2	LOSS OF NO. 2 POWER INPUT-ONE AILERON	SAME AS 1	SAME AS 1	SAME AS 1	SAME AS 1 EXCEPT CHANNEL NO. 2 LOCKED	SAME AS 1	III
3	LOSS OF NO. 3 POWER INPUT-ONE AILERON	SAME AS 1	SAME AS 1	SAME AS 1	OUTPUT OF CHANNEL NO. 3 LOCKED. SERVOACTUATOR OPERATES AT 50% MAX. RATE	VEHICLE ROLL RESPONSE 75% OF NORMAL MAX.	III
4	LOSS OF NO. 1 & NO. 2 POWER INPUT - ONE AILERON	SAME AS 1	SAME AS 1	SAME AS 1	OUTPUT OF NO. 1 & NO. 2 LOCKED. SERVOACTUATOR OPERATES AT 50% MAX. RATE	SAME AS 3	II
5	LOSS OF NO. 1 OR NO. 2 & NO. 3 POWER INPUT-ONE AILERON	SAME AS 1	SAME AS 1	SAME AS 1	OUTPUTS OF AFFECTED CHANNELS LOCKED. SERVO-ACTUATOR OPERATES AT 25% MAX. RATE	VEHICLE ROLL RESPONSE 62.5% OF NORMAL MAX	II
6	LOSS OF NO. 1 OR NO. 2 POWER INPUT -BOTH AILERONS	SAME AS 1	SAME AS 1	SAME AS 1	OUTPUTS OF AFFECTED CHANNELS LOCKED. SERVO-ACTUATOR OPERATES AT 50% MAX RATE	SAME AS 3	II
7	LOSS OF NO. 1 OR NO. 2 - ONE AILERON LOSS OF NO. 3 - ONE AILERON	SAME AS 1	SAME AS 1	SAME AS 1	SAME AS 5	SAME AS 5	II
8	LOSS OF NO. 3 - BOTH AILERONS	SAME AS 1	SAME AS 1	SAME AS 1	EACH SERVOACTUATOR OPERATES AT 50% MAX. RATE	VEHICLE ROLL RESPONSE 50% OF NORMAL MAX	II

ASC CONFIGURATION 2, AILERON - ORBITER SHEET 2

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
9	HARDOVER SIGNAL	•ELECT HARDOVER •ELECT OPEN •OPEN FEEDBACK	FAULT INDICATION - ELECTRONIC VOTER & AVERAGER	FAULTY SERVO AMPLIFIER CHANNEL SWITCHED OFF	3 CHANNELS STILL AVAILABLE FOR FAULT DETECTION	NO EFFECT	III
10	HARDOVER SIGNAL -ONE CHANNEL OFF	SAME AS 9	SAME AS 9	SAME AS 9	2 REMAINING CHANNELS IN CONTROL. CANNOT SUSTAIN ANOTHER LIKE FAILURE	NO EFFECT	II
11	ONE SOLENOID - STUCK IN ENERGIZED POSITION OR OPEN	•CONTAMINATION •MECH. JAM •ELECT OPEN	SAME AS 1	SAME AS 1	WORST CASE -SAME AS 3	SAME AS 3	III
12	SPRING CLUTH - JAMMED	•BEARING FAILURE	SAME AS 1	SAME AS 1	WORST CASE - SAME AS 3	SAME AS 3	III
13	SPRING CLUTCH - OPEN	•BROKEN SPRING	SAME AS 1	SAME AS 1	NO OUTPUT FROM DIFFERENTIAL IN ONE DIRECTION. 2 GOOD SYS. BACKDRIVE THE DISCONNECTED SHAFT	SERIOUS DEGRADATION, ONE AILERON DISCONNECTED IN ONE DIRECTION. ROLL RESPONSE 50% OF NORMAL IN ONE DIRECTION & 75% IN OPP. DIRECTION	I
14	LOSS OF DETECTION FEEDBACK	•ELECT OPEN	SAME AS 1	SAME AS 1	WORST CASE - SAME AS 3	SAME AS 3	III
15	JAMMED OUTPUT	•DIFFERENTIAL BEARING SEIZURE, STRUCT. FAILURE •OUTPUT ACTUATOR CONTAMINATION BEARING SEIZURE STRUCT. FAILURE	SAME AS 1	DETECTION SWITCHES OFF INPUT POWER TO ALL CLUTCHES	NO OUTPUT FROM DIFFERENTIAL. AILERON JAMMED.	•IF JAM IS AT TRAIL, VEHICLE ROLL RESPONSE 50% OF NORMAL •IF JAM IS AT OR NEAR EXTREME AILERON POSITION, LOSS OF CONTROL	II I

NOTE: WITH SELF-MONITORING (INTRA-CHANNEL), SURFACE HINGE MOMENT SATURATION OR SURFACE AGAINST THE STOPS FOR A PERIOD EXCEEDING DETECTION TIME DELAY WILL CAUSE MASSIVE SHUTDOWN AS SHOWN IN ITEM 16. ADDITIONAL CIRCUITRY MUST BE INTRODUCED TO PREVENT THIS OR RELIANCE MUST REST ON RESET OF INPUT POWER BY CREW.

TVC CONFIGURATION 1, BOOSTER AND ORBITER USAGE: 2 PLACES - PITCH AND YAW SHEET 1

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
1	LOSS OF ACTIVE HYD. SYSTEM	• FLUID LOSS • PUMP FAILURE	LOW PRESS WARNING	ENGAGE VALVE SHIFTS AT LOW PRESS. LOCKOUT VALVE SHIFTS TO PREVENT RE-ENGAGE-MENT.	ACTIVE CIRCUIT BYPASSED. STANDBY ENGAGED. NO PERFORMANCE DEGRADATION	NO DEGRADATION	III
2	LOSS OF STANDBY HYD. SYSTEM	SAME AS 1	SAME AS 1	SHUTOFF & BYPASS VALVE IN STANDBY CIRCUIT SHIFTS TO BYPASS. LOCKOUT, SAME AS 1.	STANDBY CIRCUIT CANNOT BE ENGAGED	NO EFFECT	III
3	LOSS OF TWO HYD SYSTEMS	SAME AS 1	SAME AS 1	AFTER 2ND FAILURE, BOTH CIRCUITS BYPASSED. CENTER-ING CURCUIT SWITCHED IN. LOCKOUT, SAME AS 1	• 1 TVC CENTERED ON ORBITER • 2 TVC SUBSYSTEMS CENT-ERED ON BOOSTER	• ORBITER CONTROL-LED BY REMAINING TVC. PERFORMANCE DEGRADED • BOOSTER CONTROL-ED BY 9 REMAINING TVC. PERFORMANCE DEGRADED	II II
4	LOSS OF CENTER-ING SYSTEM	(ACCUMULATOR FED BY APU HYD. CIR-CUIT)					
		• LOSS OF GAS PRESS	NONE	NONE	NONE BY ITSELF	NO EFFECT	III
		• FLUID LEAK TO AMBIENT	SAME AS 1	DEPENDING ON CIRCUIT LOSS, VALVES SHIFT PER 1 OR 2	SAME AS 1 OR 2	SAME AS 1 OR 2	II
	LOSS OF REMAIN-ING HYD CIRCUIT (2ND FAILURE)	SAME AS 1	SAME AS 1	AFTER 2ND FAILURE, BOTH CIRCUITS BYPASSED	• 1 TVC INOPERATIVE ON ORBITER - CANT CENTER • 2 TVC INOPERATIVE ON BOOSTER - CANT CENTER	• POSSIBLE LOSS OF CONTL ON ORBITER • POSSIBLE COLLI-SION OF ADJACENT ENGINES-BOOSTER	I I
5	HARDOVER SIGNAL -ACTIVE OR MONI-TOR CHANNEL	• ELECT HARDOVER • PLUGGED NOZZLE • OPEN FEEDBACK	FAIL INDICATION - COMPARATOR SPOOL	COMPARATOR SPOOL SHIFTS, DUMPING PRESS. ENGAGE VALVE SHIFTS AT LOW PRESS. LOCKOUT, SAME AS 1.	SAME AS 1.	SAME AS 1	III
6	HARDOVER SIGNAL STANDBY OR MONITOR CHANNEL	SAME AS 5	SAME AS 5	COMPARATOR SPOOL SHIFTS, CYCLING SHUTOFF & BYPASS VALVE. LOCKOUT SAME AS 1	SAME AS 2	NO EFFECT	III
7	HARDOVER SIGNAL-ONE CHANNEL OFF	SAME AS 5	SAME AS 5	SAME AS 3	SAME AS 3	SAME AS 3	II
8	POWER SPOOL JAM -ACTIVE OR MONI-TOR	CONTAMINATION	SAME AS 5	SAME AS 5	SAME AS 1	SAME AS 1	III

TVC CONFIGURATION 1, - BOOSTER AND ORBITER SHEE

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILU CATEG
9	POWER SPOOL JAM-STANDBY OR MONITOR	SAME AS 8	SAME AS 5	SAME AS 6	SAME AS 2	NO EFFECT	III
10	COMPARATOR SPOOL JAM - NEUTRAL ACTIVE CHANNEL	•CONTAMINATION •BROKEN SPRING	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	HARDOVER SIGNAL-ACTIVE CHANNEL (2ND FAILURE)	-	NOT DETECTED	NONE	•OUTPUT FOLLOWS COMMAND. REMAINING TVC CORRECTS ERROR. MINIMUM CONTROL	•LOSS OF CONTROL •POSSIBLE COLLISION OF ADJACENT ENGINES-BOOSTER	I
11	COMPARATOR SPOOL JAM - NEUTRAL - STANDBY CHANNEL	SAME AS 10	SAME AS 10	NONE	NONE	NO EFFECT	III
	HARDOVER SIGNAL ACTIVE OR STANDBY (2ND FAILURE)	-	SAME AS 5 - ACTIVE NOT DETECTED - STANDBY	SAME AS 5 - IF ACTIVE NONE - IF STANDBY	SAME AS 5 - IF ACTIVE NONE - IF STANDBY	SAME AS 1 NO EFFECT	III III
12	COMPARATOR SPOOL JAM-END POSITION	SAME AS 10	SAME AS 5	NONE	THIS FAILURE RESULT OF PREVIOUS CHANNEL FAILURE. HAS NO EFFECT ON SUBSEQUENT FAILURES	NO EFFECT	III
13	SOL. LOCKOUT VALVE-STUCK IN ENERGIZED POSITION	CONTAMINATION	NOT DETECTED	NONE	NONE BY ITSELF. CANNOT PREVENT ERRATIC CHANNEL FROM RE-ENGAGING	NO EFFECT UNLESS NUISANCE TRIPPING OCCURS - THEN TRANSIENT SWITCHING	III
14	SOL. LOCKOUT VALVE - OPEN	•BROKEN WIRE •OPEN COIL	NOT DETECTED	NONE IN FLIGHT	NONE IN FLIGHT. AFFECTED CHANNEL CANNOT BE RE-ENGAGED DURING GRD. CHECKOUT	NO EFFECT	III
15	ENGAGE VALVE JAM	CONTAMINATION	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	HARDOVER SIGNAL-ACTIVE CHANNEL (2ND FAILURE)	-	SAME AS 5	NONE. STANDBY CAN'T BE SWITCHED IN	OUTPUT FOLLOWS COMMAND REMAINING TVC'S CORRECT ERROR.	•LOSS OF CONTROL ON ORBITER •ADJACENT ENGINE COLLISION-BOOSTER	I I
16	INTERNAL LEAKAGE - HIGH RATE	•PISTON SEAL •EROSION/WEAR LAPPED SPOOLS	NOT DETECTED	NONE	•FLUID HEATING •LOWER SERVO GAIN •LOWER LOAD CAPABILITY	•REDUCED VEHICLE RESPONSE-ORBITER •NO DEGRADATION-BOOSTER	III
17	EXTERNAL LEAKAGE - HIGH RATE	•ROD DYN. SEAL STATIC SEAL TO AMBIENT	SAME AS 1	SAME AS 1 - IF ACTIVE CIRCUIT SAME AS 2 - IF STANDBY	SAME AS 1 OR 2. POTENTIAL SAFETY HAZARD WITH OIL SPILLAGE	•SAME AS 1 OR 2	III

TVC CONFIGURATION 2, - ORBITER | USAGE: 2 PLACES - PITCH AND YAW | SHEET 1

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
1	LOSS OF ONE HYD. CIRCUIT	• FLUID LOSS • PUMP LOSS	• LOW PRESS WARNING • FAULT INDICATION -DETECTION & SWITCHING LOGIC	PASSIVE FAILURE DETECTED BY CROSS MONITORING. SHUT OFF VALVE DE-ENERGIZED. SECONDARY ACTUATOR BY-PASSED.	STANDBY CIRCUIT SWITCHED INTO POWER ACTUATOR. PASSIVE SECONDARY ACTUATOR DRIVEN BY 2 GOOD CHANNELS.	NO DEGRADATION	III
2	LOSS OF BOTH ACTIVE HYD SYSTEMs	SAME AS 1	SAME AS 1	ALL CHANNELS BYPASSED. SEC. ACTUATORS CENTER.	STDBY CIRCUIT SWITCHED INTO POWER ACTUATOR. POWER ACTUATOR CENTERED & LOCKED	ORBITER CONTROLLED BY REMAINING TVC. PERFORMANCE DEGRADED	II
3	LOSS OF STANDBY HYD SYSTEM	SAME AS1	SAME AS 1	SAME AS 1	PASSIVE SECONDARY ACTUATOR DRIVEN BY 2 GOOD CHANNELS. NO CHANGE IN OUTPUT	SAME AS 1	III
4	HARDOVER SIGNAL	• LOSS OF SIGNAL • ELECT. HARDOVER • OPEN FEEDBACK • PLUGGED NOZZLE	FAULT INDICATION -DETECTION & SWITCHING LOGIC	ACTIVE FAILURE DETECTED BY CROSS MONITORING. SHUT OFF VALVE DE-ENERGIZED. SEC. ACTUATOR BYPASSED	PASSIVE SECONDARY ACTUATOR DRIVEN BY 2 GOOD CHANNELS	SAME AS 1	III
5	HARDOVER SIGNAL WITH ONE CHANNEL DE-ACTIVATED	SAME AS 4	SAME AS 4	SAME AS 2	SAME AS 2, EXCEPT ACTIVE HYD CIRCUITS CENTER THE POWER ACTUATOR	SAME AS 2	II
6	JAMMED BYPASS SPOOL (WONT BY-PASS)	• CONTAMINATION • BROKEN SPRING	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	HARDOVER SIGNAL (2ND FAILURE)	-	SAME AS 4	DETECTION SIGNAL DE-ENERGIZES SHUTOFF. SEC. ACTUATOR WONT BYPASS	2 GOOD CHANNELS OVER-POWER FAILED CHANNEL. LOSS IN SERVO GAIN, RESPONSE	DEGRADATION IN TVC CONTROL SENSITIVITY	II
7	JAMMED SECONDARY ACTUATOR	CONTAMINATION	NOT DETECTED	• EACH SEC. ACTUATOR HAS APPROX 400 LBS NET FORCE TO SHEAR CONTAMINATE OR 1200 LBS TOTAL.	• NO EFFECT IF JAM IS CLEARED	NO EFFECT	III
		STRUCTURAL FAILURE	NOT DETECTED	• JAM WOULD NOT CLEAR ONLY IF MASSIVE STRUCTURAL FAILURE OCCURRED	• SERVOACTUATOR CANT FOLLOW COMMAND. REMAINING TVC CORRECTS ERROR	POSSIBLE LOSS OF CONTROL-DEPENDENT ON JAM POSITION	I

TVC CONFIGURATION 2, - ORBITER SHEET 2

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
8	MAIN SPOOL-JAM	SAME AS 7	SAME AS 7	SAME AS 7	SAME AS 7	SAME AS 7	III OR I
9	DETECTION & SWITCHING LOGIC ONE OPEN	BROKEN OR OPEN CONNECTION	SAME AS 4	SAME AS 4	SAME AS 4	SAME AS 1	III
10	SOL SHUTOFF VALVE - STUCK IN ENERGIZED POSITION	CONTAMINATION	SAME AS 6	SAME AS 6	SAME AS 6	SAME AS 6	III OR II
11	SOL SHUTOFF - VALVE - OPEN	BROKEN OR OPEN CONNECTION	SAME AS 4	SAME AS 1	SAME AS 4	SAME AS 1	III
12	CONTROL MIS-MATCH - PRESS, VOLTAGE, GAIN RESPONSE	-	NOT DETECTED UNLESS MISMATCHES REACH FAILURE DETECTION THRESHOLD (NUISANCE TRIP)	NONE UNTIL FAILURE THRESHOLD IS REACHED, THEN RESULT IS SAME AS 4	EQUALIZING CIRCUITS BETWEEN CHANNELS PROVIDE AVERAGING BIAS SIGNALS TO EACH SERVO AMP TO FORCE ALL CHANNELS TO A COMMON NULL.	NO EFFECT	III
13	INTERNAL LEAKAGE - HIGH RATE	• FAILED SEAL ACTUATOR PISTONS • EROSION/WEAR LAPPED SPOOLS NOZZLE	NOT DETECTED UNLESS CHANNEL PERFORMANCE DEGRADED TO FAILURE THRESHOLD	NONE UNTIL FAILURE THRESHOLD IS REACHED, THEN RESULT IS SAME AS 4	• FLUID HEATING • LOWER SERVO GAIN • LOWER LOAD CAPABILITY ON ONE CIRCUIT	SAME AS 1	III
14	EXTERNAL LEAKAGE - HIGH RATE	• ROD DYNAMIC SEAL • STATIC SEAL TO AMBIENT	LOW PRESS WARNING WHEN FLUID CIRCUIT IS DEPLETED	SECONDARY ACTUATOR	SAME AS 1. POTENTIAL SAFETY HAZARD EXISTS WITH OIL SPILLAGE	SAME AS 1	III
15	SWITCHING VALVE - INTERSYSTEM LEAKAGE	• EROSION/WEAR • SEAL	SAME AS 1	SAME AS 1	FLUID FROM ONE SYSTEM LOST THRU LOW PRESS RELIEF OF 2ND SYSTEM. EFFECT SIMILAR TO ITEM 1	SAME AS 1	III
	LOSS OF HYD CIRCUIT (2ND FAILURE)		SAME AS 1	SAME AS 2	ACTUATOR CENTERED BY REMAINING HYD. CIRCUIT	SAME AS 2	II
16	SWITCHING VALVE-JAMMED, NORMAL POSITION	CONTAMINATION	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	LOSS OF ACTIVE HYD. CIRCUIT (2ND FAILURE)	-	SAME AS 1	SAME AS 1	STANDBY CIRCUIT CAN'T SWITCH IN. POWER ACTUATOR DRIVEN BY GOOD CIRCUIT	NO EFFECT	II

TVC CONFIGURATION 3, ORBITER — USAGE: 2 PLACES - PITCH AND YAW — SHEET 1

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
1	LOSS OF ONE HYD SYSTEM	•FLUID LOSS •PUMP FAILURE	•LOW PRESS WARNING •FAULT INDICATION -DETECTION & SWITCHING LOGIC	PASSIVE FAILURE DETECTED BY CROSS MONITORING. SHUT OFF VALVE DE-ENERGIZED. SECONDARY ACTUATOR CENTERED & LOCKED	INNER LOOP GAIN & OUTPUT RATE ON SERVOACTUATOR REDUCED 50%. OUTPUT CIRCUIT TO POWER ACTUATOR BYPASSED	VEHICLE RESPONSE LOWERED IN ONE PLANE 25%	III
2	LOSS OF BOTH ACTIVE HYD SYS.	SAME AS 1	SAME AS 1	ALL SHUTOFF VALVES DE-ENERGIZED. SEC. ACTUATORS CENTER & LOCK. CENTERING VALVE OPENS	NO CONTROL OUTPUT. CENTERING ACTUATOR CENTERS & LOCKS TVC	ORBITER CONTROLLED BY REMAINING TVC. PERFORMANCE DEGRADED	II
3	LOSS OF CENTERING HYD SYSTEM	SAME AS 1	SAME AS 1	MONITOR CHANNEL SHUTOFF-SECONDARY ACTUATOR CENTERED	NO EFFECT	NO EFFECT	II
	FAILED CHANNEL OR HYD CIRCUIT (2ND FAILURE)	-	SAME AS 1	ACTIVE CHANNELS SHUTOFF. ALL SEC. ACTUATORS CENTERED & LOCKED	SERVOACTUATOR BYPASSED. POSITION CONTROLLED BY LOAD. REMAINING TVC CORRECTS ERROR	POSSIBLE LOSS OF CONTROL	I
4	HARDOVER SIGNAL-ACTIVE CHANNEL	•ELECT HARDOVER •ELECT OPEN •PLUGGED NOZZLE •OPEN FEEDBACK	FAULT INDICATION -DETECTION & SWITCHING LOGIC	ACTIVE CHANNEL SHUTOFF. SECONDARY ACTUATOR CENTERED & LOCKED	SAME AS 1	SAME AS 1	III
5	HARDOVER SIGNAL WITH ONE CHANNEL OFF	SAME AS 4	SAME AS 4	SAME AS 2	SAME AS 2	SAME AS 2	II
6	JAMMED BYPASS-SECONDARY ACTUATOR(WONT BYPASS)	CONTAMINATION	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	HARDOVER SIGNAL (2ND FAILURE)	-	SAME AS 4	CHANNEL SHUTOFF. SECONDARY ACTUATOR WONT CENTER. REMAINING ACTIVE SECONDARY GOES HARDOVER IN OPP DIRECTION	OUTPUT WILL RESPOND TO COMMAND IN ONE DIRECTION ONLY	UNSYMMETRICAL VEHICLE RESPONSE	II
7	JAMMED BYPASS-POWER ACTUATOR (WONT BYPASS)	CONTAMINATION	NOT DETECTED	NONE	NONE	NO EFFECT	III
	HARDOVER SIGNAL (2ND FAILURE)	-	SAME AS 4	SAME AS 4	SAME AS 1 EXCEPT POWER ACTUATOR WONT BYPASS	NO EFFECT	III

