WADC TECHNICAL REPORT 55-29
PART 3
ASTIA DOCUMENT No. AD 118285

HYDRAULIC SERVO CONTROL VALVES

PART 3

STATE OF THE ART SUMMARY OF ELECTROHYDRAULIC SERVO VALVES AND APPLICATIONS

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APRIL 1957

AERONAUTICAL RESEARCH LABORATORY
CONTRACT AF 33(616)-3341
PROJECT 1385
TASK 70522

WRIGHT AIR DEVELOPMENT CENTER
AIR RESEARCH AND DEVELOPMENT COMMAND
UNITED STATES AIR FORCE
WRIGHT-PATTERSON AIR FORCE BASE, OHIO

FOREWORD

This report, State of the Art Summary of Electrohydraulic Servo Valves and Applications, was prepared by L. Schmid, P. Weiss, and B. Johnson under the supervision of G. Leslie, Administrative Engineer of the Automatic Control Systems Section and D. C. McDonald, Assistant Director, Cook Research Laboratories, Skokie, Illinois, under Air Force Contract No. AF 33(616)3341, Hydraulic Servo Control Valve Analysis. This is the third in a series of reports to be published for the Air Force under the general category of electrohydraulic servo valves, although this is the first report under this particular contract.

The report reviews the general state of the art of electrohydraulic servo valves and their application in airborne vehicles. The work was administered under the direction of the Aeronautical Research Laboratory, WADC, with Mr. R. W. Rautio as supervising task scientist. The document is unclassified.

ABSTRACT

A state of the art summary describing the electrohydraulic valves which have been brought out in the last two years is presented. Valves brought out prior to this are described in WADC TR 55-29 Part 1. Some of the manufacturing problems are reviewed.

A summary of information compiled from visits to aircraft and missile manufacturers is also presented. Their comments as to the performance of the valves presently used in their vehicles are reviewed. In addition, a general discussion of system design procedures plus a listing of typical system characteristics is presented.

PUBLICATION REVIEW

This report has been reviewed and is approved.

FOR THE COMMANDER:

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CHAPTER I

INTRODUCTION

A. Purpose

This report covers the results of the second state of the art survey conducted under the valve study program. This was a survey of valve manufacturers and valve users (primarily missile manufacturers) and had four primary objectives.

First, several new electrohydraulic servo valves have been placed on the market since the completion of the first survey; in addition, several new valves are under development. Many of these valves are characterized by new designs and manufacturing techniques, which on the surface at least, appear promising. In light of the limitations of the valves previously investigated, particularly in regard to high temperature operation, it was decided to obtain information on the newer valve developments. In this same vein, at the time of our previous survey, most of the missile manufacturers had just begun the use of electrohydraulic servo valves and had not logged sufficient time and experience with the valves to permit their making concrete conclusions with respect to the valves capabilities. During the last year and a half, they have had ample time to evaluate the valves experimentally. In addition it was understood that they had system tested at least one or two of the newer valves. A summary of this experience was considered of great value to the valve study project.

A second objective of this survey was to obtain certain information necessary for writing a finalized electrohydraulic servo valve specification for the Air Force. The needed information was related primarily to the servo valve environment (e.g., vibration, noise, temperature, pressure pulsations) since sufficient facilities were not available for investigating these factors completely in our Laboratories. In addition,

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Manuscript released by the Author in December 1956 for publication as a WADC Technical Report.

since a specification implies certain standardization, it was considered desirable to determine the valve manufacturers' and users' views as to the area and extent to which this standardization should be carried. The information relative to the specifications will be covered in a separate report.

A third objective of this survey was to investigate the various techniques used in the design and test of electrohydraulic servo control systems utilizing servo valves, especially from a dynamic standpoint and to determine how successful these techniques have been. Particular attention was to be paid to the analogical approaches that are being used. The techniques used to simulate loads and the degree to which analogs are employed in system testing were determined.

The fourth objective of this survey was to obtain actual detailed data on system characteristics, that is, design features used in the servo valve-actuator control loop. This information will be categorized and tabulated and be used as the basis for selecting a typical system for the analogical studies to be performed in a later phase of this study.

B. Method of Conducting Survey

For the most part, the information contained in this report was obtained via direct visits to the organizations listed over the period extending from March to June 1956. However, some information was obtained from Armed Services Technical Information Center reports and some obtained from brochures published by valve manufacturers. Most of the valve manufacturers visited during this contract were not visited during the previous survey. The visits to valve users were largely confined to missile manufacturers who have had extensive experience with the use of electrohydraulic servo valves. Many of the latter have actually assisted in the development of electrohydraulic servo valves by pointing out to the valve manufacturers the valve limitations encountered in system application and suggesting means for correcting them. It has been found that the people using the valves are actually more acquainted with the characteristics and peculiarities of the servo valve than the manufacturers, and thus a better source of information as to problems and limitations than are the manufacturers. Generally speaking, this information is not available from Air Force and other service installations.

C. Sources of Information

The following companies have contributed information to this state of the art study during the period of March through July 1956. The designations (M) and (U) after each organization indicate whether they are primarily manufacturers or users of servo valves, respectively, or both.

Bell Aircraft Co., Buffalo, New York	M & U
Bendix Pacific Division, North Hollywood, California	M & U
Bendix Research Laboratory, Detroit, Michigan	M & U
Boeing Aircraft Co., Seattle, Washington	M & U
Cadillac Gage, Detroit, Michigan	M
Consolidated Vultee Aircraft Co., Pomona, California	U
Consolidated Vultee Aircraft Co., San Diego, California	U
Douglas Aircraft, Santa Monica, California	U
Dalmo Victor, San Carlos, California	M
General Electric Company	U
Lear, Inc., Grand Rapids, Michigan	M
Lockheed Aircraft Co., Burbank, California	U
Glenn L. Martin Co., Baltimore, Maryland	U
Midwestern Instruments, Tulsa, Oklahoma	M
Minneapolis-Honeywell, Minneapolis, Minnesota	U
Moog Valve Co., East Aurora, New York	M
National Water Lift Co., Kalamazoo, Michigan	M
North American Aviation, Autonetics Division Downey, California	U
Northrop Aircraft, Hawthorne, California	U
Pegasus Laboratories, Berkeley, Michigan	M
Sanders Associates, Nashua, New Hampshire	M

CHAPTER II

COMPILATION OF CHARACTERISTICS OF NEW ELECTROHYDRAULIC SERVO VALVES

A. Scope

The initial survey contained data on some 21 different servo valve designs. A number of these were development valves and have since been discontinued. Notable among these are the Cadillac Gage CG-I, 1/ Peacock valve, Cadillac CG-II, 1/ Drayer-Hanson valve, NAA time modulated valve and the Minneapolis-Honeywell XMG36A1. (The Drayer-Hanson design was bought by Lear, Inc., and they presently are producing the valve.) A considerable number of new valve manufacturers have entered the field since that time. In addition, several of the original valve manufacturers have brought out new designs or designs modified for a particular application. This report will cover only the new valves and will not include any of those listed in the previous survey.

As was done in the aforementioned report, the operational and physical characteristics of the various valves are tabulated (see Table 1) to facilitate comparison and data reduction. In addition, the method of operation of each of the valves is briefly described with the aid of a schematic where one is available, or necessary.

In Table 1 no attempt has been made to classify the valves according to the number of stages, type of first stage, etc., as was done in WADC TR 55-29, Part 1. This is because most of the valves are nozzle-flapper two-stage valves and the small number of the other types does not warrant this classification. However, in all cases, the types of valves are grouped together, that is, the single-stage valves are listed first, then the two-stage valves with spool first stages, etc. The actual type of construction is, of course, covered under the detailed description of each valve.

All the servo valves listed in Table 1 with the exception of the National Water Lift-3301 valve are designed to operate at supply pressures up to 3000 psi. The minimum supply pressure has not been listed but most of the valves will not operate satisfactorily below 500 psi. All of the

1/ Cook designations, not actual model numbers.

TABLE 1 - VALVE CHARACTERISTICS

No.	Mfg. & Model Series	Over-All Size (in.)	Weight (lb)	Max Flow at 1000 psi Valve Drop (gpm)	Differential Current (ma)	Quiescent Flow (Max) (gpm)	Hysteresis and Threshold % Max Signal	Frequency Response (Flow) vs. Differential Current: ± 3 db Amp Ratio Freq. (cps)	± 3 db Amp Ratio Phase (deg)	Freq. at 90° Phase Shift cps	Max Filtration Recommended (microns)	Torque Motor Resistance (ohms)	Torque Motor Inductance (henries)	Remarks
1.	Lear, Inc. Model 5201A	1.7 x 2.0 x 3.2	0.66	0.87	20	0.065	± 3	320	250	120		1570	<10	Has 8 db peak at 380 cps
2.	National Water Lift Model 3301	1.75 2.0 x 3.0	0.66	0.65	20	0.03	2					1575		Operates at 1500 psi supply pressure Also make two stage valve
3.	Midwestern Instruments Model 7	2.1 x 2.4 x 3.5	2.3	2.2	40	0.1		95	155	52		3400	7	Hole and plug porting operates with dirty oil
4.	Lear, Inc. 5202A	1.7 x 1.9 x 3.8	1.37	2-10	20	0.13	± 2	70	50	85	100	1570	<10	
5.	Sanders Assoc. Model SA-17D	1.1 x 1.1 x 3.97	0.9	0-6	10		±3.5	220		65	10	1000	0.15	Advance information only
6.	Bendix Pacific HR Series 8	2.32 x 2.65 x 2.14	0.92	4.0	8	0.18		190	68	270	10	1000	1	
7.	Bendix Pacific HR Series 16	2.32 x 2.70 x 2.14	0.99	9.0	6 to 20	0.15		170	47	170	10	1000	1	Stale torque motor
8.	Dalmo Victor Model 10	1.75 x 2.15 3.06	0.87	6.0 gpm	8	0.15	4	120	70	180	10	1000	1	
9.	Moog 1800	1.75 x 2.5 x 3.06	1.5	0.5 to 10 gpm	5.0	0.35	3.0	62	70	92	10	Approx. 1000	1	Designed for operation under oil temperatures up to 400°F
10.	Cadillac Gage 200	2-5/8 x 1-29/32 x 2.312	0.95	8-85	6-20	≈0.15	<± 2				10	1000	1	Stainless steel construction dry coil
11.	Hydraulic Research 20	2.54 x 2.44 x 176	1.0	0.15 to 3.0	8	0.15	± 2.7	90		140	10	1100		Dry torque motor
12.	Hydraulic Research 21	2.54 x 3.20 x 176	1.1	2 to 10	8	0.25	± 2.7	60		100	10	1100		Triple filtration
13.	Robert shaw Fulton		0.7	8	8		± 3.5	50		100				Stale torque motor
14.	Bendix Research 241		3.62	Approx. 90	20		<3	150	-18	350	10	2500	23	Convergent flow, tapered lands on pilot stage
15.	Bendix Research 356	2.2 x 2.5 x 2.7	1.1	7.75	40		1	200	-64	280	10			Convergent flow, dry torque motor
16.	Pegasus Model 20	3.25 x 1.93 x 1.75	1.1	4.7	5	0.40	2	50		100	10	1500	1.0	
17.	Hagan Corp 80		130	18 54 80 @ 2000 psi drop	50		0.05					500	10	Spinning first stage spool

remaining data, such as maximum flow, leakage, dynamic response, etc., are based on a supply pressure of 3000 psi. It is, of course, realized that these are all nominal figures provided by the manufacturer, and because the figures were not always obtained under the same test conditions, they do not provide a concrete basis for comparison.

