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PART I

HYDRAULIC SERVO CONTROL VALVES

Part I. A Summary of the Present State of the Art of Electro-Hydraulic Servo Valves

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FOREWORD

This report, A Summary of the Present State of the Art of Electro-Hydraulic Servo Valves, was prepared by R. E. Boyer, B. A. Johnson, L. Schmid, and L. S. Weinstein, under the supervision of J. Warshawsky Technical Director of the Automatic Controls Systems Section, Cook Research Laboratories, Skokie, Illinois, under Contract No. AF 33(616)-2447, "Hydraulic Servo Control Valve Analysis". The work to be accomplished under this contract has been divided into three phases. This report covers the first phase and is primarily a tabulation of the characteristics and requirements of the various electro-hydraulic servo valves currently in use or under development. During the second phase, several of the most promising servo valves, that were investigated in Phase I, will be tested and evaluated. The third phase is an analysis phase, the purpose of which is to establish a standard for testing and evaluating servo valves in general. This work was administered under the direction of the Aeronautical Research Laboratory with Mr. P. P. Cerussi as supervising Task Scientist.

In order to present an unclassified report, all references to military missiles and aircraft have been excluded. Servo valves that have been built for a particular missile, often take on the name of that missile as their designation. In this report, all such valves have been assigned the following code numbers:

1. Bendix, B-1
2. Bendix, B-II
3. Cadillac Gage, CG-I
4. Cadillac Gage, CG-II
5. General Electric, GE-1

All other servo valves bear the identification numbers assigned by their manufacturers.

ABSTRACT

A summary of the present state of the art of electro-hydraulic servo valves suitable for aircraft and missile application is presented. This information was obtained from direct visits to both the valve manufacturers and the valve users, namely: the missile and aircraft manufacturers. The valve manufacturers were contacted to obtain detailed design and performance data on the valves; the valve users were contacted to determine the system and valve requirements and to determine how adequately the valves presently being used meet these requirements.

A brief description of the principle of operation of servo valves, in addition to a discussion of the various types, is also presented to provide a background for interpreting the data.

PUBLICATION REVIEW

This report has been reviewed and is approved.

FOR THE COMMANDER:

Aldro Lingard

ALDRO LINGARD, Colonel, USAF
Chief, Aeronautical Research Laboratory
Directorate of Research

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INTRODUCTION

A. Purpose

During the last few years, the development and use of hydraulic servo systems has increased at a rapid rate. This is especially true in the aircraft industry where aircraft and guided missile designers are extensively utilizing hydraulic servo systems to operate control surfaces. One of the most important single components in these systems is the servo valve which receives the electrical signal from the flight control system and controls the hydraulic output to the control surface actuator. The development of the servo valves has occurred over a span of a very few years; various different valve configurations have been designed for specialized purposes and have then been used for other applications. Due to the rapid growth and development of these servo valves, generalized design data on the various valves are not available at present. Many of the valve manufacturers use different means to describe the performance of their valves, especially from a dynamic standpoint. This presents a difficult task to the designers of the control system because they lack a common denominator upon which they might base the selection of a particular valve.

It is apparent that the cataloging of characteristics of existing valves along with the establishing of a standard for evaluating servo valves is necessary to expedite the development of new flight control systems. In addition, such information would serve as a foundation for a concrete program of research and development to support this item of equipment.

Cook Research Laboratories are presently conducting such a program for the Air Force. This report covers the first phase of this program and is primarily a tabulation of characteristics and requirements of the various servo valves currently in use or under development. Later phases will be concerned with the testing of certain valves and with the establishment of a standard for testing and evaluating servo valves.

B. Method of Conducting the State of the Art Study

The information covered in this report was obtained from two general sources: the valve manufacturers and the valve users. The manufacturers were contacted to obtain the pertinent design information on the valves; the users were contacted to determine the capability of the valves and to determine the valve and system requirements. In many cases the manufacturer and user were the same. Only valves suitable for missile and aircraft application were investigated in detail.

Every effort was made to contact all major valve manufacturers and users. In most cases contacts were made by direct visits, even when published data were available, to make the survey as complete as possible.

C. Organization of the Report

The report begins with a brief description of the principle of operation of electro-hydraulic servo valves followed by a general discussion of the different valve configurations currently in use to provide a background for interpreting the information in the subsequent sections. The information obtained in the survey is then presented in two sections. The first section contains a summary of the physical, operational, and environmental characteristics of the various valves along with a brief description of their method of operation. The second section contains a summary of the system and valve requirements as specified by the users (missile and aircraft manufacturers) along with their comments as to the capability of the valves they currently use or have previously tested. The major portion of the information in each section is presented in tabular form at the end of the section to facilitate comparison.

D. Sources of Information

A tabulation of all the organizations from which information was obtained is presented below. The designation (M) and (U) opposite the organization name indicates whether they manufacture or use valves respectively, or both.

Aerojet, Azusa, Calif.	
APL, Johns Hopkins University, Silver Spring, Md.	U
Argonne National Laboratories, Lemont, Ill.	U
Bell Aircraft Co., Buffalo, N. Y.	M & U
Bendix Aviation Corp., Mishawaka, Ind.	M & U
Bendix Eclipse Pioneer Division, Teterboro, N. J.	M & U
Bendix Pacific Division, North Hollywood, Calif.	M & U

Bendix Research Laboratory, Detroit, Mich.	M
Bertea Products, Pasadena, Calif.	M
Boeing Aircraft Co., Seattle, Wash.	U
Cadillac Gage, Detroit, Mich.	M
Consolidated Vultee Aircraft Co., Pomona, Calif.	U
Douglas Aircraft, El Segundo, Calif.	U
Douglas Aircraft, Santa Monica, Calif.	U
Drayer Hanson Co., Los Angeles, Calif.	M
Fairchild Aircraft Co., Wyandanch, N. Y.	
Firestone Tire and Rubber, Los Angeles, Calif.	
General Controls, Glendale, Calif.	
General Electric Co., Guided Missiles Dept.	M & U
Goodyear Aircraft Co., Akron, Ohio	U
Hughes Aircraft, Culver City, Calif.	M & U
Hydraulic Controls Co., Roxbury, Mass.	M
Lockheed Aircraft Co., Burbank, Calif.	U
Glenn L. Martin Co., Baltimore, Md.	U
Massachusetts Institute of Technology, Boston, Mass.	M & U
Midwestern Geophysical Labs., Tulsa, Okla.	M
Minneapolis-Honeywell, Minneapolis, Minn.	M & U
Moog Valve Co., East Aurora, N. Y.	M
Naval Air Missile Test Center (Point Mugu), Oxnard, Calif.	M & U

North American Aviation, Downey, Calif.	M & U
Northrop Aircraft, Hawthorne, Calif.	U
J. C. Peacock Machine Co., Los Angeles, Calif.	M
Pegasus Laboratories, Berkley, Mich.	M
Purdue University, West Lafayette, Ind.	U
Raytheon Mfg. Co., Newton, Mass.	M & U
Ryan Aviation Co., San Diego, Calif.	U
Sanders Associates, Nashua, N. H.	M
Sperry Gyroscope Co., Great Neck, N. Y.	M & U
Standard Controls Co., Seattle, Wash.	M
Westinghouse Air Arm., Baltimore, Md.	M & U

CHAPTER 2

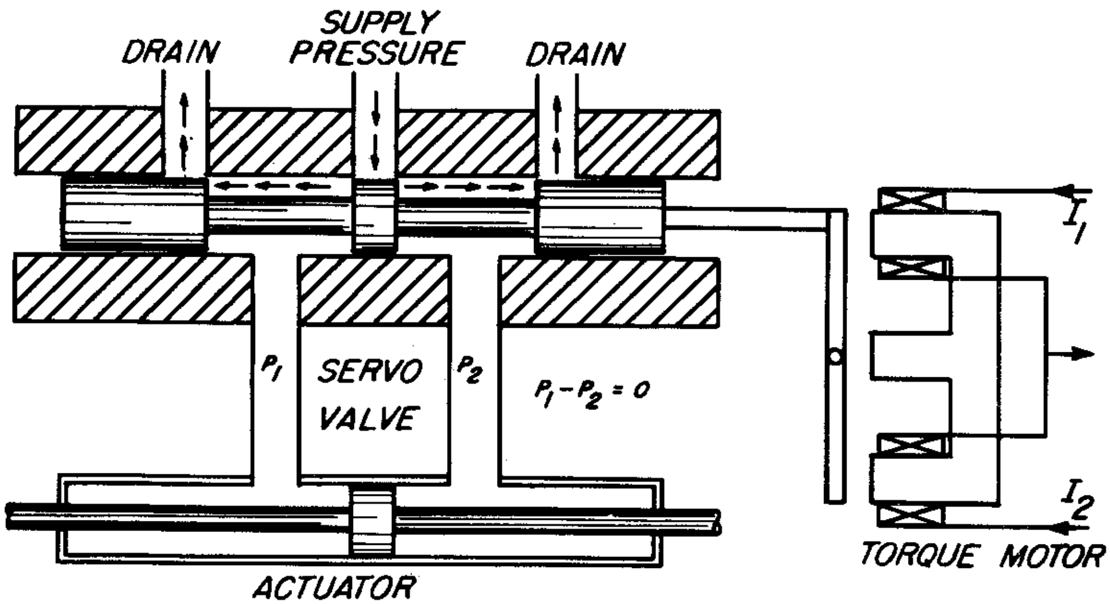
A BRIEF DESCRIPTION OF ELECTRO-HYDRAULIC SERVO VALVES AND THEIR OPERATION

The electro-hydraulic servo valve is a device which converts an electrical signal into a controlled fluid flow and pressure which, in turn, may be subsequently utilized to move a mechanical actuator. Principally the servo valve consists of two parts, the electromagnetic driver, which converts the electrical signal into a mechanical displacement, and the metering section, which is controlled by the electromagnetic driver and which regulates the output flow. For the majority of valves, when supplied from a constant pressure source, the output flow is approximately proportional to the electrical input signal, the direction depending upon the sense of the signal. In order to better understand its function, the operation of a single stage servo valve, driving a ram-type actuator, will be described with the aid of Fig. 1.

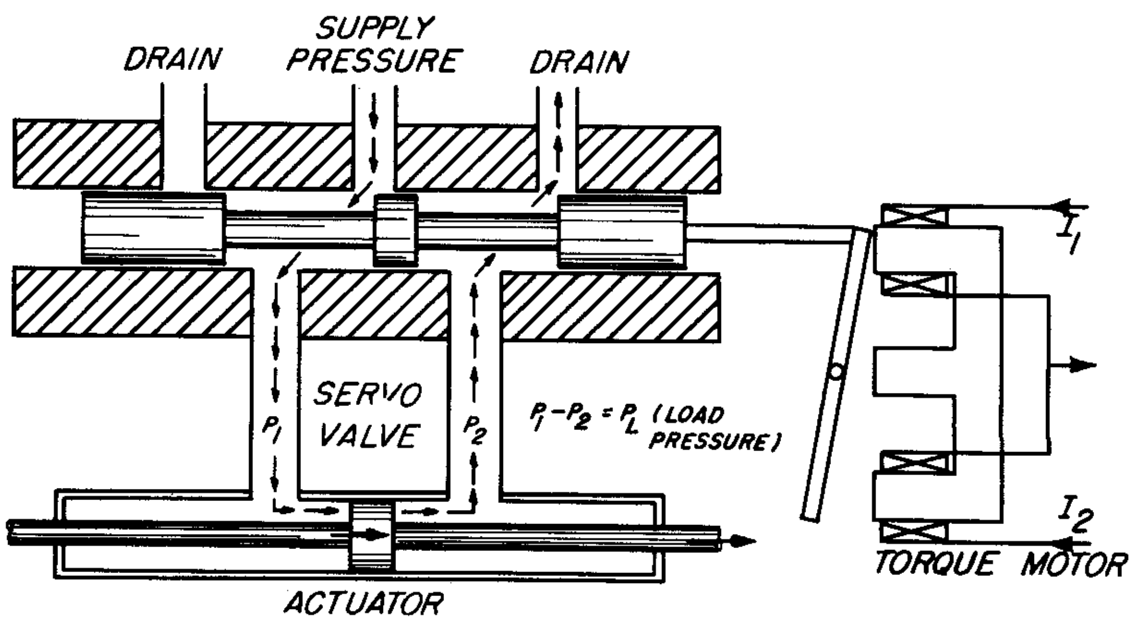
In Fig. 1a, the valve is shown in its neutral position; the input current to the electromagnetic driver is zero. The only flow is leakage flow from the supply port, through the spool clearances and out each drain. The differential pressure across the output ports is zero.

If a DC current is applied to the coils of the electromagnetic driver, the armature will rotate, displacing the spool from its neutral position. The direction of spool movement depends on the polarity of the input signal. Let us assume that the polarity is such that the spool is moved to the right. The resulting condition is shown in Fig. 1b, where the differential pressure at the output ports is such that the load flow is to the right and the ram is moved to the right. It is evident that if the input signal had been of the opposite polarity the output flow would be reversed and the ram would be moved to the left. It is interesting to note that the flow through each orifice is unidirectional; a different set of orifices is utilized for each direction of flow.

If a signal proportional to the actuator displacement, as shown in Fig. 2a, is fed back and compared with the input signal, the servo valve becomes an integral component of a servo positioning system. In this case the actuator is a linear ram and the positioned element is a control surface. The actuator also might have been a rotary motor as shown in Fig. 2b. The positioned element might have been a machine tool cutter, a gun turret, a ship's rudder, a radar antenna, etc. The servo valve and actuator thus may be utilized in many applications where electric



a.



b.

Fig. 1 - Single Stage Servo Valve Driving Actuator

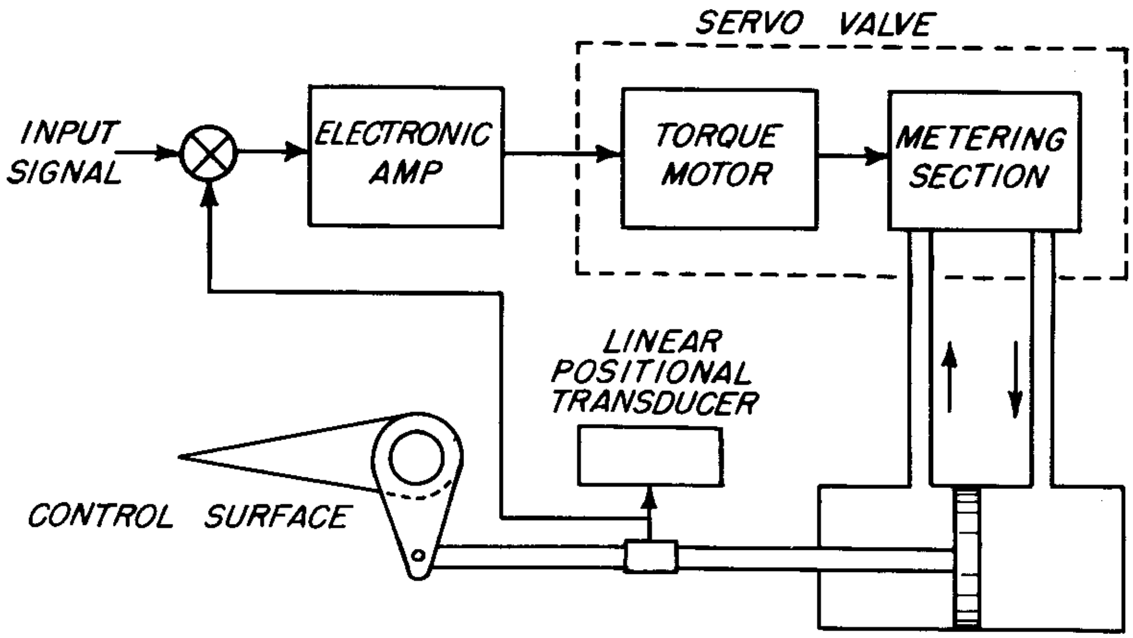


Fig. 2(a) - Servo Valve Positioning Control Surface

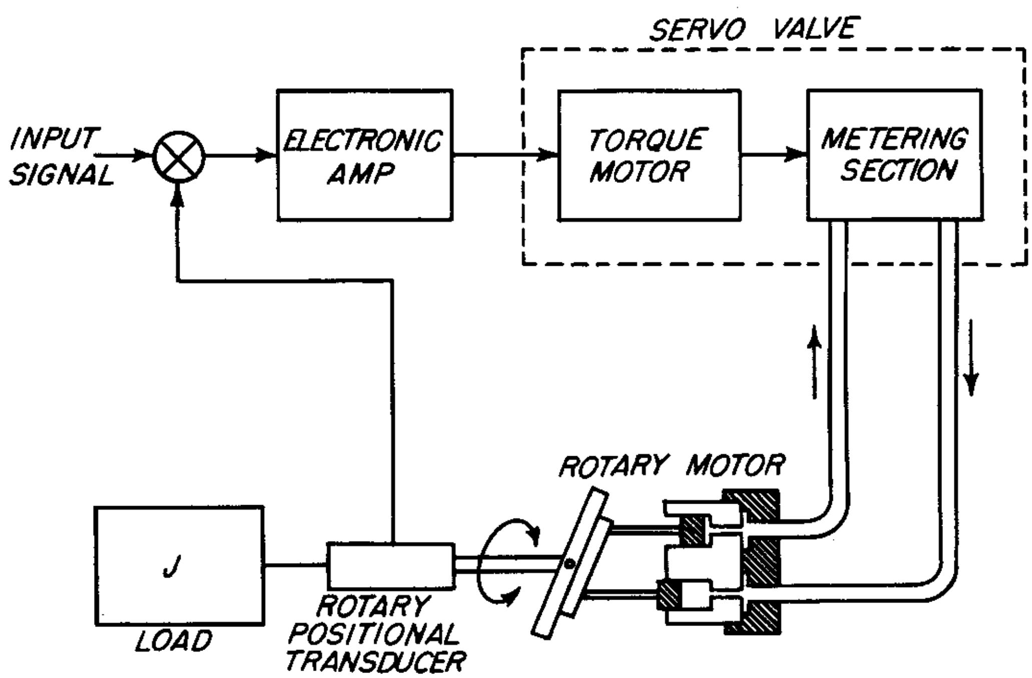


Fig. 2(b) - Servo Valve Driving Rotary Actuator

motors are employed. When compared with the electric motor, the hydraulic servo provides higher dynamic response and lower size and weight to power ratio, factors especially important in airborne applications.

