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PART VI

HYDRAULIC SERVO CONTROL VALVES
PART VI
RESEARCH ON ELECTROHYDRAULIC SERVO VALVES DEALING WITH OIL CONTAMINATION, LIFE AND RELIABILITY, NUCLEAR RADIATION AND VALVE TESTING

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FOREWORD

This report, Research on Electrohydraulic Servo Valves Dealing with Oil Contamination, Life and Reliability, Nuclear Radiation, and Valve Testing, was prepared under the direction of B. A. Johnson, Project Engineer, and T. J. Dunsheath, Technical Director of the Automatic Control Systems Section of Cook Research Laboratories, Morton Grove, Illinois. The work was done under Air Force Contract No. AF 33(616)-5136, and this report is the second of two parts to be prepared under this contract. It is also a continuation of previous work on related subjects and is Part 6 of the series entitled, Hydraulic Servo Control Valves. The report includes presentation of information gained in the design and performance of tests intended to study the effects of various contamination levels in the oil on valve operation, presentation of available information on servo valve life and reliability, the design of a servo valve-actuator combination for use in neutron and gamma radiation environments, together with results of testing this combination under gamma radiation, and results of testing three new servo valves.

Personnel involved in the program, in addition to Mr. B. A. Johnson, included Messrs. P. A. Weiss, W. L. Kinney, E. R. Schumann, and A. D'Andrea. Mr. Weiss conducted the effects of oil contamination phase of the program, and Mr. Kinney conducted the life and reliability phase. Mr. Schumann performed a large portion of the work in both the radiation and valve test phases and Mr. D' Andrea assisted in the laboratory work on all phases of the project.

The work was administered under the direction of the Flight Control Laboratory, WADC, with Mr. V. R. Schmitt as initiator. This document is unclassified.

ABSTRACT

This report is divided into four sections. Design and operation of a test fixture intended for analysis of servo valve operation under various levels of oil contamination are described. Information on servo valve life and reliability, as obtained from valve manufacturers and users, is presented, and the various design features influencing reliability are discussed. Design and construction of a valve-actuator assembly for use in a nuclear radiation environment are described, and test results on this unit under gamma radiation are presented. A series of evaluation tests was run on one unit each of three new types of valves, and the test results are presented and discussed.

PUBLICATION REVIEW

This report has been reviewed and is approved.

FOR THE COMMANDER:

Roy E. Tavasti

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CHAPTER I

INTRODUCTION

A. Purpose

This is a final report and covers the remaining four tasks to be completed under Air Force Contract No. AF 33(616)-5136. These are:

(1) Investigation of the effects of oil contamination on electrohydraulic servo valve operation

(2) Study of life and reliability of servo valves

(3) Investigation of nuclear-radiation effects on operation of a servo valve-actuator assembly

(4) Evaluation tests of three new servo valves.

The other tasks required under the above contract were covered in WADC Technical Report 55-29, Part 5.

Considerable interest has been generated on the subject of the effects produced on servo valve operation by various amounts and types of oil contamination. Much sporadic and sometimes contradictory information, based on unorganized laboratory and field experience, exists on this subject. Because of its importance as a major factor in causing valve malfunction, the oil contamination investigation was undertaken in an effort to make a controlled study of the problem. The purpose was to design a suitable test setup and to study valve operation under various measurable levels of contaminated oil.

The use of servo valves in flight control systems is becoming increasingly prevalent and, as such, knowledge of valve and associated equipment reliability is assuming considerable importance. The purpose of the life and reliability phase of the project was to obtain available information from

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servo valve manufacturers and users on the subject of valve reliability, to analyze and discuss same, and to design a test setup for evaluating valve life as a function of wear.

Possible use of nuclear propulsion plants in aircraft dictates the need for knowledge of radiation effects on all components of aircraft equipment including the servo valve-actuator assembly. It was the intent under the nuclear-radiation phase of the contract to investigate this problem in a program that would include a survey of previous work by others on related questions, an analysis of radiation effects on the materials presently used in servo valves, actuators, feedback transducers and related equipment, the design and assembly of a suitable valve-actuator assembly, and the testing of this assembly under various levels of gamma radiation.

The purpose of the valve test phase was to evaluate performance of three new servo valves which had become available since the last report. Tests which were to be made on these valves were those outlined in WADC Technical Report 55-29, Part 2, except where such tests were incompatible with the method of operation of the valve being tested.

B. Organization of the Report

Each of the various tasks is included as a separate chapter of this report. Chapter II, and Appendices I and II, cover the work on investigation of oil contamination effects. Chapter III covers the life and reliability study and Chapter IV, plus Appendix III, the nuclear-radiation investigation. Chapter V and Appendix IV present the results of tests on the three new servo valves.