Considerable latitude is generally available as to the maximum flow range of the valve and the electromagnetic driver characteristics. For instance, the manufacturer can increase the flow of a particular valve by grinding wider porting slots or he can reduce the input current requirements by using a more powerful permanent magnet. Maximum leakage can be reduced by increasing the overlap of the second stage, providing the attendant increase in threshold can be tolerated.

All of the valves listed in Table 1 can be manifolded directly to the actuator or through a manifold block. The filtration figures shown refer to nominal figures quoted by the filter manufacturers. As will be discussed in detail in a later section of this report, these figures do not necessarily mean that the filters remove all particles larger than the size listed.

Most single-stage valves, and two-stage valves with spool first stages, must be dithered to obtain smooth operation from the valve. Dither is generally not required in flapper-nozzle valves, although it may improve their operation. For instance it has been found that the null shift characteristics of flapper-nozzle valves are improved significantly with the application of dither.

The frequency response and phase lag figures listed in Table 1 refer to the valve in an unloaded condition. These are generally obtained by applying a sinusoidal input signal of variable frequency and constant amplitude to the electromagnetic driver and measuring the amplitude and phase of the output flow over a range of frequencies. The phase shift and frequency at ± 3 db amplitude ratio and the frequency at 90 degrees phase shift have been tabulated. For further general background information on servo valves, the reader is referred to WADC TR 55-29, Part 1.

In addition to the companies listed in Table 1, the following organizations are also developing valves, but published data on their valves are not yet available:

(1) Adel

(2) Weston Hydraulic Ltd.

(3) Standard Controls (previous designs purchased by Cadillac Gage)

(4) Boeing Aircraft

(5) Minneapolis-Honeywell.

B. Detailed Valve Descriptions

1. Lear, Inc., Model 5201A

The Lear, Inc. Model 5201A Servo-Valve (see Fig. 1) is a

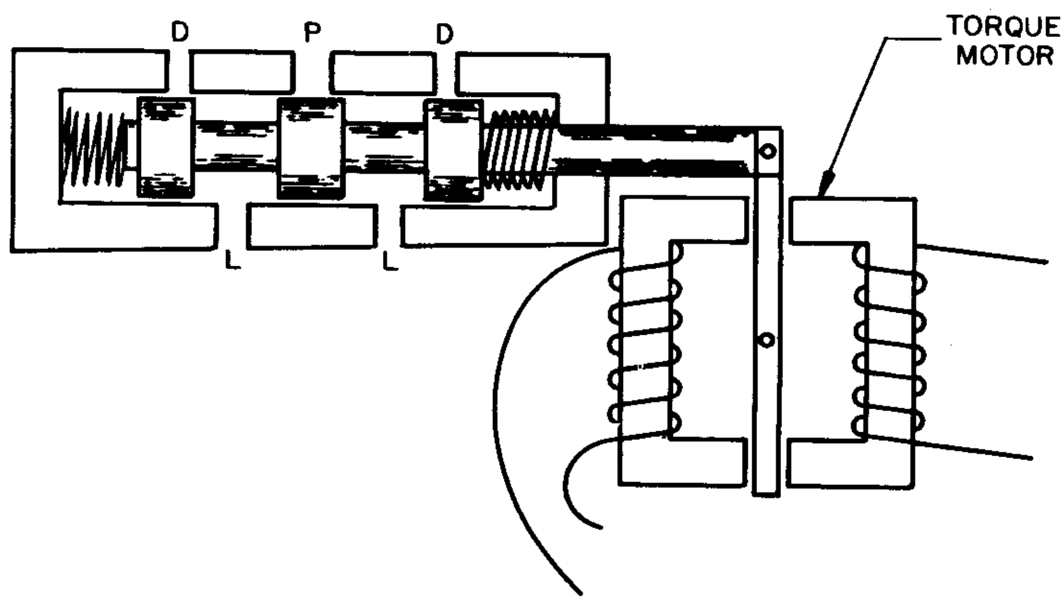

Figure 1 Lear Single-Stage Servo Valve Schematic

single-stage valve. It consists of a four-way spool which is driven by a Lear Model 5203A torque motor. The position of the spool is controlled as a function of the differential current in the two coils of the torque motor. The spool in turn controls the amount and the direction of the flow to the actuator. The torque motor is isolated from the drain and leakage oil of the output spool.

The valve spool and sleeve are made of hardened stainless steel. The valve is normally supplied with pigtailed leads; however, it is also available with a Viking VR4/2AB1 connector.

2. National Water Lift Model 3301

The National Water Lift Servo Valve Model No. 3301 (similar to the Lear valve of Fig. 1) is a single-stage, four-way valve utilizing spool and sleeve type construction. The spool is driven by a Lear Model 5203A torque motor. The position of the spool is controlled by the torque motor as a function of the differential current in the torque motor coils. The valve is supplied with a Viking Electric VR412 AG1 connector.

3. Lear, Inc. Model 5202A

The Lear, Inc. Model 5202A servo valve (see Fig. 2) is a two-stage, four-way valve utilizing spools both in the first stage and the output stage. The valve loop is closed by a feedback spring connecting the output spool to the torque motor armature.

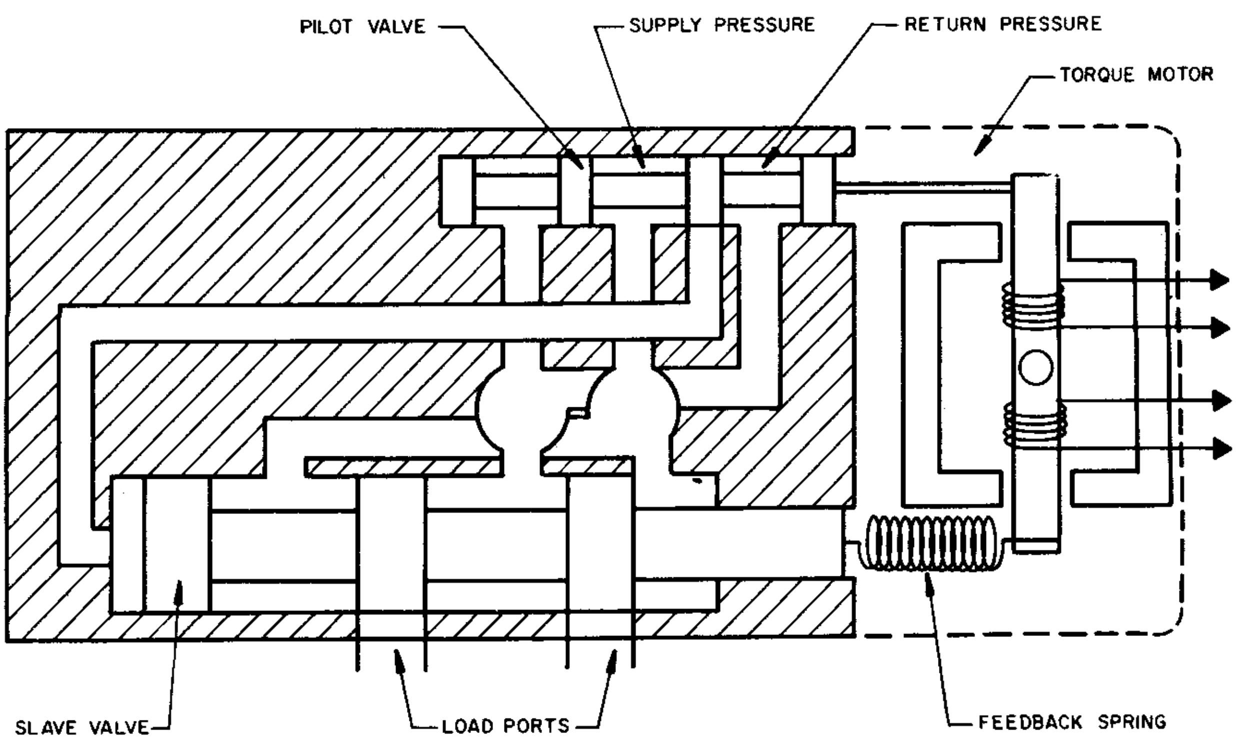

Figure 2 Lear Two-Stage Servo Valve Schematic

This input current to the torque motor develops a proportional torque which moves the first-stage spool. The flow from the latter imparts a velocity to the output spool which deflects the feedback spring, producing a torque on the motor opposite in phase to the input torque of the differential current. When the feedback torque equals the input torque, the first stage spool is centered and the output spool comes to rest at a displacement proportional to the torque motor input current.

The valve spools and seals are made of hardened stainless steel. The torque motor is completely isolated from the oil environment. According to the manufacturer, its torque level is sufficiently high that the first-stage spool is not greatly affected by oil contaminents, and 10 micron filtration is adequate. The valve is supplied with either a pigtail electrical connector or a Viking VR4/2AB1 connector.

4. Midwestern Instrument Model 7

The Midwestern Model 7 hydraulic servo valve (see Fig. 3) is a single-stage four-way valve that utilizes a rotary flat plate with hole and plug porting. (For a detailed description of the construction of hole and plug porting see TR 55-29, Part 1). A Midwestern Model 9 torque motor controls the plate movement.

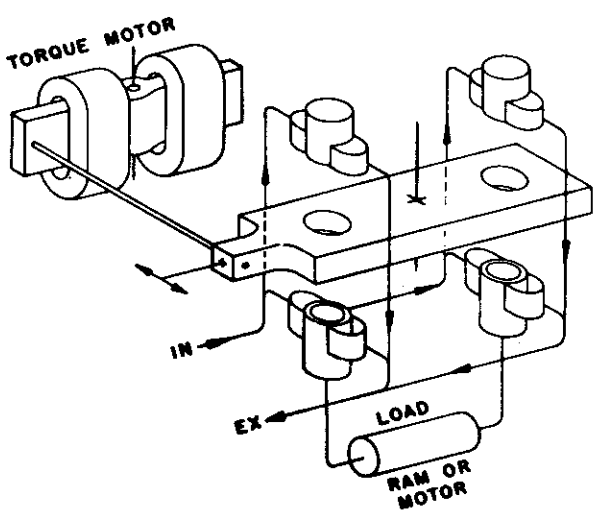

Figure 3 Schematic Diagram Model 7 Servo Valve

When the differential current to the torque motor is zero, the armature and the rotating valve plate are centered so that the two holes in the plate line up with the two porting bushings, and the milled cavities leading to supply and drain are blocked by the plate. The only flow through the valve is leakage flow and the pressure at both load ports is the same. When a differential current is applied, the armature is displaced causing the plate to rotate. Each hole in the plate is now partially over the hole in the adjacent porting bushing and partially in line with the milled cavities, providing a fluid path from the milled cavities to the load ports. Thus a differential pressure is applied across the two load ports.

Flow force compensation can be incorporated by adding baffles inside the bushings. The valve is made of hardened steel alloy throughout. The torque motor is not isolated from the drain oil.

5. Bendix Pacific HR Series

This is a two-stage servo valve using a double nozzle-flapper first stage that drives a spring restrained spool output stage. Its principle of operation is similar to other flapper-nozzle valves with spring restrained output stages. The construction of the first stage is considerably different from most other types, however, (see Fig. 4). In this valve the two nozzles are directed in the same direction rather than toward each other and the flapper is positioned to float above them. This construction permits the flapper to move a greater distance from the nozzles than in the other designs, allowing larger dirt particles to be washed to drain.