CHAPTER 3

GENERAL DISCUSSION OF TYPES OF SERVO VALVES

A. Flow and Pressure Control

Ideally a servo valve can be constructed to be either a flow control device or a pressure control device. If the valve were a perfect flow control device, the flow would be directly proportional to input signal and would be unaffected by extraneous effects such as load pressure variations. If the valve were a perfect pressure control device, the pressure output (across the load) would be directly proportional to the input signal and would be unaffected by variances in the output flow.

In practice however, most servo valves are neither perfect flow nor perfect pressure control devices since the controlling element in each case is a variable area orifice which exhibits a square law relation between differential pressure and flow ($q = k\sqrt{P}$) for any constant opening. Thus any change in load pressure will affect the flow through the orifice and vice versa. This can be seen by inspecting the solid lined curves of Fig. 3, which represent the pressure flow characteristics for an uncompensated valve. Each curve represents a constant value of input signal. The only manner whereby either the output flow or the output pressure can be maintained proportional to the input signal for a range of operation, is by compensating for load variations by varying the orifice area. A few valves are presently being compensated in this manner. Those which are compensated to provide flow proportional to input current are generally termed "gain compensated" valves; those which are compensated to provide flow proportional to pressure are generally termed pressure control valves. Both these types are illustrated in Fig. 3. In many cases gain compensation can be applied to a standard valve configuration, while pressure control generally necessitates a special design. The merits of pressure and flow control and the types of loads most ideally suited for these type of valves will be explored in more detail in the later phases of this study.

B. Valve Configuration

As mentioned previously, the servo valve controls both magnitude and direction of flow. For this purpose either three-way or four-way valves may be employed. The three-way valve has two metering orifices in series with the supply pressure and drain. The pressure between the two orifices constitutes the output pressure to one end of the load actuator.

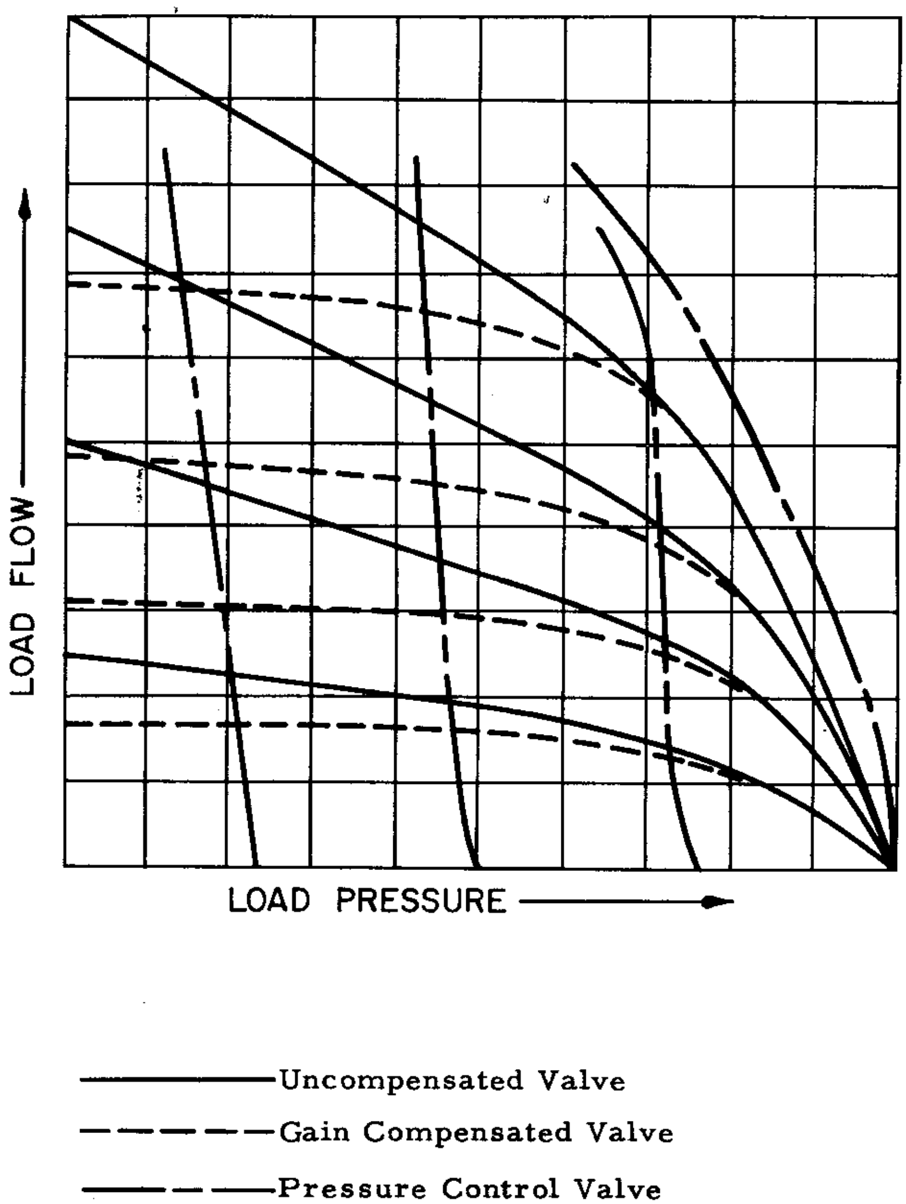


Fig. 3 - Static Pressure-Flow Curves for Various Classes of Valves

To obtain bidirectional operation, the restraining force at the other end of the actuator may be either a spring or an external pressure (i.e. supply pressure).

Principally, the four-way valve consists of a pair of three-way valves operating in push-pull. The valve shown in Fig. 1 is a four-way valve; it has four metering orifices, two for each direction of flow. Because of the push-pull action, the four-way valve provides bidirectional action without the need of a spring or external pressure source. When compared with a three-way valve, the four-way valve provides a greater force output for the same size actuator, is more linear over a larger range, especially when the valve is underlapped, and is less susceptible to supply pressure variations. It has the disadvantage of being more costly to build because of the greater number of orifices.

A valve which has greatly underlapped metering orifices, that is orifices that permit a sizeable percentage of maximum flow when in the closed or center position, is termed an open-center valve. Conversely, a closed-center valve permits very little flow, if any, when in the closed position. Nearly all the valves presently being used are of the closed center type with a very small amount of either overlap or underlap (approximately ±0.0001 in.) at the output stage. This construction is used to maintain the leakage flow at a minimum, to provide minimum dead zone, and to provide maximum dynamic response. (Since the open centered valve has relatively large flows around null, the damping forces are relatively large in comparison to those of the closed center valve).

The metering orifices may be provided by various means but the configurations most commonly used are the sliding spool (See Fig. 1) and the nozzle flapper. These and others will be discussed later in this section.

There are many different servo valve configurations currently in use and the different types will now be discussed in some detail.

1. Stages of Amplification

a. Single Stage Valve

Servo valves in all instances employ one or two stages of hydraulic amplification. The valve illustrated in Fig. 1 is a single stage valve. The flow to the load actuator is controlled by a member directly connected to the torque motor. In nearly all single stage valves this member takes the form of a single spool or a pair of spools which controls the fluid flow by opening ports in the valve body. Two spools are sometimes used to minimize the effects of acceleration on the valve. Because of the power required to overcome the inertial, friction, and flow (Bernoulli) forces on the spool, the single stage valve is generally limited to relatively low flows. Most single stage valves are flow force compensated to reduce the load on the electromagnetic driver; this technique will be described later.

b. Two Stage Valve

(1) General

In order to provide greater power amplification and improved response, two stage valves are employed. In the two stage valve the first stage member, which is connected directly to the torque motor, controls the main fluid flow to the second stage member and thus positions the second stage member hydraulically. This second stage valve then controls the fluid flow to the load actuator in a manner similar to the operation of a single stage valve. (Note that there are two parallel paths for the fluid flow). The two stage valve thus incorporates a stage of hydraulic amplification. Since the fluid flow required from the first stage is limited, the first stage member can be small, and likewise the electromagnetic driver can be small, all of which permit improved response. The first stage member may be in the form of a spool, a flapper nozzle assembly or a jet pipe; the second stage member is nearly always a spool.

The jet pipe is a relatively low response device and is used more for industrial applications. It differs from the flapper nozzle in that it provides the entire controlling pressure to the next stage in the form of a velocity head, while the flapper nozzle acts as an impedance in a bridge circuit to proportion a certain percentage of the supply pressure to the next stage. Thus, the jet pipe has a slower response than the flapper nozzle due to the inherent time lags in recovering the velocity head and due to the larger inertia of the jet pipe in comparison to that of the flapper. Both the flapper nozzle and spool receive common usage. The flapper nozzle has the advantage of higher response, no dead zone, and less friction when compared to the spool. The spool has the advantage of lower quiescent flow and higher power gain. Generally, dither is required in those valves in which the torque motor is driving a spool, either to reduce coloumb friction or to minimize the dead zone due to overlap.

The Dyanmic Analysis and Control Laboratory of MIT has been experimenting for some time with valves which have a hole and plug type of construction. These valves consist of a movable valve plate moving in respect to a fixed valve body. Metering is accomplished by either rotating or translating the plate with respect to the body, thereby displacing matched holes with respect to each other. According to MIT, this type of construction simplifies the problem of machining to the close tolerances required at the metering orifices.

(2) Feedback

The second stage spool in the two stage valves may be spring restrained or unrestrained. In the cases where it is unrestrained, the output flow is approximately proportional to the double integral of the input signal and instability will result in a closed loop system. Therefore in all cases where the second stage spool is unrestrained, feedback is required to position the second stage spool. The feedback may be accomplished mechanically, hydraulically, or electrically.

Several configurations are possible with mechanical feedback. As used in some of the earlier valves, the position of the second stage spool may be fed back to the sleeve of the first stage spool through a mechanical lever arrangement. Another method commonly used is to feed back the position of the second stage spool to a spring opposing the torque motor movement. The latter might be called force feedback while the former might be called translation feedback. One example of hydraulic feedback is that used in a pressure control valve where the load pressure is compared with the output pressure from the first stage, which is proportional to the signal input.

In electrical feedback, a signal proportional to the second stage spool position (obtained with a linear motion potentiometer or differential transformer), is compared with the input signal, and the resulting signal is used to drive the torque motor; a balance or null is obtained when the two signals are equal in magnitude and opposite in sense.

2. Compensation

As mentioned previously, certain valves employ either flow force or gain compensation to alter their flow-pressure characteristics. Flow force compensation consists of modifying the geometry of the valve spool (reshaping the valve ports and spool) so that the flow forces are either totally eliminated or greatly reduced. Gain compensation (see dotted curves in Fig. 3) involves altering the nonlinear static flow characteristics of the servo valve such that the output flow is made directly proportional to the input signal over most of the operating range. Thus, flow force compensation tends to make the pressure flow characteristics more parabolic, while gain compensation tends to make them more linear and less dependent on load pressure. Presently conceived methods of gain compensation consist of utilizing the flow forces on the second stage spool in combination with a restraining spring to reduce the nonlinearity of the valve. It is obvious, therefore, that by the above methods an improvement in flow force compensation means a degradation of gain compensation and vice versa. It also appears possible to gain compensate valves with unrestrained second stage spools by feeding back some type of signal proportional to flow but this is a more difficult problem and, to our knowledge, is not being used. In single stage valves where limited torque motor drive is available, flow force compensation is generally used; in two stage valves flow forces are generally not a problem and gain compensation may be desirable.

A third type of compensation, termed amplitude compensation, is sometimes employed to obtain more linear static flow characteristics in the region of small spool displacements with valves which have flow force compensation. The need for amplitude compensation arises from the incomplete counteraction of the flow forces which results in an extremely nonlinear residual force. This compensation may be accomplished by making the developed force of the electromagnetic driver nonlinear in a direction opposite to this remaining force.

3. Electromagnetic Driver

The term electromagnetic driver is used because all the valves encountered in this survey were of this type. Piezoelectric crystals have been used on certain experimental models to obtain improved response. However, they have not been accepted to date because of high susceptibility to vibration, temperature changes, and electrical noise and because of the difficulty in obtaining sufficiently large displacements from the crystals.

The electromagnetic driver generally takes the form of a torque motor or solenoid, DC excited, and spring restrained; in one case a voice coil is employed. The torque motor is most commonly used. It consists of an electromagnet and an armature which is coupled to the spool or flapper of the valve, whichever is used. It differs from the solenoid in that the moving element (armature) rotates about a pivot point instead of moving in a linear manner as in the solenoid. As such, the torque motor is generally shorter and smaller. Both the torque motor and the solenoid are spring restrained so as not to act as an integrator (for stability reasons) and to provide a means of obtaining a displacement proportional to the input signal. The dynamic response of the driver is closely related to its size. Generally the larger the driver the larger the mass of the moving element and the greater the inductance in the input coils. Both these reduce the dynamic response. Every effort is generally made to reduce the load on the driver as much as possible to minimize its size.

4. Integral Valve-Actuator Combination

As mentioned previously, the actuators may be either of the linear ram type or of the rotary type. The linear ram type actuators may be either single ended or double ended. As the name implies, a single ended actuator is one with a single shaft extending from one end of the actuator; the two working areas of the ram are asymmetrical differing by the area of the shaft. This type of actuator is commonly used in conjunction with a three-way valve in such a manner that the supply pressure is ported into the shaft side of the ram and the control pressure on the opposite side. If the shaft area is one-half the ram area the ram will remain stationary when the valve is in a null position. Examples of these are the Hughes and Minneapolis-Honeywell actuators. (See Table 1C). Of course four-way valves are also used with single ended actuators but the combinations are generally not of the integral type.

CHAPTER 4

COMPILATION OF CHARACTERISTICS OF CURRENTLY MANUFACTURED VALVES

A. Method of Presentation

This section of the report presents the detailed information on the electro-hydraulic servo valves currently being manufactured or under development. Only those valves suitable for aircraft and missile application, or flight table application, are described. Valves suitable for flight table application are included because they generally have very high response and are of interest from a design standpoint. The information concerning these valves was obtained directly from the manufacturers either via their published brochures or directly from company representatives.

The major portion of the information on the different servo valves has been tabulated for easy reference and comparison and is contained in Table 1. In addition a brief discussion of each valve is presented describing its principle of operation, type of compensation, if any, and obvious advantages or disadvantages. A schematic of each valve is included at the end of this section to assist in the description of its principle of operation. These schematics do not necessarily show the actual construction of the valve or subcomponents.