TVC CONFIGURATION 3, - ORBITER SHEET 2

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
8	SECONDARY ACTUATOR JAM - ACTIVE CHANNEL	• STRUCTURAL FAILURE	SAME AS 4	ACTIVE CHANNEL SHUTOFF. SECONDARY ACTUATOR WONT CENTER	INNER LOOP GAIN & OUTPUT RATE REDUCED IN DIRECTION OPP. JAM	UNSYMMETRICAL VEHICLE RESPONSE	II
		• CONTAMINATION	NOT DETECTED	EACH SEC. ACTUATOR HAS SUFFICIENT FORCE TO SHEAR CONTAMINANT	NO EFFECT IF JAM IS CLEARED WITHIN FAILURE DETECTION(POSITION)THRESHOLD	NO EFFECT	III
9	JAMMED POWER SPOOL	• STRUCTURAL FAILURE	SAME AS 4	MONITOR CHANNEL SHUTOFF. SEC. ACTUATOR CENTERS	SERVO ACTUATOR FIXED. REMAINING TVC CORRECTS ERROR	LOSS OF CONTROL IF JAM OCCURRED AT ACTUATOR EXTREME POSITION	I
		• CONTAMINATION	NOT DETECTED	EACH SEC. ACTUATOR HAS SUFFICIENT FORCE TO SHEAR CONTAMINANT	NO EFFECT IF JAM IS CLEARED	NO EFFECT	III
10	DETECTION AND SWITCHING LOGIC-ONE OPEN	BROKEN OR OPEN CONNECTION	SAME AS 4	SAME AS 4	SAME AS 1	SAME AS 1	III
11	SOL SHUTOFF - STUCK IN ENERGIZED POSITION	CONTAMINATION	NOT DETECTED	NONE	NONE	NO EFFECT	III
	HARDOVER SIGNAL (2ND FAILURE)	-	SAME AS 4	SAME AS 6.	SAME AS 6	SAME AS 6	II
12	SOL SHUTOFF - OPEN	• BROKEN OR OPEN CONNECTION	SAME AS 4	SAME AS 4	SAME AS 1	SAME AS 1	III
13	INTERNAL LEAKAGE-HIGH RATE	• FAILED SEAL ACTUATOR PISTONS. • EROSION/WEAR LAPPED SPOOLS NOZZLE	NOT DETECTED UNLESS CHANNEL PERFORMANCE DEGRADED TO FAILURE THRESHOLD	NONE UNTIL FAILURE THRESHOLD IS REACHED, THEN RESULT IS SAME AS 4	• FLUID HEATING • LOWER SERVO GAIN • LOWER LOAD CAPABILITY ON ONE CIRCUIT	NO DEGRADATION UNLESS ONE CHANNEL SHUTOFF, THEN SAME AS 1	III
14	EXTERNAL LEAKAGE-HIGH RATE	• ROD DYNAMIC SEAL • STATIC SEAL TO AMBIENT	SAME AS 1 WHEN FLUID CIRCUIT IS DEPLETED.	SAME AS 1	SAME AS 1. POTENTIAL SAFETY HAZARD EXISTS WITH OIL SPILLAGE	SAME AS 1	III

TVC CONFIGURATION 2 - BOOSTER | USAGE: 2 PLACES - PITCH AND YAW | SHEET 1

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
1	LOSS OF ACTIVE HYD SYSTEM	•FLUID LOSS • PUMP FAILURE	LOW PRESSURE WARNING	POWER PISTON BYPASSED. CENTERING VALVE OPENED	NO OUTPUT FROM 1 TVC. TVC CENTERED & LOCKED	VEHICLE CONTROLLED BY REMAINING TVC SUBSYSTEM. SMALL DEGRADATION IN VEHICLE RESPONSE	III
2	LOSS OF CENTERING SYSTEM	SAME AS 1	SAME AS 1	NONE	NONE BY ITSELF	NO EFFECT	III
	LOSS OF ACTIVE SYS. (2ND FAILURE)	SAME AS 1	SAME AS 1	SAME AS 1	NO OUTPUT FROM AFFECTED TVC & WONT CENTER	POSSIBLE COLLISION OF ADJACENT ENGINES	I
3	HARDOVER SIGNAL INPUT- CONTROL FEEDBACK	•ELECT HARDOVER •PLUGGED NOZZLE	NOT DETECTED	ACTIVE FAULT OPPOSED BY 2 GOOD CHANNELS	SMALL NULL SHIFT OF POWER SPOOL. SLIGHT OUTPUT POSITION CHANGE	NEGLIGIBLE DEGRADATION	II
4	OPEN INPUT CONTROL FEEDBACK	•LOSS OF SIGNAL •OPEN CABLE •BROKEN FEEDBACK WIRE	NOT DETECTED	PASSIVE FAULT OPPOSED BY 2 GOOD CHANNELS	REDUCTION IN SENSITIVITY ABOUT NULL	NEGLIGIBLE DEGRADATION	II
5	2 HARDOVERS IN OPP. DIRECTION	SAME AS 3	NOT DETECTED	2 CHANNELS CANCEL EACH OTHER OUT. OUTPUT CONTROLLED BY 3RD CHANNEL	RESPONSE 33% OF NORMAL ON AFFECTED SERVOACTUATOR	SAME AS 4	II
6	2 HARDOVERS IN SAME DIRECTION	SAME AS 3	NOT DETECTED	GOOD CHANNEL OVERCOME BY 2 LIKE FAILURES.	SERVOACTUATOR GOES HARDOVER	POSSIBLE COLLISION OF ADJACENT ENGINES	I
7	2 OPENS	SAME AS 4	NOT DETECTED	SAME AS 5	SAME AS 5	SAME AS 4	II
8	1 OPEN AND 1 HARDOVER	–	NOT DETECTED	-	DEPENDING ON FAILURE COMBINATIONS, OUTPUT WILL BE LIMITED TO CONTROL IN ONE DIRECTION ONLY	POSSIBLE COLLISION OF ADJACENT ENGINES	I
9	JAMMED POWER SPOOL	•CONTAMINATION	NOT DETECTED	FORCE (PRESS) ON SPOOL AREA SUFFICIENT TO SHEAR CONTAMINANT	NONE	NO EFFECT	III
		•STRUCTURAL FAILURE	NOT DETECTED	NONE	OUTPUT FIXED AT JAMMED POSITION	SAME AS 8	I

TVC CONFIGURATION 2 - BOOSTER

SHEET 2

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
10	JAMMED BYPASS SPOOL-WONT BYPASS	CONTAMINATION	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	LOSS OF ACTIVE HYD SYSTEM (2ND FAILURE)	-	SAME AS 1	CENTERING VALVE OPENED	NO OUTPUT. CENTERING RATE IS SLOW, CONTROLLED BY LEAKAGE RATE ACROSS POWER SPOOL	SAME AS 1	III
11	INTERNAL LEAKAGE - HIGH RATE	• FAILED SEAL ACTUATOR PISTON • EROSION/WEAR LAPPED SPOOLS NOZZLE	NOT DETECTED	NONE	• FLUID HEATING • LOWER SERVO RESPONSE • LOWER LOAD CAPABILITY	NEGLIGIBLE DEGRADATION	III
12	EXTERNAL LEAKAGE - HIGH RATE	• ROD DYNAMIC SEAL • ANY STATIC SEAL TO AMBIENT	SAME AS 1	SAME AS 1	SAME AS 1	SAME AS 1	III

TVC CONFIGURATION 3 - BOOSTER USAGE: 2 PLACES - PITCH AND YAW SHEET 1

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
1	LOSS OF ACTIVE HYD SYSTEM	• FLUID LOSS • PUMP FAILURE	• LOW PRESS WARNING • FAULT INDICATION-COMPARATOR	SHUTOFF & BYPASS VALVE SHIFTS AT LOW PRESS. SECONDARY ACTUATOR CENTERS. CENTERING VALVE OPENS	POWER ACTUATOR CENTERED BY CENTERING CIRCUIT & LOCKED	VEHICLE CONTROLLED BY REMAINING TVC SUBSYSTEMS. SMALL DEGRADATION IN VEHICLE RESPONSE	III
2	LOSS OF CENTERING HYD SYSTEM	SAME AS 1	SAME AS 1	SAME AS 1	POWER ACTUATOR CENTERED BY ACTIVE CIRCUIT. LOCK DOES NOT ENGAGE	SAME AS 1	III
3	HARDOVER SIGNAL ACTIVE OR MONITOR CHANNEL	• ELECT HARDOVER • ELECT OPEN • PLUGGED NOZZLE • OPEN FEEDBACK	FAULT INDICATION-COMPARATOR	COMPARATOR SPOOL SHIFTS, DUMPING PRESS TO BYPASS. SECONDARY ACTUATOR CENTERS. CENTERING VALVE OPENS	SAME AS 1	SAME AS 1	III
4	JAMMED SERVO SPOOL	CONTAMINATION	SAME AS 3	SAME AS 3	SAME AS 1	SAME AS 1	III
5	JAMMED BYPASS VALVE (WONT BYPASS)	• CONTAMINATION • BROKEN SPRING	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	II
	HARDOVER SIGNAL (2ND FAILURE)	-	SAME AS 3	COMPARATOR SPOOL SHIFTS, DUMPING PRESS TO BYPASS. SECONDARY ACTUATOR CANT CENTER. CENTERING VALVE OPENS	OUTPUT FOLLOWS COMMAND HARDOVER	COLLISION OF ADJACENT ENGINES	I
6	FAILED HYDRAULIC MONITOR	• BROKEN FLAPPER • EXCESSIVE WEAR • BLOCKED RESTRICTOR	SAME AS 3	SAME AS 3	SAME AS 1	SAME AS 1	III
7	JAMMED COMPARATOR SPOOL-NEUTRAL	SAME AS 3	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	II
	HARDOVER SIGNAL (2ND FAILURE)	—	NOT DETECTED	NONE	OUTPUT FOLLOWS COMMAND HARDOVER	COLLISION OF ADJACENT ENGINES	I

TVC CONFIGURATION 3 - BOOSTER							SHEET 2
ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
8	JAMMED COMPARATOR SPOOL-END POSITION	SAME AS 5	SAME AS 3	NONE	THIS FAILURE RESULT OF PREVIOUS FAILURE. HAS NO EFFECT SINCE POWER ACTUATOR IS CENTERED & LOCKED	NO EFFECT	III
9	JAMMED SECONDARY ACTUATOR	•CONTAMINATION	NOT DETECTED	•SECONDARY ACTUATOR HAS SUFFICIENT FORCE TO SHEAR CONTAMINANT	NONE	NO EFFECT	III
		•STRUCTURAL FAILURE	NOT DETECTED	•JAM WOULD NOT CLEAR ONLY IF MASSIVE STRUCTURAL FAILURE OCCURRED	SERVOACTUATOR CANT FOLLOW COMMAND	POSSIBLE COLLISION OF ADJACENT ENGINES	I
10	SOL LOCKOUT VALVE-FAILED OPEN	•BROKEN OR OPEN CONNECTION	NOT DETECTED	NONE IN FLIGHT	NONE IN FLIGHT. CONTROL CANNOT BE RE-ENGAGED DURING GROUND CHECKOUT	SAME AS 1	III
11	SOL LOCKOUT VALVE - STUCK IN ENERGIZED POSITION	CONTAMINATION	NOT DETECTED	NONE	NONE BY ITSELF. CANNOT PREVENT ERRATIC CHANNEL FROM RE-ENGAGING	NO EFFECT UNLESS NUISANCE TRIPPING OCCURS - THEN TRANSIENT SWITCHING	III
12	CENTERING VALVE - FAIL OPEN OR INTERNAL LEAKAGE	•FAILED SEAL •EROSION/WEAR LAPPED SPOOL	NOT DETECTED	NONE	•FLUID HEATING	NO EFFECT	III
13	INTERNAL LEAKAGE - HIGH RATE	SAME AS 12	NOT DETECTED	NONE	•FLUID HEATING •LOWER SERVO GAIN •LOWER LOAD CAPABILITY	•NEGLIGIBLE DEGRADATION	III
14	EXTERNAL LEAKAGE-HIGH FLOW	•ROD DYNAMIC SEAL •ANY STATIC SEAL TO AMBIENT	SAME AS 1	SAME AS 1	SAME AS 1	SAME AS 1	III

ASC DIGITAL CONFIGURATION - AILERON - ORBITER					USAGE: 2 PLACES-LEFT AND RIGHT		SHEET 1
ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
1	LOSS OF ONE HYD SYSTEM	•FLUID LOSS •PUMP FAILURE	•LOW PRESS WARNING	POWER ACTUATOR BYPASSED	NO OUTPUT FROM ONE CIRCUIT. REDUCTION IN ACTUATOR STIFFNESS	SMALL DEGRADATION IS SYSTEM STIFFNESS	III
2	LOSS OF TWO HYD SYSTEMS	SAME AS 1	SAME AS 1	2 ACTUATORS BYPASSED	NO OUTPUT FROM TWO CIRCUITS. 67% HINGE MOMENT AVAILABLE & REDUCTION IN SERVOACTUATOR STIFFNESS	DEGRADATION IN SYSTEM STIFFNESS & ROLL RESPONSE	II
3	HARDOVER INPUT	•SHORTED SWITCH •JAMMED TORQUE MOTOR ARMATURE •JAMMED PILOT STAGE SPOOL	FAULT INDICATION -OVERPRESSURE SENSOR	OUTPUT ACTUATOR OPPOSED BY 3 GOOD CIRCUITS-OVER PRESSURE SENSOR SIGNALS SHUTOFF VALVE TO OPEN. POWER ACTUATOR BYPASSED	SAME AS 1	SAME AS 1	III
4	2 HARDOVERS	SAME AS 3	SAME AS 3	SAME AS 3	SAME AS 2	SAME AS 2	II
5	LOSS OF CHANNEL OUTPUT	•LOSS OF SIGNAL •BROKEN WIRE	SAME AS 3	SAME AS 3	SAME AS 1	SAME AS 1	III
6	JAMMED DIGITIZER SPOOL - ANY POSITION	CONTAMINATION	SAME AS 3	OUTPUT ACTUATOR CANT RESPOND TO COMMAND. RESULT SAME AS 3	SAME AS 1	SAME AS 1	III
7	JAMMED POWER SPOOL-BLOCKING POSITION	CONTAMINATION	SAME AS 3	SAME AS 3	SAME AS 1	SAME AS 1	III
8	JAMMED POWER SPOOL-END POSITION	CONTAMINATION	SAME AS 3	SAME AS 3 WHEN COMMAND DRIVES GOOD CIRCUITS TO OPPOSE AFFECTED ACTUATOR OUTPUT	SAME AS 1	SAME AS 2	III
9	SOL SHUTOFF - FAILED OPEN	BROKEN OR OPEN CONNECTION	NOT DETECTED	SAME AS 1	SAME AS 1	SAME AS 1	III

ASC DIGITAL CONFIGURATION - AILERON - ORBITER

SHEET 2

ITEM	FAILURE MODE	PRIMARY CAUSE	DETECTED BY	FAILURE CORRECTION	EFFECT ON SYSTEM	EFFECT ON VEHICLE	FAILURE CATEGORY
10	JAMMED PILOT SPOOL- NEUTRAL	CONTAMINATION	SAME AS 3	SAME AS 6	SAME AS 1	SAME AS 1	III
11	JAMMED PILOT SPOOL- END POSITION	CONTAMINATION	SAME AS 3	SAME AS 3	SAME AS 1	SAME AS 1	III
12	SOL SHUTOFF - STUCK IN ENERGIZER POSITION	CONTAMINATION	NOT DETECTED	NONE	NONE BY ITSELF	NO EFFECT	III
	HARDOVER SIGNAL (2ND FAILURE)	-	SAME AS 3	OVERPRESSURE RELIEF OPERATES WHEN ACTUATOR IS DRIVEN BY GOOD CIRCUITS, ACTUATOR CANT BE BYPASSED	67% HINGE MOMENT CAPABILITY. REDUCTION IN STIFFNESS	SAME AS 2	II
13	INTERNAL LEAKAGE - HIGH RATE	• FAILED SEAL ACTUATOR PISTON • EROSION/WEAR POWER SPOOL	NOT DETECTED	NONE	ONE CIRCUIT LOSES EFFECTIVE STIFFNESS. SURFACE POSITION HELD BY REMAINING CIRCUITS	SAME AS 1	III
14	EXTERNAL LEAKAGE - HIGH RATE	• ROD DYNAMIC SEAL • STATIC SEAL TO AMBIENT	SAME AS 1	SAME AS 1	SAME AS 1	SAME AS 1	III

SECTION 6

PARAMETERS AND DATA

6.1 TRADE-OFF PARAMETERS

This section deals with the parameters used in the trade-off evaluation and the origin of data used. Weight is displayed where possible in parametric form and serves as the criteria for establishing the point design weights in Section 7 for the various configurations. All other parameters listed below are used to make qualitative comparisons.

Weight

Reliability

Maintainability

Performance

Checkout capability

Cost

6.2 WEIGHT

6.2.1 HYDRAULIC TRANSMISSION. Figure 6-1 is a parametric curve of specific weight of transmission tubing versus flow. The two lines represent the end limits of 0 ft length and 200 ft length. The 0 line represents a minimum for short runs. The 200 ft line is based on 500 psi allowable pressure drop. This line was modified from Vickers data.[20] The Vickers data were based on 3000-psi circuits where maximum efficiency (total vehicle minimum weight impact) occurred with approximately 33% of maximum pump discharge lost in tubing pressure drop at maximum flow. For this study, the ground rule circuit pressure is 4000 psi. The ΔP allowed in the tubing is 500 psi or 12% of maximum pump discharge pressure. This is to conserve power because the pump discharge pressure drops to 2500 psi at maximum flow. Limiting the tubing drop would impose sizable weight penalties if performance were required at low fluid temperatures. For the space shuttle, the operating oil temperature is assumed to be 70°F or warmer where friction (viscosity) losses are low.

The tubing weights are based on using AM 350 high strength stainless steel and permanent joint fittings.