CHAPTER II

OIL CONTAMINATION STUDY

A. Introduction

Electrohydraulic servo valves designed for current military airborne applications are required to operate at supply pressures between 1500 and 3000 psi. Systems are presently under development which will operate at supply pressures of 4000 psi. The employment of such high pressures requires that the clearances between moving parts within the valve must be maintained extremely small to minimize leakage and steady-state power losses. This has been done. Unfortunately, in order that the valves operate satisfactorily, it is imperative that the oil be devoid of any particles or contaminant approaching the size of the clearances or else erratic operation or malfunction may occur. This has not always been done. As a result, servo valves have acquired a stigma, perhaps unwarranted, of having questionable reliability for control application.

Actually, this problem has probably arisen due to lack of knowledge as to what contamination level the valves can tolerate and as to what contamination level actually exists in the system. This is evidenced by the fact that a good many systems have operated successfully without oil contamination difficulties. For example, in four years of performance testing of various types of servo valves at these Laboratories, only one case of a valve sticking has been encountered, and this was quickly remedied without valve disassembly. The situation is of course different in the field where similar control over oil cleanliness is more difficult to achieve.

The questions then arise; what is the maximum level and size distribution of contamination the present production valves can tolerate, and is this level compatible with the degree of cleanliness that can be obtained with the filters that are available today? In addition, what types of contaminant are the most troublesome?

It was the purpose of this particular phase of the Valve Study contract to formulate experimental procedures and design test equipment for determining the answers to the above questions. In addition, some actual tests were to be made to establish the general limits of contamination under which current valves are capable of operating.

Before the actual design work on the contract was initiated, a survey of various valve manufacturers and users was made to determine the nature of difficulties various organizations have encountered with regard to contamination. The results of this survey are discussed in the next section.

In subsequent sections, the general approach that was taken in formulating the experimental program is discussed, followed by a description of test procedures and test equipment that were used. Finally, a discussion of the test results and current filter capabilities is presented.

B. Review of Survey Gathered Data

Both valve manufacturers and valve users were visited. The valve manufacturers were contacted primarily for the purpose of obtaining information on other phases of this project but did furnish some data worthy of consideration here. The valve users visited were primarily manned aircraft manufacturers, although some data on missile applications were also provided. Data were obtained on life, reliability, and oil contamination problems. The life and reliability data are reported in Chapter III.

1. Valve Manufacturers

Across the board, all valve manufacturers stated that their valves were not susceptible to first-stage clogging in any reasonably clean system, one that employs good quality nominally rated 10 micron filters. All manufacturers stated that they had increased the power capacity of their torque motors and the first stage, so as to minimize stiction effects on the output spool. Internal valve filters of various configurations and sizes are employed in all valves to protect the critical first-stage orifices. Dry coil or stale coil designs are being used or are under development throughout the industry to eliminate the problem of magnetic contamination in the first stage.

To illustrate how the general contamination problem is being attacked by various valve manufacturers, some of the features, in addition to those already mentioned, which were emphasized by the manufacturer as important factors for reducing the susceptibility of their particular valves to contamination will be briefly reviewed. This discussion does not cover all valves currently being manufactured, as all valve manufacturers were not visited.

a. Cadillac Gage FC-200 Valve

The main feature emphasized was the filter design. The filter inside the valve filters all the oil to the first stage and therefore obviates the situation where one of two separate filters may become clogged and cause valve unbalance, a condition considered more serious than a loss in valve gain. In addition, since the filter area is quite large, it should not be easily clogged. Another feature of the filter, the usefulness of which can only be determined from field experience, is that the load flow passes around the outside surface of the filter, supposedly tending to flush off dirt particles that may be attached to its surface.

The feedback arrangement of the FC-200 provides automatic clearance of any particles that tend to clog the upstream orifices of the first stage. This is desirable in this valve since the orifices are not circular and are probably more susceptible to dirt than round orifices. The valve employs a dry torque motor

b. Cadillac Gage FC-2

The main emphasis was placed in the internal design of this valve. Since the FC-2 valve employs only one nozzle, the flapper is not restricted in movement as is a balanced valve with nozzles on each side of the flapper. Thus, any clogging of the nozzle tends to force the flapper out sufficiently far to clear itself. In addition, because of the mechanical feedback between the spool and nozzle, any clogging creates a differential pressure across the spool which forces it to move in such a direction as to extend the flapper from the nozzle. The valve employs a stale torque motor. Another factor to be considered is that for the same quiescent flow a single nozzle valve has a larger orifice area and correspondingly larger critical dimensions than a double nozzle valve.

c. Moog Valve Company 2000 Series Valve

This is a dry torque motor valve but otherwise is similar in design to the earlier wet torque motor types. Moog engineers emphasized the fact that valve returns for repair from the field had practically been reduced to zero for reasons directly related to oil contamination since the dry coil designs had been placed in production. Conversely, a relatively high percentage of the wet torque motor type had been returned for this reason.

d. Bendix Pacific HR Series

The standard Bendix Pacific valves employ no isolation of the torque motor from the drain oil. However, two new modified designs are presently available that provide varying degrees of isolation. One of these employs a baffle which diverts the main flow away from the torque motor chamber but still allows a small amount of indirect flow to the chamber. The second isolates the flow from the chamber completely by means of a diaphragm. A test program is presently underway which will evaluate all three designs under identical operating contamination conditions.