The valves listed in Table 1 are categorized according to the following classes:

(1) Single Stage Valves

(2) Two Stage Valves - Nozzle Flapper First Stage

(3) Two Stage Valves - Spool First Stage

(4) Two Stage Valves - Sliding Plate First Stage

(5) Integral Valve - Actuator Combination

The physical, static, dynamic, and reliability characteristics listed are those which are considered most important to the designer of an airborne hydraulic control system. The physical characteristics (size, weight, and mounting) are important to any airborne application where space and weight are a premium. The mounting information is not tabulated because

TABLE 1A

Single Stage Valves

No	Mfg & Model Series	Over-all Size (in.)	Wt. (lb)	Pressure Range (psi)	Flow Range @ 1000 psi Valve Drop (gpm)	Max Diff Current Input (ma)	Max Quiescent Flow (gpm)	Hysteresis & Threshold (% Max Signal)	Dither Req (cps)	Freq Resp (Flow vs Diff Current) ±3 db Amp Ratio Freq (cps)	Phase (deg)	Freq @ 90° Phase (cps)	Filtration Recommend. (mu)	Life (Hrs or Cycles)	Special Features	Remarks
1a	Bendix B-I (3)	1.1 x 1.1 x 4.5	0.75	2000 & 3000	* 1.9	18	0.08		250				10			* Rated at 667 psi valve drop
1b	Bendix B-II (3)			3500 2000 (nom. 2800)	2.0	18	0.07		315							
1c	Cadillac CG I (2)	1.25 x 1.25 x 1.5		2000	1	11			210							
1d	General Electric GE I	2.5 x 2.75 x 3.5	2.5	100 & 1600	1.12 & 0.75	40	1.12	10					None		Open Center Design	
1e	Midwestern Model 3	3.3 x 3.25 x 4.6	5	200 - 3000		40	0.3			60	40	66			Part. flow force compensated; has some gain compensation	
1f	Midwestern Model 4	2.9 x 2.5 x 4	2.8	200 - 3000	4.3	40	0.2			50	60	60			Gain compensation built in	
1g	Peacock Machine Co. Peacock (4)	2.25 x 2.25 x 5	5	800 - 1500	2	45	0.2						10			Replaced by Moog 902A at Convair

(1) Valve made by Cadillac Gage to Standard Controls Design
(2) Valve made by Cadillac Gage to Sperry's Specifications
(3) Valve made by Bendix for Douglas
(4) Valve made by Peacock Machine for Convair (Pomona)

Open Data Blocks Indicate Information Not Available (See page 19)

TABLE 1B

Two Stage - Nozzle-Flapper First Stage

No	Mfg & Model Series	Over-all Size (in.)	Wt. (lbs)	Pressure Range (psi)	Flow Range @ 1000 psi Valve Drop (gpm)	Max Diff Current Input (ma)	Max Quiescent Flow (gpm)	Hysteresis & Threshold (% Max Signal	Dither Req (cps)	Freq Resp (Flow vs Diff Current) ±3 db Amp Ratio Freq (cps)	Phase (deg)	Freq @ 90° Phase (cps)	Filtration Recommend. (mu)	Life (Hrs or Cycles)	Special Features	Remarks
2a	Bell SV-6C	1.4 x 2.6 x 3	1.0	Up to 3000	6.35	10	0.25		None	*60 **170	90	*60 **195	10	1000 hr	Torque Motor coil not in the oil	*Min Fig Most valves have much higher freq response **Most recent version with modified torque motor. Same change in all lines planned.
2a	Bell SV-7C		2	Up to 3000	12.5	10			None				10	1000 hr	Torque Motor coil not in the oil	
2a	Bell SV-9C	1.4 x 2.6 x 3.3	1.0	Up to 3000	1.9	6	0.18		None	*60	90	*60	10	1000 hr	Torque Motor coil not in the oil	
2b	Cadillac PC-2 (1)	1.9 x 1.9 x 3.7	1.1	500 - 3000	4.0	10	0.15	Hys - 2 Thr - 1	None	*70	*90	*70	10			*Spool Pos. vs Diff Current
2c	Cadillac PC-2 (Pressure Control) (1)	2.0 x 2.0 x 3.2	1.3	500 - 3000	7.7	8	0.25	Hys - 2 Thr - 1	None	*100	*30	*180	10			*Zero load flow (Diff pressure vs Diff current)
2d	Moog 500	1.75 x 2.5 x 3.06	0.8	1000 - 3000	0.5 to 8.0	2.0 - 40 Usually 8.0	0.1 + 2% of rated flow	Hys - 7.5 Thr - 1.3	None	62	70	92	10			Gain compensation can be built into valves
2d	Moog 900	1.75 x 2.5 x 3.06	0.8	1000 - 3000	0.2 to 5.0	1.0 - 40 Usually 8.0	0.1 + 2% of rated flow	Hys - 7.5 Thr - 1.3	None	170	80	193	10			
2d	Moog 1400	1.75 x 2.5 x 3.5	0.85	1000 - 3000	0.5 to 10.0	2.0 - 40 Usually 8.0	0.1 + 2% of rated flow	Hys - 7.5 Thr - 1.3	None	62	70	92	10			
2d	Moog 2000	1.75 x 1.895 x 3.06	0.70	1000 - 3000	0.5 to 8.0	2.0 - 40	0.1 + 2% of rated flow	Hys - 5.0 Thr - 1.3	None	59	70	100	10		Has dry solenoid	
2e	Pegasus Model 120-B	1.9 x 1.5 x 9	1.5	to 3000	2.1	30	0.2	3	None	*185	*100	*135	5		Solenoid coils not in the oil	*Spool Pos. vs Amp Voltage Input

TABLE 1C

Two Stage - Spool First Stage

No	Mfg & Model Series	Over-all Size (in.)	Wt. (lbs)	Pressure Range (psi)	Flow Range @ 1000 psi Valve Drop (gpm)	Max Diff Current Input (ma)	Max Quiescent Flow (gpm)	Hysteresis & Threshold (% Max Signal)	Dither Req (cps)	Freq Resp (Flow vs Diff Current) ±3 db Amp Ratio Freq (cps)	Phase (deg)	Freq @ 90° Phase (cps)	Filtration Recommend. (mu)	Life (Hrs or Cycles)	Special Features	Remarks
3a	Bertea (Pt. Mugu)	(approx) 12 x 6 x 6	20	3000	15	50			1st Stage spool spun @ 1800 rpm	1st Stage 1700 2nd Stage 600			10		Voice coil drive	Not for airborne application
3b	Cadillac CG II (2)	(approx) 3 x 3 x 4	2												Concentric 1st & 2nd Stage spool	
3c	Drayer Hanson	1.75 x 2.0 x 3.5	1.0	to 5000	2 to 20	20			None					2 million cycles		
3d	Hydraulic Controls DS 2	2.6 x 3.8 x 7.25	5.75	3000	5.5	40	0.2			100						
3e	N.A. Time Modulated		2	3000	5.0	40	0.347		180	*50	*90	*50	10 plus magnetic			Replaced by Moog 534 * Spool Pos. vs Diff Current
3f	Raytheon Model MRTG65-1	1.9 x 2.25 x 5	1.25	to 3500	to 5	15	0.040		400	*35	*70	*45	20		Chip shearing principle for clearing 1st stage	
3g	Sanders Model 10	1.3 x 2.5 x 5.5	1.75	to 3000	to 12	15	0.026			50			20 - 25	4 million cycles	Chip shearing principle for clearing 1st stage	

TABLE 1D

Two Stage - Sliding Plate First Stage

No	Mfg & Model Series	Over-all Size (in.)	Wt. (lbs)	Pressure Range (psi)	Flow Range @ 1000 psi Valve Drop (gpm)	Max Diff Current Input (ma)	Max Quiescent Flow (gpm)	Hysteresis & Threshold (% Max Signal)	Dither Req (cps)	Freq Resp (Flow vs Diff Current) +3 db Amp Ratio Freq (cps)	Phase (deg)	Freq @ 90° Phase (cps)	Filtration Recommend. (mu)	Life (Hrs or Cycles)	Special Features	Remarks
4	M.I.T.	2.25 x 4.9 x 1.9	2.75	3000	10	30	0.3		None				10		Hole and plug type of construction	

TABLE 1E

Integral Valve - Actuator Combination

No	Mfg & Model Series	Over-all Size (in.)	Wt. (lbs)	Pressure Range (psi)	Flow Range @ 1000 psi Valve Drop (gpm)	Max Diff Current Input (ma)	Max Quiescent Flow (gpm)	Hysteresis & Threshold (% Max Signal)	Dither Req (cps)	Freq (cps)	Phase (deg)	Freq @ 90° Phase (cps)	Filtration Recommend. (mu)	Life (Hrs or Cycles)	Special Features	Remarks
5a	Hughes Actuator	1.4 x 2.75	2	2000	0.125	14	0.016		150				25 (use specially processed oil)	100 hr	Complete Actuator Package	
5b	Minn. Honeywell MMG 36A1 Actuator		2.9	3000		16	0.10		400	*77	*105		10		Complete Actuator Package	*Actuator Response w/load

all valves except those built by Midwestern Geophysical Laboratory can be manifolded directly to the actuator.

The static characteristics (pressure range, flow range, electrical input, quiescent flow, hysteresis, threshold, and dither requirement) provide information necessary for the servo design. Many of the valve manufacturers specify a range of flows for a particular series valve. To change the flow capacity of the valve, they vary the size of the valve ports. Thus in Table 1, the maximum flow range at 1000 psi valve drop (supply pressure minus load pressure plus drain pressure) specifies the various flows that can be obtained from a particular valve by varying the size of the valve ports. For systems which use a 3000 psi supply pressure, these figures can be quickly converted into maximum load horsepower by multiplying by 1.16. The quiescent flow ratings listed in Table 1 include the steady state flow of the first stage in addition to the leakage of the second stage for two stage valves.

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 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 the ± 3 db amplitude ratio point and the frequency at the 90° phase shift point have been tabulated.

The reliability characteristics are covered by the recommended filtration and minimum life estimate. Important unique features are stated in the special features column. Pertinent remarks concerning the performance data are included in the remarks column. The significance of an asterisk in any column in the Tables is also explained in the remarks column.

Some of the reasons for having the open data blocks are that the people contacted did not know or could not furnish this information, it had never been measured or the spcification had not been established, the information appeared unreliable or the information was of such a form to be unsuitable for tabulation, or the information was too highly classified for release.

B. Valve Descriptions

1. Single Stage Valves

a. Bendix Aviation Corp. B-I Valve

The Bendix Aviation Corp. (Pacific Division) manufactures the B-I servo valve to Douglas Aircraft Co. specifications. This valve uses a single spool driven by two opposed solenoids (See Fig. 4). The solenoids are spring-restrained making the spool position proportional to differential current. The spool is flow force compensated to reduce the driving force needed from the solenoids.

b. Bendix Aviation Corp. B-II Valve

The Bendix Aviation Corporation manufactures the B-II valve for Douglas Aircraft. This is a single stage valve using dual spools driven by a torque motor and is similar to the Midwestern valve shown in Fig. 5. The spools are flow force compensated and the arrangement in which a spool is attached to each end of the torque motor makes this valve insensitive to transverse accelerations.

c. Cadillac Gage Co. CG-I Valve

The CG-I servo valve is manufactured by the Cadillac Gage Co. to Sperry Gyroscope Co. specifications. This valve uses a single spool driven by a torque motor. Thus the spool position is proportional to differential current input. Principally this valve is similar to the B-I valve shown in Fig. 4.

d. General Electric Co. GE-I Valve

The General Electric Co. GE-I servo valve is a completely open center, spool type valve driven by a Midwestern Geophysical torque motor. In this open center design the flow through the valve is always constant. Whenever the spool is displaced from its neutral position, flow is shunted through the load to provide the controlling force. This open center design has the disadvantage of high quiescent power consumption. No schematic is available for this valve.

e. Midwestern Geophysical Model 3 Valve

The Midwestern Geophysical Model 3 servo valve is a dual spool, single stage valve, driven by a torque motor. (See Fig. 5). The spools are flow force compensated to reduce the driving force needed from the torque motor. The valve also exhibits gain compensation to a partial extent which indicates

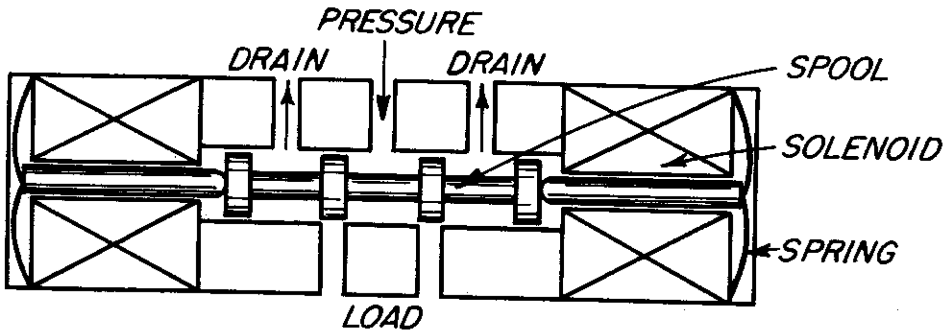


Fig. 4 Schematic of Bendix B-I Valve

the flow force compensation is not complete. External porting of this valve is for 3/8 in. tubing using an Ermeto type fitting at the port. The valve uses all steel construction.

f. Midwestern Geophysical Model 4 Valve

The Midwestern Geophysical Model servo valve is a dual spool, single stage valve, driven by a torque motor. (See Fig. 6). The spools are not flow force compensated; the valve is gain compensated. External porting of this valve is through use of AND 10050-5 ports for 5/16 in. tubing. The valve uses all steel construction.

g. Peacock Machine Co. Valve

The valve made by Peacock Machine Co. to Convair (Pomona) specifications is a single stage, single spool valve, driven by opposed type solenoids. It is similar in configuration to the B-I valve of Fig. 4. The spool is not flow force compensated; the solenoids are large and heavy and the valve has limited response. For these reasons the Convair designers consider this valve obsolete for their applications and have replaced it.

2. Two Stage - Nozzle Flapper First Stage

a. Bell Aircraft Company Valves

The Bell Aircraft Co. servo valves (SV-6C, SV-7C, SV-9C) use a double nozzle-flapper first stage driving a spring-restrained

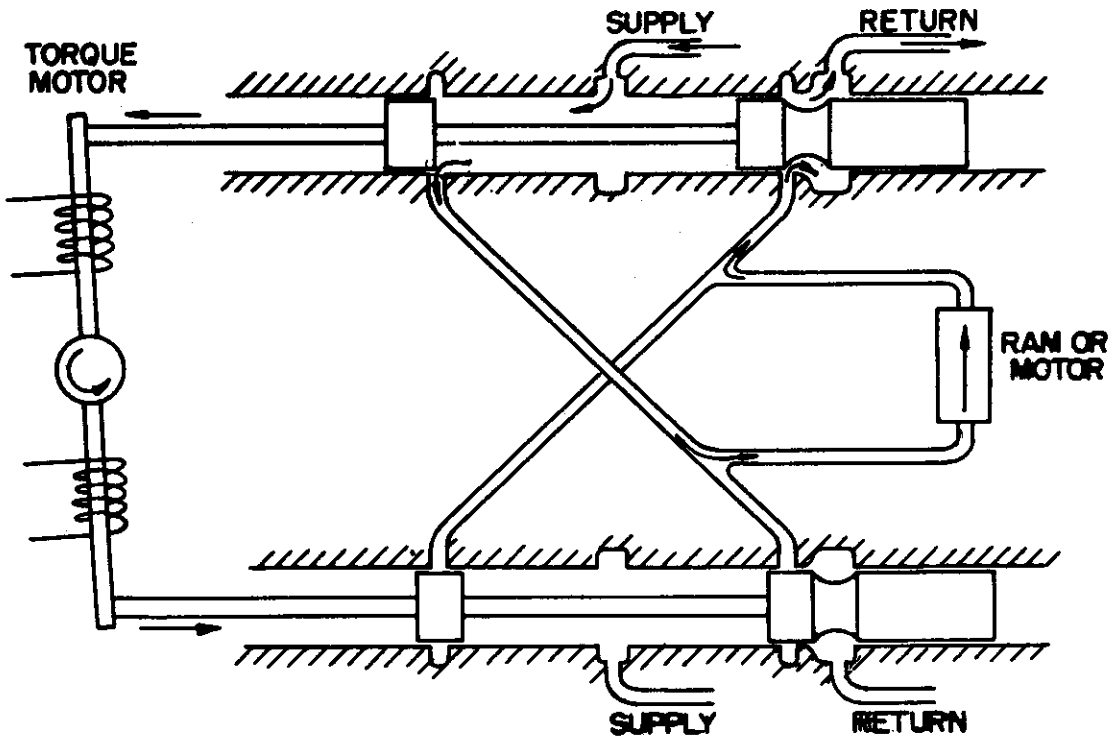


Fig. 5 - Schematic of Midwestern Model 3 Valve

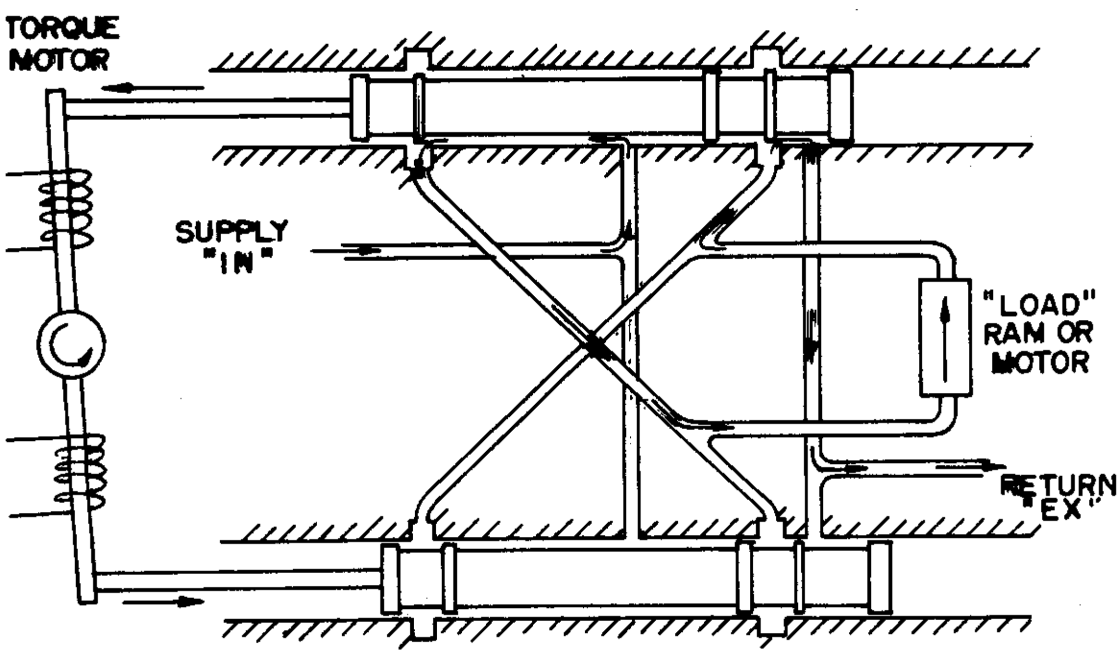


Fig. 6 - Schematic of Midwestern Model 4 Valve

spool type second stage. (See Fig. 7) The double nozzle-flapper assembly produces a differential pressure proportional to the differential current input. This pressure produces a force on the second stage spool which is balanced by the compression force of the restraining springs thus giving a spool position proportional to differential current. A unique feature of the Bell valves is the diaphragm seal between the flapper-nozzle and the torque motor armature and coils. In this way the torque motor is kept out of the fluid and thus magnetic particle difficulties are eliminated. An integral 800 mesh strainer is used to filter the oil supplying the first stage.