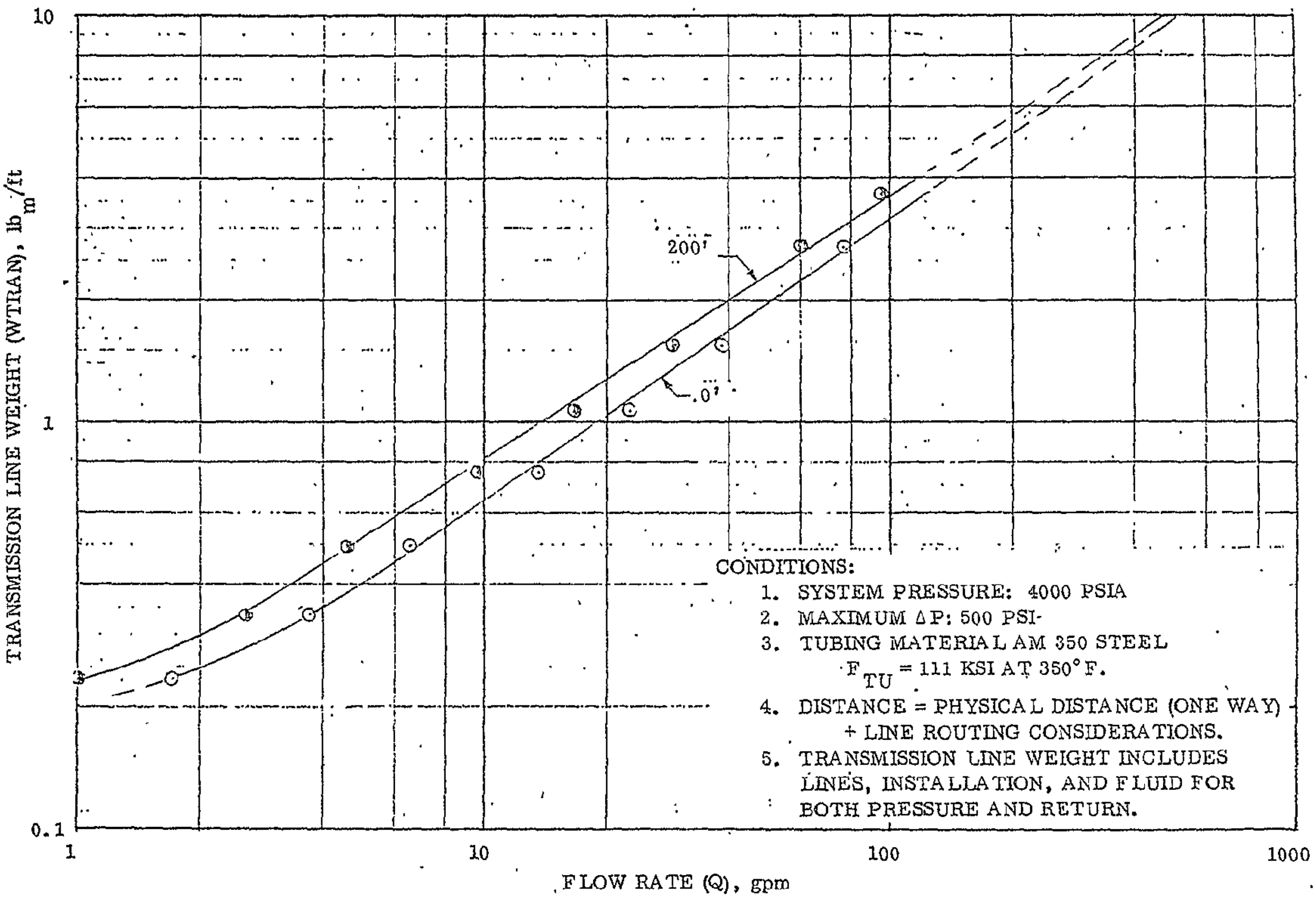


Figure 6-1. Transmission Line Weight

6.2.2 HYDRAULIC POWER GENERATION. A typical hydraulic power generation circuit schematic is presented in Figure 6-2. The total weight of a circuit is broken down into the following elements:

a. Tubing and fluid in tubing

b. Pump (dry)

c. Reservoir (dry)

d. Miscellaneous components (dry)

e. Fluid in components

6.2.2.1 Power Generation Tubing. The tubing within the power generation circuit cannot be estimated from Figure 6-1 because less pressure drop is allowed. Figure 6-1 applies to transmission lines only, where up to 500 psi drop is allowed. The power generation tubing is based on an allowable pressure drop of 1 psi/ft. Figure 6-3 shows this relationship. Comparing Figure 6-3 to Figure 6-1, one can see the specific weight of tubing is heavier. Another contributing factor to the increased specific weight is the addition of a pump case drain line.

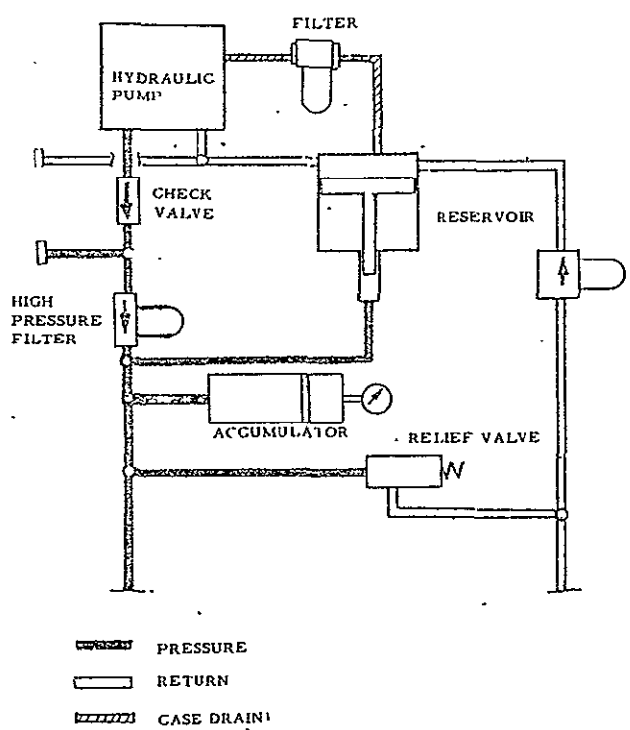


Figure 6-2. Hydraulic Power Generation Circuit

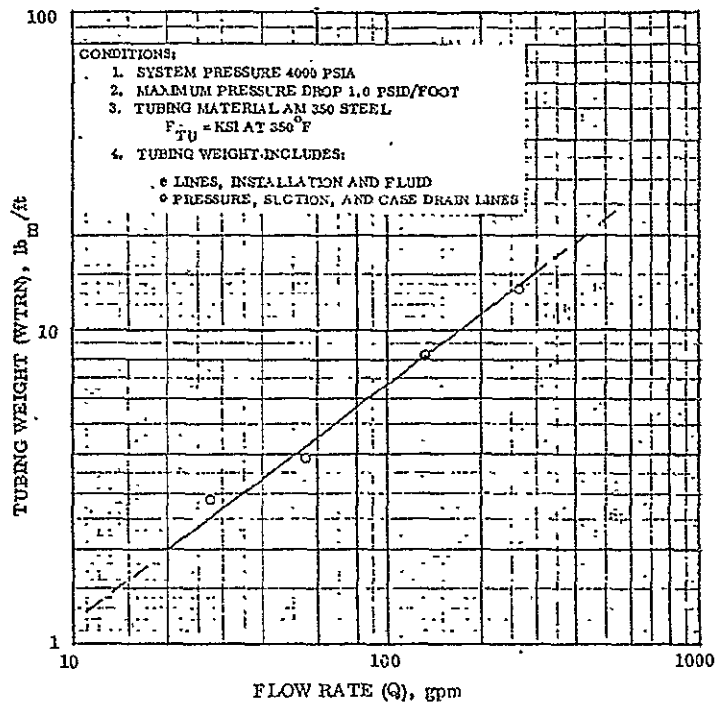


Figure 6-3. Hydraulic Power Generation Tubing Weight

6.2.2.2 Pump Weights. The hydraulic pump weight was established as a function of rated flow. See Figure 6-4. Pump weights were obtained using Vickers data for inline pumps and adding a correction factor for operation at 350°F maximum and 4000 psig.

6.2.2.3 Reservoir. The hydraulic reservoir weights are shown as a function of swept volume. See Figure 6-5. The reservoir is assumed to be a piston type. These data are based on previous designs used at Convair. Reservoir weight is more closely associated with total oil volume and differential displacement than with circuit flow. A reservoir size for any given circuit is approximately 15% of total circuit fluid volume plus differential volumes (accumulators only for flight control systems). The 15% includes allowance for thermal contraction, fluid compressibility, thermal expansion, and leakage.

6.2.2.4 Miscellaneous Components. The remaining components in the hydraulic power generation circuit include such items as filters, valves, ground connections, accumulator, and instrumentation. The weight of these miscellaneous components is shown in Figure 6-6. This curve is based on a fixed weight of 10 lb and a variable weight that is a function of flow rate.

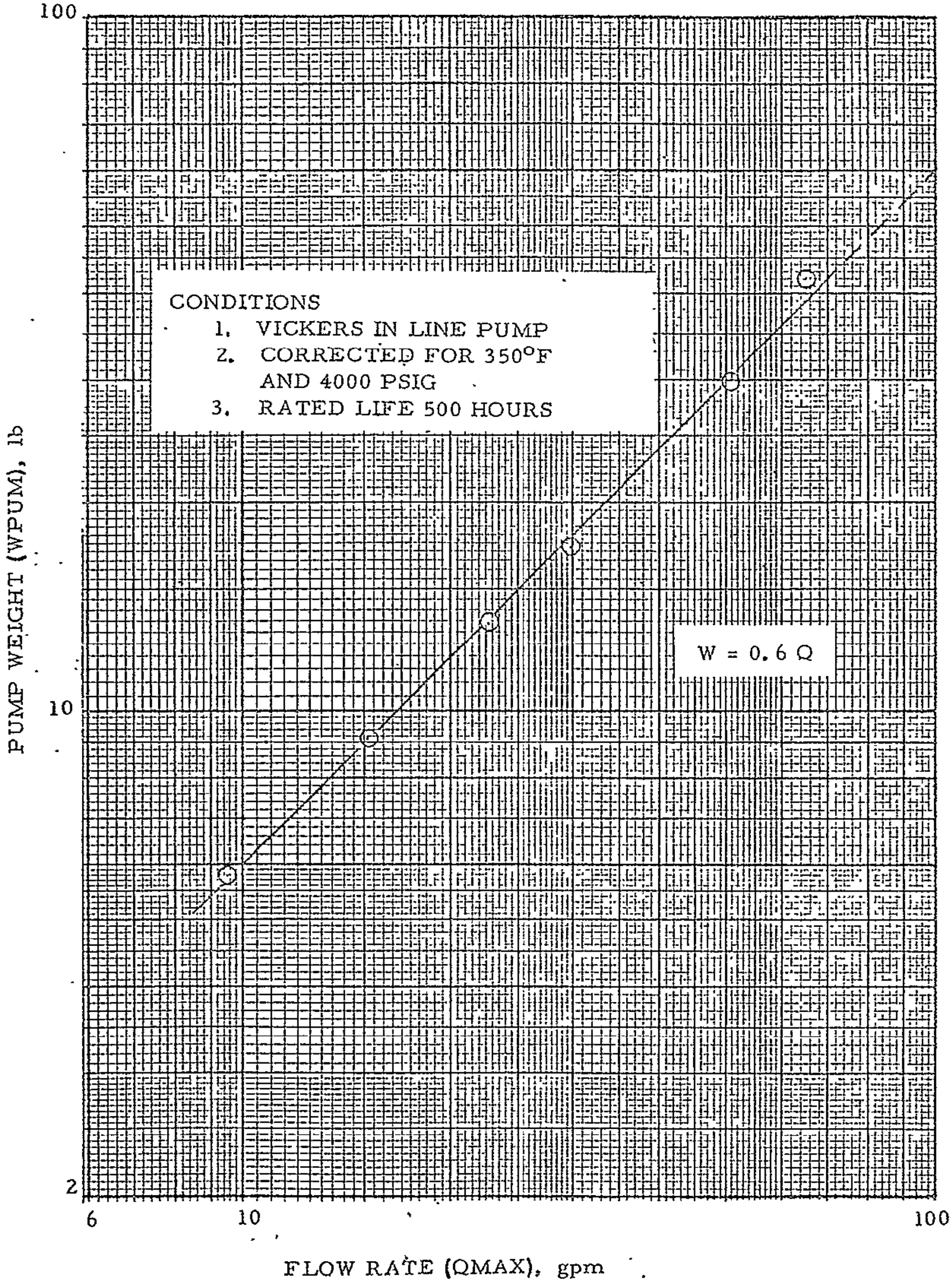


Figure 6-4. Hydraulic Pump Weight

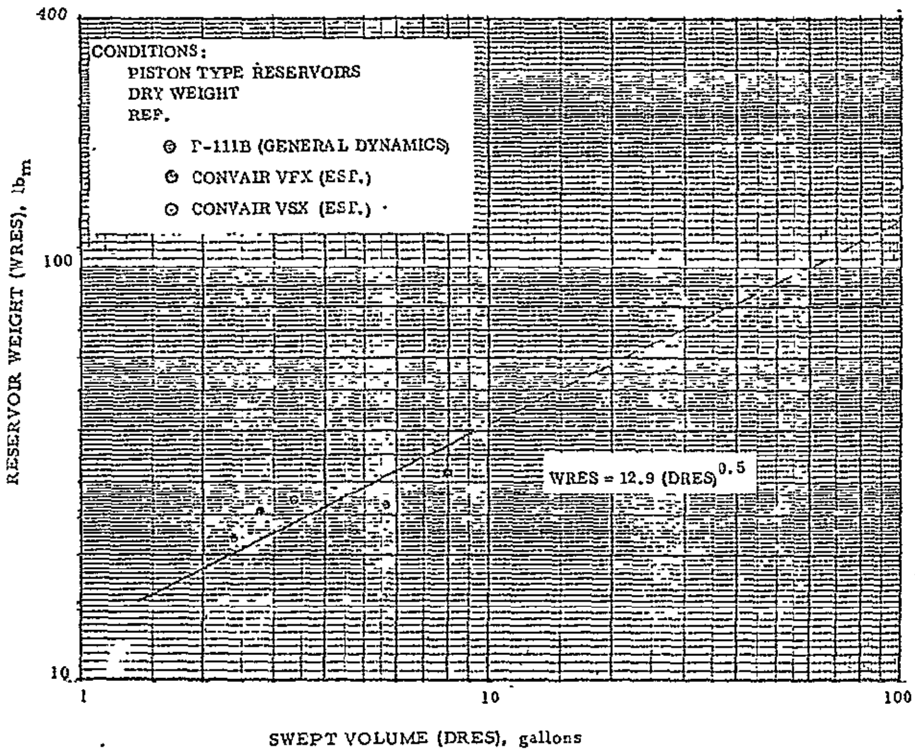


Figure 6-5. Reservoir Weight

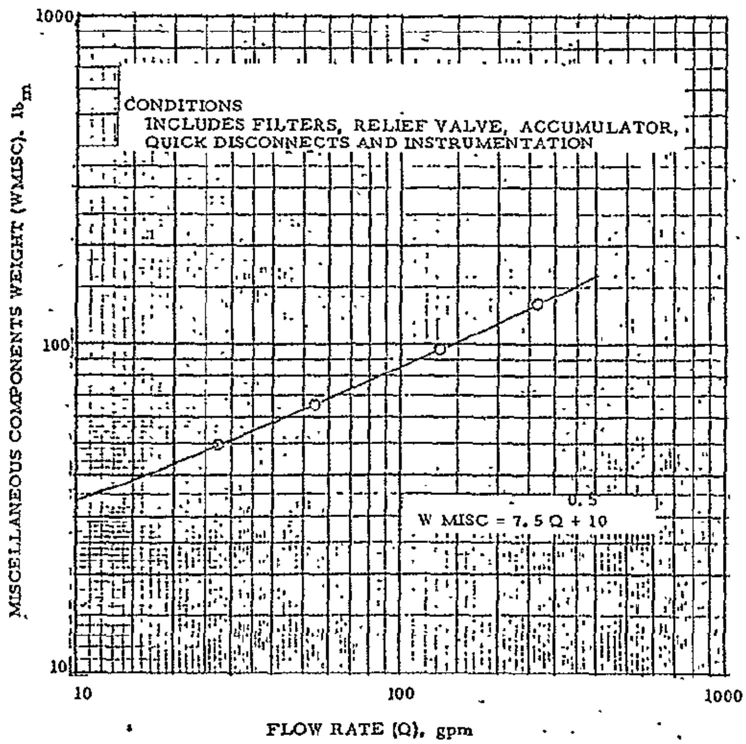


Figure 6-6. Miscellaneous Component Weight

6.2.2.5 Summary. The preceding data on power generation elements are shown to give background information that was used to develop the total hydraulic power generation weight shown on Figure 6-7. The curve is not truly parametric. The change in slope of the curves between the booster and orbiter circuits is primarily due to the following factors:

a. Longer transmission lines for the booster. (Transmission lengths assumed are shown in Section 7.) This influences reservoir and reservoir fluid weights.

b. Longer power generation lines for the booster. The lengths are assumed as follows:

   1. Orbiter: 25 ft each of pressure, return, and pump case drain.

   2. Booster: 40 ft each of pressure, return, and pump case drain.

   The booster line lengths are longer primarily due to larger power circuits and distance to ground connections.

Also shown on Figure 6-7 is weight for a rocket engine driven TVC hydraulic circuit. This falls on the orbiter curve because of the similarity of the assumed tubing lengths.

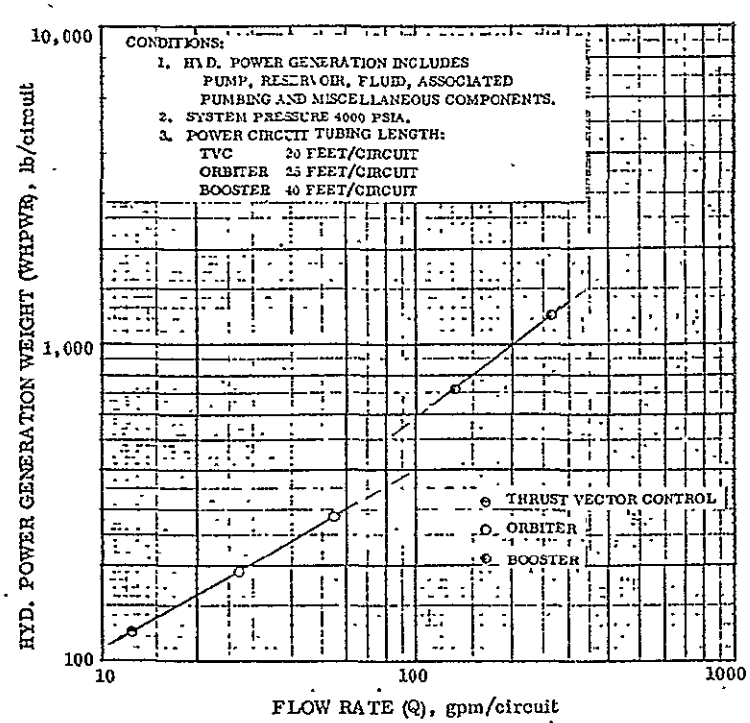


Figure 6-7. Hydraulic Power Generation Weight

6.2.3 HYDRAULIC CONDITIONING. The hydraulic conditioning weight includes the heat exchanger (hydrogen-to-oil) and associated control valves. The heat exchanger size and heat load was determined for four cases:

a. Orbiter with four hydraulic circuits.

b. Orbiter with three hydraulic circuits.

c. Booster with four hydraulic circuits.

d. Booster with three hydraulic circuits.

Figure 6-8 displays hydraulic circuit flow rate and conditioning weight versus thermal rate. The ratio of environmental heat rate to hydraulic generated heat rate (constant) varies from approximately 0.66 for a 27 gpm circuit to 0.28 for a 265 gpm circuit.