One factor, which has been emphasized in regard to this valve design, is that the flapper is located not between the nozzles as with many other types, but floats above them. This arrangement permits much greater flapper movement to pass particles that reach the nozzles.

e. Hydraulic Research Valves

The main features emphasized by the Hydraulic Research engineers were the valves' powerful torque motor, dry coil design, and their internal filtering arrangement. Five filters are used to protect the first stage. The first-stage flow is tapped from the second-stage flow at an angle of 90 degrees. Thus, the inertia of the largest particles in the oil tends to keep them from turning the corner, so to speak, and entering the first stage. One filter, located just beyond the tap, filters all the first-stage flow. In addition, filters are located before each fixed orifice and nozzle, primarily to prevent clogging from contamination which may not be cleaned out during manufacture.

f. Raymond Atchley Valve

This valve, as described in Chapter V, uses the jet pipe principle for first-stage control. Thus, no fixed orifices need be employed, at least for pressures up to 2000 psi, according to the manufacturer. The critical orifice opening is a circular area capable of passing particles larger than 200 microns. This contrasts considerably with the critical cylindrical openings of the conventional flapper nozzle valves where the clearances between flapper and nozzle cylinder length are of the order of 25 to 50 microns for the same quiescent flow.

g. Airesearch Time Modulated Valve

The comments made here apply, in general, to all time modulated valves. The main features of this valve that are important to contamination insensitivity are the high controlling forces available for controlling the second stage and the constant dithering. High forces are available because the torque motor can be designed to maximize force output without compromising linearity. Dithering maintains the particles in motion and tends to reduce agglomeration tendencies.

h. Vickers Inc. Valve

The features cited by the Vickers engineers that reduced their valves' sensitivity to contamination were the dry coil construction and the arrangement of the flapper below the nozzles rather than between them.

2. Valve Users

Eight aircraft and missile manufacturers were visited during the survey. Their experience with regard to the oil contamination problem varied considerably. A number of companies have had very little difficulty. Others have had considerable difficulty but solved their problems by elaborate oil purification programs. Others are still experiencing difficulties. A brief summary of these problems and the approach used in solving the problems will be presented for each of the organizations visited. Further information on the general subject of life and reliability is available in Chapter III of this report.

a. Minneapolis Honeywell

Minneapolis Honeywell uses wet torque motor Moog valves in their autopilot applications. Filtering is provided by Purolator 10 micron paper filters. This combination has provided very satisfactory operation from the standpoint of oil contamination. System oil temperatures are generally low, less than $160^{\circ}F$.

No formalized contamination specification has been set up for their system, primarily because it has not been needed to date. However, they have made some particle counts of contaminant in special systems, listed below, which might be of interest. Since there are no particle sizes given, these data are valuable from a comparison standpoint only.

System	No. of Particles/mm^3 fluid
MH Hydraulics Laboratory	30-100
Commercial Airplane (Northwest Airlines)	700
Army Tank Hydraulic System	5,000

b. Northrop Aircraft

Northrop's aircraft experience with servo valves has been entirely with the yaw damper of the F-89. The system has limited authority and its malfunction would not cause loss of control of the aircraft. Wet coil Moog valves and Purolator paper filters are employed in this system. The main system filter is nominally rated at 10 microns while a 2 to 5 micron filter is used before the supply inlet to the valve. In three years' experience and 170,000 hours of flight, only two failures have occurred which were attributed to oil contamination. In fact, a total of only five valve failures was reported. As far as could be determined, the maximum operating time on any valve was 250 hours. Oil temperatures are around 100°F. No contamination specification has been issued for this system.

c. Douglas Aircraft

Douglas employs dry coil Moog valves on the yaw dampers of the F4D and F5D. Filtering is provided by a 10 micron wire-woven Purolator filter. Oil temperatures may reach as high as 225°F. Here again, little difficulty has been experienced with oil contamination in servo valves; however, a number of manually-operated valves have stuck. No specifications on oil contamination have been issued.

d. North American Aviation

North American has accumulated a great deal of experience with servo valves and oil contamination over the years. Most of this was obtained in connection with the Navaho program. As a result of this experience, an oil contamination specification has been set up (see Table I) as constituting the maximum allowable contamination distribution in an operational system. This does not mean that greater amounts of contamination are detrimental. It does mean, however, that oil contamination is not

TABLE I

CONTAMINATION LEVEL SPECIFICATIONS (50 ml SAMPLE)

Company	Particle Size (microns)	Type of Contamination	Maximum Count Allowable
Bell Aircraft Buffalo	100 26-100 11-25 6-11 1-5	All All All All All	None 5 20 81 No Count Specified
Bendix Products Division	300 0-500 All Sizes 25 All Sizes 25	Fiber Fiber Airborne Dust Airborne Dust Metal Metal	5 25 25,000 400 25 8
North American	>80 40-80 20-40 10-20	All All All All	50 175 688 1,338
Convair San Diego	100 51-100 21-50 11-20 1-10	All All All All All	17 42 84 170 253
Hughes	≥150	All	None
	(Average of 10 largest particles 50 microns)		

likely to be a problem in NAA systems if these specifications are followed.