b. Cadillac Gage Company FC-2 Valve

The Cadillac Gage Co. FC-2 servo valve uses a single nozzle flapper first stage driving a spool type second stage. (See Fig. 8). The pressure from the first stage is applied to an area A at one end of the spool and supply pressure is applied to an area one-half A at the other end of the spool. Feedback is accomplished by means of a lever and spring arrangement between the spool and the flapper. The feedback, by providing a force proportional to spool position, closes the loop making a positional servo with the spool following the electrical differential current input. The single nozzle design has the advantage of being less susceptible to jamming than the double nozzle because the flapper can move further away from the nozzle to clear itself of particles. The first stage is protected by integral 40 micron sintered bronze filters and also by magnetic filters. The latter do not remove all the magnetic particles.

c. Cadillac Gage Company PC-2 Valve

The Cadillac Gage Co. PC-2 pressure control servo valve uses a double nozzle-flapper first stage driving the two freely floating pistons of the second stage. (See Fig. 9). This valve is unique in that it controls the load pressure rather than the load flow. This pressure control is accomplished through the design that uses a pressure proportional to, and derived from the load pressure to balance the second stage spools against the driving pressure (proportional to differential current) from the first stage. This balancing pressure is built up across the fixed orifice, which is situated in the passage connecting the dead ends of the spool chambers, by the flow provided by the load pressure through the hollow spools. The first stage is protected by 40 micron sintered bronze filters and also

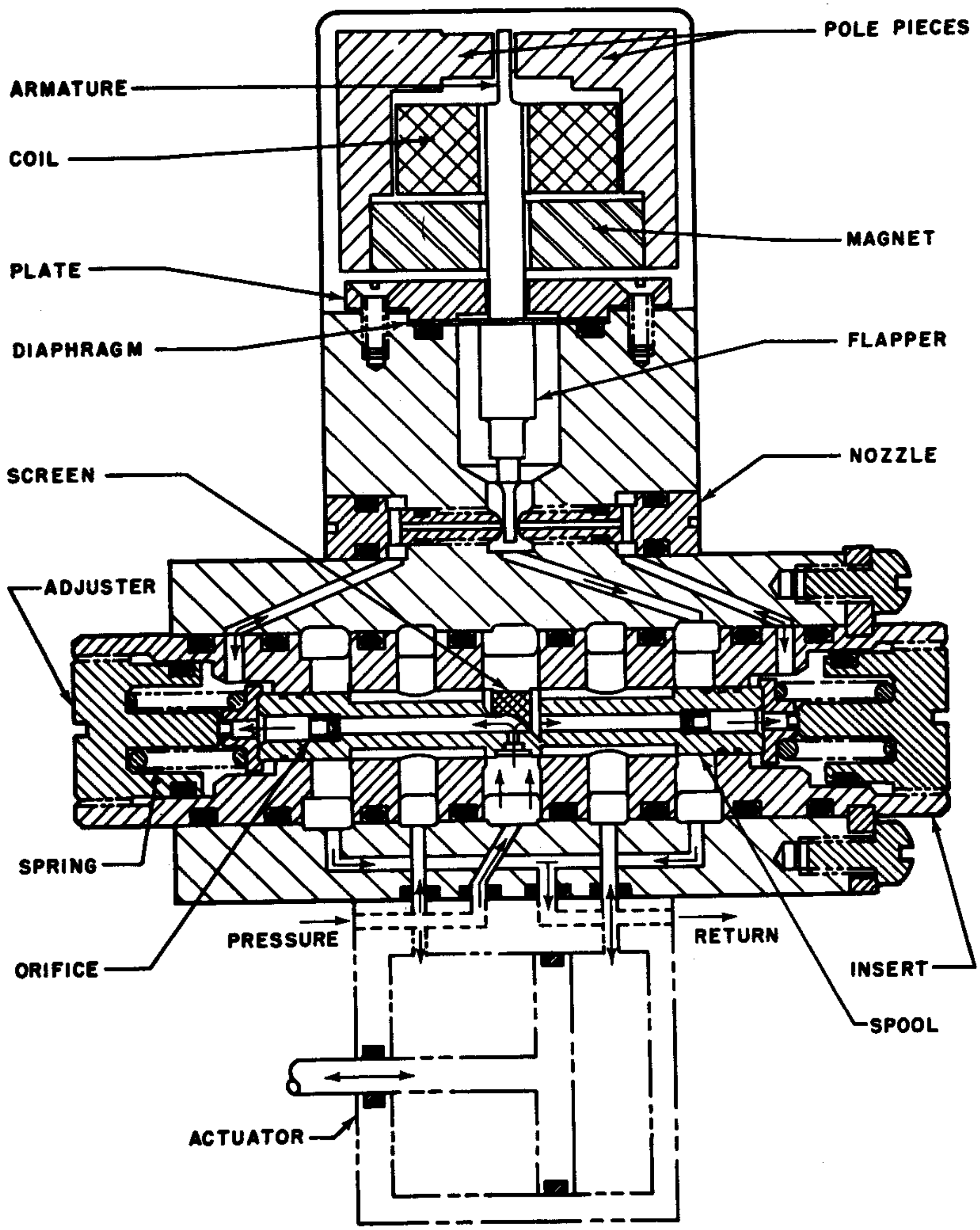


Fig. 7 - Schematic of Bell Valve

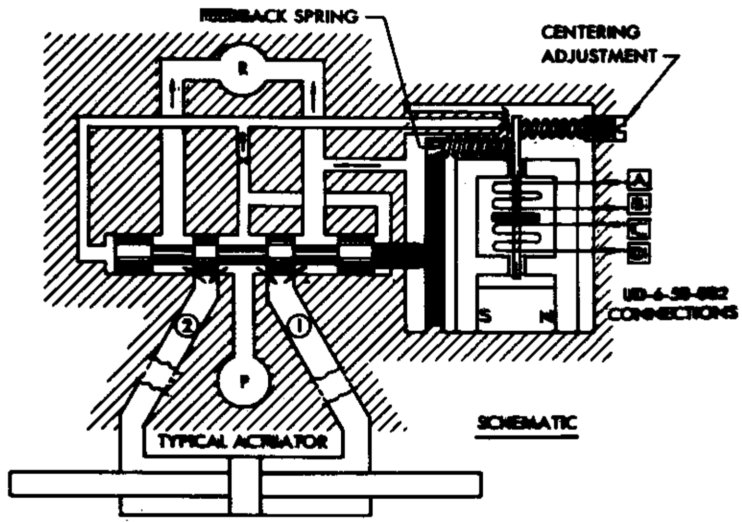


Fig. 8 - Schematic of Cadillac FC-2 Valve

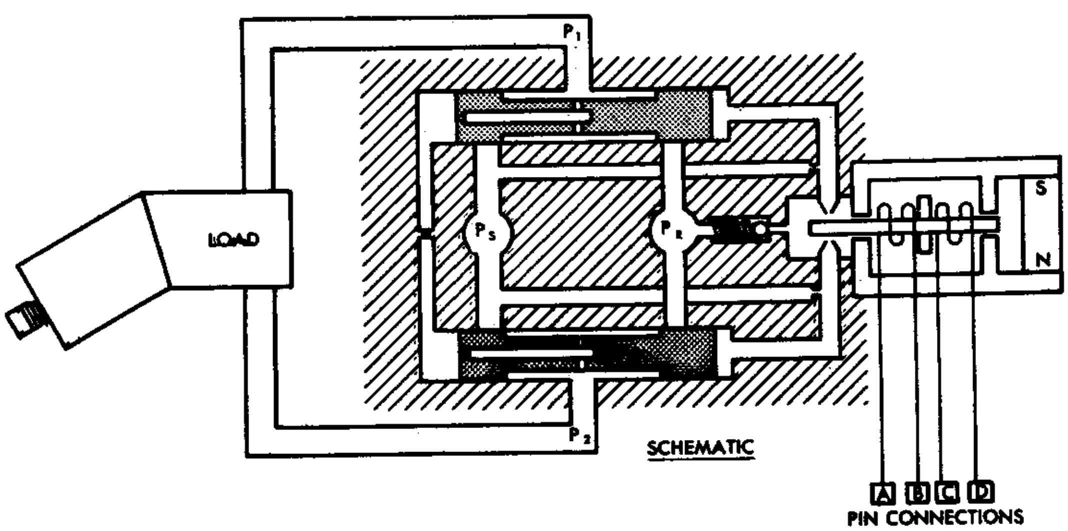


Fig. 9 - Schematic of Cadillac PC-2 Valve

magnetic filters. The latter do not remove all the magnetic particles.

d. Moog Valve Company Valves

The Moog Valve Co. servo valves (500, 900, 1400, series) use a double nozzle-flapper first stage driving a spring-restrained spool type second stage (See Fig. 10). The double nozzle-flapper assembly produces a differential pressure proportional to the differential current input. This pressure produces a force on the second stage spool which is balanced by the compression of the restraining springs, thus resulting in a spool position proportional to differential current. The second stage spool is 1/4 in. in diameter in the 500 and 1400 series and 3/16 in. in diameter in the 900 series, a fact which probably accounts for the higher response of the 900 series valves. The first stage of the valves is protected by integral 40 micron sintered bronze filters and also by magnetic filters. The magnetic filters remove only approximately 50% of the magnetic particles. The Moog Valve Co. can build gain compensation into their valves by proper proportioning of the spring constants and driving force with respect to the flow forces on the second stage spool.

A series 2000 valve has recently been introduced which features dry torque motor coils. This valve is shown in Fig. 11. Its principle of operation is similar to other Moog valves.

e. Pegasus Laboratories Model 120-B

The Pegasus Laboratories Model 120-B servo valve is a two stage valve using a single nozzle-flapper first stage driving a spool type second stage. The design of the valve is unique in that the nozzle is attached to the spool and is thus free to follow the flapper movement. (See Fig. 12) The flapper is driven by a solenoid type driver which is sealed off from the nozzle flapper chamber by a bellows. The controlled pressure from the nozzle is fed to one end of the second stage spool. The other end of the spool receives a pressure, derived from two fixed orifices in series, which is equal to one-half the supply pressure. A self balancing bridge configuration is thus produced with the spool position following the flapper movement. The sealed solenoid eliminates magnetic particle difficulties. Pegasus Laboratories

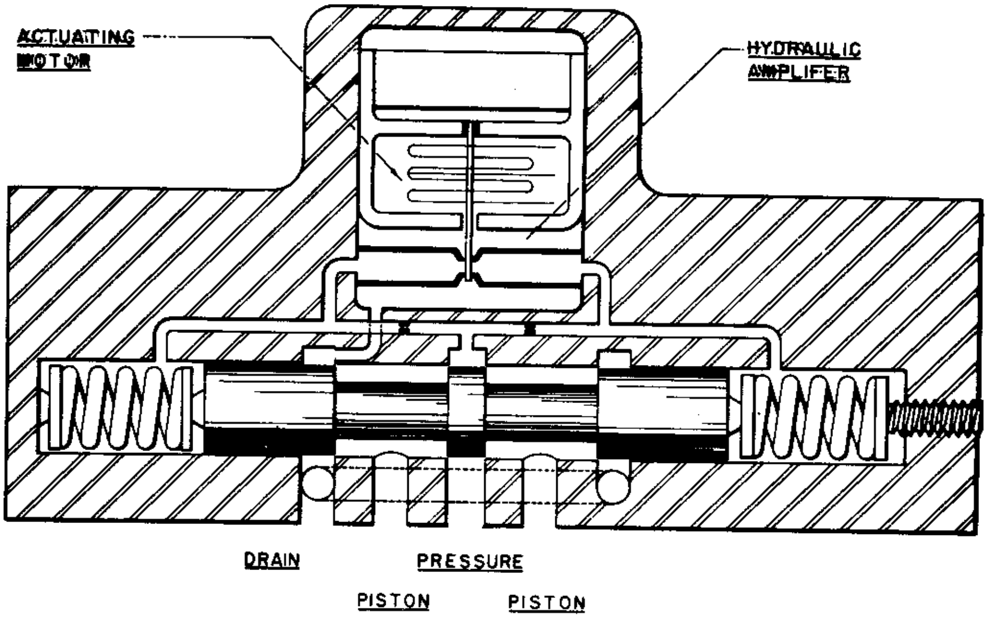


Fig. 10 - Schematic of Moog Valve (Models 500, 900, 1400)

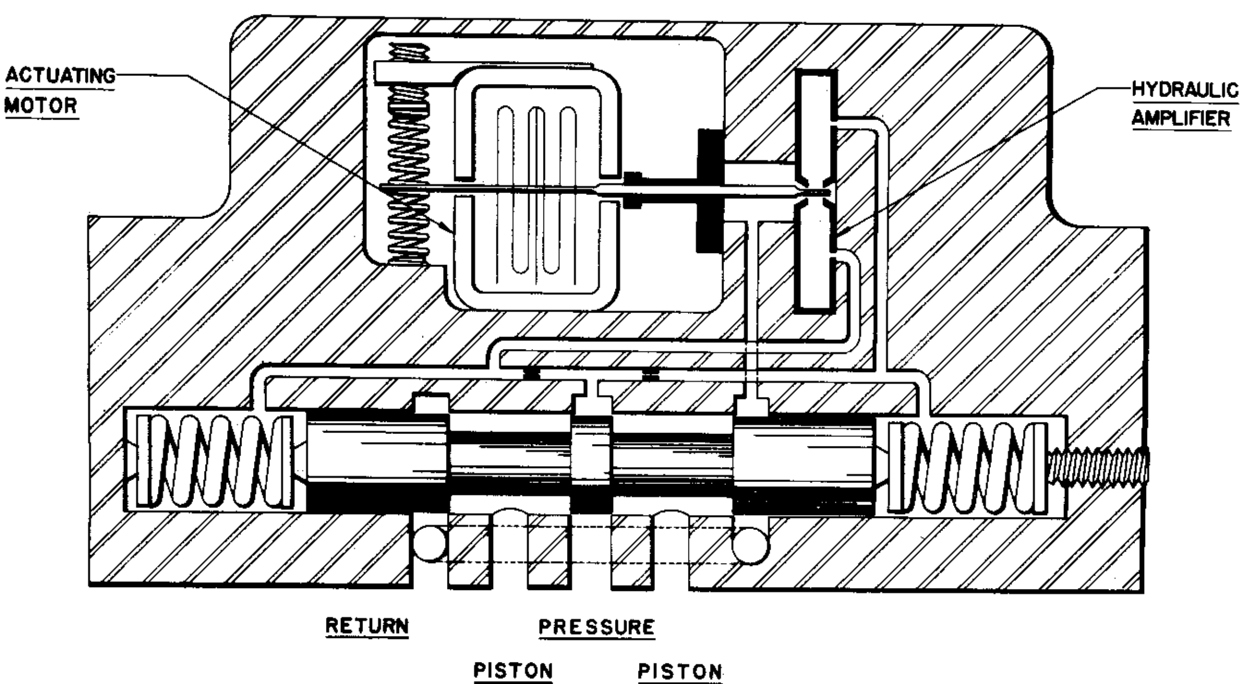


Fig. 11 - Schematic of Moog Model 2000 Valve

makes an additional spool type power stage model 127 with a built-in linear differential transformer pickoff (Linearsyn), the signal from which is used to close the loop. This power stage is about the same size and weight as the 120-B unit and has a flow rating of 10 gpm at 125 psi valve drop. The pressure rating on the 127 unit is 500 psi.