The thermal load was determined using the following assumptions:

a. Maximum ambient temperature of the horizontal stabilizer is 600°F.

b. Only elevator equipment (lines and components) contribute to environmental load. [The wing hydraulics also contribute but are much smaller and are not included so that vehicle weight impact becomes a function of elevator systems only.]

c. Total length of tubing exposed to high ambient temperature:

   Orbiter: 50 ft (horizontal stabilizer runs)

   Booster: 75 ft (horizontal stabilizer runs)

d. Emissitivity of lines and components in high temperature area = 0.6.

e. Nominal temperature of hydraulic system is 300°F.

f. Hydraulic equipment is insulated from hot structure (e.g., conductive heat load is neglected).

g. Total heat load consists of:

   Environmental heat load

   Pump losses (Case drain flow 5% of rated flow)

   Leakage losses (5% of rated flow)

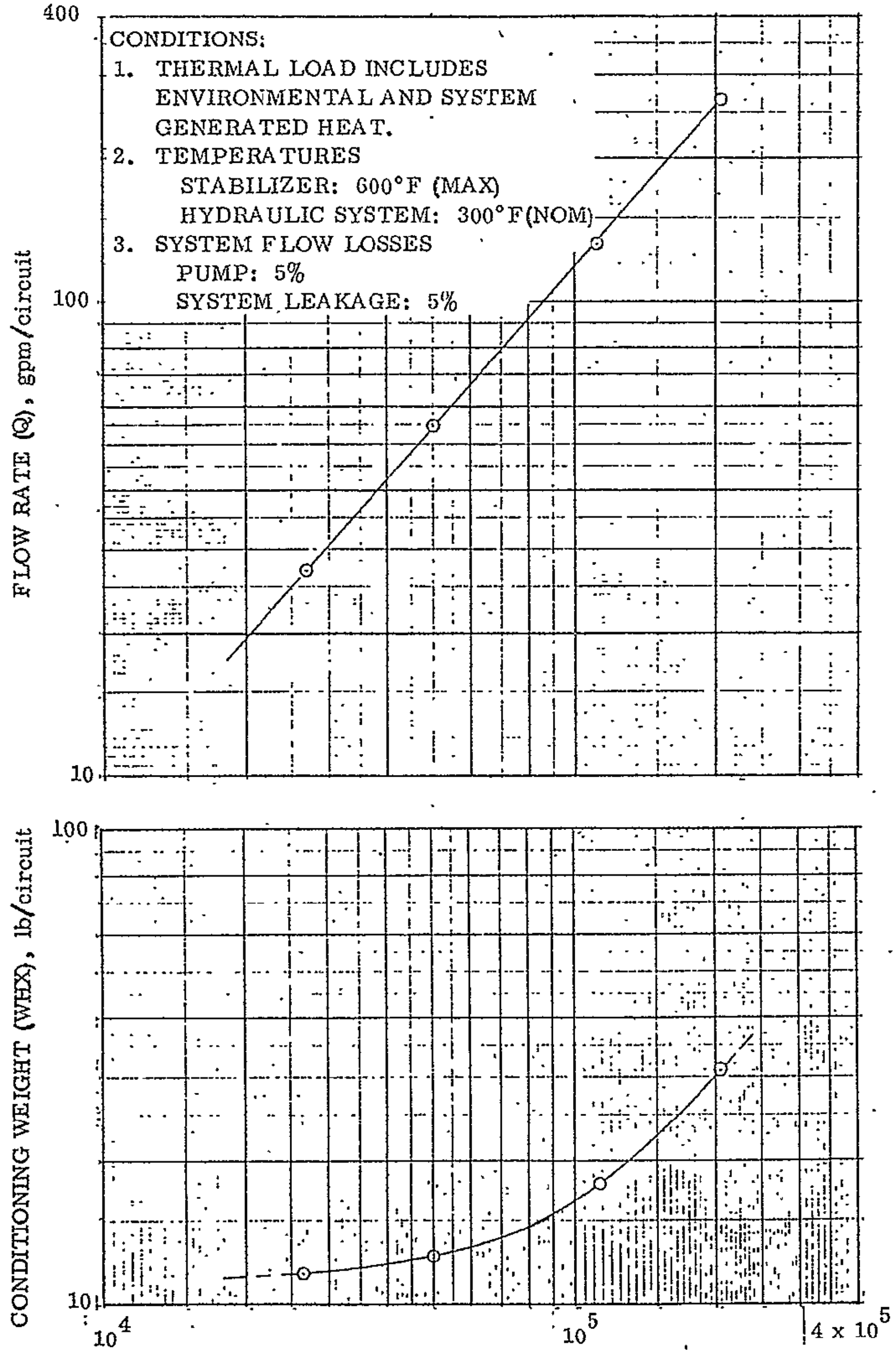


Figure 6-8. Power Conditioning Weight

The heat exchanger size and weight were based on:

a. Cross/counter flow heat exchanger.

b. Hydrogen gas is available as a heat sink (200°R and 100 psia).

c. Mean ΔT in heat exchanger: outlet oil to inlet hydrogen = 360°R
inlet oil to outlet hydrogen = 100°F.

d. The weight is a fixed weight of 10 lb plus an incremental weight which is a function of equivalent tube weight to provide sufficient surface area.

6.2.4 POWER SOURCE AND FUEL. Figure 6-9 shows APU weight versus shaft horsepower. The basic APU is shown as well as the total APU installation weight. The total weight is based on an installation weight of 50% of the APU weight plus a fixed weight of 15 lb for fire detection and fire extinguishing equipment. Figure 6-10 shows specific reactant (fuel) consumption (SRC) versus per cent rated capacity. The SRC varies slightly with APU size but is not significant. Per cent rated capacity is significant, however. The average hp as established in Section 3 (38% for orbiter and 30% for booster) converted to hp-hr and used with Figure 6-9 determines the fuel weight required for each elevator configuration. See Section 7.

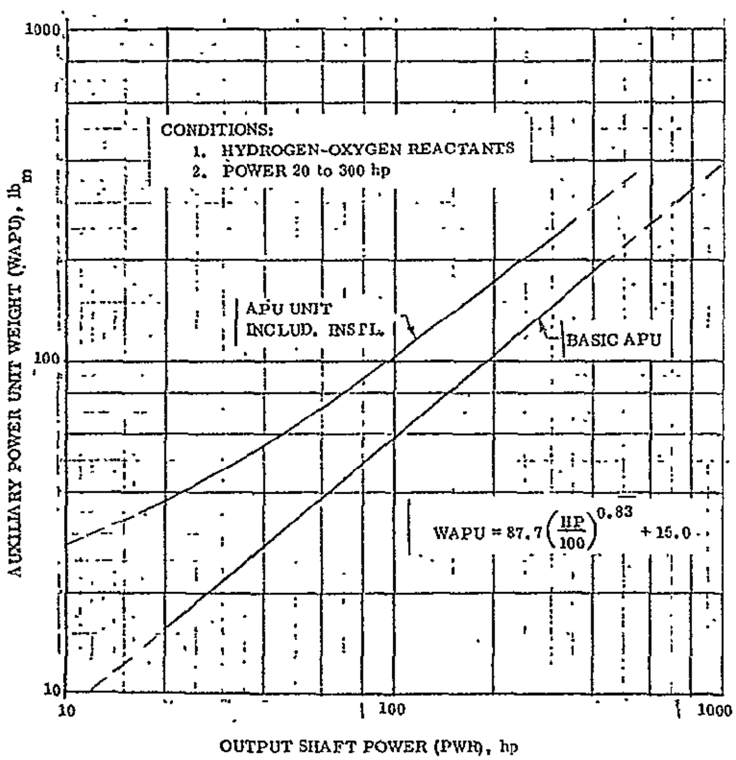


Figure 6-9. APU Installation Weight

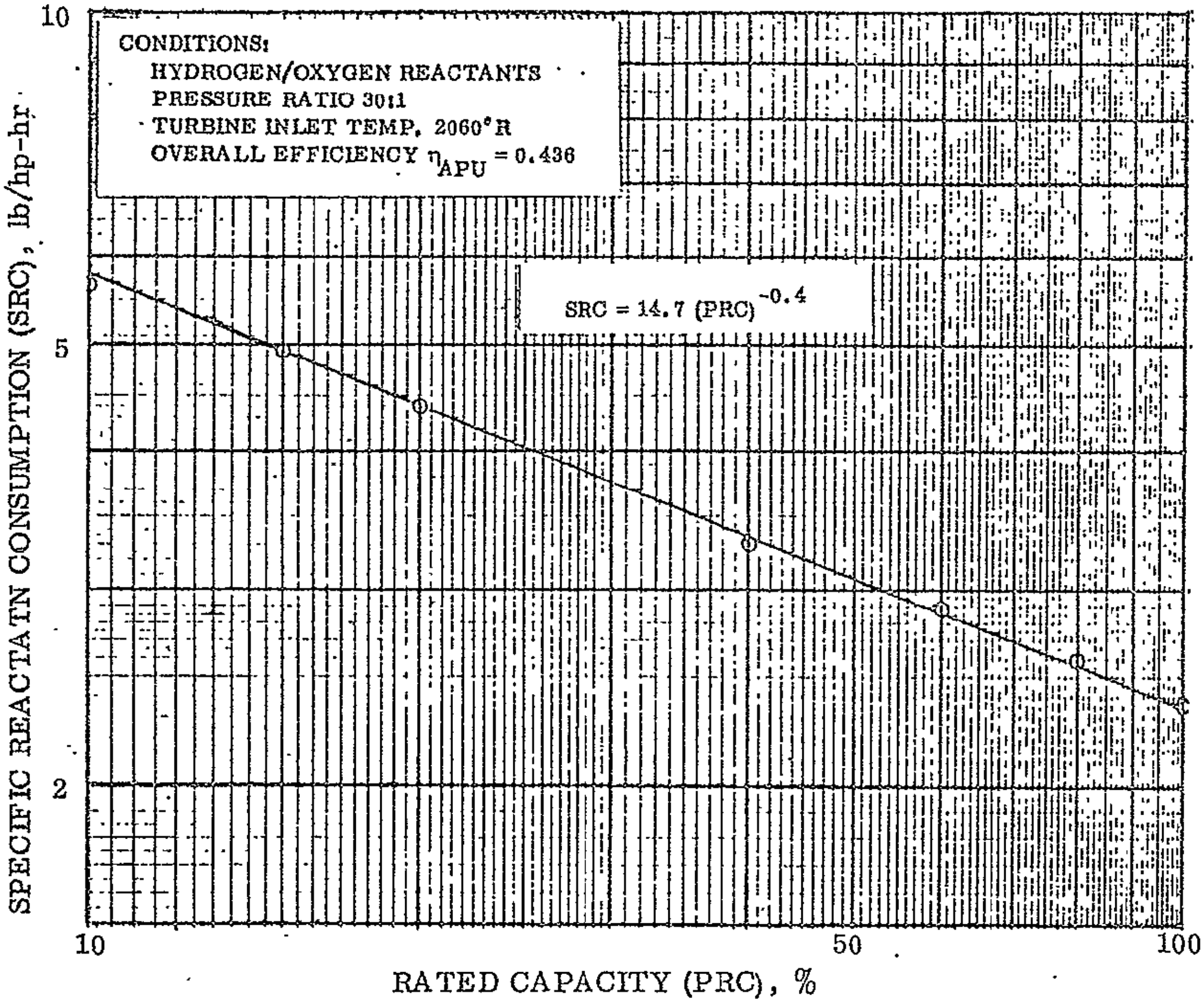


Figure 6-10. Part Load Specific Reactant Consumption

6.2.5 HYDRAULIC ACTUATORS. Figure 6-11 shows actuator weight versus work capability. This is essentially the base weight of the actuator including fluid but not valving or other integrated equipment. Work capability is used as the most convenient measurement for sizing actuators. The curve is independent of pressure and actuator geometry. It is assumed that geometric efficiency (ratio of moment arm length at maximum load point to bellcrank length) is approximately 90%.

6.2.6 HYDRAULIC VALVES AND VALVE MANIFOLDS. Two curves are shown in Figures 6-12 and 6-13. The first curve lists electrohydraulic servovalve weights versus a flow function as shown. The upper portion (high flow rates) of the curve is not applicable because in high flow applications additional valve stages are added to keep the electrohydraulic servo relatively small. The second curve, Figure 6-13, identifies power valve and manifold weights. A 6.5-lb fixed minimum weight is assumed.

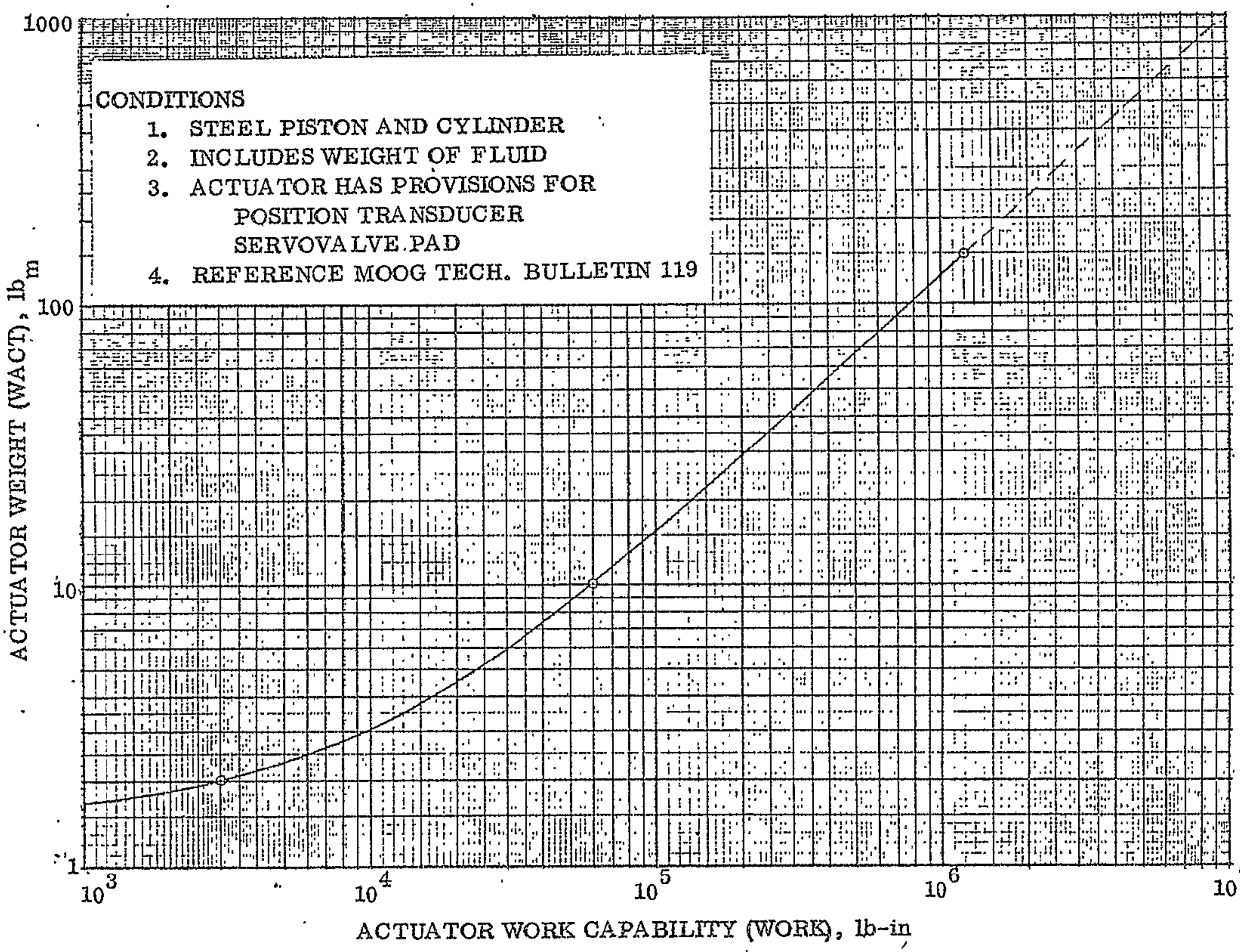


Figure 6-11. Hydraulic Actuator Weight

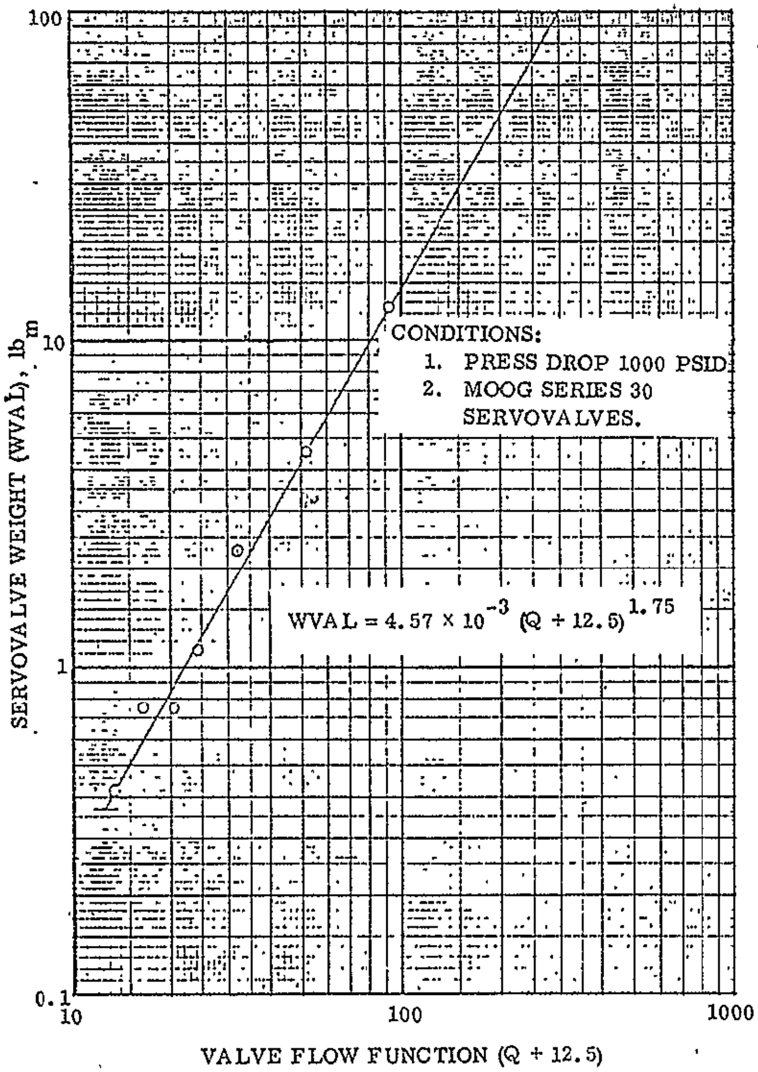


Figure 6-12. Electrohydraulic Valve Weight

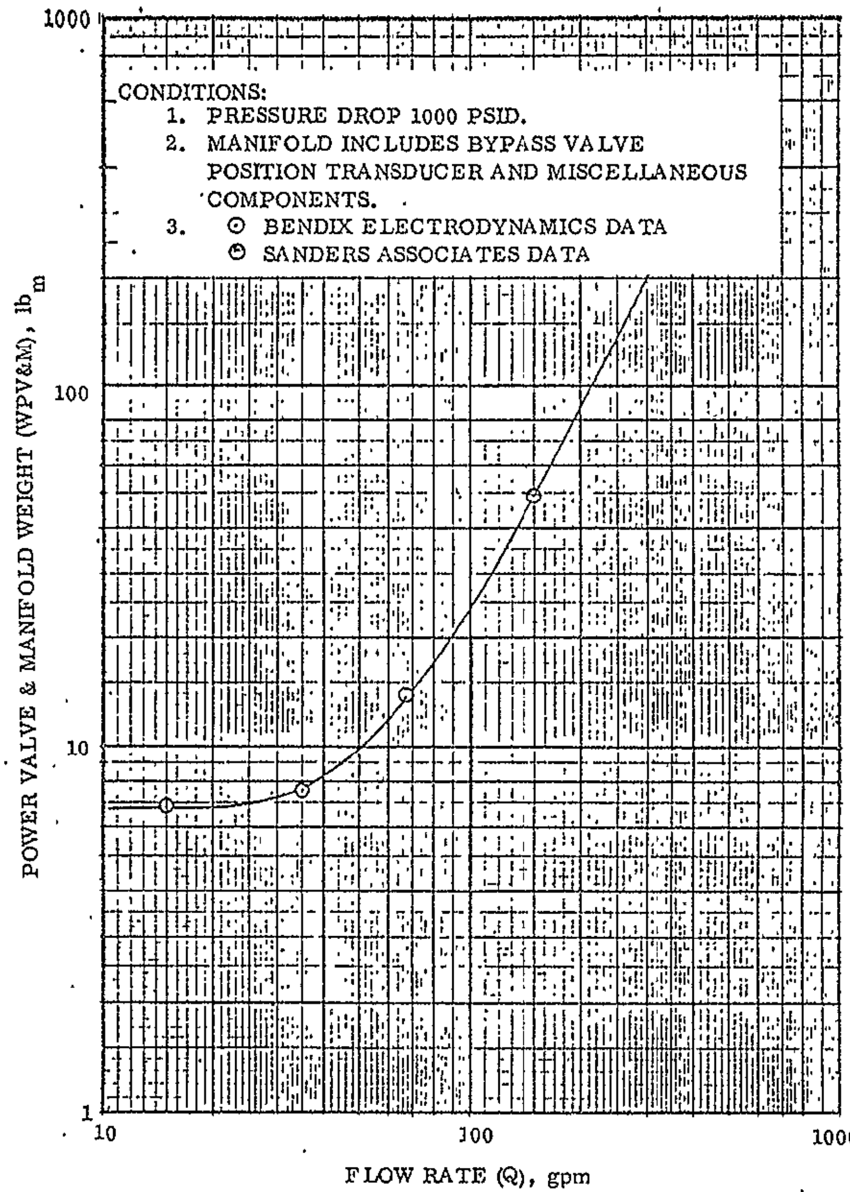


Figure 6-13. Power Valve and Manifold Weight

6.2.7 ELECTRICAL. Two basic elements are defined: motors and amplifiers. Amplifier weights are shown because of the gross differences in power levels required between electrohydraulic and electromechanical control. Additionally, some configurations employ more servoamplifiers than others. The delta weight differences are small when compared to overall weight but are displayed here so that a comparison can be made of servoactuator control portions. Figure 6-14 shows ac motor weights versus hp. The curve is based on Westinghouse data. Figure 6-15 shows amplifier weight versus power output and is derived from information within Convair.