Valve clogging has not been the only problem with contaminated oil. An even more serious problem has been increased valve wear and thus shorter valve life. This would be especially serious in manned aircraft.

Much of the wear problem can probably be attributed to the fact that many of the systems have had to operate at high oil temperatures, some 400°F or higher, where the rate of generation of contamination in the system is high.

North American has tested and used most types of filters. They prefer the wire-woven, unsintered type provided by Purolator. However, a number of systems have been operated with Permanent Filter Corporation sintered bronze filters and Porous Media sintered wire-woven filters. Metallic filters must be used because of the high temperature requirements. Both Moog wet and dry coil valves and Cadillac Gage FC-2 valves are employed in their systems.

e. Bendix Missile Division

Bendix employs a number of Moog wet coil valves on the Talos missile. The filters are nominally rated at 5 microns and are of the sintered stainless steel wire-woven type manufactured by Aircraft Porous Media.

About two years ago Bendix experienced a great deal of difficulty with oil contamination which they attributed to fibrous particles agglomerating and clogging the first stage of the valves. To obviate the condition, they set up an oil contamination specification (see Table I) which has proved very satisfactory. In addition, considerable attention was given to maintaining the assembly and test areas clean, and personnel were required to wear special laboratory coats.

It is interesting to note that no problem has been experienced with magnetic particles even though the valves are of the wet torque motor variety. Magnetic filters are not employed.

f. Bell Aircraft Corporation

Bell Aircraft employs their own valves on the Rascal missile. They estimate that in the past, 70 percent of valve failures could be attributed to oil contamination. To combat this problem, they have developed an oil contamination specification for their hydraulic systems including the test stands (see Table I). As in the case of most other specifications, these particular specifications have not been optimized butappear to assure freedom from contamination problems when followed.

Bell has experienced most of the contamination problems that have been mentioned thus far. Contamination resulting from manufacture has been a problem and has necessitated the placement of internal filters before each critical orifice. Accumulation of magnetic particles on the torque motor in gaps necessitated the switch from wet coil to dry coil valves. They have not experienced difficulties from particle agglomeration. Most of the contaminant has been found to be in the form of metallic and rubber chips and dust particles.

g. Lockheed Aircraft Corporation

Lockheed employs servo valves on the F104, Constellation, and Electra. Both Cadillac Gage FC-2 and Hydraulic Research SV414C valves are used. Until recently, Permanent Filter Company 10 micron sintered bronze filters were employed on the F104. A recent change to Aircraft Porous Media stainless steel and wire-woven filters was authorized by the Air Force. Operating temperatures may reach as high as 275°F.

Considerable difficulty has been experienced with contamination on all three systems, one of the reasons they gave for switching filters. No specification has been submitted to date on oil cleanliness although a visual check of photographed filter disks is made.

h. Hughes Aircraft Corporation

Hughes employs their own valve designs on the Falcon missile. Although they have experienced many difficulties in the past, the present systems do not have an oil contamination problem. One of the reasons for this is that they employ a special low viscosity oil, Bray 761, which is prefiltered before it is injected into the system. Another factor which they consider important relates to the use of a single-stage valve with a very powerful torque motor. The system is ultrasonically cleaned before the oil is added. No filters are used since the time of operation is too short for self-generated contamination to be a problem.

The Hughes particle size specifications on the supply oil requires that the average size of the 50 largest particles should be no greater than 50 microns and that the largest particle should be no greater than 150 microns. All oil is passed through a Luberfiner, consisting of a 10 micron filter in series with a 2 micron filter, thus eliminating the bulk of all particles.

i. Summary of Survey Data

It is quickly evident in reading the above that the systems that have the least contamination problems are those operating under relatively low temperatures. At temperatures below 160°F, very little difficulty has been experienced even though paper element filters have been used. At higher temperatures the situation is entirely different. Because the resins used in paper filters tend to disintegrate at temperatures above 200°F, clogging difficulties become significant. Most companies have therefore switched to metallic filters, which are no "cure alls" either, as will be discussed in a later chapter. Apparently, the particle generation is quite high at the higher temperatures and problems arise from the mere fact that none of the filters are 100 percent effective.