3. Two Stage Valves - Spool First Stage

a. Bertea Products Point Mugu Valve

This valve is made by Bertea Products to the specifications of the Naval Air Missile Test Center at Point Mugu and is for a flight table application. The spool of the first stage is driven by a voice coil. (See Fig. 13). The second stage spool has a linear differential transformer (Sheavitz) pickoff providing an electrical signal proportional to position and this signal is used to close the loop. The first stage spool also has a Sheavitz pickoff measuring its position and this signal is considered a measure of the velocity of the second stage spool and is used for stabilizing purposes. Both spools are flow force compensated. The first stage spool is rotated at about 1800 rpm by the turbine action of the leakage oil. This rotary dither eliminates stiction in the first stage.

b. Cadillac Gage Co. CG-II Valve

The CG-II valve is made by Cadillac Gage Co. to Sperry Gyroscope Co. specifications. This valve uses concentric first and second stage spools with the first stage spool inside the second stage spool. (See Fig. 14). There is a hydraulic follow-up between stages, making the second stage follow the position of the first stage. The first stage is positioned by a torque motor. A disadvantage of this valve is that the first stage spool has to move the same distance as the second stage spool necessitating a long stroke torque motor with an attendant reduction in response capability.

c. Drayer-Hanson Co. Valve

The Drayer-Hanson Co. valve uses a torque motor driven, spool type first stage driving a spool type second stage (See Fig. 15). The first stage valve may have either three-way or four-way action depending on which external ports are used.

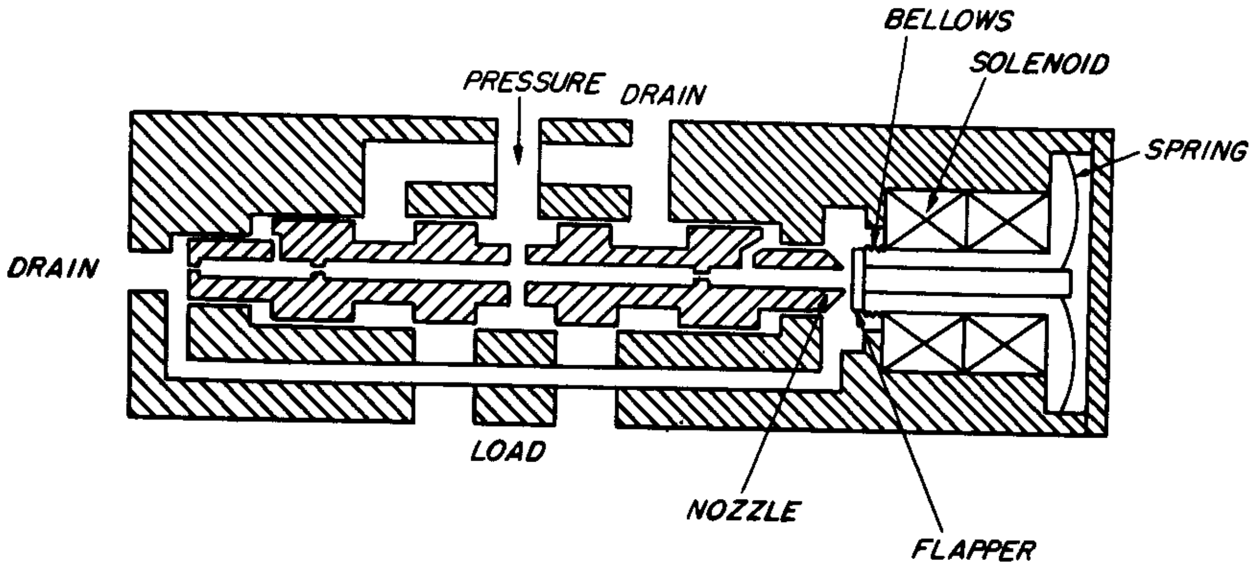


Fig. 12 - Schematic of Pegasus 120-B Valve

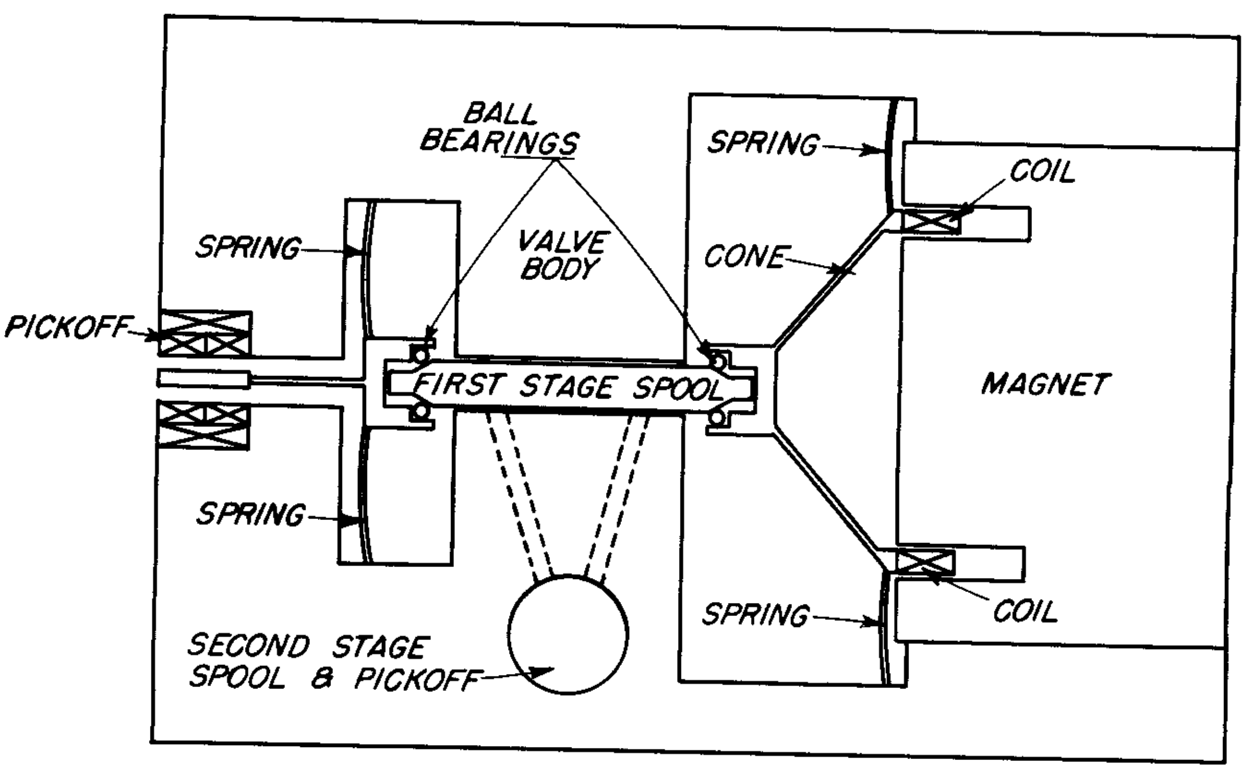


Fig. 13 - Schematic of Bertea (Point Mugu) Valve

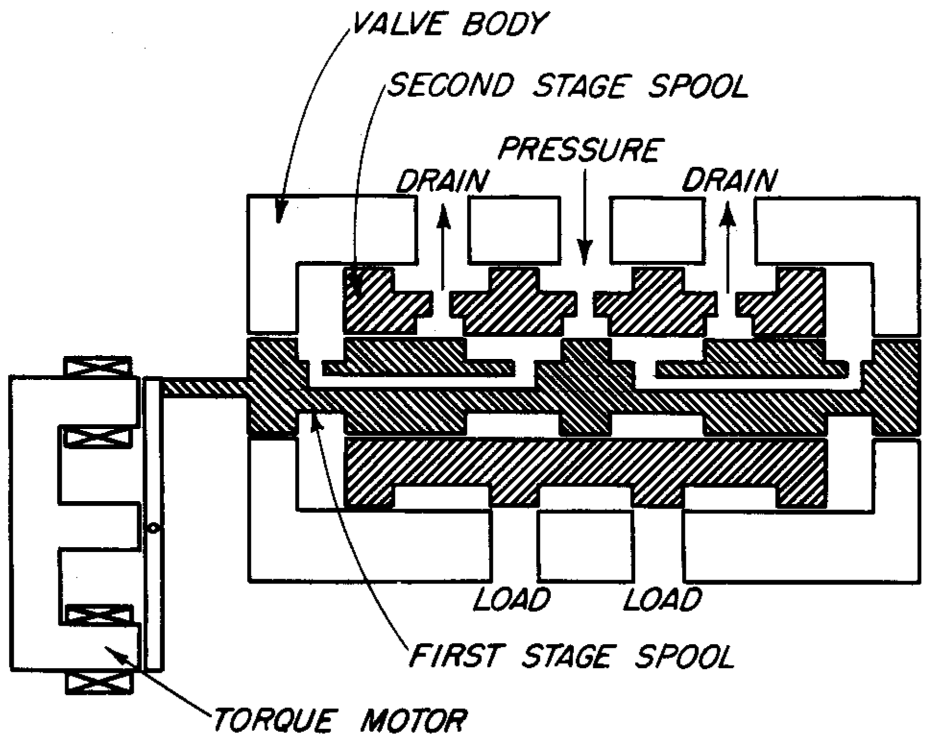


Fig. 14 - Schematic of Cadillac Gage CG-II Valve

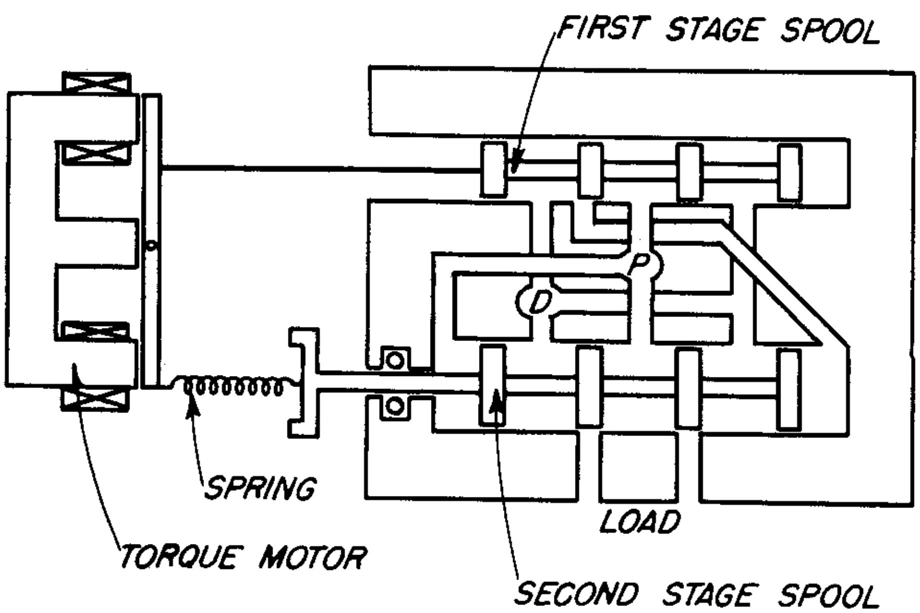


Fig. 15 - Schematic of Drayer-Hanson Valve

In the two stage valve, the first stage acts as a three-way valve supplying oil to one end of the second stage spool. The resultant driving force is balanced by supply pressure applied to a half area on the opposite end of the spool. Internal feedback is accomplished through a spring between the second stage spool and the torque motor. The spool position is fed back through the spring as a torque opposing the torque produced by the differential current; this feedback tends to recenter the torque motor and the first stage spool. In this manner the position of the second stage spool is made to follow the differential current input. The force type feedback is similar to that used in the Cadillac Gage Co. FC-2 valve. The valve is so constructed that the torque motor, first stage, or second stage can be used separately. The torque motor and first stage are used as a single stage four-way valve in some applications.

d. Hydraulic Controls DS-2 Servo Valve

The Hydraulic Controls DS-2 Servo valve uses a torque motor driven, spool type first stage controlling a spool type second stage (See Fig. 16). A linear differential transformer pickoff provides an electrical signal proportional to the second stage spool position, which is used to close the loop. This signal is subtracted within the amplifier from the electrical input signal and the difference is amplified and used to drive the torque motor. In this way the second stage spool position is made proportional to the electrical input to the amplifier. This type of feedback allows the use of a short stroke, high response torque motor. A disadvantage is that additional electronics are necessary for the electrical feedback, e. g., oscillator, AC amplifier, and demodulator.

e. North American Aviation Time Modulated Valve

North American Aviation's Time Modulated Valve is a two-stage type with an electromechanically driven, spool type first stage and a spring restrained, spool type second stage (See Fig. 17). The first stage spool and driver combination is not spring restrained, a unique feature of this valve. The spool is stroked continuously at a frequency of 133 cps. The input current to the electromechanical driver is a square wave with the positive portion energizing one coil and the negative portion energizing the other coil of the driver. Control is accomplished by changing the width of the positive and negative portions of the wave which results in a shift of the average position of the first stage spool.