6.2.8 MECHANICAL. Spring clutch weight versus output torque is shown in Figure 6-16. The curve is based on Curtiss-Wright data for bi-directional clutch assemblies without brake provisions. The curve is modified to incorporate braking.

Differential weights, shown in Figure 6-17, are derived from data that established weights of gear trains in a previous study conducted at Convair.[21]

Figure 6-18 is used to size the output ballscrew actuator used in the orbiter aileron configuration 2 and the secondary actuators used in the electromechanical control portions of orbiter aileron configuration 3, and orbiter and booster elevator configuration 3. Determining output ballscrew actuator weights is straightforward, with output force, stroke, and rate determined by surface hinge moment requirements. Sizing secondary actuators is not straightforward in that the ballscrew is sized by hydraulic power spool flow forces, friction, control loop maximum actuator rate and gain, power spool stroke, and secondary actuator/power spool geometry. The most predominant factor is power spool flow forces, especially in the very large servoactuators where each of three power spools may be controlling up to 75 gpm. An equation used for correlation between the secondary actuator and hydraulic power spool size is:

$$F_{PV} = 9.1Q + 400$$

where

F_{PV} = power valve force

Q = flow rate, gpm.

The term 9.1Q is a function of flow forces and 400 is established as a minimum force capability.

Ballscrew actuator weights in Figure 6-18 are shown in relation to force for different strokes. Unlike a hydraulic linear actuator, the ballscrew weights cannot be shown in terms of work capability because within reasonable values of strokes and rotational speed, the ballscrew weight is more sensitive to force.

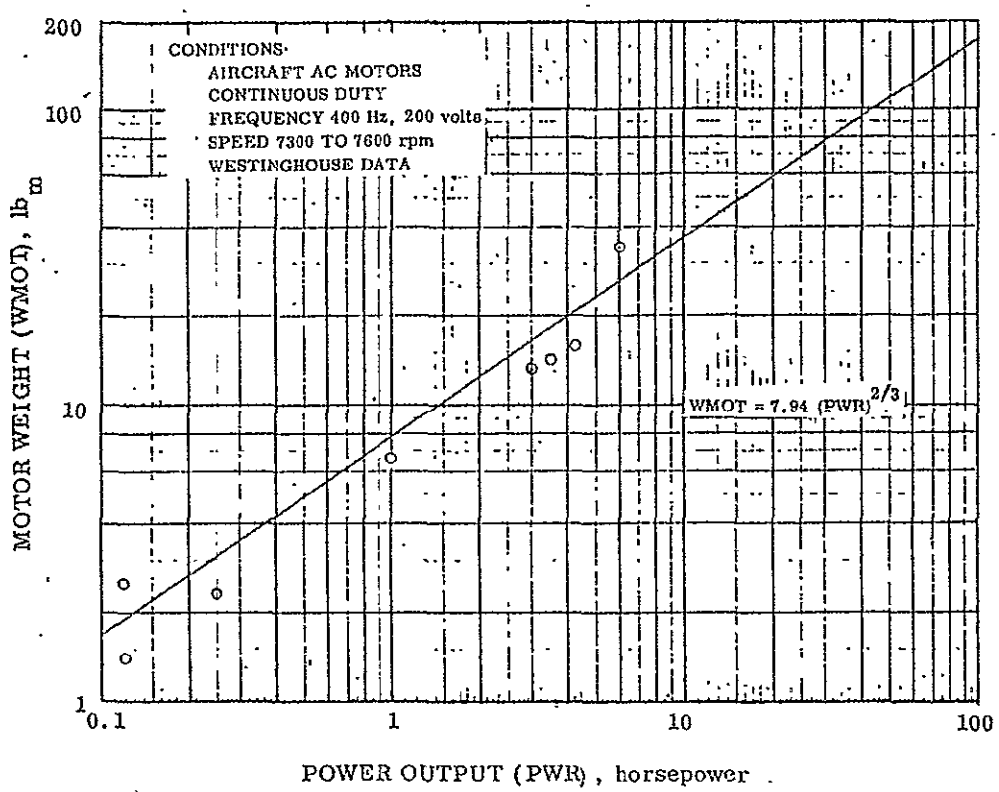


Figure 6-14. Electric Motor Weight

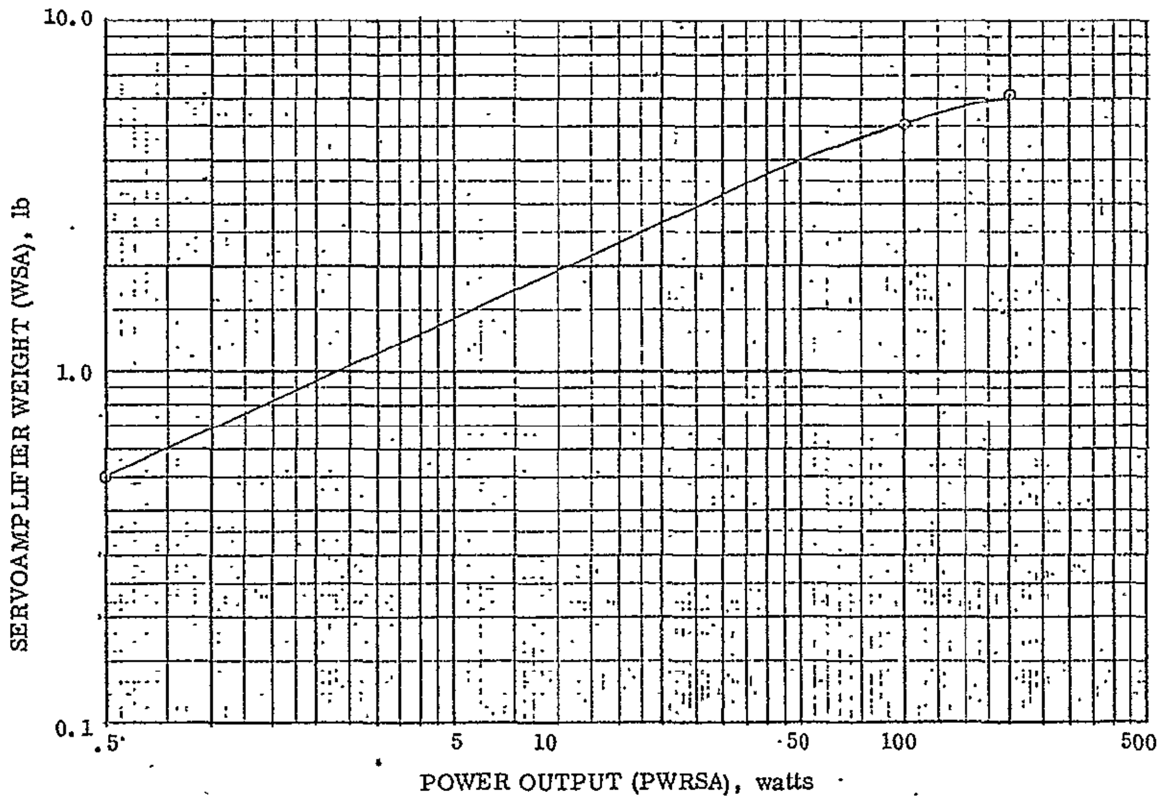


Figure 6-15. Servoamplifier Weight

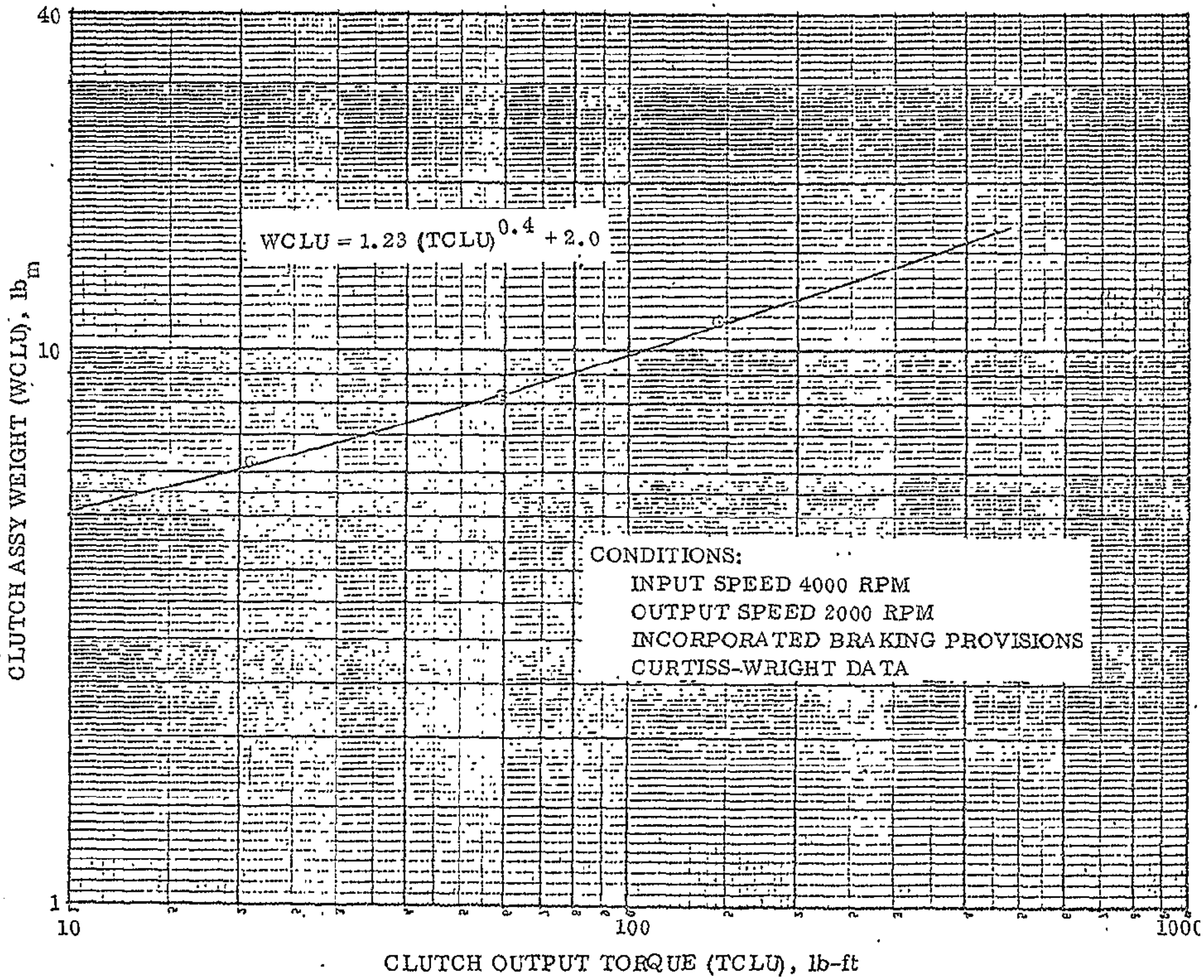


Figure 6-16. Spring Clutch Weight

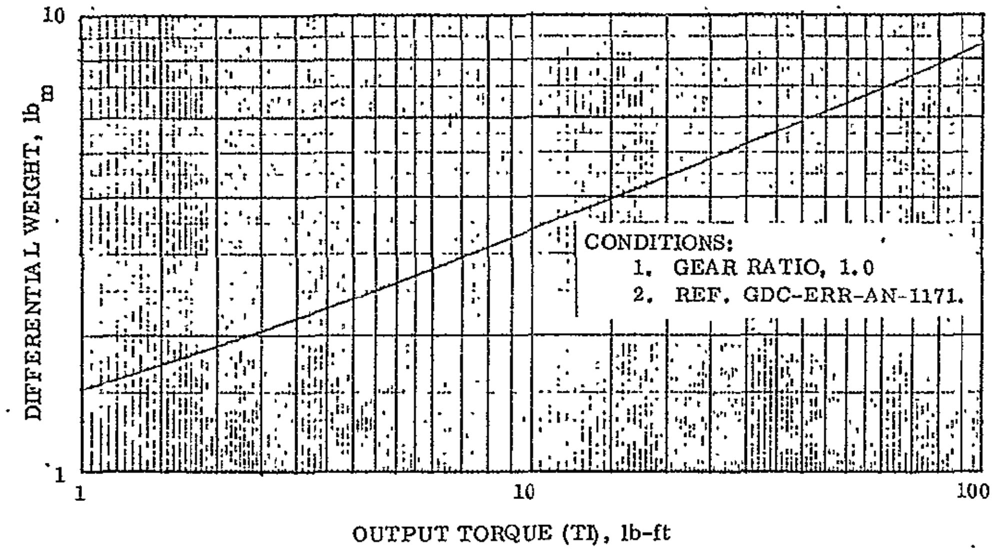


Figure 6-17. Differential Weight

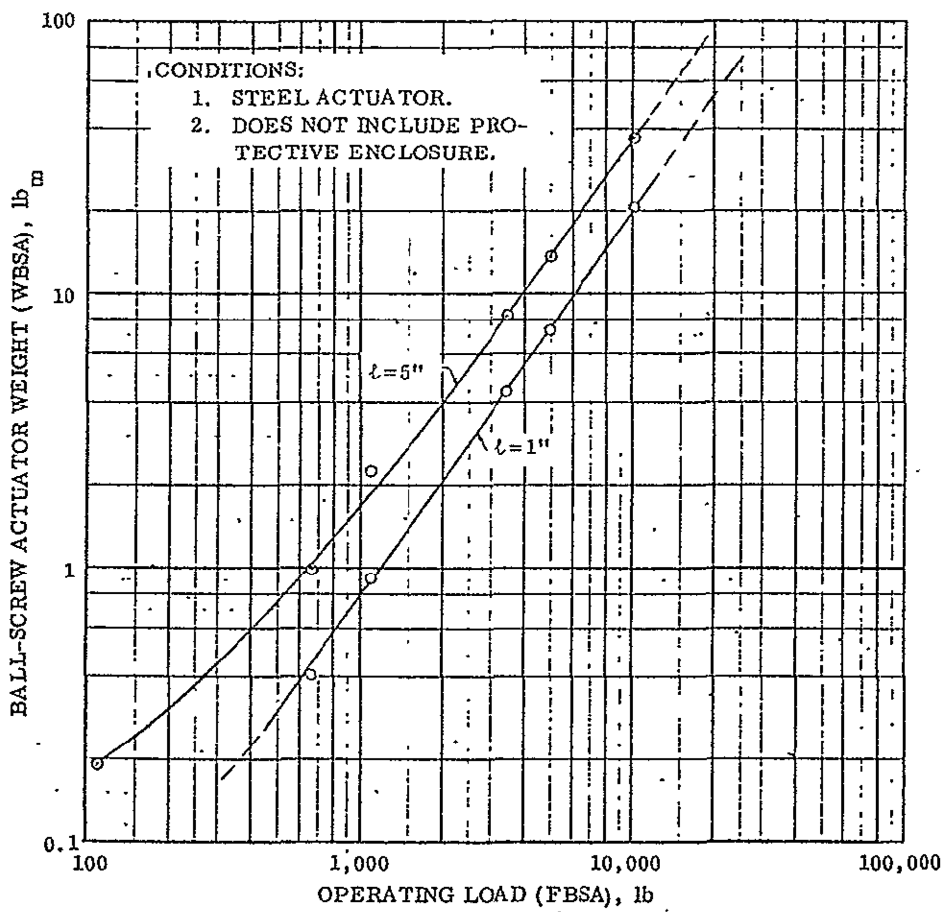


Figure 6-18. Ball Screw Actuator Weight

6.3 RELIABILITY

A computer program in existence at Convair was used to generate the reliability comparisons shown in Section 7. The analysis consisted of the following basic steps:

a. Determine failure rate data from existing sources.

b. Convert failure rates to failures/flight.

c. Computer programming.

d. Normalize results.

6.3.1 <u>FAILURE RATE DATA</u>. Two basic sources of failure rate data were used. They are RADC-TR-68-114[22] and FARADA.[23] Low, high, and average failure rates were taken from these sources. The failure rates covered a wide range, but for this study the failure rates picked were, in general, conservative or representative of the high failure rates listed.

6.3.2 <u>FAILURES/FLIGHT</u>. Table 6-1 lists the failure rate data used and the conversion to failures/flight. The column headings are identified as:

λ = failure rate per 10^6 hours (from reliability data sources)

λ_{tB1} = expected failures $\times 10^{-6}$/flight - boost phase - booster (.05 hr)

λ_{tB2} = expected failures $\times 10^{-6}$/flight - total flight time - booster (2.5 hr)

λ_{tO1} = expected failures $\times 10^{-6}$/flight - boost phase - orbiter (2.1 hr)

λ_{tO2} = expected failures $\times 10^{-6}$/flight - flight time - orbiter (2.9 hr)

The above mission times are not necessarily actual operating times for the various servoactuators but include total time that the units may be exposed to extreme environments and/or be powered up.

The failure rate data shown for hydraulic actuators, item 6, must obviously include all discrepancies that require maintenance action, but not necessarily serious failures. Two computer runs were made, one using the failure rate data as shown to establish maintainability comparisons. The failure rate of $765/10^6$ hours was reduced to 76.5 on the second run to establish reliability comparisons. Item 3 failure rates are shown divided into two groups. The first number is "fail to open" failure mode and the second number represents "fail to close" failure mode.

6.3.3 <u>COMPUTER PROGRAMMING</u>. The servoactuator is broken down in functional sets where all components in a set are in series. For example, consider a servoactuator that employs three servo channels, three hydraulic circuits, a tandem power

Table 6-1. Failure Data

Item	Nomenclature	λ	λ_{tB1}	λ_{tB2}	λ_{tO1}	λ_{tO2}
1	Hydraulic Pump	85	4.2	212	179	246
2	E/H 2 Stage Servovalve	125	6.2	311	262	362
3	Solenoid Valve	287/143	14.4	717/358	603/301	832/416
4	Spool Valve	62	3.1	155	130	180
5	Spool Valve, Spring Loaded	62		155	130	180
6	Hydraulic Actuator	765	38.2	1912	1607	2220
7	LVDT	38	1.9	94	79	109
8	Servo Amplifier	160	8	400	336	464
9	Power Amplifier	320		800		929
10	Servo Motor	107		268		310
11	Differential	41		103		119
12	Ball Screw	23		58		67
13	Electronic Voter	263		658	552	763
14	Solenoid	72				209
15	Spring Detent	20		50		58
16	Spring Clutch	28				81
17	Gear Reduction	46				133
18	Tachometer	126		315		365
19	Motor, A.C.	470				1363
20	Pressure Switch	368	18	920	811	1067
21	ΔP Transducer	368		920	811	1067
22	Hydraulic Monitor	30	1.5	75		87
23	Hydraulic Comparator	62	3.1		130	180
24	Time Delay Relay	103		258		300
25	Valve Controller	4		10		12
26	Integrator	96		239		278

actuator, cross monitoring, detection, and correction. It is fail operate whereby it can lose one channel and/or one hydraulic circuit. The functional sets might be:

SET	
1	Servo Channel No. 1
2	Servo Channel No. 2
3	Servo Channel No. 3
4	Hydraulic Supply No. 1
5	Hydraulic Supply No. 2
6	Hydraulic Supply No. 3
7	Tandem Actuator (and Power Spool)
8	By-Pass Function No. 1
9	By-Pass Function No. 2
10	By-Pass Function No. 3
11	Monitoring and Detection

A servo channel includes a servo amplifier, two-stage servo valve, LVDT feedback, and a secondary actuator. A reliability number for the servo channel is assigned using the failure rate data of all the components in the set and considering that all the components are in series. This procedure is repeated for each functional set. For example, reliability of the bypass function is the reliability of a solenoid shutoff valve (to shut off) times the reliability of a pressure/spring-operated bypass valve (to by pass).