Particle size specifications have been initiated by many companies and proved effective. However, it is doubtful that these specifications give the true picture as far as contamination susceptibility is concerned, as systems have been operated successfully under much higher rates. The operating time of the valve is also important, and while one system can operate under a large contamination dose for a short period, it probably cannot do so indefinitely. For one thing, internal filters within the valve would become clogged. Nevertheless, if a judicious field servicing procedure is followed, consisting of periodic replacement or cleaning of filters, a fairly high contamination level probably can be tolerated.

Although it was not discussed above, the contamination generation capability of systems differ. Boeing Aircraft, for instance, has had considerable difficulty, largely because they employ many contamination generators within their hydraulic systems. These are in the form of hydraulic rotary motors. The use of such components should be avoided if possible.

The lack of good field cleanliness is another factor. Allowing lines and hoses to be exposed to air and dirt can present many headaches. As field experience testifies, rubber hose and valves with graphite packings should also be avoided.

C. Discussion of Oil Contamination Effects on the Operation of Electrohydraulic Servo Valves

1. Valve Design Limitations

In order to provide the reader with a clear picture of how contamination in the oil can affect the servo valve, a detailed analysis of a typical valve will be made. In this analysis, the contamination susceptibility of designs will also be discussed where their characteristics are such as to lead to different conclusions.

The typical flapper nozzle valve, as illustrated in the schematic of Figure 1, has a number of critical points where particles in the oil can cause valve malfunctions because of their size. These are the fixed orifices upstream from the nozzles of the first stage, the orifices formed by the nozzles and flapper, and the four orifices on the output spool.

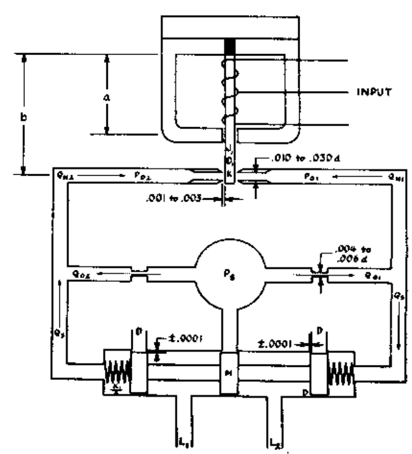

Figure 1. Critical Flow Restrictions in a Typical Two-Stage Valve

The fixed orifices of the first stage generally consist of a circular hole drilled into a flat plate which is inserted in the flow-path. The flow through the orifice may be laminar or in the transition range, depending on the particular first-stage design. The orifice diameters are generally in the range of 0.004 to 0.007 inch, depending on the first-stage flow requirements.

The diameters of the nozzles adjacent to the flapper are generally in the 0.015 to 0.020 inch range. The clearances between the flapper and each nozzle vary from 0.0005 to 0.003 inch, the total permissible travel being twice this value. Micronic filters nominally rated anywhere from 5 to 50 microns are employed upstream.

The diameter of the output spool depends on the rated flow of the valve and can vary anywhere from 0.10 inch to an inch or greater, a typical value being 0.25 inch. Both the axial clearances and the radial clearances are of the order of 0.0002 inch.

Generally, radial slots are used as flow passages rather than the complete periphery of the spool. This is done to permit longer spool strokes and thus improve linearity and reduce leakage at high pressures. Such practices also improve the contamination susceptibility of the valve to an extent.

From the above, it is quickly evident that the most critical points in the first stage are the clearances between the flapper and nozzle. On the typical valve, particles as small as 25 microns (0.001 inch) can create a temporary blockage of fluid. However, the flapper can yield and allow particles as large as 50 microns to pass, with only a transient disturbance to the valve.

The fixed orifices will allow particles as large as 125 microns to pass if such are able to penetrate the filter.

The shape and distribution of the particles, of course, enter the picture. Most particles are not symmetrical in shape and, if they approach the critical points singly, will probably pass openings slightly larger than their smallest dimension. However, if they enter in groups, blockage through "log jamming" can occur even though the minimum dimensions are only a fraction of the orifice opening. Particles with maximum dimensions smaller than the orifice opening can also clog the orifice by the process of agglomeration if present in sufficient numbers.

The smallest openings in the valve actually occur at the valve filters where the nominal ratings are generally around 10 microns. Therefore, it appears that most clogging problems should originate at the filters and this is generally the case. However, because of their small capacity, it is not desirable to use valve filters with absolute particle size ratings too small. Consequently, most valve filters allow a small percentage of larger particles to pass.

Therefore, the more contaminated the oil, the more particles will pass.

Some of the techniques discussed in the previous section for improving the filtering are:

(1) Tapping the first-stage flow at 90 degrees from the direction of the second-stage flow, such that the momentum of the large particles prevent them from entering the first-stage tap. Unfortunately, this technique is of limited value because the valve normally operates around null where the load flow is near zero, and the only particle momentum is that contributed by leakage flow.