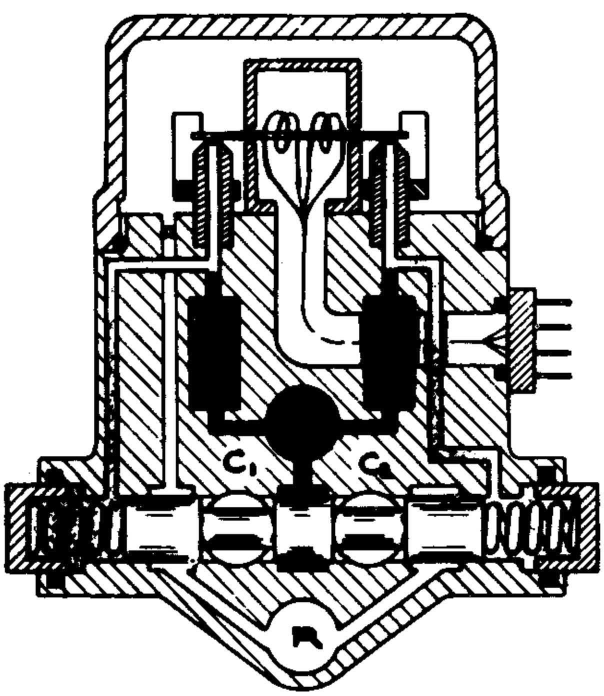

Figure 4 Bendix Pacific HR Series Servo Valve Schematic

Additional protection against dirt clogging is provided by a 5 micron filter ahead of each nozzle. The valve has an all-steel body with hardened steel inserts. The torque motor is not isolated from the drain oil. Future versions will incorporate a stale torque motor which permits oil in the torque motor chamber but does not allow the drain oil to circulate through it.

6. Dalmo Victor Model 10 Series

The Dalmo Victor Model 10 electrohydraulic servo valve is a two-stage four-way valve. The first stage utilizes a flapper-nozzle type construction and the second stage is a spool sleeve construction.

This valve's operation and construction are very similar to the Moog series 500 valves, except for the way the output stage is manufactured.

The valve body is machined from high tensile strength aluminum. The spool and sleeve are fabricated from high grade bearing steel, hardened and cycled for purposes of stabilization. The valve sleeve is constructed of five segments held in precise alignment with a retaining sleeve. Rectangular flow ports are produced by grinding radially across the segment face. Filtering is accomplished by use of 10 micron sintered metal filters and magnetic traps. The torque motor is not isolated from the drain oil. Electrical connection is made through a Winchester Connector.

7. Sanders Associates SA17D

This is a two-stage four-way valve employing a dual nozzle-baffle first stage driving a spool type output stage (see Fig. 5). The

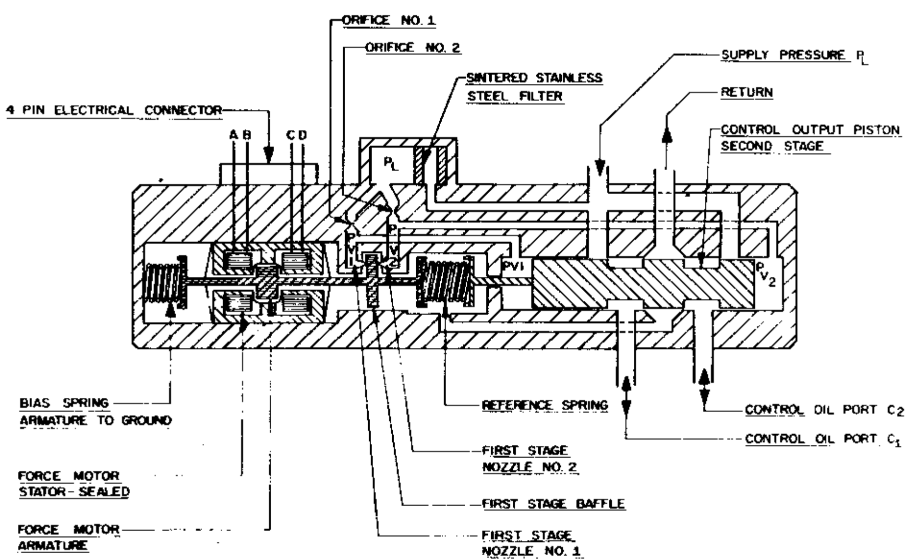

Figure 5 Sanders Associates SA17D Servo Valve Schematic

baffle is stroked by a force motor which is located in line with the baffle and the output spool. The position of the output stage is fed back as a force to the force motor via a reference spring located between the output spool and the baffle. When a differential current is applied to the force motor, the baffle is moved with respect to the twin nozzles, creating a pressure drop across them. Thus, differential pressure is applied across the output spool which moves until the force of the reference feedback spring balances out the combined force due to the differential current and bias spring.

This is an all-steel valve, the entire body of which is constructed of segments which are brazed together. Prior to the brazing operation each of the segments is machined to provide the necessary porting. The SA17D contains two sintered stainless steel filters located before each nozzle and orifice of the first stage; these are readily accessible for change. The force motor is completely isolated from the oil. Sanders also manufactures large flow valves of similar design to that described in TR 55-29 Part 1. These valves are capable of providing flows up to 200 gpm.

8. Moog 1800 Servo Valve

The Moog 1800 servo valve is a two-stage valve with a double nozzle-flapper first stage driving a spring restrained output stage and is similar from a functional standpoint to the earlier Moog valves (series 500, 900, and 1400) (see schematic of Fig. 6).

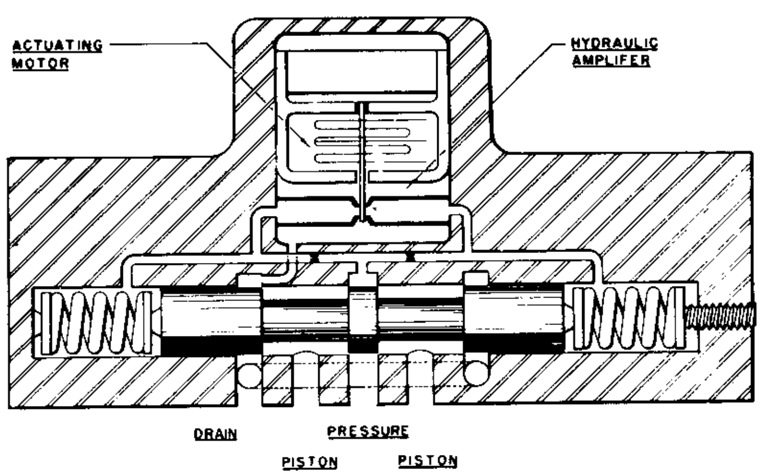

Figure 6 Moog 1800 Servo Valve Schematic

The distinguishing feature of the Series 1800 is that it is designed for operation under oil temperatures up to 400°F. All-steel construction is employed with a steel sleeve pressed in the steel body. It has been designed to operate with OS-45-1 synthetic fluid.

9. Cadillac Gage FC-200

The Cadillac Gage FC-200 series servo valve uses a dual nozzle-flapper first stage which drives a four-way spool second stage. The valve contains no external null adjustments. It utilizes dry torque motor construction (see Fig. 7).

In operation the valve uses a hydraulic followup system. For example, if a differential current is applied to the torque motor so that the flapper is forced to move closer to nozzle No. 1, the flow from that nozzle is restricted while the flow from nozzle No. 2 is increased. This results in a pressure differential across the spool. The spool moves in the direction of the lower pressure. As the spool moves, it reduces the area of orifice No. 1 and increases the area of orifice No. 2. The spool continues to move until the percentage change in area of the inlet orifices equals the initial percentage change in area of the outlet nozzles. At this point the differential pressure across the spool is again zero and the spool is at equilibrium in its new position.

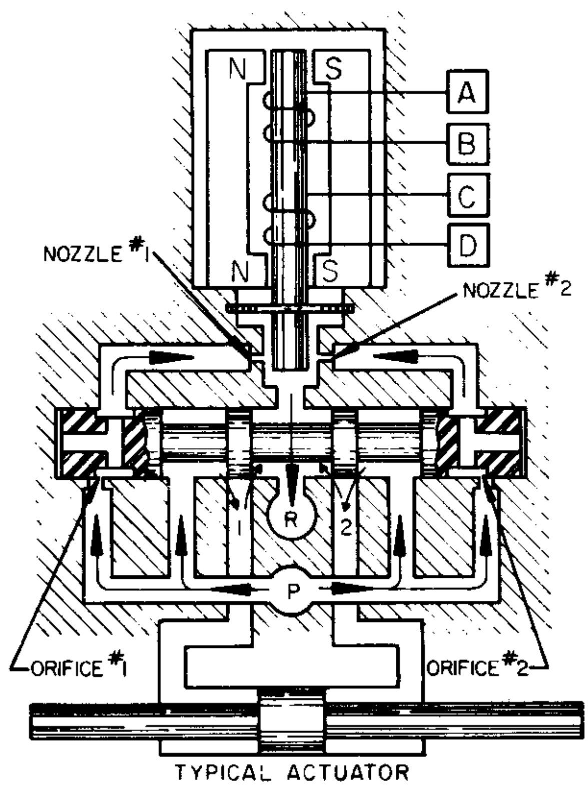

Figure 7 Cadillac Gage FC-200 Servo Valve Schematic

The valve employs a hardened steel sleeve floated on 0-rings, in an aluminum body. The first stage uses 40 micron sintered bronze and magnetic filters. The valves use dry torque motor construction and are designed to operate at temperatures up to 300°F.

10. Bendix 241

The Bendix 241 is a two-stage servo valve utilizing spools for both stages and electrical feedback for stabilization. This valve is rather large, having been designed for flight table application and not intended for aircraft application. However, because it has a few unique features which may also be applicable to aircraft valves, it is included in this report.

The output stage is designed around the "convergent flow" principle. In this four-way design (see Fig. 8) the porting is arranged such that the flow through the orifices always reacts against the valve body; that is, the flow leaving the orifices is directed toward the body at both the supply and drain side orifices. As discussed later, this technique provides added dynamic stability to the output stage, eliminating the necessity of an orifice between spool ends for damping.

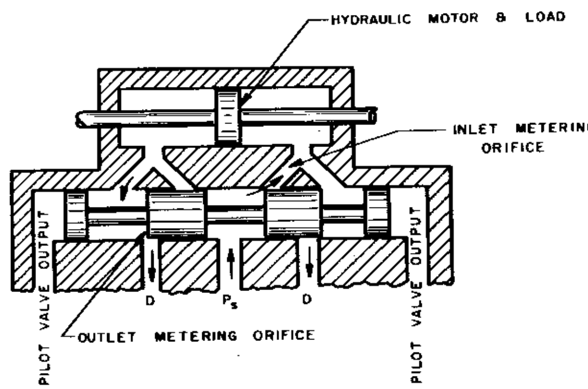

Figure 8 Bendix No. 241 Valve Output Stage

The first stage utilizes tapered lands to reduce the friction level. A scavenging groove subjected to a negative pressure is employed between the spool end and metering orifices of the first stages to reduce the leakage of oil to the torque motor chamber. The valve is of all-steel construction.

Bendix Research is also developing a nozzle-flapper type valve which will be suitable for aircraft application. This valve is in an early stage of development; its design characteristics are listed in Table 1.

11. Robertshaw Fulton Valve

This is a two-stage flapper-nozzle with a spring restrained output stage and operates in a manner similar to other valves of the same general type. However, it has several distinguishing design features. First the valve employs a "stale coil" torque motor.

Although the torque motor is immersed in oil, there is no oil flow through the magnetic circuit. A stainless steel diaphragm acts as the flapper pivot and as a fluid barrier. The section of the flapper that lies between the two nozzles is nonmagnetic; thus, there is no magnetic flux through the hydraulic circuit.