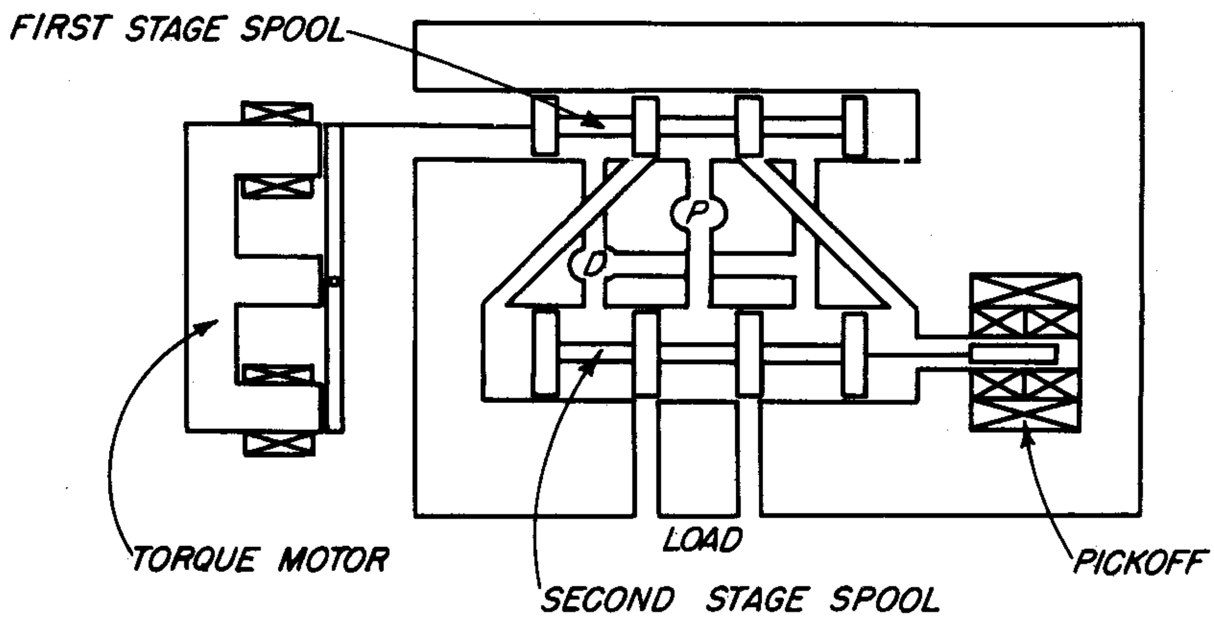


Fig. 16 - Schematic of Hydraulic Controls DS-2 Valve

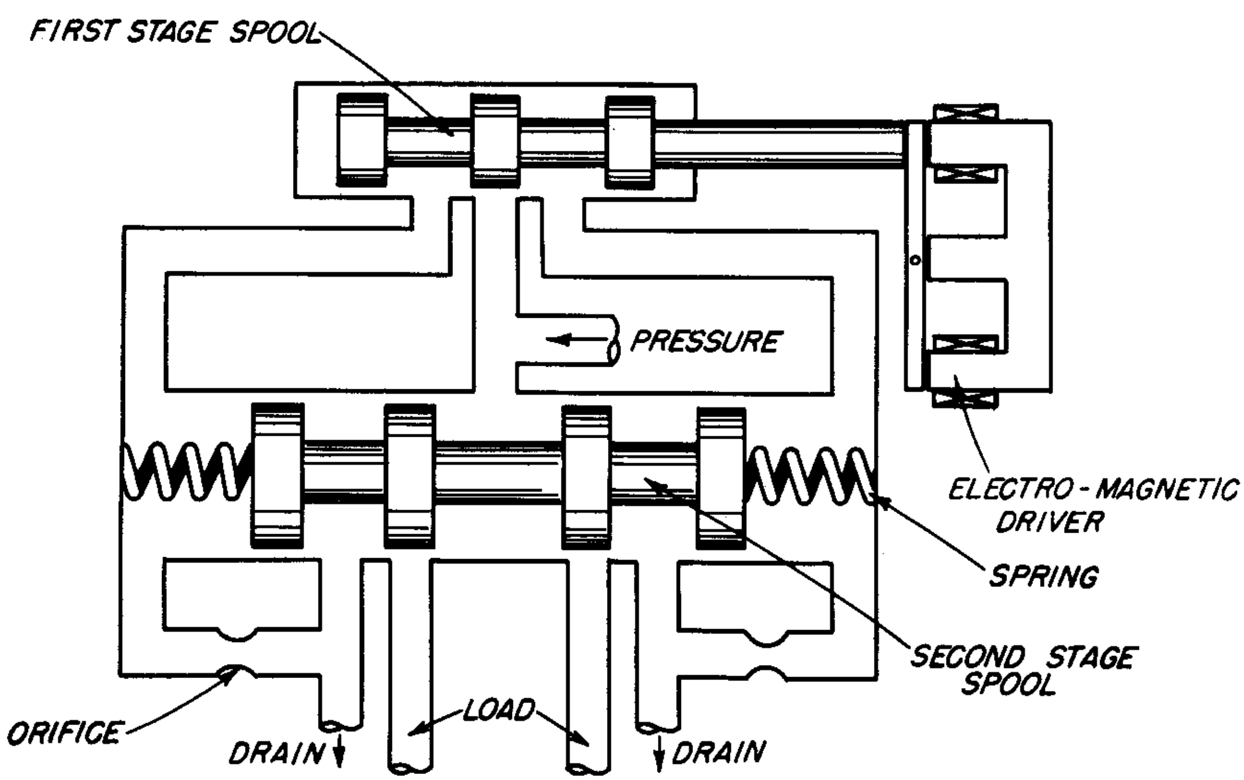


Fig. 17 - Schematic of North American Valve

This shift of average position of the first stage spool produces an average flow from the first stage which is transformed into an average pressure unbalance across the spring-restrained second stage by the orifices connected to drain. The pressure unbalance produces an average displacement in the second stage spool position which opens one metering orifice longer than the other metering orifice. There is no feedback from the second stage spool and its position depends upon the average pressure unbalance and the centering springs. This valve has been replaced because of the complexity of the electronics needed to operate it .

f. Raytheon Manufacturing Co. Model MRTG 65-1 Valve

The Raytheon Manufacturing Co. Model MRTG 65-1 servo valve uses a pair of opposed solenoids to drive the spool type first stage which drives the spool type second stage. (See Fig. 18). The principle of operation of the valve is identical to that of the Sanders Associates valve. Chip shearing in the first stage is also a feature of the Raytheon valve. The first stage of this valve, using a four-way type spool, is used as a single stage valve in some low power applications.

g. Sanders Associates Model 10 Valve

The Sanders Associates Model 10 servo valve comprises a torque motor driven, spool type first stage driving a spool type second stage. (See Fig. 19). The first stage spool is a three-way type and ports oil to one end of the second stage spool, creating a driving force. The second stage spool is connected to the sliding sleeve of the first stage by a feedback lever. Supply pressure is continuously impressed upon the area on the opposite end of the sleeve. The resulting force, operating in conjunction with that from a bias piston (not shown) acting on the second stage spool, opposes the driving force and forces the second stage spool position to follow the differential current input. This arrangement also provides a stroke amplification between first and second stages. A chip shearing arrangement which utilizes the forces moving the second stage spool and first stage sleeve to automatically free the first stage spool if it becomes jammed is also provided in the valve. This is accomplished by limiting the travel of the first stage spool by mechanical stops while permitting the sleeve to move further; thus enabling the valve to clear itself

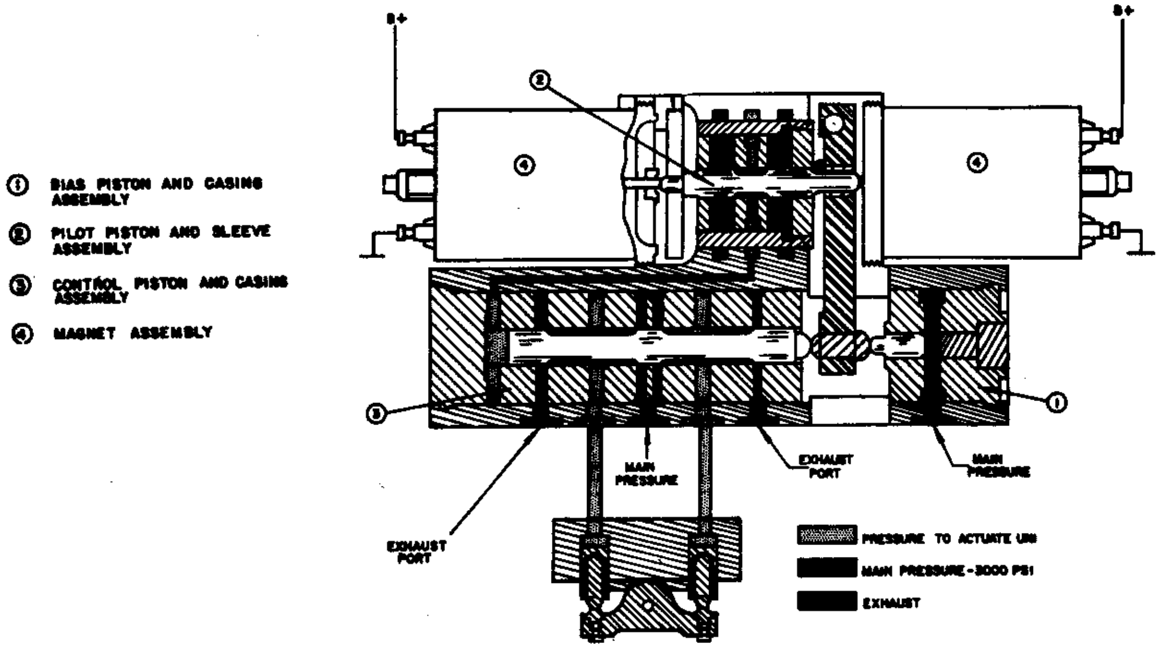


Fig. 18 - Schematic of Raytheon MRTG 65-1 Valve

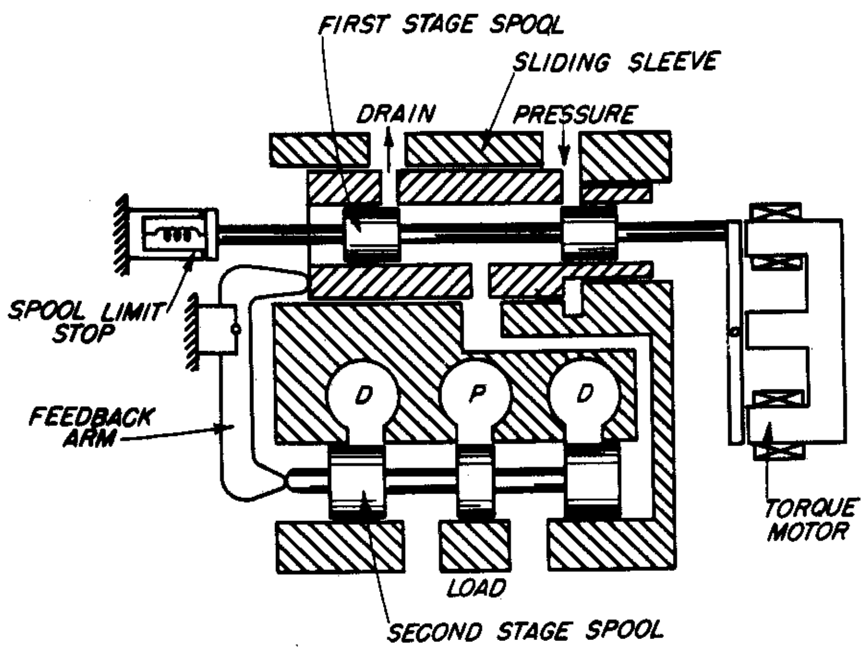


Fig. 19 - Schematic of Sanders Model 10 Valve

of an obstruction.

4. Two Stage Valves - Sliding Plate First Stage MIT DACL Model 10HP-A Valve

The Dynamic Analysis and Control Laboratory of MIT has designed a valve using a suspended sliding plate first stage driving a spool type second stage (See Fig. 20). Both stages use the hole and plug porting arrangement. In this type of porting construction, the two parts, the movable valve plate or spool (hereafter called movable member) and the fixed valve body, are clamped together and two holes are drilled, bored and finish lapped through both parts. The movable member is removed from the body and a groove of width less than the hole diameters is milled in the movable member between the two holes. Bushings are then pressed into the holes in the movable member and finished off flush with the surface. The edges of these bushings which are exposed by the longitudinal groove, together with the corresponding portions of the holes in the valve body, form the four metering orifices of a four-way valve.

In this particular valve the first stage is a separate unit which acts as a positional servo. A force feedback is accomplished through a spring connecting the ram and torque motor armature. In this manner a force proportional to ram position balances the force due to differential current and thus recenters the torque motor armature and the first stage sliding plate. The second stage spool is connected to the ram by a connecting rod which extends through the ram. The valve is an experimental model built to test and prove principles and would have to be redesigned to meet the size and weight requirements of missiles and aircraft.

5. Integral Valve Actuator Combinations

a. Hughes Aircraft Co. Actuator

Hughes Aircraft Co. integral valve-actuator combination consists of a servo valve, feedback potentiometer and single ended ram. (See Fig. 21). The valve is of the three-way type and consists of two spools driven by a torque motor. The actuator is positioned by applying control pressure from the valve to the plain end of the actuator and supply pressure to the shaft end of the actuator. Clearances are so

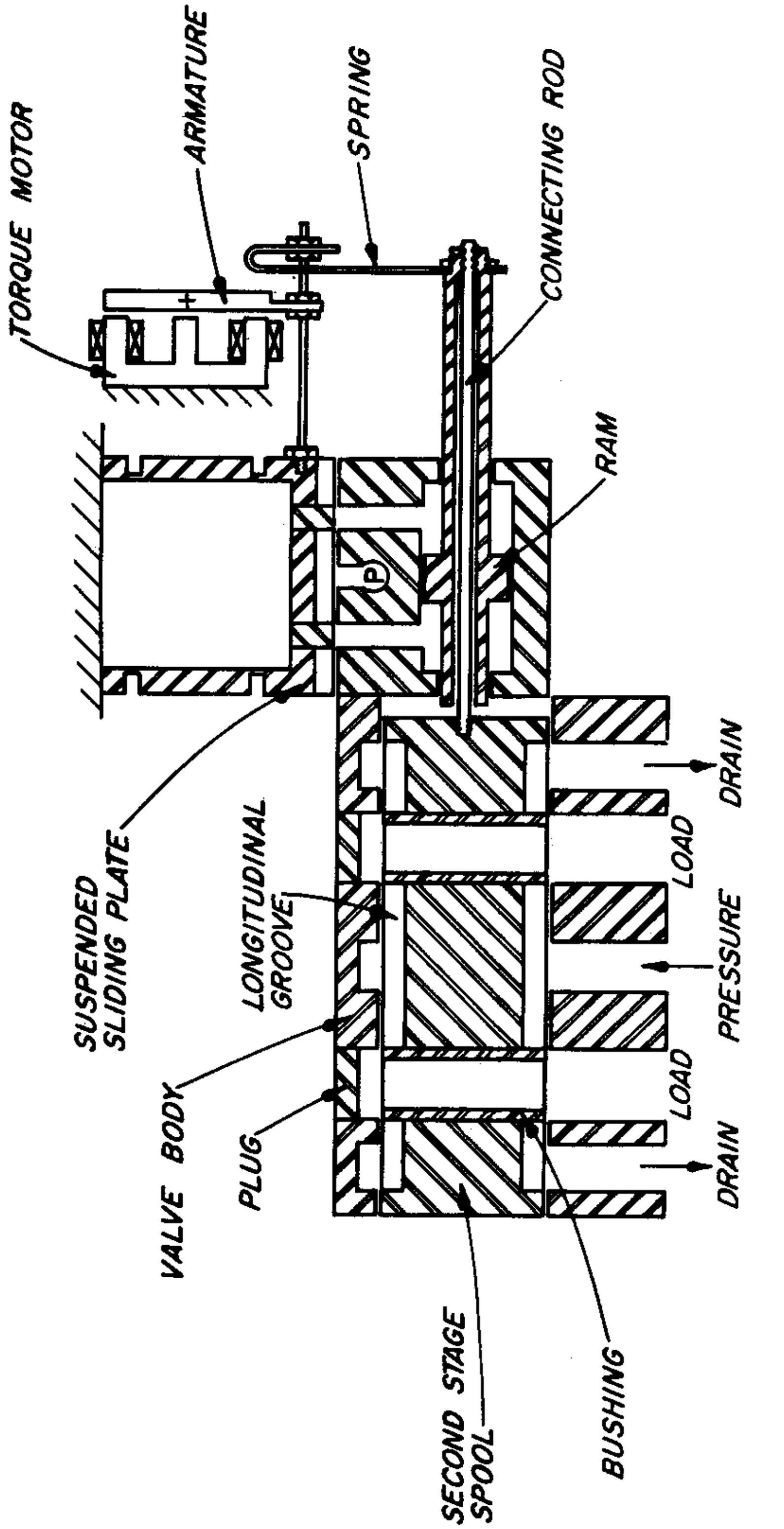


Fig. 20 - Schematic of MIT 10 HP-A Control Valve

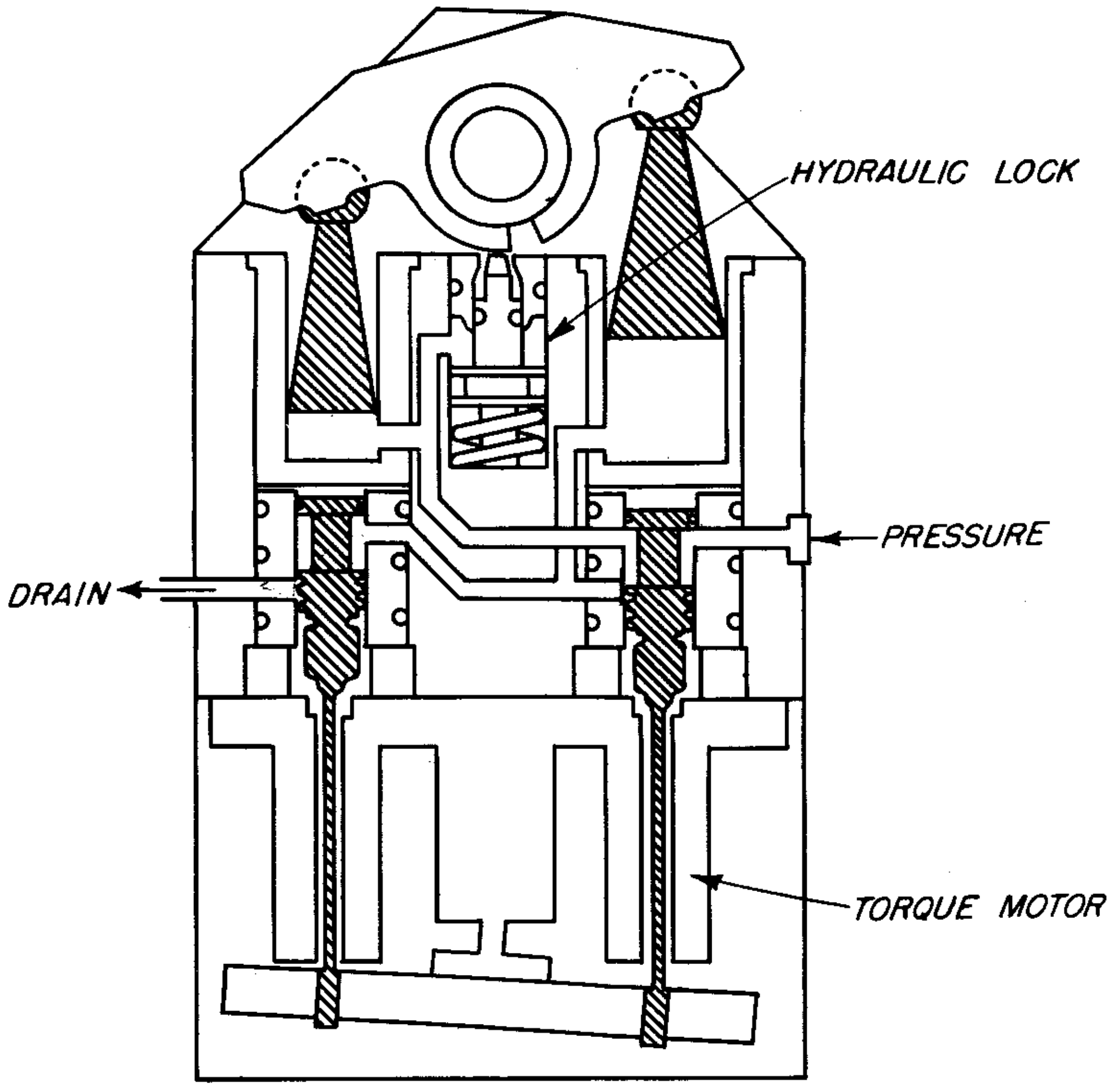


Fig. 21 - Schematic of Hughes Actuator

small that MIL-0-5606 oil has to be specially processed through Fullers earth type filters to keep it from gumming up and jamming the valve. This type filter removes the gum and gelatinous material from the oil in addition to the solid hard particles that regular filters remove.

b. Minneapolis-Honeywell Actuators

The Minneapolis-Honeywell XMG 36A1 and XMG 42A1 actuators use the same type servo valve. This valve is a single stage two spool three-way action valve driven by a torque motor. Its operation is similar to that of the Hughes Aircraft actuator shown in Fig. 21. In this three-way, two spool type construction, the spool positions are adjustable axially thus making it unnecessary to hydraulically check laps while grinding.