A "minimum truth table" is then developed (based on the failure modes and effects analysis) that describes all of the minimum ways in which the servoactuator can function and still be successful. The minimum truth table is prepared in binary form where a 1 means the set must function and a 0 means the set is not required to function.

Shown below is an example of the table where the sets correspond to those shown above and the rows represent the combinations (minimum number of ones) that can exist for success.

SET

1	2	3	4	5	6	7	8	9	10	11
1	1	0	1	1	0	1	0	0	1	1
1	0	1	1	0	1	1	0	1	0	1
0	1	1	0	1	1	1	1	0	0	1

The reliabilities for the sets and the minimum truth tables were then submitted to the DCS and 7094/7044 computer to calculate the system reliability. The computer determines <u>all</u> the combinations that can be successful and computes the probability of any successful combination that might exist. For example, the computer calculates the probability of all sets functioning correctly, then proceeds downward to a minimum limit established by the minimum truth table. The summation of the probabilities of all the successful combinations that can exist is the reliability of the unit.

6.3.4 NORMALIZE RESULTS. Since absolute reliability numbers are only as good as the original failure rate data used, an additional step was taken. The reliability numbers are normalized based on the least reliable configuration (in a group being compared) reduced to 1. Therefore all comparisons are in whole numbers and the larger the number the better the ranking.

6.4 QUALITATIVE COMPARISONS

Maintainability, checkout capability, performance, and cost are treated qualitatively. Major features or characteristics are shown in tables in Section 7. The following paragraphs list criteria that was used for comparison.

6.4.1 MAINTAINABILITY. Criteria for comparison.

a. Mean flights between maintenance action (MFBMA).
   The probability (P_S) that all functions work successfully within a servoactuator was determined from the first computer run using the high failure rate data for actuators. $1 - P_S$ is then the probability of failure (maintenance action). The reciprocal $1/1 - P_S$ = MFBMA. The MFBMA data is normalized in the same manner as described in Section 6.3.4.

b. Number of components.
   For aerodynamic surface control actuators only servoactuator components are totaled. Electronic logic where used was assumed to be equivalent to many components (up to 20) and the actual number was estimated based on apparent complexity. TVC servoactuator component count includes hydraulic power circuit components where rocket engine driven circuits are used. Twelve components per circuit were assumed.

c. Installation complexity.

6.4.2 CHECKOUT CAPABILITY. Comments in Section 7 are based on capability of performing three checkouts listed below in order of importance.

a. Normal operation.

b. Health status of each servo channel.

c. Health status of fault detection and switching.

6.4.3 COST. Relative comments in Section 7 are based on the three factors listed below:

a. Unit procurement cost.

b. Operational cost.

c. Development cost.

SECTION 7

TRADE-OFF EVALUATION

7.1 GENERAL

This section contains the tabulation of all results and comments and is arranged as follows:

7.2 BOOSTER AND ORBITER ELEVATOR

7.2.1 BOOSTER WEIGHT

7.2.2 ORBITER WEIGHT

7.2.3 ALTERNATE ORBITER ELEVATOR ARRANGEMENT

7.3 ORBITER AILERON

7.4 DIGITAL SERVOACTUATOR COMPARISON

7.5 ORBITER TVC

7.6 BOOSTER TVC

7.2 BOOSTER AND ORBITER ELEVATOR

7.2.1 BOOSTER WEIGHT

Basic data used to generate weight:

	3 Hydraulic Power Circuits	4 Hydraulic Power Circuits
Hinge Moment Total Req'd, Ft-lb	650,000	650,000
Hinge Moment/Power Circuit, ft-lb	650,000	325,000
Hinge Moment/Actuator/Side, ft-lb	325,000	162,500
Flow to Elevator System/Circuit, gpm	265	133
Flow to Each Actuator/Side, gpm	133	67
Length - APU to Aft Vehicle ℄, ft	25	25
Length - Actuators to Vehicle ℄, ft	75	75
Max APU HP/Hydraulic Circuit, hp	430	215
Ave Hp/Circuit - % of Max, &	30	30
Flight Hours - Operating Time, hr	1.72	1.72
Hp-Hr	222	111

Table 7-1. Weight - Elevator, Booster

Nomenclature	Configuration 1			Configuration 2			Configuration 3		
	No.	Unit Wt. Lb	Total Wt. Lb	No.	Unit Wt. Lb	Total Wt. Lb	No.	Unit Wt. Lb	Total Wt. Lb
Servoactuator:	8								
Actuator		315		4	630	2520	6	585	3510
Pwr Valve & Manifolds		13			52			120	
Control Portion		4.4			18			52	
		332.4	2659	2	70	140	2	172	344
Synch. Shaft	2	10	20						
Hyd. Power Generation	4	716	2864	4	716	2864	3	1222	3666
Hyd. Transmission	4	285	1140	4	285	1140	3	455	1365
Hyd. Conditioning	4	18	72	4	18	72	3	30.5	92
APU	4	182	728	4	182	728	3	325	975
APU Fuel	4	416	1664	4	416	1664	3	832	2496
Total Vehicle Weight			9147			9128			12448
ΔWeight			0			-			+3300

7.2.2 ORBITER WEIGHT

Basic data used to generate weight:

	3 Hydraulic Power Circuits	4 Hydraulic Power Circuits
Hinge Moment Total Req'd, ft-lb	133,000	133,000
Hinge Moment/Power Circuit, ft-lb	133,000	66,500
Hinge Moment/Actuator/Side, ft-lb	66,500	33,250
Flow to Elevator System/Circuit, gpm	54	27
Flow to Each Actuator/Side, gpm	27	13.5
Length - APU to Aft Vehicle ℄, ft	15	15
Length - Actuators to Vehicle ℄, ft	50	50
Max APU Hp/Hydraulic Circuit, hp	88	44
Ave Hp/Circuit - % of Max, %	38	38
Flight Hours - Operating Time, hr	0.83	0.83
Hp-Hr	28	14

Table 7-2. Weight - Elevator, Orbiter

Nomenclature	Configuration 1			Configuration 2			Configuration 3		
	No.	Unit Wt. Lb	Total Wt. Lb	No.	Unit Wt. Lb	Total Wt. Lb	No.	Unit Wt. Lb	Total Wt. Lb
Servoactuator	8								
Actuator		76		4	152	608	6	142	852
Pwr Valve & Manifolds		6.8			27			21	
Control Portion		3.2			14			48	
		86	688	2	41	82	2	69	138
Synch. Shaft	2	6	12						
Hyd. Power Generation	4	191	764	4	191	764	3	285	855
Hyd. Transmission	4	61	244	4	61	244	3	99	297
Hyd. Conditioning	4	12	48	4	12	48	3	13	39
APU	4	62	248	4	62	248	3	95	285
APU Fuel	4	48	192	4	48	192	3	96	288
Total Vehicle Weight			2196			2186			2764
ΔWeight			0			-			+ 540

Table 7-3. Reliability Comparison — Elevator, Orbiter and Booster

Criteria	Configuration 1	Configuration 2	Configuration 3
Probability of Mission Success	.99995	.99927	.99918
Relative Ranking	16.4	1.12	1

Table 7-4. Maintainability Comparison — Elevator, Orbiter and Booster

Criteria	Configuration 1	Configuration 2	Configuration 3
MFBMA (Mean Flights Between Maintenance Action)			
a. Probability of All Elements Working	.9639	.9748	.9929
b. MFBMA	28	39.7	141
c. Relative Ranking	1	1.4	5
Components/Vehicle	134	120	132
Installation	8 Separate Valve Packages, Integral with Actuators. Difficult to Troubleshoot Installation.	2 Unitized Valve Packages, Remote From Actuators.	• 2 Unitized Valve Packages, Remote From Actuator. • Large and Heavy Units

Table 7-5. Performance Comparison - Elevator, Orbiter and Booster

Criteria	Configuration 1	Configuration 2	Configuration 3
Normal Performance	• Lower Static Stiffness and Positional Accuracy Due to Power Spool Synch. Method • More Adaptable To Spreading Actuators Along Rear Spar	• Stiffness Better Than 1 If Dual Tandem Spools Can Be Synchronized By Fabrication.	• Stiffness Better Than 1 If Triple Tandem Spools Can Be Synchronized By Fabrication. • May Be Subject To Small Amplitude Limit Cycling If Tach Feedback Gain Can't Hold Channels At Null.
Redundancy Performance	• Nuisance Tripping May Be Problem. • Negligible Degradation After Channel Failure.	• Less Nuisance Tripping Than 1 Due to Cross Equalization. • Negligible Degradation After Channel Failure.	• Same as 1.
Versatility And/Or Commonality	Fault Correction Can Be Manual Function of Crew Due to Force Summing Mechanization.	Same as 1.	• Requires Automatic Fault Correction. • Can Be Used With 2, 3 Or 4 Hydraulic Power Systems With No Change.

Table 7-6. Checkout Capability — Elevator, Booster and Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
Normal Operation	Add Surface Position XDCR Or Surface End Position Limit Switches To Provide Output Signal To The Checkout/Monitoring Function.	Use Actuator Position XDCR Signal.	Same as 2.
Single Channel Operation	• Add Switching To Disable Self Equalization And Fault Detection. • Add Switching to De-energize Solenoid Shutoff Valves. Operate One On At A Time.	• Add Switching To Disable Cross Equlization & Fault Detection Logic. • Add Switching to De-energize Solenoid Shutoff Valves. Operate One On At A Time.	• Add Switching To Disable Fault Detection Logic. • Add Switching to Shut Off Elect Power To Each Channel.
Fault Detection	• Add Program to Introduce Hardover Failures • Add Reset Provisions.	• De-energize Shutoff Valves in Sequence and Observe Failure Indication • Add Reset Provisions	• Switch Off Elect Power To Servos In Sequence And Observe Failure Indication. • Add Reset Provisions.

Table 7-7. Cost Comparison — Elevator, Booster and Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
Unit & Instl. Costs	High — Due To 4 Power Circuits And 4 Power Actuators.	High — Same As 1.	Lower Than Configuration 1 Or 2 Due To 3 Power Circuits And 3 Power Actuators.
Operational Cost	High - Complexity And Low MFBMA Means High Maintenance.	High — Same As 1	Less Than Baseline. Fewer Power Circuits, Higher MFBMA.
Development	Low — Similar Unit Of Same Size Range Under Development.	Medium — Similar Unit Of Small Size Under Development. Problem With Power Spools' Synchronization.	Medium — Control Portion Under Development. Problem With Power Spools' Synchronization.

7.2.3 ALTERNATE ORBITER ELEVATOR ARRANGEMENT. The previous three configurations for the orbiter elevator required one servoactuator package in each left and right horizontal stabilizer. There are many alternate physical installation options to consider, but one that deserves attention is a center fuselage installation using four power actuators. The servoactuator has many advantages:

a. Less severe environment (vibration and temperature).

b. Only four valve/actuators instead of eight.

c. More installation space.

d. Less weight in the actuators and elimination of horizontal stabilizer, transmission lines.

e. Better maintenance.

A key structural consideration in determining the required actuator configuration is flutter. For the center-fuselage-mounted arrangement the moment will be reacted in the body, whereas for the actuators mounted in the horizontal stabilizer, the moments will be reacted there. In the case of the fuselage-mounted actuators, the elevator torsional stiffness must be sufficient to prevent excess aeroelastic losses due to elevator twist. In the case of actuators mounted in the stabilizer, the elevator torsional stiffness is not as critical since it is effectively clamped in torsion at the actuator points. The horizontal stabilizer twist due to the elevator hinge moment in this case should be small because the horizontal stabilizer structure is sized for max αq.

By actuating the surface at the root only, a large structural weight increase may be required in the elevator to provide enough stiffness to prevent flutter. It is not the intent of this discussion to determine what that weight increase is, but to point out that with a center-fuselage-mounted servoactuator grouping, a weight savings of approximately 250 lb can be realized within the servoactuator and power supply subsystems.

7.3 ORBITER AILERON

Basic data used to generate weight:

	3 Power Systems: Active/Standby Hydraulic or Elect/Mech	3 Power Systems: Load Sharing
Hinge Moment Req'd, ft-lb	9,600	9,600
Hinge Moment Req'd/Power Circuit, ft-lb	9,600	6,400
Hinge Moment/Actuator/Side, ft-lb	4,800	3,200
Flow to Aileron System/Hyd Circuit, gpm	5.2	3.5
Flow to Each Actuator/Side, gpm	2.6	1.75
Length - APU to Mid Vehicle ℄, ft	50	50
Length - Actuators to Vehicle ℄, ft	100	100
Aileron Max. No Load Rate, deg/sec	40	40

Table 7-8. Weight — Aileron, Orbiter

Nomenclature	Configuration 1 No.	Configuration 1 Unit Wt. Lb	Configuration 1 Total Wt. Lb	Configuration 2 No.	Configuration 2 Unit Wt. Lb	Configuration 2 Total Wt. Lb	Configuration 3 No.	Configuration 3 Unit Wt. Lb	Configuration 3 Total Wt. Lb
Servoactuator:									
Power Actuator	6	10	60	2	50	100	6	7.2	43
Control Portion		22.5		2	7.0	14		45.0	
Pwr Valve &/or Manifolds		13.5						19.5	
	2	36	72				2	64.5	129
Differential				2	11	22			
Solenoids & Spring Clutch			—	6	7.7	46			—
			132			182			172
Hyd. Transmission	3	53	159				3	42.5	128
Elect. Motor and Gearing			—	6	36.2	217			—
Total Weight			291			399			300
ΔWeight			0			+108			+9

Table 7-9. Reliability Comparison — Aileron, Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
Probability Of Mission Success	.999998	.999027	.99918
Relative Ranking	.486	1	1.19

Table 7-10. Maintainability Comparison — Aileron, Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
MFBMA (Mean Flights Between Maintenance Action)			
a. Probability Of All Elements Working	.9736	.9866	.9929
b. MFBMA	37.9	74.6	141
c. Relative Ranking	1	2	3.7
Components/Vehicle	98	128	132
Installation	Smallest Envelope. Most Feasible To Install And Remove.	Integrated Power Supply Makes Unit Large And Heavy.	Similar to 1 But Larger Envelope And Heavier.

Table 7-11. Performance Comparison — Aileron, Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
Normal Performance	Good. No Fight Between Outputs, Only 1 Active At A Time.	Poor. Subject To Limit Cycling, High Threshold And Dead Zone.	May Be Subject To Small Amplitude Limit Cycling If Tach Feedback Gain Can't Hold Channels At Null.
Redundancy Performance	• May Be Subject To Nuisance Tripping. • No Degradation After Failure.	• Output Rate Is Degraded After Failure. • Dependent On No Jamming At Output.	Same As 1.
Versatility And/Or Commonality	• Fault Correction Hyd. Logic Must Remain Within Servoactuator Interface. • Requires Automatic Fault Correction.	• Requires Automatic Fault Correction.	• Requires Automatic Fault Correction. • Can Be Used With 2, 3 Or 4 Hydraulic Power Systems With No Change.

Table 7-12. Checkout Capability - Aileron, Orbiter

Criteria	Configuration 1 (Baseline)	Configuration 2	Configuration 3
Normal Operation	Use Actuator Position XDCR To Provide Output Signal To Checkout/ Monitoring Function.	Use Actuator Position XDCR Signal.	Use Actuator Position XDCR Signal.
Single Channel Operation	Must Start Up And Shut Down Hydraulic Systems In Sequence Or Add Hydraulic Shutoff Valves And Control Switches.	Add Switching To Shut Off Electrical Power In Sequence.	o Add Switching To Disable Fault Detection Logic. o Add Switching To Shut Off Electrical Power To Each Channel.
Fault Detection *Shutting Off Hydraulic Power Will Not Check Status Of Monitors And Comparator.	o *Add Program To Introduce Hardover Signals Sequentially To Servoamplifiers Downstream of Voters. o Must De-pressure And Repressurize Hydraulic Circuits For Reset Or Add Shutoff Valves.	o Introduce Command Input With Electrical Power Off To All Clutches An Observe Power Stage Failure Indication. o Add Program To Introduce Hardover Signals (Electrical Power Off To Power Stages) Sequentially And Observe Command Failure Indication. o Add Reset Provisions.	o Switch Off Electrical Power To Servos In Sequence And Observe Failure Indication. o Add Reset Provisions.

Table 7-13. Cost Comparison — Aileron, Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
Unit & Instl. Costs	Low. Smaller Envelope, Fewer Parts.	Medium. Difficult Instl. Fabrication.	Low.
Operational Cost	High Due To High Maintenance Required (Low MFBMA).	Less Than Configuration 1. (Less Maintenance Required).	Low. (High MFBMA).
Development Cost	Low. Similar Units Under Development.	High. Requires Extensive Development.	Medium. Control Portion Under Development. Problem With Power Spools' Synchronization.

7.4 DIGITAL SERVOACTUATOR COMPARISON

A digital configuration weight was established for the orbiter aileron, orbiter elevator, and booster elevator. They are compared to the analog configurations as shown in Figure 7-1. The weight is approximately equal to Configuration 1, aileron orbiter, but tends to become heavier than analog electohydraulic when compared at the elevator applications (+70 lb for the orbiter, +260 lb for the booster). In the case of the aileron, the digital configuration has less weight in the actuators, but more weight in transmission lines. This is due to the four power circuits it has compared to three. (The digital configuration uses four power circuits, analog Configuration 1 uses three.) The flow rates are small allowing small digitizers to be used; thus they have little influence on weight but become sensitive to fabrication tolerances.

For the larger applications, the increased weight trend of the digital configuration over that of analog Configurations 1 and 2 is due primarily to the digital valving (all use four power circuits). The valving becomes unwieldy at high flows and the assumption of a 30-Hz cycling rate may not be valid for the large digitizer spools.

The reliability of the digital configuration compares favorably with the analog configurations even though the upper stage servo failure rates were assumed to be 10 times that of an equivalent analog. This assumption was made to compensate for the increased number of cycles imposed on a torque motor for digital applications.

The reliability of the digital configuration remains high, apparently, due to the multiple redundancy and failure effects, which are less severe in digital systems.

The maintainability suffers drastically in the digital configuration. The duplication of servos and digitizers in each channel coupled with high failure rates result in an estimated MFBMA of 7.5. This compares with analog configurations that range from 28 to 140.

The critical comparisons between the digital and analog configurations are in the area of performance and development. Performance of the analog type has been verified by development, testing, and usage. As stated before, the digital configuration in this report is a concept only and does not represent any previous development effort with regard to redundancy mechanization. Without detailed investigation, there remains considerable doubt that pulse synchronization and exact positioning between channels can be achieved.

In summary, the comparisons show that a digital configuration can be competitive in most areas with analog types, but more investigation and development effort is needed.