The output spool sleeve is constructed of five steel inserts pressed into another sleeve floated in 0 rings in an aluminum body. The segments are not brazed. Peripheral porting is employed at the sleeve segments.

12. Hydraulic Research Model 20, 21 Valves

This is a double nozzle-flapper valve with a spring restrained output stage (see Fig. 9). It is similar to the earlier Bell Aircraft valves except for the technique used to isolate the torque motor coils from the drain oil. Stainless steel is used throughout the valve except, of course, in the flux carrying parts.

13. Pegasus Model 20 Aircraft Servo Valve

This is a two-stage valve in which the first stage nozzles are attached to the ends of the output spool (see Fig. 10).

Two baffles mounted on a shaft extending through the spool and nozzles are located adjacent to the nozzles to perform the valving action. The porting between the first and second stage is arranged so that when the shaft connecting the two baffles is moved in one direction, thereby enlarging one nozzle orifice and decreasing the other, a pressure differential is produced to move the output spool a corresponding amount. Thus the

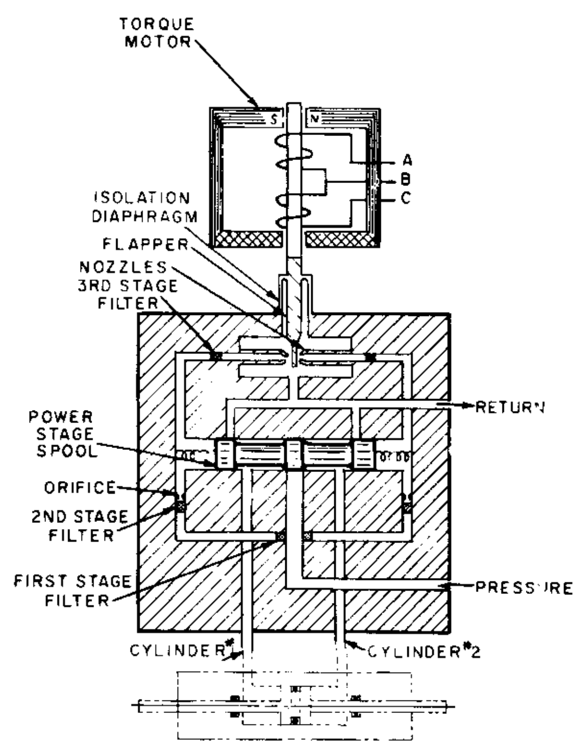

Figure 9 Hydraulic Research Mod. 20 & 21 Schematic

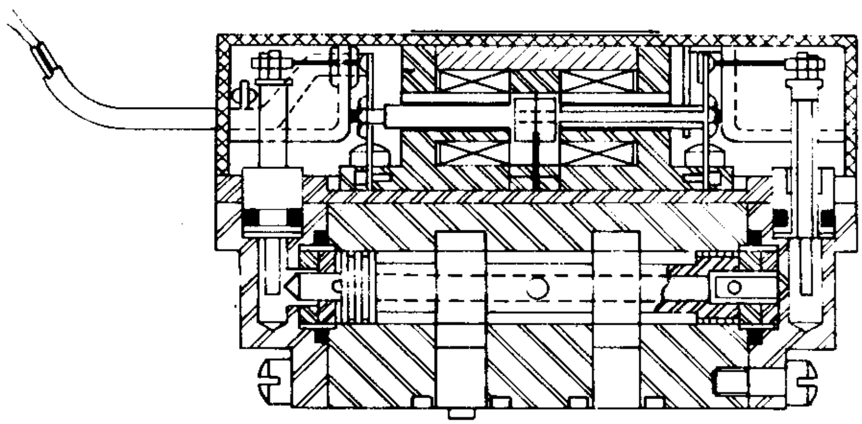

Figure 10 Pegasus Servo Valve Schematic

output spool always has the same absolute displacement as the baffle shaft.

The baffle shaft is stroked by a force motor or solenoid through drive arms, which in combination with the force motor linkage provide a mechanical advantage between the baffle shaft and the force motor displacement to reduce the displacement of the force motor.

The force motor is isolated from drain oil by the sealed points of the drive arm. These are of integral stainless steel construction with no braze points to reduce the effects of extreme temperature variations.

Triple filtration is employed. Two filters are located before the fixed orifice and one before the nozzle. The porting is arranged such that the flow to the first stage is topped off at an angle of 90 degrees to the flow to the output stage, thereby utilizing the inertia of the dirt particles to discourage them from entering the first stage.

14. Hagan Corporation Electric Hydraulic Pilot Valve Model 80

This is a two-stage valve employing spool and sleeve type construction in both the first and second stage. The unique feature of this valve is that it uses a spinning first stage (see Fig. 11). It is however, too large for aircraft use.

C. Discussion of Manufacturing Problems and Valve Design

A number of the problems which the valve manufacturers faced and solved during the infancy of the servo valve industry were discussed in WADC TR 55-29, Part 1. Since that time, numerous other problems have arisen and some of these will be discussed here.

1. Null Shift

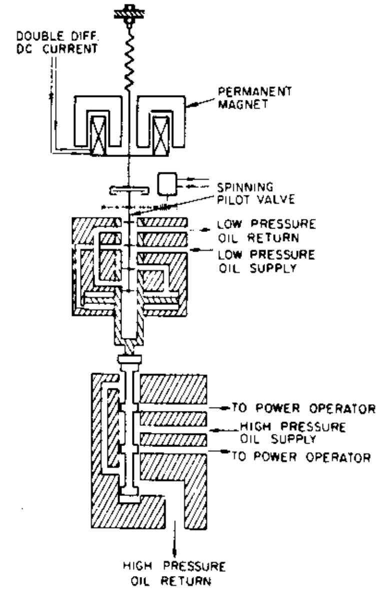

Figure 11 Hagan Mod 80 Servo Valve Schematic

One of the most difficult problems valve manufacturers have had to contend with, and are still seeking to solve, has been the null shift problem. This problem has been quite serious in the nozzle-flapper valves, particularly the open loop or spring restrained type. Most servo valves are provided with a null centering adjustment for "recentering" the valve to correct for this null shift. Unfortunately, the centering adjustment operates on the output stage spool and spring assembly, which rarely, if ever, is the source of the null shift. Therefore, while this adjustment provides a means of synchronizing the flow null and the differential current nulls, it does nothing to actually relieve the internal unbalance and can only be carried so far before the operation of the valve becomes nonlinear. Another factor of even more importance is that if the null shift occurs during a missile flight, it cannot be recentered. Small null shifts can, of course, be compensated for by the use of sufficient loop gain.

The major source of null shift appears to occur in the torque motor and flapper. Enginners at Boeing Aircraft Co. have studied this problem on an analog computer and have concluded that the major null shift is not the result of changes in the nozzle and orifice coefficients, but rather, it is primarily the result of changes in the torque motor and the flapper. This is borne out by investigations carried out by other prople visited during the survey.

The major contributing sources of null shift in the torque motor appear to be the magnetic core material and the permanent magnet. The materials used are sensitive to changes in temperature and begin to lose their magnetic properties at temperatures above 500°F. The core materials are also predominant contributors to the hysteresis of the valve. At present Alnico magnets and Hypernik core material are most commonly used in the servo valve electromagnetic driver.

During manufacture, the technique commonly used for centering the first stage of the valve is to bend the top of the flapper structure, a procedure apparently first initiated by Moog Valve Co. This procedure, called Moogizing by one organization, causes stresses in the flappers which are released either at higher temperatures or over a period of time, causing flapper displacements and thus null shifts. One technique presently used to circumvent this problem is to accelerate this process by boiling the valves in MIL-0-5606 for a period of several hours. Thus, the stresses are relieved, and the attendant flapper movement is compensated for by recentering the output stage. The procedure does not remedy the difficulty, but merely provides a means of coping with it. Engineers at G. L. Martin have found that the gain compensated valves are more susceptible to null shifts than uncompensated valves.

2. Dirt

The present design trend appears to be moving toward the use of dry torque motor valves to circumvent the magnetic dirt contamination problem. In these valves the flapper is generally fastened in some way to a diaphragm, or insulating disk, which seals the drain oil from the torque motor chamber. Because this diaphragm has a different environment on each side, it is very susceptible to environmental changes. Any movement of the diaphragm causes corresponding movements of the flappers, resulting in valve null shift. Recently a number of companies have brought out a stale torque motor valve which permits oil to exist in the torque motor chamber but does not allow oil to circulate through it. Thus the initial oil remains in the chamber and the contaminants cannot enter. This design appears to be less susceptible to pressure and temperature changes than the purely dry torque motor type but we will have to wait for experimental evidence to bear this out. In another design, a negative pressure source is provided between the first stage and the torque motor thus drawing off the drain oil before it gets to the torque motor chamber.

3. Erosion

Another problem encountered with the first stage of the valve has been erosion, both in the nozzles, and the flapper surface. The erosion is caused by minute contaminants in the oil, traveling at high velocities, and is responsible for changing the orifice coefficients. To circumvent this problem most manufacturers construct the nozzles of beryllium copper and the flapper of hardened steel.

4. Construction

Normally in the construction of the valve, the sleeve for the output spool is made up of a number of segments or washers which are brazed together and then the central bore is lapped to the correct diametrical size. The spool is then ground to fit the axial dimensions of the sleeve, and the entire assembly is fitted into the valve body with "O" rings. The ports in the sleeve may be either peripheral slots or circumferential annuli. Experience has shown that the segmented sleeve is simpler to construct than a solid sleeve. However, because of the considerable length of time required to fit the spool to the varying axial dimension of the sleeve, efforts are continually being made to find a simpler way.

One technique presently being used by Dalmo Victor is to use a segmented sleeve without brazing, the segments merely being pressed together to prevent leakage and movement. In this construction, the sleeve is made up of five segments which are lapped to an axial dimension of 50 μin. Three of these segments are slotted to form ports. The five segments are then arranged in the proper order and orientation and arbor pressed into the valve body. The center bore is then ground and diamond honed to the necessary radial tolerances. The spool, which can now be machined to predetermined axial and radial dimensions, is then placed in the sleeve and centered with the restraining springs. The Robertshaw Fulton valve also utilizes a similar type of construction.

Boeing is using a technique where the valve sleeve and body are each made of five segments. The axial dimensions of the segments are maintained at the same value by lapping all the segments at one time, or at least, by lapping adjacent sleeve and spool segments at the same time. Previous to this, four axial slots, spaced equally apart radially, are ground into those spool segments which will be located across the portless segments of the sleeve. The sleeve ports are circumferential. The sleeve and spool segments are then arranged in the proper order and brazed together; no more grinding is done on the axial dimensions of either the sleeve or spool. A problem has been encountered with excessive underlaps or overlaps and research is presently being directed toward a better bonding alloy to reduce growth during the brazing operation.

Sanders Associates employ a building block design in which the entire valve body is built out of a number of stainless steel blocks. All internal porting is performed on the axial ends of each block and then the entire assembly is brazed together. This eliminates the necessity of a separate sleeve with the valve body.

It is interesting to note that only one manufacturer is using the hole and plug porting method developed at M. I. T. The company is Midwestern Instruments. They have built a rotary flat plate valve which apparently has operated satisfactorily. To our knowledge, no one as yet, outside of M. I. T., has built a spool plug and hole valve which appears at first glance to be somewhat easier to construct than the flat plate version. However, engineers at Minneapolis-Honeywell have attempted to build this valve and found it very difficult to drill the holes through the valve body and spool and maintain sharp edges on the metering orifices. Burrs accumulate on these edges and cannot be removed without undercutting the spool, or enlarging the bore. As a result, leakage is difficult to control.