(2) Flushing the outside of the filter by the second-stage flow. This probably is of some value. However, it has the disadvantage that the filter is subjected to full valve flow and is thus subjected to a large number of particles.

(3) Utilizing single filters of large capacity to eliminate unbalance problems created by partial filter blockage.

Designs to maximize the internal clearances involve:

(1) The use of jet pipe principles. Here, for the same leakage, the minimum dimension is a circular hole at the end of the jet pipe. This provides a much larger dimension than the cylinder area between the typical nozzle and flapper.

(2) The placement of the flapper above both nozzles rather than between them to reduce the maximum particle clearing displacement of the flapper.

(3) Use of a single nozzle which allows a greater total area between nozzle and flapper for a given leakage and, also, permits larger flapper displacements.

The clearances on the output spool may be the smallest in the entire valve when the spool is at null. Here, particles as small as 2 microns can accumulate on the metering edges, reducing the leakage flow and changing the load pressures. This is called silting. Some of these particles can wedge between the spool and sleeve in certain areas and increase the friction level or threshold of the spool or actually jam the spool.

This is probably one of the main reasons valve designers have gone to two-stage valves. The latter provide much larger forces for controlling the spool displacement than even the largest torque motors and are not as susceptible to the stiction problems. Another reason, of course, is that the Bernoulli forces increase at higher flows, further decreasing the forces available for moving the spool.

The problem of removing particles in the 2 micron size category apparently defies solution, at least with presently known filtering techniques. However, the effects of silting can be greatly reduced by dithering; that is, applying a low amplitude sinusoidal signal to the valve and oscillating the spool so as to prevent the particle buildup. However, dithering has the disadvantage of increasing spool wear and increasing leakage. These effects can be compensated somewhat by using overlapped spool metering orifices. However, this procedure tends to "soften" or reduce the valve's sensitivity to small inputs near null, thus reducing the servo response.

The best solution for the output stage appears to be the use of as large driving forces as practical from the first stage. In this connection, considerable advantage may be obtained by use of time modulation principles. The latter combine high torques with a dithering action. Large overlaps can be employed to reduce leakage and wear without the penalty of reducing sensitivity.

2. Particle Type Considerations

In addition to the particle size and distribution problem, other difficulties arise due to the nature and composition of the particles. The agglomeration tendency of some particles was mentioned before. Perhaps the most serious problems can be attributed to ferrous particles in the oil in valve designs where the oil is in contact with the magnetic circuit. These particles are attracted by the magnetic field of the torque motor and tend to adhere and embed themselves in the sides of the coil and the air gaps of the magnetic circuit. In the former case, the main flux paths tend to be short circuited, thus reducing the valve's response. In the latter case, the reluctance of the air gaps is reduced, increasing the torque motor and valve gain and presenting stability problems. The magnetic field may also tend to agglomerate the ferrous particles in other areas of the valve.

Most valves employ magnetic traps to filter the magnetic particles from the oil. However, these filters are very inefficient and of questionable value. The magnetic particles that provide the most difficulty are those that are too small to be captured by the mechanical filters in the system and too light to be easily captured by the magnetic filters. Highly efficient but cumbersome magnetic filters have been developed, but even these may not be adequate if the magnetic particle content is too large.

The best solution to the problem is to prevent the particles from coming in contact with the magnetic circuit. This may be accomplished with dry or stale torque motor designs. In the former case, positive seals such as diaphragms are used to keep the drain oil from the nozzles out of the torque motor. In the latter case, oil is allowed in the torque motor chamber but the actual nozzle oil is restrained from entering the torque motor. The restraint may take the form of a baffle which is not positive but which allows only a small percentage of the total oil flow to enter, or it may take the form of a positive seal separating the drain oil and the oil in the torque motor chamber.

The hardness of the particles is also a factor in controlling spool wear and friction. However, there does not seem to be suitable methods of separating or filtering particles based on their hardness.

D. Experimental Procedures for Determining Contamination Susceptibility of Servo Valves

1. Plan of Attack of Experimental Program

It is very difficult to predict theoretically the critical combination of contaminant size, shape, and particle distribution per unit oil volume for a particular valve or design of valve. This depends to a large extent on the fluid flow pattern, particle distribution statistics, tendency of particles to agglomerate, manufacturing tolerances, etc. Therefore, it is desirable to determine the contamination susceptibility of valves by controlled experimental techniques.

The questions then arise; how should such a program be conducted and how should the valve be tested? What type of contamination should be used and in what manner should it be applied to the valve?

First, it appears desirable to satisfy two conflicting requirements. The valve should be operated in much the same manner as it would in an actual system but should be isolated from contamination introduced by the system as far as possible. The reason for the latter is quite obvious in that any contamination introduced by the system complicates the problem of controlling the contamination added externally. The reasons for operating the valve under actual system conditions is that many deleterious effects may not manifest themselves sufficiently except under closed loop conditions. For example, the silting action described previously is not particularly significant when the valve is operated alone but may cause limit cycle instability when operated as part of a closed loop. Therefore, it was concluded that the valve should be tested in a simple closed loop servo comprising a minimum number of hydraulic components, particularly those which would be susceptible to relatively rapid wear and deterioration.