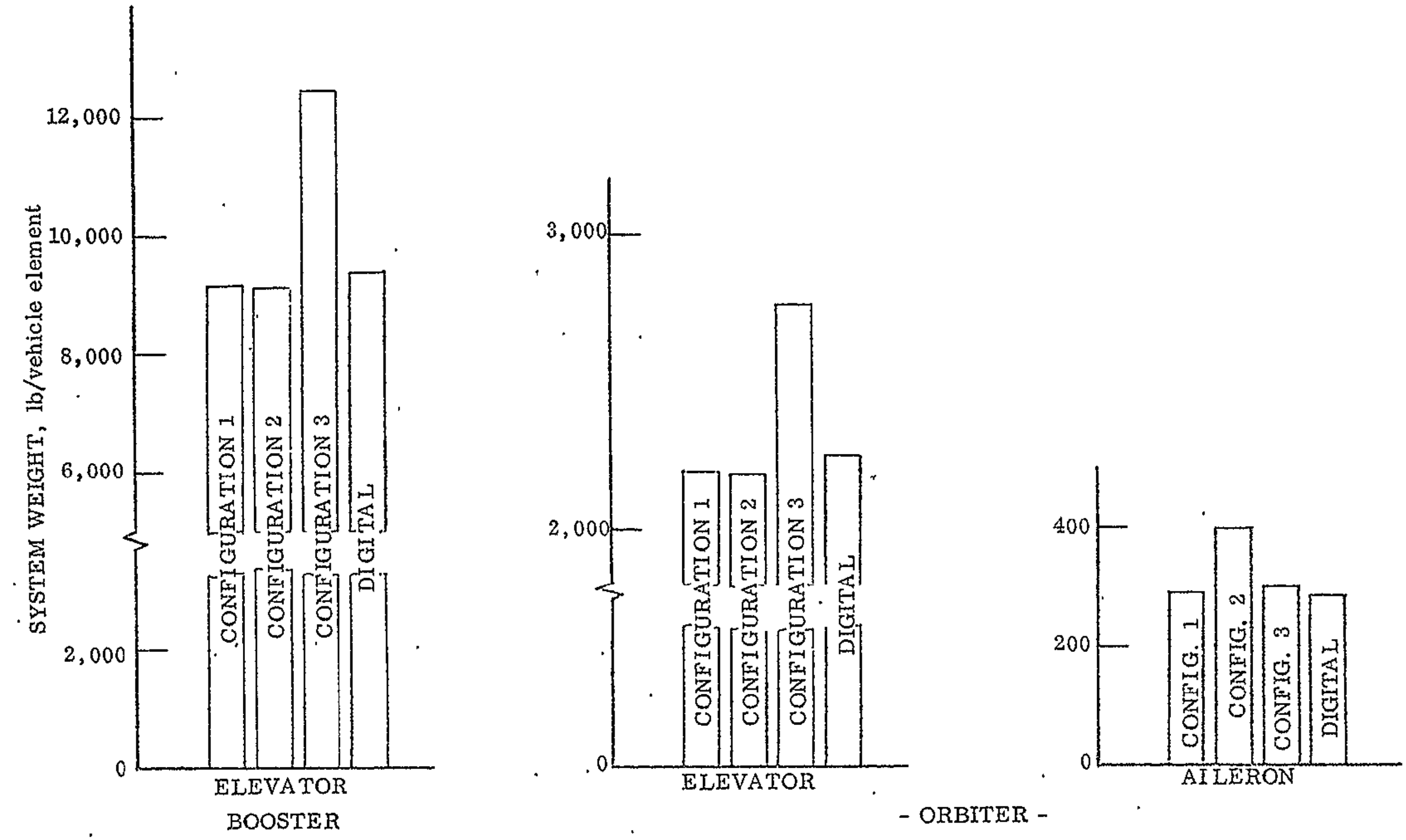


Figure 7-1. Analog vs. Digital Weight Comparison

7.5 ORBITER TVC

NOTE: Configuration 1 for the orbiter is the same as Configuration 1 for the booster. These configurations operate off the vehicle APU driven hydraulic circuits.

Basic data to generate weight:

Max Hinge Moment/Axis/TVC, ft-lb	65,000
Hinge Moment at Max Rate/Axis/TVC, ft-lb	44,000
Max Flow Rate/Axis/TVC, gpm	8.8
Max Flow Rate/TVC (1.41 x Max Flow/Axis), gpm	12.3
Flow Rate/Circuit/Engine Mounted Hyd. Supply, gpm	12.3
Flow Rate/Circuit/APU Driven Hyd. Supply, gpm	12.3
Length - ASC Junction to Engine Bulkhead, ft	21
Length - Bulkhead to Actuators/TVC/Circuit, ft	17
Length - Engine Mounted Hydraulic Transmission/TVC/Circuit, ft	16

Table 7-14. Weight Comparison - TVC, Orbiter

Nomenclature	Configuration 1			Configuration 2			Configuration 3		
	No.	Unit Wt. Lb	Total Wt. Lb	No.	Unit Wt. Lb	Total Wt. Lb	No.	Unit Wt. Lb	Total Wt. Lb
Servoactuator	4			4			4		
Power Actuator		58			60			58	
Control Portion		18.5			15			14	
Pwr Valve And/Or Manifold		9			13			16	
		85.5	342		88	352		88	352
Centering									
Actuator	4	20	80				4	20	80
Accumulator	2	20	40						
Valve	2	2	4	2	2	4	2	2	4
Hyd. Power Generation				6	125	750	4	124	496
							2	83	166
Hyd. Transmission	4	30	120	6	13	78	6	13	78
Total Weight			586			1184			1176
ΔWeight			0			+598			+590

Table 7-15. Reliability Comparison — TVC, Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
Probability of Mission Success	.99998	.99941	.99904
Relative Ranking	48.5	1.65	1

Table 7-16. Maintainability Comparison — TVC, Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
MFBMA			
Probability Of All Elements Working	.9915	.9862	.9876
MFBMA	118	73	81
Relative Ranking	1.62	1	1.11
Components/Vehicle	128	252	212
Installation	Difficult Actuators' Instl., But Requires No Engine Mounted Hydraulic Circuits.	Poor. Due to Separate Engine Mounted Hydraulic Circuits.	Poor. Difficult Actuators' Instl. Plus Engine Mounted Hyd. Circuits.

Table 7-17. Performance Comparison — TVC, Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
Normal Performance	• Good. No Fight Between Outputs, Only 1 Active At A Time.	Good	May Be Subject To Small Amplitude Limit Cycling.
Redundancy Performance	• No Degradation After Failure. • Nuisance Tripping May Be Problem	• Slight Degradation After Failure.	• Output Actuator Rate Reduced 50% After Failure. • Nuisance Tripping May Be Problem.
Versatility And/Or Commonality	Common To Booster Configuration.		

Table 7-18. Checkout Capability — TVC, Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
Normal Operation	Use Actuator Position XDCR To Provide Output Signal To Checkout/ Monitoring Function.	Add Actuator Position XDCR Or End Position Limit Switches.	Same as 1.
Single Channel Operation	Must Start Up And Shut Down Hydraulic Circuits In Sequence Or Add Hydarulic Shut Off Valves And Control Switching.	o Add Switching To Disable Cross Equalization & Fault Detection Logic. o Add Switching To De-energize Shut Off Valves.	o Add Switching To Disable Fault Correction Logic. o Add Switching To De-energize Shut Off Valves.
Fault Detection	o Add Program To Introduce Hardover Signals Sequentially To Servo Amplifiers.	o De-energize Shut Off Valves In Sequence And Observe Failure Indication. o Add Reset Provisions.	o Same as 2.

Table 7-19. Cost Comparison — TVC, Orbiter

Criteria	Configuration 1	Configuration 2	Configuration 3
Unit & Instl. Costs	Low. Fewer Parts.	High. Due To Multiple Engine Mounted Hydraulic Circuits.	High. Same As 2.
Operational Costs	Low. Same As Above.	High. Same As Above.	High. Same As Above.
Development Cost	Low.	Low.	Low.

7.6 BOOSTER TVC

NOTE: Configuration 1 for the booster is the same as Configuration 1 for the orbiter. These configurations operate off vehicle APU driven hydraulic circuits.

Basic data to generate weight:

Max Hinge Moment/Axis/TVC, ft-lb	65,000
Hinge Moment at Max Rate/Axis/TVC, ft-lb	44,000
Max Flow Rate/Axis/TVC, gpm	8.8
Max Flow Rate/TVC (1.414 x Max Flow/Axis), gpm	12.3
Flow Rate/Circuit/Engine Mounted Hyd. Supply, gpm	12.3
Flow Rate/Circuit/APU Driven Hyd. Supply, gpm	74
Length - ASC Junction to Engine Bulkhead, ft	35
Ave Length - Bulkhead to Actuators/TVC.(APU Hyd.), ft	30
Length - APU Hyd. Transmission (6 Engines/Circuit, ft	180
Length - Engine Mtd. Hyd. Transmission/TVC/Circuit, ft	16

Table 7-20. Weight Comparison – TVC, Booster

Nomenclature	Configuration 1			Configuration 2			Configuration 3		
	No.	Unit Wt. Lb	Total Wt. Lb	No.	Unit Wt. Lb	Total Wt. Lb	No.	Unit Wt. Lb	Total Wt. Lb
Servoactuator	22			22			22		
Power Actuator		58			51			51	
Control Portion		18.5			5			7.5	
Pwr Valve And/Or Manifolds		9			6.5			13	
		85.5	1881		62.5	1375		71.5	1573
Centering									
Actuator	22	20	440						
Accumulator	11	20	220	11	20	220			
Valve	11	2	22	11	2	22	11	2	22
Hyd. Power Generation				11	125	1375	11	125	1375
							11	83	915
Hyd. Transmission	4	258	1032	11	13	143	22	13	286
Total Weight			3595			3135			4171

Table 7-21. Reliability Comparison — TVC, Booster

Criteria	Configuration 1	Configuration 2	Configuration 3
Probability of Mission Success	.999996	.999996	.999899
Relative Ranking	2.5	2.5	1

Table 7-22. Maintainability Comparison — TVC, Booster

Criteria	Configuration 1	Configuration 2	Configuration 3
MFBMA			
Probability Of All Elements Working	.99984	.99991	.99983
MFBMA	6250	11,111	5888
Relative Ranking	1.06	1.89	1
Components/Vehicle	766	444	720
Installation	Difficult Actuators Instl., But Requires No Engine Mounted Hydraulic Circuits.	Fewer Actuators To Maintain, But Requires 11 Engine Mounted Hydraulic Circuits.	Poor. Tandem Actuator Instl. Plus 22 Engine Mounted Hydraulic Circuits.

Table 7-23. Performance Comparison — TVC, Booster

Criteria	Configuration 1	Configuration 2	Configuration 3
Normal Performance	Good.	Good.	Good.
Redundancy Performance	• No Degradation After Failure (Fail Operate Capability). • Nuisance Tripping May Be Problem..	• Fail Operate In Upper Servo Stage. Some Degradation After Failure. • Not Fail Safe After 2 Like Failures.	• Fail To Null Capability Only.
Versatility And/Or Commonality	• Common To Orbiter Configuration.	• No Monitoring - Health Status Cannot Be Verified In Flight.	

Table 7-24. Checkout Capability — TVC, Booster

Criteria	Configuration 1	Configuration 2	Configuration 3
Normal Operation	Use Actuator Position XDCR To Provide Output Signal To Checkout/ Monitoring Function.	Add Actuator Position XDCR Or End Position Limit Switches.	Same As 2.
Single Channel Operation	Must Start Up And Shut Down Hydraulic Circuits In Sequence Or Add Hydraulic Shutoff Valves And Control Switching.	Add Program To Introduce Opposite Hard-over Signals To 2 Servo Channels Sequentially To Disable All But One.	Same As Normal Operation.
Fault Detection	Add Program To Introduce Hardover Signals Sequentially To Servo Amplifiers.	Not Applicable. Servoactuator Is Monitorless (No Detection, Correction)	Add Program To Introduce Hardover Signal To Either Servoamplifier.

Table 7-25. Cost Comparison — TVC, Booster

Criteria	Configuration 1	Configuration 2	Configuration 3
Unit & Instl. Costs	High. To Be Common To Orbiter And Also Use APU Power, Excess Redundancy Is Applied At All 22 Actuator Instls.	Lower Than 1. Simpler Actuator Partially Offset By The 11 Engine Mounted Hydraulic Circuits.	High. Excess No. Of Engine Mounted Hydraulic Circuit Instls.
Operational Costs	High. Due To Number Of Components To Stock & Maintain.	Low. Fewer Parts.	Same As 1.
Development Costs	Low.	Minimum. Similar Unit Fully Developed.	Low.

SECTION 8

DISCUSSIONS AND RECOMMENDATIONS

8.1 DISCUSSION

8.1.1 WEIGHT TRENDS

8.1.1.1 Intermediate Applications. Figure 8-1 is a plot of servoactuator and transmission line weights for three and four power circuits through all hinge moments of interest for aerodynamic surface controls. This trend is only valid if the ground rules and requirements of this study apply. That is, there is no alternate function to back up an aerodynamic control surface, and fail operate, fail safe (e.g., fail operate) performance is required. The fail operate, fail operate rule allows each of the four actuators to be designed for 50% of the required hinge moment after two failures. This concept weighs less than a three-actuator arrangement down to the lower limit where valve and hydraulic transmission weights begin to have influence. No other power weights are included.

8.1.1.2 Electromechanical. The only pure electromechanical configuration studied was applied to the orbiter aileron. The weight did not compare favorably with hydraulic configurations primarily due to the electric motor weight. The motor sizes were determined by torque requirements resulting in oversized motors with respect to horsepower rating. Due to the limited time available for detail design analysis, no attempt was made to apply the balanced power concept.[24] This concept involves the use of flywheel inertia to absorb power peaks allowing the motors to be sized for average power demand, thus reducing overall weight. The electromechanical unit was configured as an integrated package (power and control). This configuration would be very difficult to install in the limited space in the wing and more likely would consist of control channels, clutches, and motors in the fuselage with mechanical shafting installed in the wing to an output ballscrew or power hinge. No weight penalty was assigned for mechanical shafting to offset the over-design of the motors.

8.1.1.3 Electromechanical Control. The electromechanical velocity summing control unit was used with three hydraulic circuits (Configuration 3, aerodynamic surface controls). This configuration is obviously heavy because of the use of three hydraulic power circuits. Figure 8-2 shows the comparison of just the electromechanical and electrohydraulic control components. The electromechanical weights for this study are shown by the upper solid line. The criteria used for sizing channel control outputs were conservative and affected electromechanical weights more than electrohydraulic. The dotted line represents a reduced size for electromechanical control where the output force ranges from 400 lb at small flows to 1000 lb at high flows. The reduced

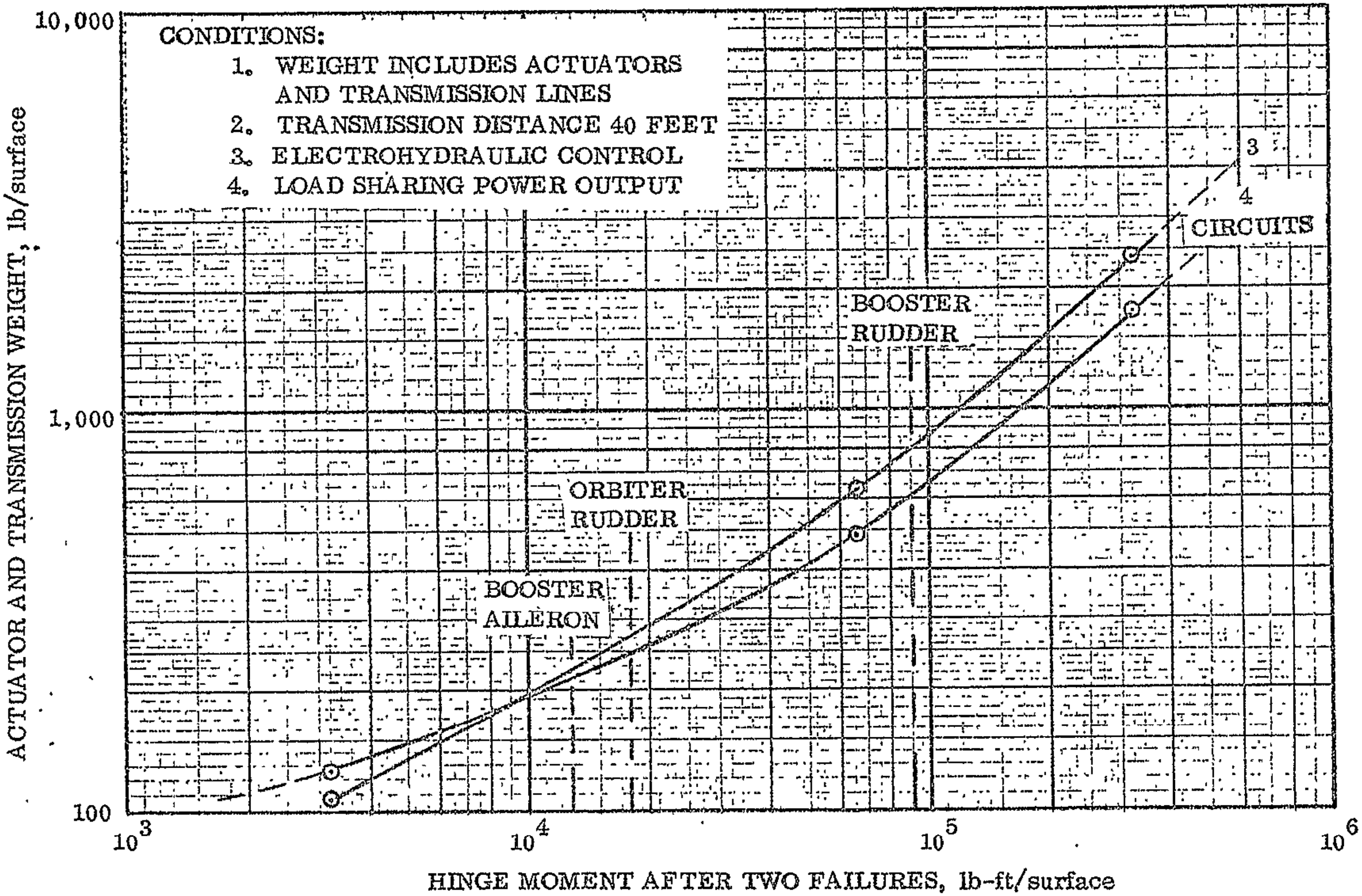


Figure 8-1. Servoactuator and Transmission Weight

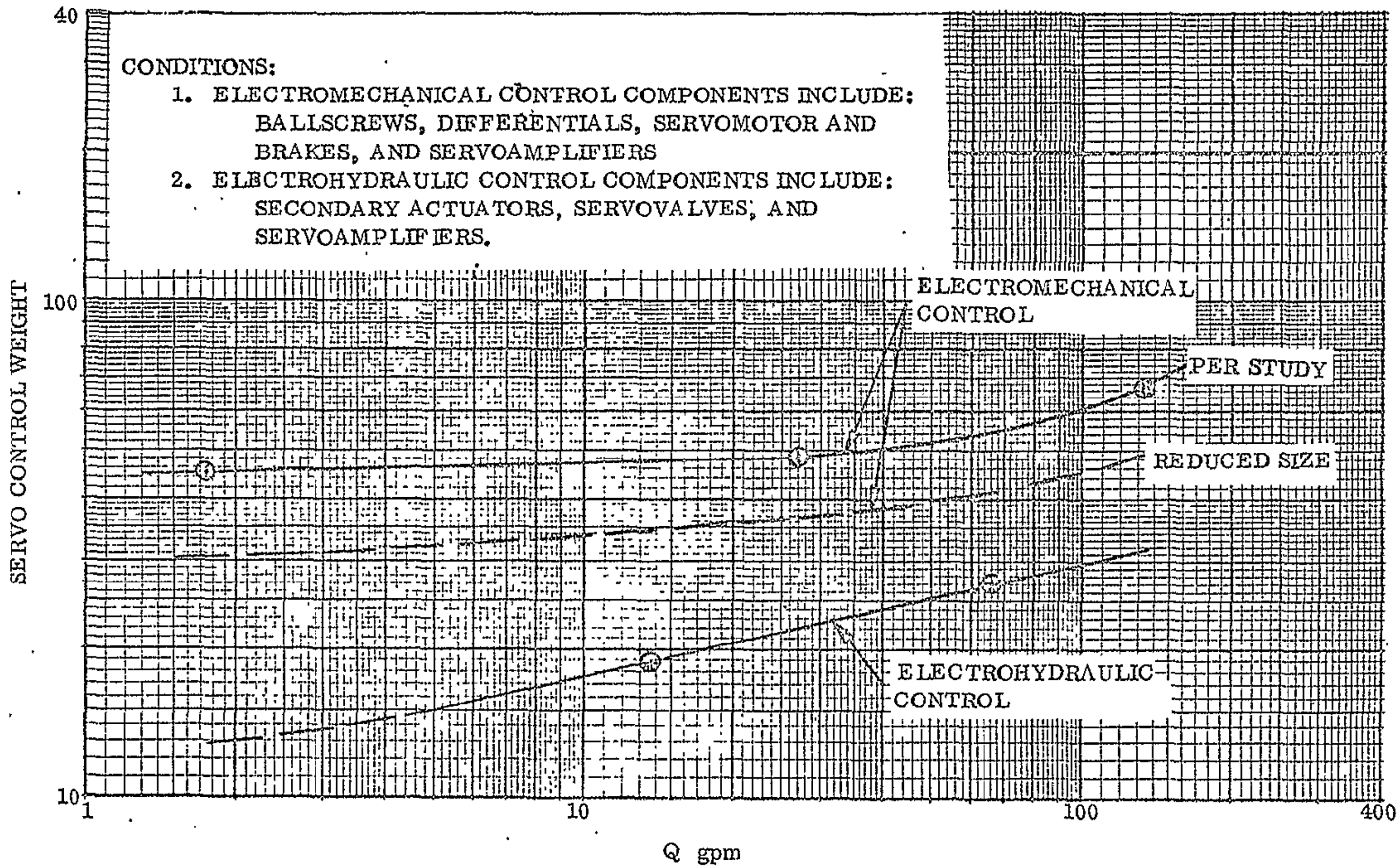


Figure 8-2. Weight - Electromechanical vs. Electrohydraulic Control

electromechanical control is still heavier than its electrohydraulic counterpart but the Δ weight is less significant in comparison to power actuator and power circuit weights.