5. Torque Motor

In the design of the servo valve torque motor, the valve manufacturer is confronted with the necessity of providing a wide range of differential current inputs to satisfy users' requirements. To accomplish this he may vary the number of turns in the torque motor, the width of the air gaps, the bias flux created by the permanent magnet, etc. The latter particularly provides an effective way of varying the differential current requirements. In fact, the differential current input can be theoretically reduced to a very low value by increasing the size of the permanent magnet, since the force output of the torque motor is proportional to the product of the permanent magnetic and differential current produced fluxes. However, if the differential current is reduced to too low a value, the output force becomes too dependent on the characteristics of the permanent magnets, which may be quite temperature sensitive, thus presenting a serious null shift problem. There probably is an optimum compromise between minimum controlling power and the susceptibility to external disturbances creating null shift.

6. Output Stage Oscillations

Another problem that has plagued valve manufacturers in the early stages of servo valve design, is instability, or oscillation (squeal) of the output stage. This phenomenon may occur as a result of a number of different factors, and may in fact, be induced by conditions external to the valve such as acoustic resonance 1/ of the oil columns in the supply and drain lines leading immediately to the valve.

Other factors 2/ hinge on the relative directions of motion of the valve spool and the oil flow through the metering orifices, and the axial distance between the incoming and outgoing flow at each orifice. In a conventional valve spool-porting arrangement, containing two orifices, or two pairs of two orifices, negative damping forces are developed at one orifice and positive damping forces at the other orifice. The magnitudes of the damping forces are dependent on the axial distances between the centers of incoming and outgoing flows at each orifice. Generally, the valve spool and sleeve are designed such that the axial lengths are greater at the orifices providing positive damping, so that the resultant damping due to both orifices is positive. Even when this is done, the valve has a tendency to oscillate because of the limited range of control, and to overcome this, many manufacturers provide additional damping of the output spool by bypassing the spool ends with an orifice. This procedure of course, reduces the bandwidth of the valve but the reduction is of negligible importance as far as most missile and aircraft applications are concerned.

Bendix Research Division has designed their valve output stage porting such that the direction of motion of the valve spool is always in the same direction as the flow from the metering orifices,

1/ Ainsworth, Frank "Effect of Oil Column Acoustic Resonance on Hydraulic Valve Squeal, ASME Transactions, Vol. 78

2/ Lee, S. Y. and Blackburn, L. "Transient-Flow Forces and Valve Instability", ASME Transactions, Vol. 74

thus providing positive damping at both orifices (see schematic of Bendix valve). According to their engineers, this design has obviated the necessity of further damping of the valve spool and their valve has a wide bandwidth.

7. High Temperatures

High temperature application presents the greatest problem to the valve manufacturers at present. The null shift problems created by temperature changes have already been discussed as have some of the problems with magnetic strength changes.

At least one valve, the Moog 1800, is presently in production for operation at oil temperatures up to 400°F. These high temperatures require valves of all steel construction and with greater clearances and tolerances. At still higher temperatures wire insulation deterioration becomes a serious problem. Aluminum oxide is presently being used for higher temperature application but is not completely satisfactory.

CHAPTER III

COMPILATION OF DATA OBTAINED FROM VALVE USERS

The objective of this phase of the survey was to obtain as much information as possible on the complete servo valve-actuator loop employed in various missile and aircraft control systems, primarily the former. For instance, information was requested on the amplifier, any compensation networks employed, the servo valves, feedback components, actuators, and the loads acting on the actuator. In addition, because this system is influenced to an extent by the supply source, some data were also obtained on the type of supply system, filtration, etc. No effort was made to cover all systems on a particular missile or group of missiles. Instead, every effort was made to obtain fairly complete information on the most representative control system on a particular missile while limited information was also obtained on some of the adjacent systems where possible.

Mose of the data obtained from these visits have been categorized and tabulated in Tables 1, 2, and 3. Table 1 lists valve characteristics, Table 2 lists general system characteristics, and Table 3 lists servo loop design data on typical control systems. For the most part, the information contained in Tables 2 and 3 was not covered in the previous survey. However, some of the information contained in the previous survey report has been repeated here for purposes of completing the information presented and to include important changes and additions. Information of the system's dynamic requirements and desired valve life was not included because the previous report adequately covered these requirements; at least the information is up to date and very little could be added.

The following discussion summarizes the information contained in Tables 1, 2, and 3 and, in addition, presents certain data which could not be easily tabulated.

A. System and Valve Data

1. Supply Pressure, Flow Characteristics

a. Pressure

Most missile and aircraft systems are operating with

TABLE 2 - GENERAL SYSTEM CHARACTERISTICS

System Requirements				Valve Characteristics						
Valve User	System	Supply Source	Supply Pressure psi	Valve Amplifier	Filtration Microns	Make and Model	Differential Current ma	Rated Flow gpm	Max Leakage gpm	Oil Temperature Range °F
Bell Aircraft	A, B, C,	3600 rpm, 9 piston pump with accumulator	3000 + 0 - 400	tubes	10 Paper	Bell SV 3B Bell SV 5C Bell SV 11 C	10 10 10	20 12 2	0.4	-65 to 200
Martin Aircraft	D, E	New York airbrake engine driven pump with no accumulator	1500 + 0 - 200	tubes (experimenting with 3200 cps magamps)	2-5 Purolator	Moog 512	8	3.25	0.37	-65 to 200
	F	pump driven at two speeds of turbo jet engine 2610 and 4350 rpm	1500 + 0 - 200	tubes (experimenting with 3200 cps magamps)	2-5 Purolator	Moog 568	8	3.25		-65 to 275
	G	constant displacement pump and 50 in.^{3} accumulator	2000	tubes	2-5 Purolator and Magnetic Filter	Moog 907 512	8 8	1.75 3.25	0.25 0.37	-65 to 160
	H, I	first stage - engine driven peroxide turbine pump; second stage - electric motor driven pump	1500	tubes	2-5 Purolator	Moog	8 8	2.6 0.8		-65 to 120
Lockheed	J, K	Vickers or New York airbrake pump with accumulator	1500 + 0 - 200	tubes push, pull	10 Perm. Filter ahead of each valve	Moog 2000 Bendix Pacific Cadillac FC-2	8.0 8.0 8.0	3.0 1.0	0.5	-65 to 275
Douglas	L, M	200 in.^{3} accumulator powered	2000 3000	tubes ac pre-amplifier	10 Purolator Sintered bronze	Bendix Nike Bendix Nike	18.0 18.0	3.0 3.7	0.08	-65 to 200
Northrop	N	pump + 75 in.^{3} accumulator	1000 3000	all dc, tubes coils series connected	2-5 Purolator Sintered Bronze	Moog 561	8.0	0.5 to 2.0	0.1	-65 to 170
NAA	O	ethylene oxide turbine driven 5 gpm pressure compensated pump and 200 in.^{3} accumulator	3000	tubes, push, pull, ac pre-amplifier stage complete transistor being developed	2-5, 10 Porous Media Purolator Puroiloy Permanent Filter	*Moog 1800 2000 *Cadillac Gage FC-2 Bendix Pacific HR Sanders SA 17 Nat'l Water Lift Dalmo Victor Robert Shaw Fulton	5.0 to 8.0	0.5 to 12.0	0.35	-65 to 275 -65 to 400 -65 to 600
Convair San-Diego	Aircraft	3.5 gpm pump with 50 in.^{3} accumulator	3000 + 500 - 200		10 No magnetics	Moog 900 and 2000	8.0	1.5		-65 to 275
	P	Vickers variable volume pump (1.519 in.^{3}/rev) driven at 3800 rpm by gas turbine 90 in.^{3} accumulator	3000	Single ended tube with coils in series, ac transistor preamplifier	10 Porous Media Woven Filter	Moog	4.0	1.5 to 10 Average = 8.0		-65 to 275
Convair Pomona	Q	Constant volume pump driven by air motor from storage bottle air pressure regulated high and low pressure accumulator	1500 + 300 - 200 1800 + 400 -100	Single ended tube driver with coils driven differentially on dc	10 Permanent Filter	Moog 902 A 2009 Bendix HR HR and M	6.0	0.9 to 3.0	0.25	-40 to 285
Boeing	R, S, T, U, V	14 gpm pressure compensated pump plus 45 hp-sec accumulator battery supplied electric motor driven	3000	Magnetic amplifier driving coils in series	2-5 Filters and Magnetics	Cadillac Gage FC-2 PC-2	10	2.3 4.7 8.8	0.2 to 0.4	-65 to 290

* in actual use other values are proposed experimental models

System	Nominal Supply Pressure psi	Max. Actuator Velocity in/sec	Inertia (Load) lb-in-sec²	Aero Dynamic Hinge Moment in-lb	Control Surface Travel degrees	Aero Dynamic Spring K in-lb/rad	Actuator Area in²	Total Actuator Stroke in	Torque Arm in	fn1 Mass & Aero cps	fn2 Mass & Oil cps	fn3 Structure & Oil cps	Loop Gain 1/sec	Friction lbs
R	* 3000	o 7.57	+ 52.6	* 54×10^3	* 25 Total	o 124000	* 4.12	* 3.00	+ 6.73	o 7.70	o 17.3	* 30	* 45	* Negligible
S	* 3000	o 476 rad/sec	o 90	none	not limit- ed	none	* Vickers 3906		* Gear Ratio 70/1		o 37.3		* 30	
W			+ 533			+ 1.08×10^6				o 7.2	+ 9.6			
X			+ 494			+ 1.07×10^6				o 7.4	+ 11.2			
H	* 1500	o 4.76	o 225.6	none	* ±5	none	* 1.48h 1.24r	1.75	* 9.1		o 188		o 50	o 658
I	* 1500	o 2.1	o 230	none	* ±3	none	o 1.47	o 0.628	* 6.0		o 64.8	o 50 or greater		* 340
D	* 1500	o 9.8	o 78.2	+ 8000	o ±31.8°	o 14.4×10^3	* 1.35h 1.20r	* 4.562	+ 18.1	o 2.16	o 179		o 137	
E	* 1500	o 4.15	+ 5.8				* 3.14h 2.89r	* 2.0						+ 480
G	* 2000	o 2.96					* 2.40h 2.15r	* 1.75						
L	* 2000	o 37.2	o 0.259	* 500	* ±15	o 1910	* 0.276	o 1.04	* 2.0	o 13.6	o 93		* 60 to 863	o 13.8
J	* 1500	o 9.3	o 1.6	* 14,400	* +15 -10	o 54.8×10^3	* 1.42r 1.62h	* 3.83	* 8.87	o 29.3	o 696		o 49.3	
K	* 1500	o 2.82	o 0.074	* 2600 per aileron	* ±22.9	o 6500 per aileron	* 1.67	* 1.5	* 1.88	o 47	o 1160		o 70.2	
O	* 2800	o 7.9	o 10.7	* 35.2×10^3 on 2 Actuators	* ±20°	o 1.01×10^5	* 2 Actuators 1.62 ea (3.24)	* 2.48	* 3.62	o 15.4	o 135	* 4-5 Structure alone		o 272
P	* 2700	o 10.4	o 4680	* 1.09×10^6 (Not Aero- Dynamic) coul frict.	* ±4		* 3.6	o 2.94	* 21		o 54.5	* 6 Structure alone	* 25	o 1430
Q	* 1500	o 3.55	o 0.0375	+ 654	* ±20	o 1880	* 1.188h 0.875r	* 1.5	o 2.2	o 35.6	o 1490	* 36 to 500 as a function of altitude		
T	* 3000	* 1.44 rad/sec (Load vel)	o 266	none		none	* 2.35	* 3.02			* 22.5	* 24		
U	* 3000	* 975 rad/sec	+ 870.68	* 45000			* Vickers 3909		+ Gear Ratio 740/1		o 249	* 41	* 45	
V	* 3000	o 1030 rad/sec	+ 302.38	* 20,000	* ±15	o 76,300	* Vickers 3906		+ Gear Ratio 770/1	o 2.2	o 223	* 58	* 45	
Y	* 3000	o 38.5	o 0.172	* 1000	* ±30°	o 1910	* 0.4	* 0.75	* 1.5	o 16.8	o 297	* Linkage & Structure <170	o 1240	o 60
A	* 3000		o 2.94	* 5000	* ±33°	o 8680	* 2.76h 2.45r	* 3.72	* 3.08	o 8.63	o 231		o 85	
B	* 3000	o 13.9	o 6.11	* 55000	* +22° -33°	o 95.5×10^3	* 5.94h 5.15r	* 6.18	* 6.00	o 19.9	o 355		o 184	
C	* 3000	o 8.3	o 17.4	* 75000	* ±17°	o 251×10^3	* 5.94h 5.15r	* 3.56	* 6.12	o 19.1	o 282		o 123	
Z	+ 3000	+ 11 11 38 13	Negligible	Driver Control Stick		none	+ 0.1 0.1 0.292 0.083			none		none	Negligible	
N	+ 3000	o 2.29	Negligible	o 134	+ ±1.12°	o 5850	+ 0.791	+ 0.360	o 0.993				o 16.7	