The problem of the kind of contamination that should be used was considered. One approach would be to test the valves in a system using typically contaminated oil. But what is a typically contaminated oil? How much do the oils in different systems vary from this value? This is a study in itself and was not included in the scope of the present contract.

Another approach, the one selected for this program, is to employ contaminants of known characteristics and determine to what degree the various valves can tolerate them. In specifying the contaminants, it is necessary that they simulate the general characteristics of known contaminants. While this can be done only to a point, the results should provide very concrete information along the following lines.

By employing a relatively small number of spherical particles of a certain size range, it appears possible to determine experimentally the maximum size of particles a valve can tolerate. The exact sizes cannot be predicted analytically because they depend to a large extent on interactions within the valve. For instance, will the flapper always clear itself when exposed to a particle of smaller size than its entire displacement?

By employing larger concentrations of spherical particles, the number of particles in various size ranges which can be tolerated by the valve may be determined. These results could then be compared with the results of tests employing asymmetrical particles to determine the effect of asymmetry on log jamming and agglomeration tendencies. It would also be desirable to use contaminants of varying degrees of hardness.

By using magnetic particles alone, the adequacy of a particular valve design in resisting this contamination can be determined. These not only cause difficulties in the first stage, as described previously, but because of their hardness provide a good means of increasing spool friction and therefore provide means for testing the valve's susceptibility to spool threshold increases.

Probably the main difficulty with the above approach is that contaminants of the proper size, shape, density, and materials are difficult to obtain commercially and in many cases are not available at all. The cost of producing and sorting them would be prohibitive for this program.

The only contaminants available with spherical characteristics (outside of magnetic contaminant) in the desired size ranges are glass beads. These were tested but found unsatisfactory because they deteriorated too rapidly, expediting the normal rate of pump wear, and creating a heavy background "noise" level which rendered particle counts difficult. Therefore, the idea of using spherical particles had to be discarded, although there is a possibility the magnetic contamination in the desired size ranges can be used on dry coil valves for these tests.

Irregular particles in the form of Arizona road dust are commercially available in only two size groups, coarse and fine. Fortunately, AC Spark Plug Division, Flint, Michigan, had a limited stock of sorted air cleaner dust available which was turned over to this project. This contamination was obtained in the following sizes and quantities:

0 - 5 microns	100 grams
5 - 10 microns	50 grams
10 - 20 microns	100 grams
20 - 40 microns	100 grams

It might be noted that when this stock of contaminants is depleted, no more will be available, thus making it imperative that the volume of oil used in the system be maintained at a minimum.

In the case of magnetic contamination, carbonyl iron particles are available in many size ranges and of varying hardness. The mixture selected for the tests were rated at a nominal 3 microns, although test results indicated that larger sized particles should also have been used in tests where the prime concern was the first stage. The small size particles (3 microns and less) quickly bind valve spools before any difficulties in the first stage are encountered.

The problem of obtaining repeatability with regard to getting the same distribution of particles into the oil in test after test was given considerable thought. This is a difficult task to accomplish, and one never knows with complete certainty whether he has accomplished it because particle measuring techniques are not very accurate.

In fact, one of the major obstacles in a contamination study of this type is the lack of any adequately accurate and reliable measuring technique or apparatus for determining the contamination level. After investigating the various methods currently in use, the particle count method using millipore filters was selected for this study. The choice was seemingly appropriate as, subsequently, the SAE-A6 Committee on Contamination selected substantially this same method as the proposed standard method. This method was chosen by the panel as being the best over-all method presently available for measuring contamination level. The panel together with its associated advisory board is made up of representatives of practically all the companies in the United States and Canada that are interested in contamination and its affect on servo valves.

Recent tests in which 16 companies actively participated showed that the standard method was not reliable. Standard deviations in particle count of as high as 6 between companies and 1.95 between samples were obtained; i.e., Statistical Variances of 36 between companies and 3.8 between samples resulted. However, at the panel meeting following the test, a majority of those present agreed that the state of the art was such that no better measuring method was presently available; a few minor revisions of the method were made in an effort to increase its accuracy. The above pointedly illustrates the lack of good means of determining contamination level.

Since the method of measuring contamination level is not sufficiently adequate, it made the problem of proving the maintenance of an equivalent contamination distribution in several different tests, as required for this program, exceedingly difficult. It was decided, therefore, to test several valves at once. In this way each valve receives approximately the same magnitude and type of contamination, providing at least more accurate means of comparison. This also reduces the total time for testing and cost of the program.