8.1.1.4 Redundancy vs. Weight - Aerodynamic Surface Controls. The configurations in this study define the limits of redundancy required to meet fail operate, fail operate criteria. The redundancy was carried through to the power stages. The result is a rather dismal weight total. It is no surprise to note that most of the weight is packed into the power systems and power actuators.

We assume that a hydraulic power actuator never jams. If redundant seals are used to reduce leakage failures and adequate margins are built in to preclude any structural failures (barrel rupture) the actuator becomes very reliable and as such may not need to be as redundant as other elements. Figure 8-3 shows the effects on weight when power actuator and power circuit redundancy are reduced. The curves are based on averaged data: 1.5 hr flight time, 35% average power and 40 ft transmission length. Curve A is the four-power circuit, four-actuator/side configuration per this study for the booster and orbiter elevator. Curve B eliminates two actuators per side but retains the four power circuits. Switching valves are used to switch in two standby power circuits. Curve C goes one step further and eliminates the two actuators/side and two power circuits. In this case each pair of actuators has to provide 100% hinge moment. The trend shows that eliminating power circuits is not effective, especially at the orbiter hinge moments, because the remaining two circuits must produce twice the output required of each of the four circuits.

8.1.1.5 Redundancy vs. Weight - TVC. Following the same procedure as mentioned in Section 8.1.1.4, the weights of all configurations could be reduced. In Configuration 3 for the booster TVC, the centering power circuit is also used in the servo channel to ensure that the system will fail to null at the first power circuit failure. This requires two hydraulic circuits for the configuration. If only one power circuit were used in the servo channels, the servoactuator could be centered by accumulator power. By eliminating one half of the tandem actuator and spool (add a switching valve) the total weight could be reduced 1256 lb from the 4157 lb tabulated in Section 7. This arrangement is not as safe as Configuration 3 because centering capability is lost after a failure combination of accumulator gas pressure and hydraulic circuit. If power actuator redundancy is eliminated in the booster TVC configurations, the Δ weight decrease is as shown in Table 8-1.

8.1.2 COMMONALITY. Configuration 1 for the orbiter and booster TVC was common and operated off vehicle APU systems. The results show that the booster TVC systems become heavy and quite complex in the servoactuator. The 22 actuator installations should not be configured by orbiter requirements for the sake of commonality alone. However, when power actuator redundancy is reduced, the absolute weight penalty between Configurations 1 and 2 (least weight configuration) for the booster is 262 lb as shown above. Later studies will have to determine the allowable penalty to attain commonality.

Table 8-1. Δ Weight Decrease

Booster TVC	Vehicle Weight (lb) Per This Study	No Redundancy in Power Actuator	Δ Weight
Configuration 1	3595	*2979	-616
Configuration 2	3135	**2717	-418
Configuration 3	4171	***2915	-1256

*Delete one half of tandem actuator. Add switching valve and incorporate centering provisions in power actuator.

**Delete centering portion of power actuator.

***Delete one half of tandem actuator and valve manifold, one engine-driven hydraulic circuit, and add switching valve and centering accumulator.

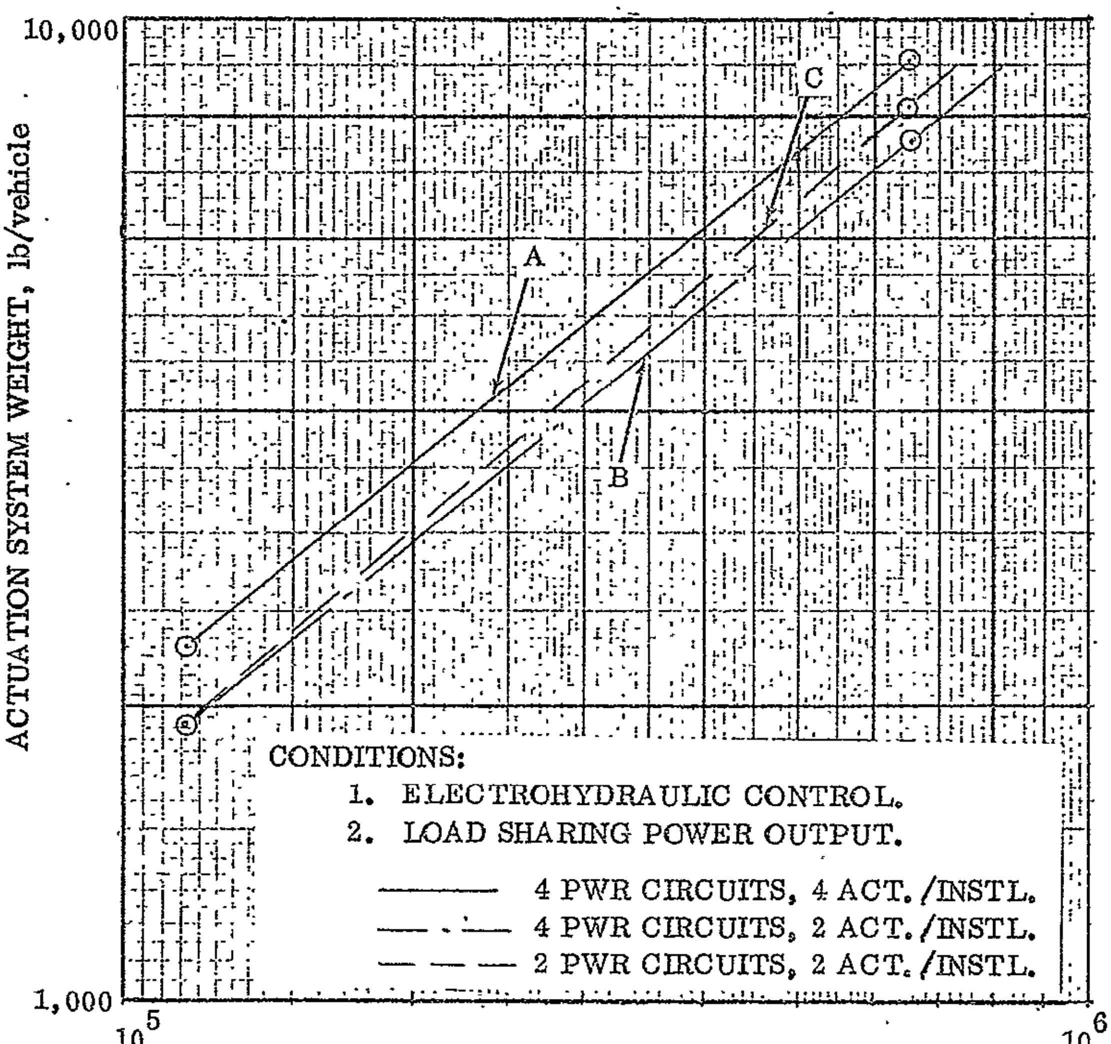


Figure 8-3. Vehicle Hydraulic Weight

The electromechanical velocity summing control is attractive because of the potential commonality on aerodynamic surface controls. It can be applied to any application independent of the number of hydraulic power circuits used. The unit is essentially the same whether it controls two, three, or four hydraulic power circuits. Hydraulic control requires small hydraulic circuits for servo power, in addition to the circuits supplying output actuators if the control is more redundant than the output.

8.1.3 RELIABILITY AND MAINTAINABILITY. A review of reliability indicates that self-monitoring techniques offer a substantial gain in reliability. This reflects the paralleling effect that self-monitoring provides. It does not require cross connections or cross monitoring voting logic which tends to place all detection and switching elements in series in a reliability model. This trend can be seen by observing the relative reliability ranking of each Configuration 1 for aerodynamic surface controls and TVC. Self monitoring is common to these configurations.

In the orbiter aileron, Configuration 1 is two orders of magnitude better than Configurations 2 and 3. This gain is due to hydraulic logic detection and switching (not dependent on electronic or electrical switching) as well as self monitoring. In this case self monitoring is accomplished by massive redundancy at the control level (six channels).

Another noticeable trend, however, is that maintainability is usually lower for self-monitoring schemes in terms of MFBMA. These results agree with the normal conflict between reliability and maintainability; that is, when channel or system redundancy is used to increase reliability, maintainability suffers.

8.1.4 AUTOMATIC VS. MANUAL SWITCHING. The force summing redundancy mechanization has the advantage over all other methods in that switching does not have to be immediate because the output of a bad channel is prevented from feeding through to the output. Thus a servoactuator (for aerodynamic surface controls) can be simplified (human factors permittting) by assigning the switching function to the crew.

8.1.5 INTEGRATED HYDRAULICS. Integrated hydraulics (power and control by wire) was not considered in this study. A cursory look at sizes for the elevator of both vehicles indicated a prohibitive package size for installation. Once central hydraulic circuits were assumed, they were then available to supply any load application. Installing integrated packages at low load applications would introduce small hydraulic power circuits in addition to the APU-driven central circuits causing an unnecessary complexity. Additionally, an integrated package would no doubt require cooling lines to route into the fuselage, thus hydraulic transmission could not be eliminated entirely.

8.2 RECOMMENDATIONS

a. Hydraulic power and analog servoactuators should be used throughout all flight control applications.

b. Four hydraulic power circuits should be available and all four used on aerodynamic surface controls where they are competitive in weight.

c. Power actuator and power circuit redundancy as defined by this study be retained on aerodynamic surface controls.

Reasons:
1. Length of operating time/flight.
2. Need to spread actuator reaction points on large single surfaces.
3. Number of power circuits required on board is not solely a function of flight controls. (Utility functions will probably require a minimum of four circuits.)

d. Power actuator redundancy should be reduced or eliminated on TVC and failure criteria applied to control channels only.

Reasons:
1. Short operating time reduces probability of failure/flight.
2. Save weight.
3. Difficult actuator installation.

e. Mechanical feedback be employed in TVC if possible to eliminate separate centering actuator (fail to null).

f. Central APU-driven circuits be used for TVC if power actuator redundancy is eliminated and APU driven circuits are sized by aerodynamic surface controls. If TVC sizes the APU (not true based on requirements used in this study) additional study is required to determine impact on power source and power source fuel weight. (Aerodynamic surface controls operating at lower part load will increase fuel consumption.)

g. Obvious superiority of one control redundancy mechanization method could not be established. The many different techniques under development in the industry today indicate that no one solution has a clear advantage. Quite often a particular technique evolves as a result of special requirements. However, based on the requirements and findings of this study the following techniques are recommended as best candidates:

1. Force summing control (self monitored) for aerodynamic surface controls [similar to Configuration 1 for the elevator].

   Reasons: a) Nearly fully developed concept (SST).

   b) Provides protection against jams.

c) Has potential of providing control after three control failures. (e.g., three channels may be able to provide two fail operate capabilities in a pure fly-by-wire system.)

d) Versatile: Switching can be manual.

2. Electromechanical velocity summing for aerodynamic surface controls [similar to Configuration 3 for the elevators].

Reason: Easier to achieve commonality for all load applications. Unit is functionally the same regardless of the number of hydraulic power circuits used. (Use is contingent on fabrication of multiple tandem power spools.)

3. Active/standby control (self monitored) using secondary actuator for TVC [similar to Configuration 1, TVC, except secondary actuator, and mechanical feedback from power actuator added, and power actuator redundancy reduced].

Reasons: a) Common to orbiter and booster.

b) Can use vehicle APU-driven circuits (four per vehicle).

c) Attains fail operate capability with only two hydraulic circuits.

d) Redundancy level is consistent with available command channels. Requires no voters or additional servoamplifiers.

4. Monitorless "majority voting" for booster TVC only [similar to Configuration 2 for the booster TVC].

Reasons: a) Least weight.

b) Fully developed concept.

c) Least complex servoactuator.

d) Monitorless arrangement more adaptable to booster than orbiter. (System checkout on the ground prior to flight.)

h. A single orbiter elevator package be installed in the fuselage area if at all possible. If elevator stiffness can be improved satisfactorily with a 250-lb Δ weight increase, no weight penalty is involved and the servoactuator complexity, installation, and performance are vastly improved.

8.3 ENVELOPE SIZES

Installation configurations were not possible at this time due to the preliminary state of vehicle design. To show some representative sizes, reasonable geometric proportions were used based on very preliminary vehicle configuration data. Figures 8-4 and 8-5 are examples of booster elevator and orbiter elevator envelope sizes

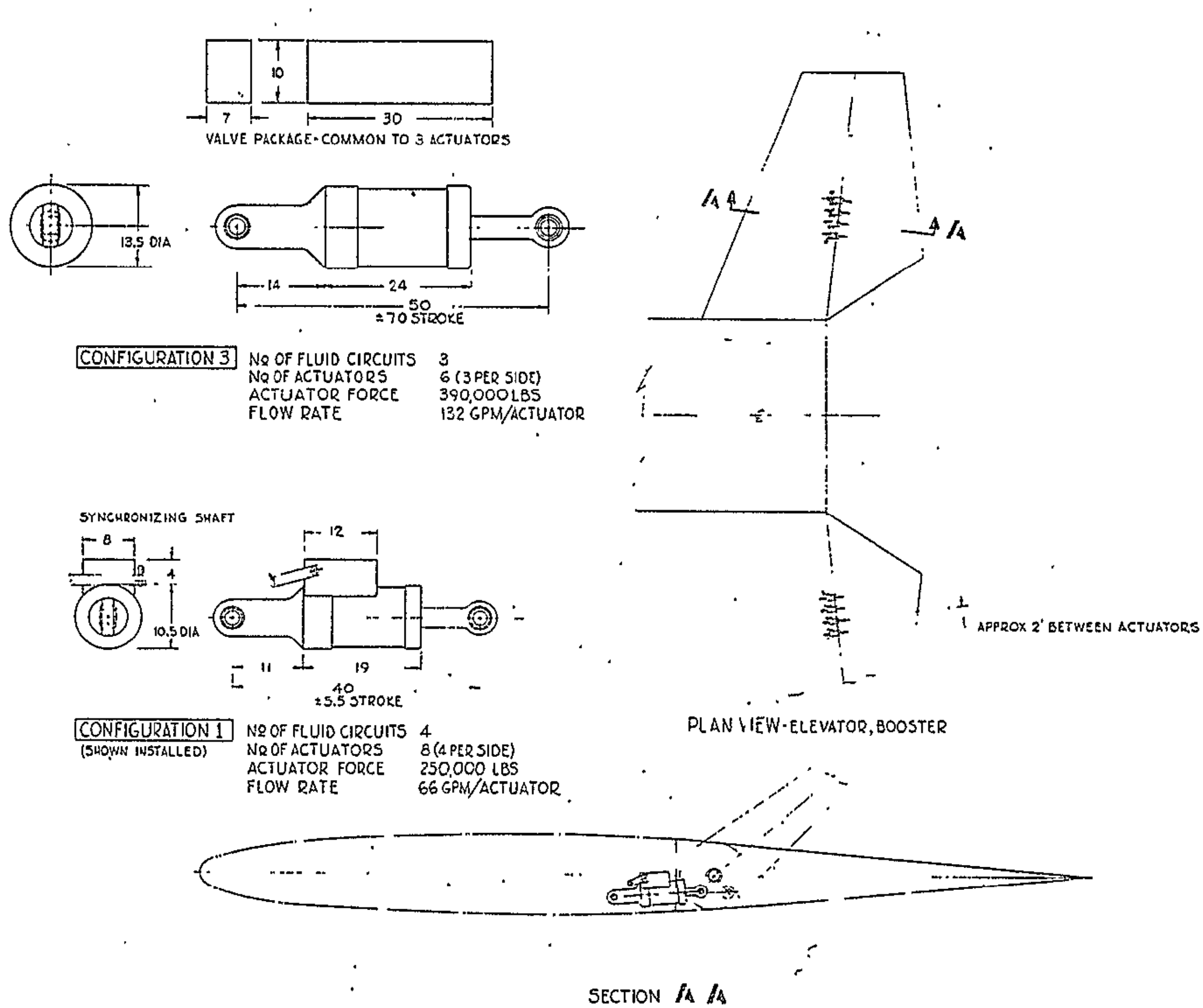


Figure 8-4. Envelope Sizes for Booster Elevator

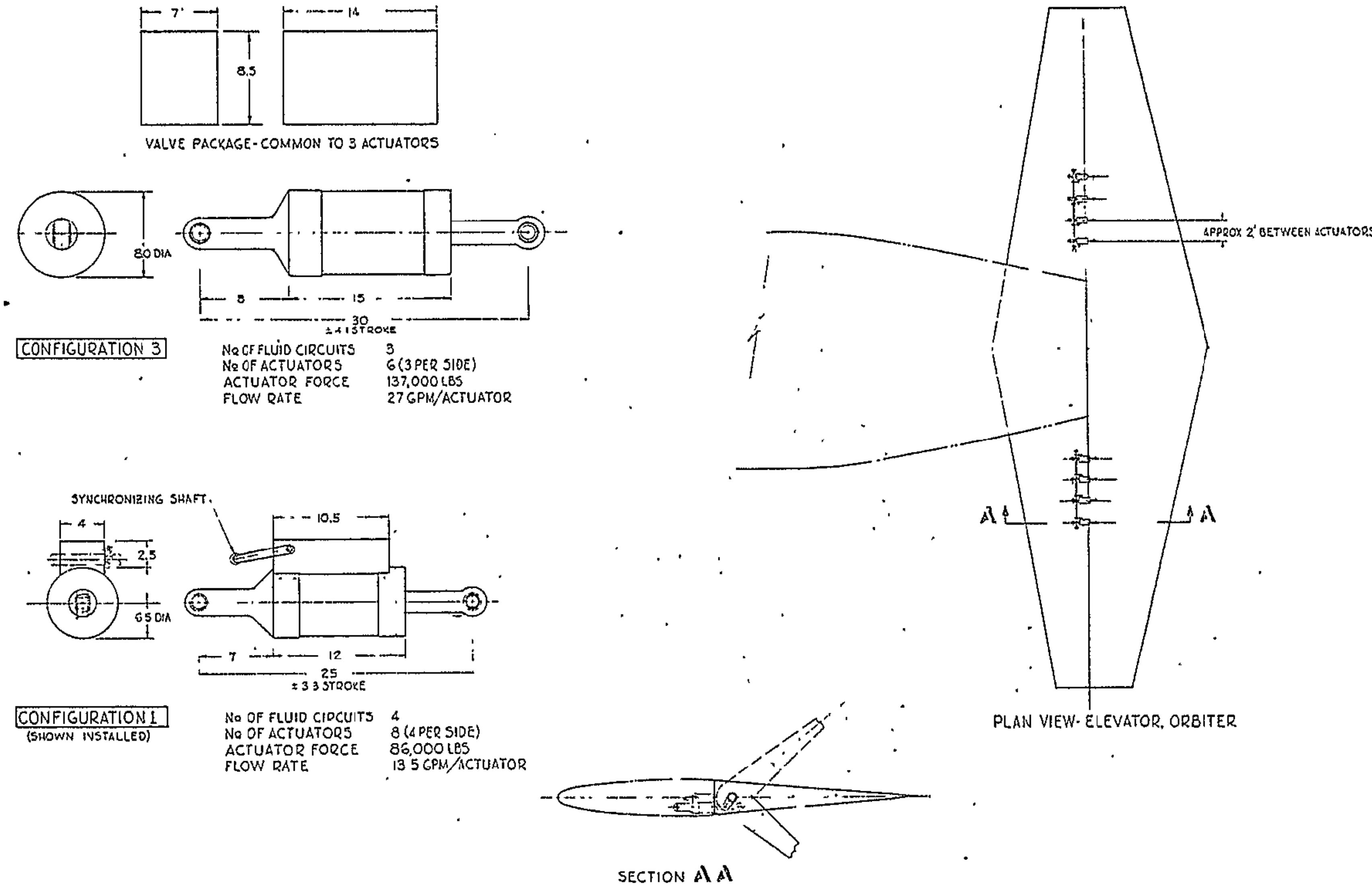


Figure 8-5. Envelope Sizes for Orbiter Elevator

respectively. These examples are shown only for the sake of ball park estimating. To pursue this exercise further was not considered productive in that actual envelope sizing requires more vehicle physical definition. One point is obvious from the figures: the installation designer will be hard pressed to install the three power circuit actuators in the available space.

SECTION 9

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