CODE:
* Data Obtained on Visits
o Data Computed
+ Data Obtained from Publications or Letters

TABLE 3 - TABLE OF CONTROL SYSTEM CHARACTERISTICS

supply pressures of 3000 psi, although a number are operating over the range of 1500 to 2000 psi. There has been considerable talk about going to higher pressures in the last two years, but the idea appears to be losing its appeal. A number of studies on this subject have been made, 1/ and the optimum pressure has been found to occur around 4000 psi from a size, weight, and strength standpoint. However, the percentage improvement is rather small and is offset to a considerable extent by the expense of developing new equipment to handle the higher pressures and converting existing equipment.

As mentioned previously, a number of people prefer to operate under 3000 psi, where possible, to reduce wear and leakage problems with servo valves and to reduce the stresses and wear on seals, 0 rings, and fittings. In fact, as will be described later, one group of missile designers bases its supply pressure requirements on the bandwidth requirements of the system.

b. Flow

The flow ratings of the valves employed in missile application vary with the size and type of missile. Generally, for vehicle control application, the flow ranges of the valves may be loosely classified as follows:

(1) Large missiles - 4 to 10 gpm

(2) Medium size missiles - 2 to 4 gpm

(3) Small anti-aircraft - up to 3 gpm.

Most autopilot and antenna stabilization applications employ valves with flow ranges less than 2 gpm, excluding dual input servo valves, which were not investigated. It is interesting to note that servo valves with maximum flows of between 40 and 160 gpm are currently being used in certain

1/ Cooke, Gessner and Smith - Theoretical Investigation of Optimum Pressures in Aircraft Hydraulic System, TR 54-189, Glenn L. Martin

Naval and industrial applications. It appears that these valves might be suitable for missile fuel control applications.

c. Leakage

The leakage requirements also vary with each application but generally are not critical except in the case of small antiaircraft missiles with outboard drains. In several of the latter applications the leakage specifications are sufficiently tight that two-stage flapper nozzle valves could not be used.

d. Supply

Most missile hydraulic supply systems are of the closed type employing pumps and accumulators. The capacities of the supply pumps are generally a fraction of the maximum output capabilities of all the servo valves combined, the accumulator handling the peak flows. Apparently, instances where all servo valves are drawing peak flow are practically nonexistent or the peak flow durations are very short. Most small antiaircraft missiles employ outboard drains with accumulator supplies.

It is interesting to note that most supply pumps are driven by internal sources such as air motors, ethylene oxide or peroxide turbines, battery powered motors, etc. Ram air driven turbines have not been used extensively because of the high temperature problems accompanying the use of ram air.

Supply and drain oil is generally conducted through stainless steel tubing, coupled by both AN and MS flareless fittings. One company, Convair, Pomona Division, is utilizing a single manifold block containing the supply accumulator to which all valves are manifolded, thus completely eliminating oil lines. This is done for reasons of economy of manufacture rather than for improving response.

2. Oil, Filtration, Seals, Etc.

Nearly all current missile and aircraft hydraulic control systems are operating under temperatures of 275°F. Only one company visited, North American Aviation, was operating at temperatures above this figure, although it is known that other companies have future requirements above this temperature. For instance, assuming a standard hot day, the boundary layer temperature rise of an aircraft flying at Mach 2 is 304°F at an altitude of 39,000 feet, 1/ and 404°F at sea level. The temperature increases approximately as the square of the Mach number ratio. A considerable portion of this temperature rise will reach the hydraulic system during long flights unless cooling is provided.

In missile application it is, of course, advantageous to operate the control systems at high temperatures rather than add bulky heavy cooling equipment to reduce the temperatures. The main problem then is in designing and developing equipment and oils to operate at high temperatures.

a. Below 275°F

The development of equipment and oils for operation up to 275°F has been accomplished. MIL-0-5606 oil operates very satisfactorily under temperatures of 275°F. The main problem was to develop seals and 0 rings for operation in MIL-0-5606 oil under these temperatures, and elastomeric seals such as MS28784 and MS28785 are adequate. Fiber and phenolic filters will operate up to temperatures of 275°F although most manufacturers are reverting to metallic filters, such as sintered stainless steel or bronze, wire woven stainless steel, etc., in an effort to circumvent the increased tendency of the paper filters to decompose and loose their fibers. The decomposition of the filters of course, is not only a problem with paper filters. All sintered filters have the problem of particles becoming loose from the mesh and thus reaching the critical points of the valves and pumps.

Another problem with all filters presently used and designed for missile and aircraft use, is that they apparently do not live up to their ratings. One prominent engineer

1/ Keller, G. R., - High Temperature Average Hydraulics, NAA.

stated facetiously that the main difference between a 2 to 5 micron filter and a 10 micron filter is that the former passes less 100 micron particles. This appears to be a problem with all filters currently in use and filter inadequency is a common complaint of all the missile manufacturers visited.

Most companies do not employ magnetic filters in their systems at temperatures below 275°F; they find that the mechanical filters remove most of the magnetic particles also. One notable exception is Boeing, where even with magnetic filtration, magnetic particles have been found to accumulate in the valve torque motor.

b. Above 275°F

At temperatures above 275°F these problems get much worse. Now the MIL-0-5606 oil becomes a limitation. While MIL-0-5606 has been operated successfully up to temperatures approaching 500°F in completely closed systems devoid of air contamination, they are highly flammable at these temperatures and a completely airless requirement is not practical.

The oils showing most promise at temperatures above 275°F are the silicate esters such as MIL-0-8200, MIL-0-8515 (same as the former but with a plastisizer added) and OS-45, OS-45-1 (latter contains additive to elevate vapor phase oxidation temperature). Of these, OS-45-1 appears to be superior for operation at temperatures up to 400°F, primarily because it forms less varnish deposits, and it is presently being used in North American high temperature systems. This oil is also being employed in test setups operating at ambient temperatures of 1150°F and oil temperatures of 600°F under sea level pressures. Table 4 lists some of the characteristics of the high temperature oils, in addition to MIL-0-5606. These data were obtained from ASME papers 56-AV-21 1/ and 56-AV-22 2/.

1/ Litzsinger, H. and Hatton, R. - The Technical Appraisal of the Performance of OS-45 and/or OS-45-1 High Temperature Hydraulic Fluids.

2/ Furby, N., Peeler, R. and Stirton, R. - Oronite High Temperature Hydraulic Fluids 8200 and 8515.

TABLE 4 - CHARACTERISTICS OF HYDRAULIC FLUIDS

Base	Petroleum Base	Disiloxane		Ortho-silicate 0S45
Fluid	MIL-O-5606	MLO-8200	MLO-8515	0S45-1
Viscosity (cs) -65°F	2130.0	2490.0	2357.0	2230.0
100°F	14.0	34.4	24.30	12.1
400°F	1.9	3.82	2.64	1.2
Pour Point	-90	100	100	
Compatability with 5606		compatible	compatible	compatible
Vaper Pressure, 400°F (mm Hg)	230	1.2	1.0	4.7
Bulk Modulus at 3000 psi (psi)	2.6×10^5	2.5×10^5	2.5×10^5	2.5×10^5
Thermal Conductivity (BTU/hr-sq ft-°F/ft)	0.089 at 86°F	0.089 at 86°F	0.090 at 86°F	2.2×10^{-5} at 82°F
Specific Heat (BTU/lb/°F	0.46 at 86°F	0.46 at 86°F	0.47 at 86°F	0.45 at 77°F
Specific Gravity at 100°F	0.84	0.92	0.915	
at 300°F	0.755	0.83	0.83	
Flash Point °F	200°F	415	410	
Fire Point °F	*245°F	450	450	
Spontaneous Combustion°F	440°F	815	770	700

*Estimate

Only metallic filters are being employed at temperatures above 275°F. The following filters are being extensively tested by a number of companies and are quoted as capable of operation up to 500°F and beyond:

Purolator Products, Rahway, N.J., sintered bronze

Permanent Filter Corp., Los Angeles, California, sintered stainless steel

Porous Media, Los Angeles, California, stainless steel wire woven

Porolloy, Pocoima, California

Cuneo Engineering Corp., Meridian, Conn., sintered stainless steel.

At the temperatures above 275°F, magnetic filters are more necessary than at lower temperatures since the wear of steel components is increased. Those most often used are supplied by S. C. Frantz, Trenton, N.J., although some of the users build their own.

All system components present problems above 275°F. Supply pumps and accumulators have greatly shortened lives at elevated temperatures, although North American has pumps which will operate 50 hours at 400°F. The problems with high temperature valves were discussed earlier.

c. Below Zero

Most companies are concerned only with the high temperature problem. The low temperature problem is not so severe because time is allowed for the oil to reach its operating temperature. However, in a few cases, as in the case of small surface-to-air missiles, no warmup time is allowed and the system is required to operate satisfactorily at temperatures as low as 40 degrees below zero. To circumvent the problem of high viscosity at this temperature, one company, Douglas, is utilizing a modified MIL-0-5606 oil with the viscosity improver <u>not added</u>. This oil is designated MPD-2067 and is performing very satisfactorily.

Their upper temperature requirement, of course, is not severe.

3. Vibration, Noise and Pressure Pulsations

Not too much information is available on these three. Most companies vibration test their servo valve-actuator systems more or less according to standard military specifications over the frequency range of 20 to 2000 cps with inputs up to 20 g. The servo valves are generally tested separately on a vibrator; performance is measured by recording the leakage flow over the frequency range, the leakage specifications covering the maximum leakage allowed. To our knowledge, vibration has not been a serious problem in the control systems, either from a fatigue standpoint, or from a dynamic control standpoint; at least the problems have been solved. Most valves actually appear to operate more satisfactorily under vibrating conditions.

The missile control systems are also tested on centrifuges under constant accelerations up to 50 g. In one application (Douglas) the servo valve was modified to include an additional unbalanced weight to compensate for the acceleration forces on the spool. However, in most applications, conventional nonacceleration-compensated valves are employed by judicious orientation of the valve's axis.