C. Discussion of Manufacturing Procedures and Difficulties

The electro-hydraulic servo valve industry is quite new, and this fact presents many unknowns and new problems to the manufacturer. In addition, the valves themselves are sometimes unpredictable and present problems to which the solutions are not obvious or even generally applicable. The following paragraphs discuss some of these problems and their solutions.

The first major problem in servo valve manufacturing is the difficulty in obtaining close tolerances and the precision to which parts must be made. Diametral clearances on spools are of the order of 0.0001 to 0.0002 in. and the spool and sleeve must be made straight and true to hold this clearance. Laps on metering edges are of the same order of magnitude which makes it necessary to hydraulically test the spool and sleeve assembly to obtain information for final grinding. In almost all cases the ports are cut in the sleeve and the spool is ground to match it.

In some cases the servo valve is made with two spools and sleeves to perform the valving action, a type of construction that eliminates one critical axial dimension. (As mentioned previously, the two spool construction is also used to minimize the effects of acceleration forces). If the valve is of the three-way type all critical axial dimensions can be eliminated with a screw type adjustment incorporated to set the underlap or overlap of the metering edges.

The characteristic of certain steels to grow after a period of time has also created a problem to valve manufacturers. The combination of extremely small clearances and steel growth caused the spools to jam in many of the earlier valves. To minimize this problem, the steel parts are thermally cycled and stabilized before the final grinding or lapping operation.

In order to reduce the weight of the servo valves for airborne application most manufacturers employ aluminum wherever possible. The type of construction where a steel sleeve is pressed into an aluminum body has created another difficulty in that the difference in thermal expansion and contraction between the aluminum and steel creates a compressive effect on the steel sleeve when the assembly is cold. This compression can be great enough to bind the spool within the sleeve. To eliminate this problem, the spool and sleeve assemblies are loosely fit into the body with "0" Rings performing the sealing function between sleeve and body. A further advantage to this type of construction is the ease with which a spool and sleeve assembly can be replaced in case they are damaged or become jammed.

It has also been noted that there is some difficulty in making valves that exhibit identical or repeatable characteristics. This is especially true with reference to dynamic characteristics where it has been observed that two valves in serial sequence vary by as much as 50% in their frequency response. The reasons for these differences are not obvious but this particular manufacturer felt that the differences would decrease when the valve went into full-scale production.

There has been some interest in a sliding or rotary plate valve (hole and plug) developed at MIT. According to MIT, one attribute of this valve is its ease of manufacture; however, companies that have made this type of valve (Midwestern Geophysical, Hydraulic Controls, etc.) have found it more difficult to construct than the spool type valve. This difficulty arises in part from the need to make and maintain four surfaces flat and parallel. These manufacturers found it more difficult to produce flat and parallel surfaces to small tolerances than to produce cylindrical spools and holes to the same small tolerances.

The nozzle flapper type valve is apparently easier to manufacture than a spool type valve. There are fewer critical dimension and for this reason the nozzle flapper type valves are also less costly to build than spool type valves.

CHAPTER 5

COMPILATION OF DATA OBTAINED FROM USERS OF SERVO VALVES

The purpose of this section is twofold; first to review the system and valve requirements as specified by the users, and second, to determine how adequately, from the users' viewpoint, the valves presently employed meet these requirements. This information was obtained, in all cases, by direct visits to missile and aircraft manufacturers employing servo valves in their vehicles.

A. System and Valve Requirements

The information in Table 2 describes the operational and environmental conditions under which servo valves must operate in the various aircraft and missiles. The manufacturer, model, and characteristics of the various valves employed are also listed. The following discussion is primarily a summary of the information included in Table 2 along with any additional pertinent information obtained during the survey. Terminology will be defined when used unless it has been defined in previous sections, or the glossary, or is self-explanatory. The valve users' comments as to the capability of specific valves will be discussed in Part B.

1. Pressure and Flow Characteristics

As can be seen from Table 2, the supply pressures currently used in aircraft and missiles vary from 1000 psi to 3000 psi, the most commonly used pressure being approximately 3000 psi. Most aircraft and missile manufacturers are going to higher pressures to reduce the size and weight to power ratio of the actuators, to reduce the flow capacity and size of the supply pumps, to reduce the size of the oil reservoirs, and to obtain higher response.

It is of interest to note the variance in supply pressures encountered by certain of the valves listed in Table 2. In many cases, the supply pressure varies over 500 psi from its normal no load value, primarily due to the drain of the many loads being supplied. This is not a desirable condition since the gain of most servo valves varies directly with supply pressure.

TABLE 2A

Tabulation of Information Obtained from Valve Users

Valve User	SYSTEM REQUIREMENTS FOR VALVES: Type of Load	Freq Resp Req	Desired Valve Life	Supply Pressure & Variation (psi)	Operating Temp (°)F Min	Operating Temp (°)F Max	CHARACTERISTICS OF VALVES EMPLOYED: Mfg & Model	RATING: Press (psi)	RATING: Flow (gpm)	RATING: Current (ma)	Quiescent Flow** (gpm)	Filtration (mu)
Bell Aircraft Co.	Inertia and spring		1000 hrs	3000 + 0 - 800	-60	+225	Bell valves	3000	1.9 - 12.5	10		10
Bendix Aviation Corp (Pacific Div) (Hydraulic components only	Inertia and spring	Valve - open loop only Max amp - -6db/oct -7db @ 2 cps (approx) 180° lag @ approx 32 cps	500 hrs	2800 + 700 (accumulator)	-65	150	Bendix B-II	2800	3.5 *NLD	18	0.07 (with dither)	10
Boeing Airplane Co.	Inertia plus spring for aileron. Inertia for gimb.	Rocket Gimbal: 3 cps cut off. Aileron: -45° @ 10 cps (Vel. servo)	20 hr @ 60 cps w/spool moving @ ½ max amplitude	3000 + 0 - 500	-65	250	Moog 1302X valves for rocket gimbal	3000	8.8	8	0.42	10 (also magnetic)
	Inertia plus spring	Sys.-45° @ 10 cps (Vel. servo)		3000 + 0 - 500	-65	250	Moog 1401X (elevator)	3000	4.7	8	0.29	10 also magnetic
	Inertia plus spring	Sys.-45° @ 10 cps (Vel. servo)		3000 + 0 - 500	-65	250	Moog 1400 (rudder)	3000	2.34	8	0.22	10 also magnetic
	Inertia			3000 + 0 - 500	-65	250	Standard Controls PC1 valve (Roll Bulkhead)	3000	2.15	8	0.20	10 also magnetic
Consolidated Vultee (Pomona)	Tested unloaded w/constant force	Moog valve flat to 40 cps Gain ratio 0.9 to 1.2 phase lag < 26°	1,800,000 cyc full rated current @ 10 cps (50 hrs)	1500 + 200 -300	0	+160	Moog 902A valve	1500	3	6	0.25	10
Douglas (El Segundo)	Inertia spring damper combination	Servo - flat to 3 to 6 cps Phase 1.5°/cps	1,600,000 cyc @ 100°F 400,000 cyc @ 200°F 20,000 cyc traveling to extreme limit	1500 ± 50	-65	+225	Moog 522 valve	1500	2.0	7 (approx)	0.25 (max)	2
	Inertia spring damper combination	Servo - flat to 8 to 10 cps Phase 1.5°/cps		1500 ± 50	-65	+225	Moog 522 valves	1500	1.0	7 (approx)		2
Douglas (Santa Monica)	Inertia and spring	Valve - open loop only Max amp - 6db/oct -7db @ 2 cps $\phi = 120°$ @ 1 cps 180° @ 32 cps	1 to 2 million cycles	2000 + 0 - 1000	110	+160	Bendix B-I valves	2000	3.25	18	0.08 (with dither)	10
Goodyear Aircraft Co.	Tail: Spring inertia Spoiler: Inertia friction		500 hrs			140	Moog 916 valves				0.22	2 - 5
	Inertia	Sys. flat to 30 cps	500 hrs	1500 ± ?		140	Moog Mod 504	1500	6.2	8		10
			500 hrs	1500 ± ?		140	Dreyer-Hanson	1500	0.37	25	0.025	10

*NLD - No load drop at listed supply pressure
** Quiescent Flow - Leakage flow plus first stage flow

TABLE 2B

Tabulation of Information Obtained from Valve Users

Valve User	SYSTEM REQUIREMENTS FOR VALVES: Type of Load	Freq Resp Req	Desired Valve Life	Supply Pressure & Variation (psi)	Operating Temp (°)F: Min	Max	CHARACTERISTICS OF VALVES EMPLOYED: Mfg & Model	RATING: Press (psi)	Flow (gpm)	Current (ma)	Quiescent Flows (gpm)	Filtration (mu)
Hughes Aircraft Co.	Spring	Sys. flat to 20 cps <11.5° lag @ 5 cps 108° lag @ 32 cps with Δi = 3 ma	100 hrs	2000 ± 150	0	160	Hughes Aircraft Co Mark 5 Mod 2 Actuators	2000	0.125	14	0.016 (w/dither)	
	Spring		100 hrs	2000 ± 150	0	160	Hughes Aircraft Co Mark 5 Mod 3 Actuators	2000	0.125	14	0.016	25
Lockheed Aircraft Corp	Inertia and spring	Servo cutoff @ 4 cps	1000 hrs	3000 (Transient of 1000 psi)		Below 225	Moog 526Y valves	3000	0.42	8	0.12 (approx)	5 to 10
	Inertia and spring						Lear Electrical Servo (dry powder clutch)					
	Inertia		1000 hrs			Below 225	Moog 521 valves in autopilot (information from Eclipse-Pioneer	3000	2	8	0.17	10
Glenn L. Martin	Inertia and spring	Sys. max phase lag 25° @ 30 rad/sec	250,000 cyc @ various freq	1500 + 50 - 150 Normal	-65	160	Moog 512 valves	1500	2.7	8	0.37	5
	Inertia and spring	Sys. max phase lag 25° @ 30 rad/sec	250,000 cyc @ various freq	1800 + 250 - 300 Peak	-65	160	Moog 512 Moog 907 valves	2000 (907)	2 (907)	8	0.25 (907)	5
Minneapolis Honeywell Regulator Co. Aeronautical Res. Dept.	Inertia, spring and friction	Typical loaded actuator response -3 db & -106° phase lag @ 8 cps w/1500 psi supply (XHG 42A1)	1000 hrs	3000 ± 100	-65	250	Minn.Honeywell (Ridgway Plant) XHG 42A1 Actuators	3000	2.6	16	0.10 (w/dither)	10
	Inertia, spring and friction	-3 db & -106° phase @ 8 cps w/1500 psi supply (XHG 42A1)	1000 hrs	3000 ± 100	-65	250	XHG 42A1 Actuators	3000	2.6	16	0.10 (w/dither)	10
	Inertia, spring and friction	Typical loaded actuator response -3 db & -105° phase lag @ 77 cps w/3000 psi supply (XHG 36A1)	1000 hrs	3000 ± 100	-65	250	Minn.Honeywell (Ridgway Plant) XHG 36A1 Actuators	3000	2.6	16	0.10 (w/dither)	10
N.A. Aviation, Inc.	Inertia and spring	Sys. flat to 6 or 7 cps	1000 hrs	3000 ± ?	0	225	Moog 534 valves	3000	6.25	8	0.3	10 plus magnetic
	Inertia and spring	Sys. flat to 6 or 7 cps	1000 hrs	3000 ± ?	0	375	Moog 534 valves	3000	6.25	8	0.3	10 plus magnetic
	Inertia,	Sys. flat to 15 cps 90° lag @ 15 cps	No spec yet	3000 ± ?	-65	250	Pilot stage of NAA's time modulated valve	3000	0.52	40	0.048 (w/dither)	10

TABLE 2C

Tabulation of Information Obtained from Valve Users

Valve User	SYSTEM REQUIREMENTS FOR VALVES: Type of Load	Freq Resp Req	Desired Valve Life	Supply Pressure & Variation (psi)	Operating Temp (°)F Min	Operating Temp (°)F Max	Mfg & Model	CHARACTERISTICS OF VALVES EMPLOYED: RATING Press (psi)	RATING Flow (gpm)	Current (ma)	Quiescent Flow* (gpm)	Filtration (mu)
Northrop Aircraft	Very small	Valve flat to 50 cps Servo - max freq 10 - 15 cps	250 hrs	3000 ± ?	-65	+165	Moog 500 series	3000	0.5 - 2			2 - 5 at valve
	Very small	Servo - max freq 10 - 15 cps Valve flat to 50 cps	300,000 cyc	3000 ± ?	-65	+165	Moog 500 series					2 - 5 at valve
Raytheon Mfg	Inertia and spring	No data	4 million cycles	3000 + 0 - 1000	-65	200	Raytheon Two Stage	3000	1 (one stage)	15		
	Inertia and spring	No data	4 million cycles	3000 + 0 - 1000		350	Raytheon valves	3000	5 (two stage	15		
	Inertia and spring	No data	4 million cycles	3000 + 0 - 1000		350	Raytheon single stage for radar	3000		15		
Ryan Aviation Co.	Inertia	No spec yet	No spec yet	Don't know			Peacock Machine Co to Ryan specs	800	2.6 *NLD	70	0.195	5
	Inertia	No spec yet	No spec yet	Don't know			Electrical Servo	800	2.6	70	0.195	5
Sperry Gyroscope	Inertia	Sys. @ 2.5 cps, < 25° phase lag -12 db or more @ 180° phase		2000 to 950 (accumulator)			Cadillac Gage CG-1	2000	0.78 *NLD	11		None at present but are going to use 10
Westinghouse Air Arm.	Inertia and spring	Sys. resonance 3 - 12 cps	1000 hrs	3000 + 0 - 900	-65	230	Moog valves	3000	1.5	8		10
	Inertia and spring	Sys. resonance 3 - 12 cps	1000 hrs	3000 + 0 - 900	-65	230	Moog valves	3000	1.5	8		10
	Inertia	Sys. resonance 25 - 50 cps	1000 hrs	3000 + 0 - 900	-65	230	Moog valves	3000	1.5	8		10 (also magnetic)
	Inertia and spring		1000 hrs	3000 + 0 - 900	-65	230	Moog valves on antenna Moog valves on turret	3000	6	8		10
Fairchild Aviation (Doesn't use servo valves)	Spring	Sys. 3 db @ 3 cps			-65	170	Nonhydraulic servo only Electrical-actuator developed by Fairchild					
	Spring				-65	170	Lear Elect. servo					
	Spring				-65	170	Pneumatic					
	Spring				-65	170	Pneumatic					

*NLD - No Load Drop at Listed Supply Pressure

The maximum load flows of the various systems vary from 1/8 gpm to 9 gpm; however, most of these are in the vicinity of 2.5 gpm. This indicates that most of the valves employed have ratings of less than five horsepower. (See Section III.) The quiescent flows of the various valves in Table 2 vary from 0.02 gpm to 0.42 gpm. There is no particular trend indicated in these figures although most users are interested in maintaining the quiescent flow as low as possible. Valve quiescent flow is an especially important factor in small guided missiles where reservoir space is limited. Hughes for instance specifies a minimum quiescent flow of 0.016 gpm, which is less than the quiescent flow rating of the commercially available valves.