Most of the noise problems, as far as the servo valve actuator loop is concerned, originate from feedback devices such as potentiometers and dc tachometers. A number of people are experimenting with Markite potentiometers which have infinite resolution, and are not as subject to noise as wirewound pots. The loop gain in most systems is sufficiently low that this noise has not been a limiting factor. Once again the greatest problems are encountered with the small air-to-air missiles which employ the highest loop gains.

Pressure pulsations or variations do not appear to be a source of trouble; at least their effects have not been diagnosed. These variations may develop from several sources. One high frequency source is the pump pistons. A nine piston pump operating near 4000 rpm will produce pressure pulsations of the order of 600 cps. There is generally sufficient accumulator action in the lines to filter out the effects of such a high frequency pulsation. Another source of pressure pulsations results from cross action of the various servos operating from the same supply pump. These generally are not appreciable unless the capacity of the pump is exceeded appreciably and the accumulator is allowed to deflate markedly. A third source of pressure variation occurs as a result of the pressure drop in the lines of a particular servo. Quite often in the design of the supply system the size of the lines is based on the normal flow that the servo is expected to operate at, it being reasoned that the peak flows do not occur often enough to warrant larger size tubes. Thus, when peak flows do occur during large step inputs, a considerable pressure drop occurs in the lines to reduce the pressure at the valves.

The significance of pressure pulsations or variations, of course, depends pretty much on the sensitivity of the valves to these variations. Experience has shown that the valves available at present do not appear to be sensitive to these pressure pulsations.

4. Electrical Characteristics, Dither

Most valve users design their servo valve amplifiers around a maximum differential current of 8 ma with a quiescent level slightly greater than one-half this value to obtain the best linearity. A few systems are operating at lower currents to reduce the power requirements of the output stage. Several companies are using higher differential currents.

Subminiature vacuum tubes are most often used in the servo amplifiers both in the preamplifier and driver stages. About 50% of the valve users that were visited utilize ac preamplifier stages followed by a phase sensitive detector before the final dc stage to reduce drift problems. In some of these cases the detector output is not filtered but used as a sort of a dithered input to the valve. This is possible because of the high frequency of the output signal, 800 cps. In some cases smoothers are also used.

The driving stage of course, must be dc and may operate as a differential amplifier or single ended, the former design most often being used. In the latter case, the coils may be connected in series, parallel or differentially, depending on the design of the circuit. The coils connected in series of course, require only one-half the maximum plate current of those connected in push-pull or parallel. (See Fig. 12 for typical output stage circuits)

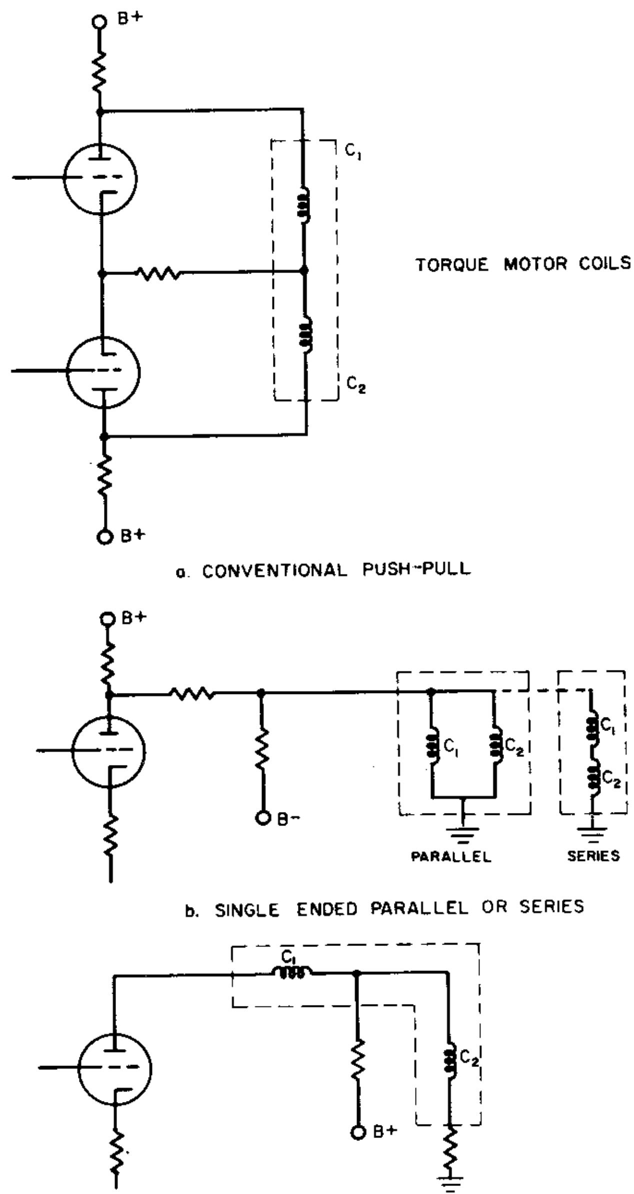

Figure 12 Typical Output Stage Circuits

Considerable experimentation is presently going on in the application of magnetic amplifiers. Carrier frequencies ranging from 400 cps to 3200 cps are being used. Boeing Aircraft was the only company visited that employs a magnetic amplifier in their present design. This was a push-pull magnetic amplifier employing feedback and using a carrier frequency of 900 cps. Output current was 10 ma.

A number of organizations are also working with transistor servo valve amplifiers. North American has developed a completely transistorized amplifier, including output stage, which they plan to use in future missile applications.

Both magnetic amplifiers and transistors, in their present state of design, are somewhat poorer from a response standpoint than vacuum tubes. However, because of the relatively low system dynamic response requirements and because of their potentially greater reliability, both magnetic amplifiers and transistors are receiving a great deal of consideration.

Most applications using conventional flapper nozzle valves do not require dither to improve valve and thus system performance. Single-stage and two-stage valves using spool first stages, however, require dither to reduce the effect of stiction on the first-stage spool. The most common dither frequency is 400 cps, although frequencies of 250, 315, 210, are used to take advantage of loop resonances to reduce power requirements. The advantage of using 400 cycles of course, stems from the fact that this is the power source frequency.

Dither will probably be used to a greater extent in higher temperature applications. This is because the valves laps must be greater due to the thermal expansion difficulties, and dither may be required to prevent bonding and instability.

5. Life, Reliability

There still is not much data on the life and reliability of servo valves. It can, of course, be said that the valves used in the manned aircraft application require greater life and reliability than those used in missile application and the valves used in the large long range missile require a greater life capability than those used in small air-to-air missiles. The specifications which are used for airplane application range around 1000 hours or one to four million cycles. The life requirements for missiles vary between 10 to 250 hours. Most valve manufacturers are confident that the latter figure is presently attainable.

Not only do the life specifications of the valve vary with the expected length of operation of the missile but also with the environment. For instance valve users do not expect valves to operate at 600 degrees for 250 hours. Engineers at North American stated that 10 hours operation at this temperature would be highly desirable and probably adequate for some of their applications.

B. Discussion of System Design and Dynamic Requirements

A general block diagram of a typical control surface or rocket gimbal servo is shown in Fig. 13. This servo accepts voltage outputs from the autopilot and guidance system and provides an output movement which is proportional in position or velocity to the input signal.

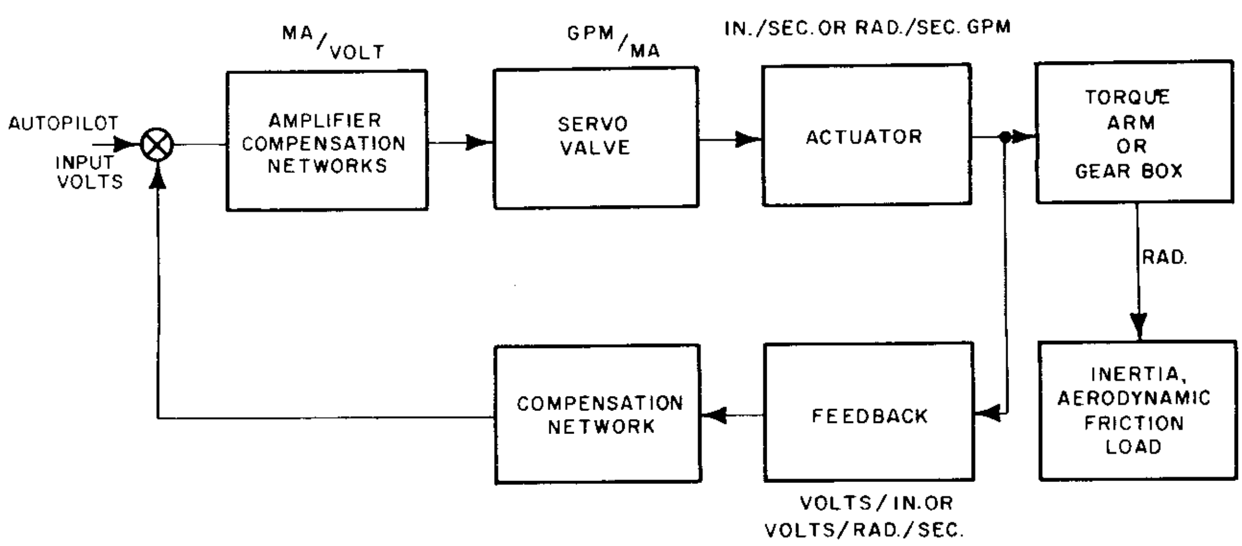

Figure 13 Block Diagram

1. Selection of Actuator and Valve

Generally the first step in the design of the control surface servo system is to determine figures for the maximum hinge moment, the maximum surface velocity, and displacement. These figures are generally provided by the analysis group as a result of preliminary analog computer studies. The former may be based on the maximum force to maintain structural integrity or possibly the maximum aerodynamic force to be encountered at the maximum control surface displacement. The latter figures are dependent on the dynamic requirements of the vehicle.

The next step is to determine the supply pressure to be employed. In most modern systems a supply pressure of 3000 psi is used to permit the use of the smaller and lighter pumps and actuators which are presently available. In cases where size and weight are not important, lower pressures are sometimes used to improve reliability and to reduce wear on seals and metering orifices.

The size of the actuator - bell crank combination is then determined. In most cases, the bell crank dimensions are pretty well defined by the space allotted for its mounting. The actuator area can then be determined by dividing the maximum force at the actuator by the supply pressure. The actuator stroke can be computed from figures on the maximum displacement and torque arm.

The next step is to select the servo valve for driving the actuator. The technique most often used is to compute the maximum flow (multiply the maximum velocity by the actuator area) and select the valve characteristics such that this flow occurs at two-thirds supply pressure, which is the point of maximum power transfer. If there is a maximum force requirement at this flow, the valve characteristics are selected such that the resulting load pressure (force divided by actuator area) occurs at this flow. Either of the last two criteria tie down the pressure-flow characteristics of the valve.

Sometimes, where there is no maximum hinge requirement (e.g., a rocket engine control servo) the actuator area may be based on the force needed at the maximum velocity, or some other velocity. In such a case the maximum power transfer criteria is used to obtain the actuator area (divide force at maximum flow by two-thirds supply pressure) and the required valve flow is determined by multiplying this area by the maximum velocity of the actuator.

In one case, supply pressure was used as a dependent variable to provide a certain oil-load mass resonance. The relationship for this resonance is:

$$\omega_n = A\sqrt{\frac{1}{K_3 M}} = \frac{Ps}{Fm}\sqrt{\frac{1}{K_3 \quad M}}$$

or $Ps = \omega_n \, Fm \sqrt{K_3 M}$

where A = area of actuator

$$K_3 = \frac{Vm}{2\beta}$$

Vm = volume of oil on one side of cylinder

β = bulk modulus of oil

M = mass load

Fm = maximum force required (static)

After the value for Ps was determined, the actuator area, A was calculated and the valve selected on the basis of the maximum power transfer at maximum velocity.