Most of the valves presently used in airborne application are of the standard type; that is, they are neither gain compensated nor of the pressure control type. However, those valves utilizing a spring centered output stage may inherently possess some gain compensation depending on the relative magnitudes of the spring constant and the flow forces. Glenn L. Martin presently employs a gain compensated Moog valve in one application, while Boeing employs a Standard Controls pressure control valve. Boeing is also considering the use of a gain compensated valve in another application. The general concensus of opinion among the people contacted is that gain compensation would be desirable but not absolutely necessary in these present applications.

The type of valve characteristics required for a particular system is, of course, very dependent on the type of load. As can be seen from Table 2, most users consider the control surface loads as pure inertia on the ground, and as a combination of spring and inertia in the air. Generally, the stability problem is most serious on the ground where pure inertia loads are encountered.

2. Dynamic Response

Closed-loop bandwidth specifications for the electrohydraulic systems listed in Table 2 vary between 3 cps and 20 cps. The greater majority of these are in the 4 cps to 10 cps range. In one case a lag network was employed to cut off the servo response at approximately 3 to 4 cps. A comparison of these values with the frequency response specifications of the various valves in Table 1 shows that the frequency response of the servo valve is not a limiting factor in most airborne applications. This is especially true of the commercially available valves which have no load frequency responses varying from 60 cps to 170 cps.

Phase shift appears to be more of a problem than amplitude response. The larger the phase shift in the valve, the less phase shift can be tolerated elsewhere in the system. Therefore, the phase shift specifications generally run out to a higher frequency than do the amplitude specifications which define frequency response. For example, Hughes specifies a maximum of 108 degrees phase shift at 32 cps while the amplitude response specifications extend only to 20 cps. Thus it appears that these systems are more critical with respect to phase margin than gain margin. Most users stated they would prefer less phase shift in their valves. However in no case did the user indicate that the dynamic response of the commercially available valves was inadequate for their application.

Many of the users have attempted to represent the valves by a mathematical transfer function relating output flow to input current to aid in their analytical work. Most of the users would or could not furnish these transfer functions. The few that were obtained are shown in Table 3. In the low response systems the valve is generally represented as an integrator, in the higher response systems as a third order function.

3. Environment

The servo valves used in airborne application must meet military specifications for hydraulic equipment in addition to special specifications required by the particular valve user. The temperature specifications appear to be the most demanding, requiring operation from -65° to as high as 350°F in some present applications; anticipated requirements are even more stringent. For instance, North American requires satisfactory valve operation at oil temperatures of 750°F in one of their future applications.

Vibration specifications generally demand that frequency response characteristics be unaffected by vibration test frequencies from 0 to 2000 cps under acceleration loadings of 4 g to 10 g.

TABLE 3

VALVE TRANSFER FUNCTIONS

Moog 522	Douglas (El Segundo)	$\frac{q(s)}{\Delta i(s)} = \frac{K}{s} \left(\frac{\text{flow}}{\text{differential current}} \right)$
Moog 512	G. L. Martin	$\frac{q(s)}{\Delta i(s)} = \frac{1.5}{s(1 + 0.006s)} \left(\frac{in^3}{\text{ma sec}} \right)$
Moog -	Lockheed	$\frac{q(s)}{\Delta i(s)} = \frac{K}{s} \left(\frac{\text{flow}}{\text{differential current}} \right)$
Moog -	Westinghouse	$\frac{q(s)}{\Delta i(s)} = \frac{K}{s(s^2 + 2\zeta\omega_n s + \omega_n^2)} \left(\frac{\text{flow}}{\text{differential current}} \right)$
Drayer Hanson	Drayer Hanson	$\frac{\theta(s)}{\Delta i(s)} = \frac{\omega_n^2}{(1 + s)^2(s^2 + 2\zeta\omega_n s + \omega_n^2)} \left(\frac{\text{gpm}}{\text{ma}} \right)$ $\tau = 0.002,\ \omega_n = 1800,\ \zeta = 0.2 - 0.3$ $\theta(s)$ position of output spool
Peacock	Ryan	$\frac{q(s)}{\Delta i(s)} = \frac{0.143}{s(1 + 0.016s)} \left(\frac{in^3}{\text{ma sec}} \right)$
Hughes Actuator	Hughes	$\frac{\omega(s)}{\Delta i(s)} = \frac{50}{s(1 + \tau s)} \left(\frac{\text{deg}}{\text{ma sec}} \right)$ $0.002 < \tau < 0.003$ $\omega(s)$ = output angular speed
Moog 1403-B	Bendix (Mishawaka)	$\frac{q(s)}{e(s)} = \frac{\log^{-1}(-1.2)}{(.005\ s + 1)(.00016\ s + 1)^3} \left(\frac{in^3}{\text{volt sec}} \right)$

4. Filtration, Dither, and Type of Oil

Filtration requirements vary from one manufacturer to another. One manufacturer (Hughes) specially processes all oil used in their servo systems by recycling the oil through a Fullers earth type filter and five micron filters. Generally, filtration requirements vary from two to ten microns with some manufacturers considering sintered bronze type filters to replace the conventional paper type cartridges used at the present time. Magnetic filtration is required in some servo systems but the magnetic filters which are incorporated internally in commercially available servo valves seem to satisfy most valve users. Although dirt has been a problem in the earlier designs, most aircraft and missile manufacturers are presently utilizing sufficient external filtering to greatly minimize this problem.

All servo valves which do not incorporate a nozzle-flapper require dither of varying magnitudes and frequencies. These dither frequencies vary from 150 cps to 400 cps. It should be noted that there are only two instances of dither being used on servo valves incorporating a nozzle-flapper in their design. (Glenn L. Martin and Westinghouse Air Arm).

Every manufacturer visited used MIL-0-5606 oil in his test stands and vehicle hydraulic systems. North American Aviation Inc. also uses NA2-2074 Silicate Ester Base Hydraulic Fluid for some of their applications.

5. Life, Reliability

Life defines the nominal duration of time or number of cycles through which a valve will operate before failure due to cumulative and predictable effects such as wear, fatigue, etc. The reliability is a measure of the probability that the valve will operate satisfactorily within its nominal life time without failure due to random and unpredictable effects such as dirt clogging, manufacturing defects, etc.

Most users were unable to quantitatively specify criteria for reliability; a few considered reliability synonymous with life. Opinions concerning desired servo valve life varied from 20 hours operation (with the power spool moving one-half maximum amplitude at 60 cps in a missile application) to 1000 hours of "normal" system operation (no definite number of cycles of operation) for aircraft applications. Except for a desired life of 1000 hours for aircraft use, there was little agreement as to the valve life requirements for current missile applications.

6. Electrical Drive Requirements

The input differential current for the various valves listed in Table 2 varies between 8 ma and 70 ma for maximum output flow. The majority of the valves operate toward the low end of this range. The general concensus of opinion of the users was that this current should be as low as possible to simplify the system electronics.

Nearly all the valves that employ a spool type first stage require artificially inserted dither signals. Some of the users indicated that they would not use valves with first stage spools since dithering requires additional circuitry complication and because dither had a tendency to shake the control surfaces at the dither frequency.

B. Users' Comments As to Capabilities of Valves

Table 4 lists all the servo valves which are presently being used, or have been tested, by the various aircraft and missile manufacturers, along with their comments (where available) as to the capability of these valves. Even in those cases where comments could not be obtained, the fact that a particular valve has been tested and not used by a number of people is significant. An asterisk before a manufacturer's name indicates the valve under consideration is being used by that manufacturer on a vehicle it produces. A double asterisk indicates the information was obtained from the valve manufacturer.

A cursory glance at Table 4 shows that the Moog Valve Company supplies most of the commercially available valves presently used for aircraft and missile application. However, since the original survey was completed, the Cadillac Gage FC-2 has received considerable application (see double asterisks). With the exception of Moog, the Cadillac FC-2 and PC-2, the Bendix B-II, and the Drayor Hanson valves, the remaining valves are either being manufactured or were designed by the user.

Apparently most valve manufacturers are quite satisfied with

Moog valves. The major complaint is null shifting and susceptibility to magnetic particles. Most users think the null shifting results from an accumulation of magnetic particles on the torque motor pole faces which are in direct contact with the oil. North American has found that the null shifting persisted even with magnetic filtration, and that a precise adjustment of the flapper with the aid of a microscope alleviated the problem.

Boeing uses the Cadillac Gage PC-2 in the roll bulkhead control circuit. The roll bulkhead constitutes primarily an inertia load and Boeing has found that the PC-2 provides more stable operation than standard type valves.

No user's comments were received on the operation of the Drayer Hanson or Cadillac Gage FC-2 valves.

TABLE 4

USERS' COMMENTS ON VALVES

Valve	Commenting Manufacturer	Comments
Moog		
902-A	*Convair (Pomona)	Most reliable valve tested; null shifts in one direction with temp. and both directions with time.
- -	*Chance Vought	- -
500-900	*Goodyear	Null shifts due to magnetic particles; otherwise reliable.
500-900	*Westinghouse	Magnetic dirt is a problem. Null shifts with pressure, not with time.
- -	Hughes	Flow and leakage too high; susceptible to sticking from particles.
534	*NAA	No difficulty other than null shift - believe they have solved problem by precisely adjusting flapper position with microscope; magnetic filtration remedied valve sticking.
512	*Glenn L. Martin	Null point shifts.
- -	*Eclipse-Pioneer	Flow and leakage too high; null point shifts with pressure and time; dislike torque motor coils in oil.
500	*Northrop	Northrop has had no null shifting problem; dead zone becomes larger probably due to unequal clogging of filter; would like

TABLE 4 (Cont'd)

Valve	Commenting Manufacturer	Comments
Moog		
		greater reliability.
522	*Douglas (El Segundo)	Magnetic particles found on torque motor pole faces cause erratic operation; have problem of dirt but not in clogging.
- -	*Lockheed	No difficulty with magnetic particles.
- -	*Boeing	In comparison to other valves tested, Moog valves are simpler; have less size and weight, are cheaper. It is the only valve to meet their flow requirements. The null shifts with temperature; could be more reliable.
1402	Bendix (Mishawaka)	Moog valves give superior performance compared with their own JX-141-10112 valve. The latter was recently replaced by Moog valves.
- -	Minneapolis Honeywell	- -
**	Sperry Gyroscope	
Cadillac Gage		
**FC-2	Boeing	
FC-2	Lockheed	
**FC-2	Douglas (El Segundo)	
**FC-2	Convair (San Diego)	

TABLE 4 (Cont'd)

Valve	Commenting Manufacturer	Comments
Cadillac Gage		
**FC-2	Convair (Pomona)	
FC	Northrop	- -
**FC-2	North American	
PC-1	Westinghouse	Valve tested had a large unbalance.
PC-2	Northrop	- -
PC-	Douglas (El Segundo)	No better than Moog valves in their application.
**PC-2	Arma	For turret application.
PC-	Raytheon	Operation unsatisfactory under load.
**PC-2	General Electric	For robot application.
PC-2	*Boeing	More stable under inertia load than other types.
**PC-2	Grumman	- -
- -	Minneapolis Honeywell	- -
- -	NAA	- -
Bendix B-I	Northrop	- -
	*Douglas (Santa Monica)	More reliable than Bendix MIT type valve or B-II valves.

TABLE 4 (Cont'd)

Valve	Commenting Manufacturer	Comments
Bell		
SV9C-1	Douglas (El Segundo)	Tests not completed.
- -	Boeing	Plan to test.
- -	*Bell Aircraft	No null shifting reported.
- -	NAA	Tests not completed.
Drayer Hanson		
first stage	Goodyear	- -
first stage	Lockheed	- -
Raytheon	*Raytheon	- -
NAA Time Mod. Valve	NAA	Not reliable - too much electronic complication.
CG-I	*Sperry	- -
Bendix B-II	*Douglas (Santa Monica)	- -
CG-II	*Sperry	- -
Hughes Actuator	*Hughes	Most difficulty in magnetic driver and potentiometer. Present valve operating satisfactorily.
Midwestern	- -	- -
- -	Hughes	- -
- -	Lockheed	Major objection: it can't be manifolded.

TABLE 4 (Cont'd)

Valve	Commenting Manufacturer	Comments
Midwestern		
- -	Minneapolis Honeywell	- -
- -	Boeing	Too heavy; did not test.
Mod 4	Convair (Pomona)	Too low frequency response.
- -	Northrop	- -
Mod 4	Westinghouse	Very susceptible to dirt clogging; too low frequency response.
Mod 3	NAA	Valve very dirt sensitive; operating range very limited.
Hydraulic Controls	Douglas (El Segundo)	Could not use because required dither shook control surfaces.
Peacock		
- -	*Ryan	- -
- -	Convair (Pomona)	Too large, low response.
M-H 1 stage, 2 spool MIT type valve		
- -	Convair (Pomona)	- -
- -	Boeing	Unacceptable
Sanders		
Mod 10	Boeing	Unable to operate

TABLE 4 (Cont'd)

Valve	Commenting Manufacturer	Comments
MIT Sliding Plate Valve	Hughes	Very susceptible to dirt; center plate buckles under temperature shocks.

GLOSSARY

1. Amplitude Compensation - Modification of the characteristics of the servo valve to obtain linear static flow characteristics in the region of small spool displacements.

2. Closed Center Valve - A valve in which the output spool lands completely overlap the valve ports when the valve is nulled or centered.

3. Dither - A low amplitude sinusoidal signal superimposed on the input signal to the servo valve to reduce "stiction" and dead zone.

4. Electromagnetic Driver - That element within the servo valve which converts the electrical input signal into a mechanical displacement of the flapper, spool, jet pipe, whichever is used.

5. Flow Force Compensation - Reshaping of the valve ports and spool to reduce the flow forces.

6. Four-Way Valve - Two three-way valves operating in push-pull. The valve has four external connections, supply, drain and two load ports.

7. Gain Compensation - Modification of the characteristics of the servo valve such that the output flow for a specific input signal is relatively independent of load pressure.

8. Hysteresis - Valve hysteresis is the difference in input signal necessary to produce flows increasing from zero to maximum from that necessary to produce flows decreasing from maximum back to zero.

9. Integral Valve Actuator Combination - A servo valve and actuator built into a single unit. Generally the valve is of the three-way type and the actuator is single ended.

10. Jet Pipe - A device sometimes used to provide a velocity head for controlling the position of the output stage. It consists of a rotatable nozzle which proportions flow between two adjacent holes which lead to opposite ends of the output member.

11. Linear Actuator - A ram operating in a cylinder and driven by some pressure source. The actuator may be either single ended or double ended.

12. Metering Section - That portion of servo valve which directly controls the flow and pressure.

13. Nozzle Flapper - A device often used as the first stage to proportion the pressure across the output spool. This is accomplished by varying the area of a variable orifice with respect to a fixed orifice by varying the distance of the flapper from the nozzle which passes the fluid.

14. Load Drop - The pressure across the load or across the actuator.

15. Open Center Valve - A valve in which the spool lands do not completely overlap the valve ports at any time.

16. Pressure Control Valve - A valve in which the output load pressure is approximately proportional to input differential current irrespective of flow.

17. Quiescent Flow - The magnitude of flow through all stages of the servo valve when the valve is closed or centered.

18. Quiescent Power Loss - The power consumed in the servo valve due to quiescent flow when the valve is closed or centered.

19. Rotary Actuator - An actuator which provides a rotary output when a pressure source is applied to its input terminals.

20. Servo Valve - A high response device which converts some form of input signal (electrical or mechanical) into a proportional fluid flow or pressure which may be used to move a mechanical actuator.

21. Single Stage Valve - A valve in which the output flow is controlled by a spool directly connected to the electromagnetic driver.

22. Solenoid - A device which provides a linear mechanical output directly proportional to input current.

23. Spool Land - That portion of a valve spool which mates with the sleeve surface and which acts in conjunction with the valve port to control flow and pressure.

24. Spool Passage - The passage between the spool lands which provides a passageway for flow between valve ports.

25. Torque Motor - A device which provides a rotary output displacement directly proportional to input current. Generally the arc of rotation is so small that the displacement can be considered linear.

26. Three-Way Valve - A valve which has two metering orifices in series with supply and drain; the pressure between the two orifices constitutes the output pressure.

27. Threshold - The minimum amplitude of input signal to a servo valve necessary to obtain measurable flow.

28. Two Stage Valve - A valve which contains two stages, the first of which is similar to a single stage valve and which positions a second stage spool hydraulically. The second stage spool then controls the output flow.

29. Valve Drop - That difference in pressure between the supply pressure and the load plus drain pressure.

30. Valve Loop - The term refers to a two stage valve and includes all the elements in the forward portion of the servo valve in addition to those making up the feedback path.

31. Valve Ports - The openings in the valve body and sleeve which connect the spool passages and the external connections to the valve.

32. Valve Sleeve - That portion of the valve which houses the valve spool.

33. Valve Null - Refers to the condition where the flow from the output load ports is zero for zero input current.