2. Flow Limiting

Flow limiting is commonly employed to limit the maximum velocity. This is done at one of two locations, in the amplifier or in the hydraulic lines. The valves are rarely used as flow limiters alone. The most common technique is to place variable orifice, flow actuated limiters in the supply, drain or load lines. The majority of design engineers evidently believe that this is the most reliable means of limiting flow although there are disadvantages to the application of the limiters in all locations mentioned. When a flow limiter is placed in the supply lines, the supply pressure to the valve will vary; when placed in the drain line, the drain pressure will vary with flow. Both of these are undesirable. When located in the load line, the limiter may resonate with the load elements; in fact, considerable trouble has been experienced due to this effect.

Limiting in the amplifier is accomplished by designing the amplifier to saturate at a certain input voltage. In this case the valve actually does the limiting although at a point dictated by the amplifier. The use of gain compensated valves would be of advantage here since the flow does not increase appreciably (and can be made to decrease) at pressures less than two-thirds supply pressure. While flow limiting at the amplifier appears to be the simplest technique, most engineers prefer the variable orifice type because the latter holds the flow to closer tolerances than can be expected from the current-flow relationship of the valves. Of the companies visited, only Boeing utilized amplifier flow limiting.

3. Dynamic Characteristics and System Synthesis

Table 3 lists the system characteristics of a number of missile and aircraft control systems. In this table maximum velocity and friction are referred to the actuator, while inertia and aerodynamic hinge moment are determined at the control surface or rocket gimbal hinge. The aerodynamic hinge moment occurs at maximum control surface deflection. To determine the maximum control surface velocity, the actuator velocity must be multiplied by the moment arm or gear ratio, whichever is given. The supply pressure listed is nominal and will vary due to drop in lines, cross coupling between servos, accumulator action, etc. (see Table 2). Data on the load actuator are self explanatory.

The symbol f_{n1} refers to the natural frequency of the load mass and the effective aerodynamic spring; thus

$$f_{n1} = \frac{1}{2\pi} \sqrt{\frac{K}{M}}$$

The symbol f_{n2} defines the natural frequency of the oil-load mass combination and is equal to:

$$\frac{A}{2\pi} \sqrt{\frac{2\beta}{VmM}} \quad \text{or} \quad \frac{d}{4\pi^2} \sqrt{\frac{2\beta}{V_m J}}$$

for linear and rotary systems respectively, where

A = actuator area (linear actuator)

d = actuator displacement (rotary actuator)

β = bulk modulus of oil

J = moment of inertia

M = mass

K = effective aerodynamic spring

V_m = oil under compression at one side of actuator

The term f_{n3} refers to the combined natural frequencies of the structural resonance and the oil-load mass natural frequency, unless otherwise listed.

Table 3 shows most of the valve actuator loop gains to be below 100. Exceptions are the small missile control systems which have a high response requirement. Another is a medium sized missile with large actuators which provide a sufficiently high mass-oil resonance to permit the use of a high gain. In most cases the servo valve actuator loop resonance is at least two octaves below the mass oil resonance. (An approximation of the loop resonance can be obtained by use of the following formula:

$$f_{\ell} = \frac{1}{2\pi} \sqrt{\frac{K}{T}}$$

where T is the time constant of the valve). A typical value for T is 0.005. The one exception is a system utilizing a pressure control valve with tachometer feedback, a system which is theoretically independent of the mass-oil resonance.

With loop gains less than 100, little is generally used in the way of compensation. Again, except for the small missiles, no compensation networks were employed in the servo valve-actuator loop; leakage around the actuator was used in one case to provide damping.

In small missile control systems, the situation is considerably different. Stabilization networks are generally provided in both the forward and feedback paths. In addition, the loop gains are varied as a function of altitude or ram pressure, the gains being highest at low altitudes; this is also done in some of the larger missiles. It is interesting to note that the compensation consisted of the lead-lag networks; no tachometer or accelerometer feedback was used. In most cases the response of the valve is a limiting factor in these small missiles. In other words the response of the system could be improved if the dynamic response of the valve were higher.

In the design of the large missile control systems it has been found that if the valve and actuator were designed to meet the static requirement such as a maximum load, velocity, acceleration, etc., then dynamic requirements would also be satisfied. The only problem would be to obtain sufficient amplifier gain to meet the static accuracy requirements. In smaller missiles and other high performance control systems this is not the case of course, and considerable time is spent in their analyses and syntheses. As a result of this survey it appears that the analogical studies during subsequent phases should be directed toward the small missile control systems for which the dynamic problems are most severe.

4. Simulation Practices

Generally the complete missile system is simulated on an analog computer to determine the required characteristics of the various subsystems. These characteristics are then used as initial design criteria in designing the control systems.

When the control systems reach the breadboard stage, they are generally tested using the control surfaces as physical loads, or using simulated loads consisting of equivalent inertias, aerodynamic springs and friction. Following this stage of development the breadboard equipment is actually included in the complete system simulation, replacing the electrical equivalent used previously. The final mechanical changes are made at this stage. Quite often, the final prototype equipment is also tested in this manner for purpose of making final gain and compensation network adjustments.

In most cases the simulated spring loads are passive and simulated by springs or hydraulic rams; in many cases the spring loads are neglected altogether since they have a stabilizing effect. Stability is more of a problem where they do not exist (e. g., when the aircraft is on the ground). Considerable work has been done by a number of organizations including Cook Research Laboratories in the investigation of active load simulators to take into account the nonlinearities of aerodynamic forces.

C. High Temperature Facilities

Because of the forthcoming high temperature problems, it appears desirable to review the facilities which are presently available or under construction for investigating this problem.

There is very little in the line of high temperature test equipment available at the valve manufacturers. At the time of our survey, only one organization had any facility at all while another had one under construction. Moog Valve Company had begun testing high temperature models in a converted electrical roaster oven combined with a gas burner and a heat exchanger for cooling the oil down to 200°F before it re-enters the pump. The valve is manifolded to an actuator placed in the oven during tests. Supply and drain lines are run to the outside of the oven. With this setup, Moog plans to run preliminary tests on valves at ambient temperatures up to 800°F and oil temperatures of 500°F. They are constructing another facility in New Jersey where they hope to do more extensive testing.

Bell Aircraft is also constructing an elaborate high temperature facility which will permit testing valves and systems at oil temperatures of 400°F and ambient temperatures of 800°F.

Of the companies visited, North American Aviation had, by far, the finest high temperature facilities. They have been conducting experimental work in the high temperature field for a number of years, during which time they have investigated both oil characteristics and component characteristics at high temperatures.

North American's present facility includes at least five large high temperature ovens where they test equipment at ambient temperatures up to 1150°F and oil temperatures up to 600°F. They are presently constructing another oven which will provide these temperatures at a simulated altitude of 100,000 feet. A system of motorized mirrors and a closed circuit TV is provided for visual observation during tests. They have developed a great number of components for high temperature application. For instance they presently use a high temperature pump which will perform satisfactorily for at least 50 hours at 400°F.

Boeing is also doing some experimental work at high temperature although their present requirements do not extend beyond 290°F.

A number of companies have conducted high temperature studies on paper but the ones listed were the only ones visited doing experimental testing above 300°F.

D. Consensus of Comments on Valves

Moog Valve Company is still the largest producer of electrohydraulic servo valves for the aircraft and missile industry. However, a number of companies have converted, or are in the process of converting to other makes of valves, notably those manufactured by Cadillac Gage and Bendix Pacific Division. Conversely, some companies, who have previously used or contemplated using other makes, are converting to Moog valves.

1. Moog Valves

Moog Valves are used by North American, Northrop, Glenn L. Martin and Convair. The criticisms of Moog valves vary although, on a whole, the valves were performing satisfactorily. Boeing discontinued the use of Moog valves because they found the valves to be considerably more susceptible to oil contaminants than their replacements, the Cadillac Gage valves. The principal oil contaminants were magnetic particles and it is important to note that Boeing was the only company visited (except North American in their very high temperature applications) that reported a problem due to magnetic dirt.

A number of companies reported a null shift problem with the Moog valves, especially with temperature and time. As mentioned previously, the procedure used by Moog to align their first stage, that of bending the flapper, causes stresses which are released at high temperatures, or after a long period of time. To overcome this, Moog is utilizing an accelerated aging process in which the valves are boiled in oil, followed by a re-entering of the second stage. These difficulties and their method of correction were reported by engineers at Glenn L. Martin and North American.

Convair of Pomona reported difficulties with external leakage with initial Moog valves which have since been corrected. Lockheed tested several Moog 2000 valves and found they could not meet their dynamic performance specifications. Reworked versions appear acceptable although the testing of these valves was not completed at the time of our visit.

Both Convair San Diego Division and Northrop reported good results with Moog valves. Moog is the only valve manufactuere producing valves at present that will operate at temperatures up to 400°F.

2. Cadillac Gage

The Cadillac Gage FC-2 valves are being used or have been selected for use by North American, Boeing, Lockheed, and Douglas; North American, however, buys a greater quantity of Moog valves.

The Cadillac Gage valve that replaced the Moog valves at Boeing was the conventional wet torque motor valve. Boeing has since tested and evaluated their stale torque motor valve, found it to be superior and plan to convert to it soon.

Lockheed found the Cadillac Gage FC-2 valve, a prototype version, to be superior to the Moog Model 2000 valve from a response standpoint. Conversely, Convair, Pomona Div. found the production version of the Cadillac Gage could not meet their dynamic requirements and reverted to Moog valves. They had previously accepted the prototype version of the FC-2 on the basis of the results of experimental tests conducted at Convair.

3. Bendix Pacific Division Valve

The Bendix Pacific Div. is being tested by a number of companies although, to our knowledge, no large production quantities have been ordered so far. The initial reports on the valve are very promising, although Bendix is having some trouble producing in quantity. Convair Pomona Division is presently evaluating both the

Bendix and Hydraulic Research and Manufacturing valves with a view toward giving a second source contract to one of these. Lockheed tested a prototype version of the Bendix Pacific HR series valve and found it to out-perform both the Moog and Cadillac Gage. It of course remains to be seen how the production version will turn out.

4. Hydraulic Research and Manufacturing Valve

This valve is used in all Bell Aircraft missile applications. Considerable research has been applied to the valve in the last couple of years and they now hope to enter the servo valve field in earnest. The valve is presently being evaluated by Convair of Ponoma as a possible second source.

5. Other Valves

Other valves in missile application are the Nike valve, a modified Nike compensated for acceleration, the Sparrow valve, all manufactured by Bendix Pacific Div. for Douglas, and the Hughes Falcon valve-actuator. According to engineers at Douglas, Santa Monica Div. these valves have operated very satisfactorily but are not used by other companies.

Valves are also presently under development by Boeing Aircraft, Sanders Assoc., Dalmo Victor, Lear and National Water Lift, but these have not been used in missile application thus far. Dalmo Victor has built a fairly large number of three-way valves for Philco for antenna applications; no data are available on these.

6. Summary

While a number of new valves have been placed on the market during the last two years, none of these are yet available in large production quantities. There is a definite difference between producing a few good or superior valves and producing a large quantity of valves of the same quality. There appears to be a delay period of around two years between the time the valve is first brought out and the time it is available for large quantity production.

The only company outside of Moog producing commercially available valves in quantity at the time of this survey was Cadillac Gage.