2. Description of Test Stand and Instrumentation

A schematic of the test stand is shown in Figure 2; a picture of the stand is shown in Figures 3 and 4. The test stand consists of a

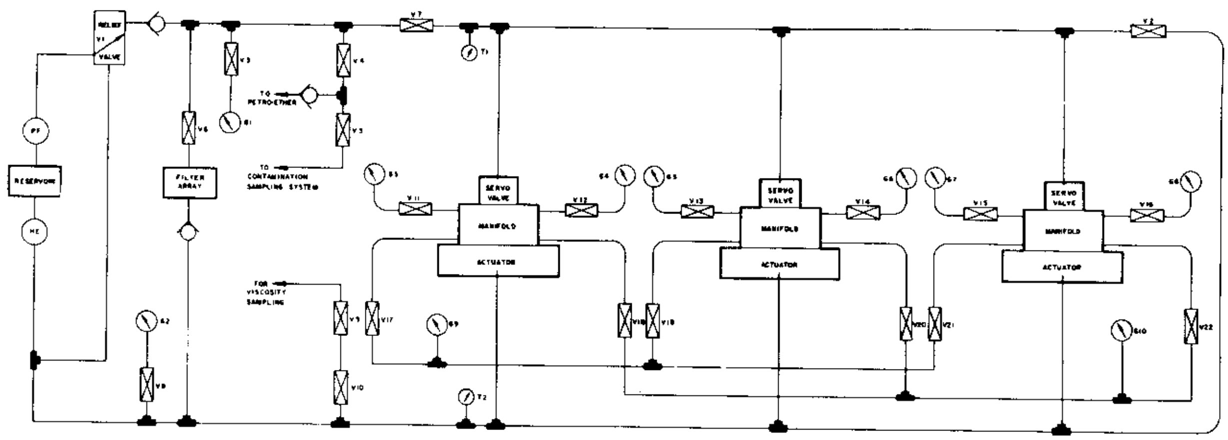

Figure 2. Schematic of Hydraulic Contamination Test Stand

hydraulic supply comprising an electric motor-driven pump, a reservoir and heat exchanger, a filter bank, a control panel, and a valve test assembly. The reservoir incorporates an agitator and contamination injection inlet.

The filter bank employs five filters rated from 2 microns to 10 microns. The control panel contains a relief valve for regulating supply pressure, pressure and temperature gages, and necessary valves and outlets for taking 75 cc contamination samples.

The valve test assembly contains three actuators, to which the valves are mounted, feedback potentiometers coupled to the actuator shafts, and necessary pressure gages and valves to permit measurement of spool end pressures and actuator chamber pressures. A more detailed description of this equipment is contained in Appendix I.

The valves are controlled from an electronic control panel. The latter contains three amplifiers for driving the valves, three recorder amplifiers for driving a pair of two-channel Brush recorders, and a low frequency oscillator plus power supplies. A picture of the control panel is shown in Figure 5.

In addition to the above, equipment consisting of a vacuum pump, filter holders, filters, and tygon tubing is used to separate contaminants from oil samples taken from the control panel during tests.

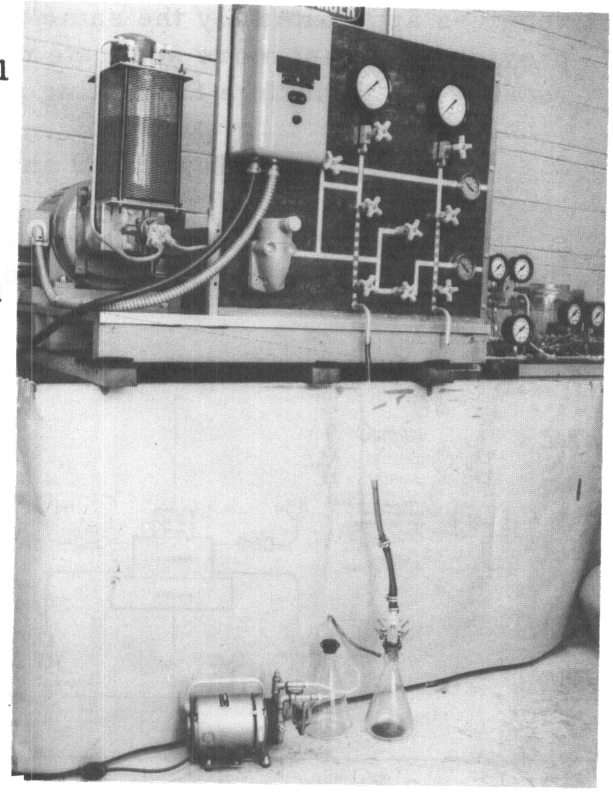

Figure 3. View of Contamination Test Stand Front Panel

3. Test Procedures

This section describes the test procedures that were developed during the first two-thirds of the test development program. As a result of the latter tests, it became evident that further changes must be made in these procedures. The recommended changes are discussed in the Conclusions section.

The step-by-step test procedures are contained in Appendix II. These will be briefly summarized below.

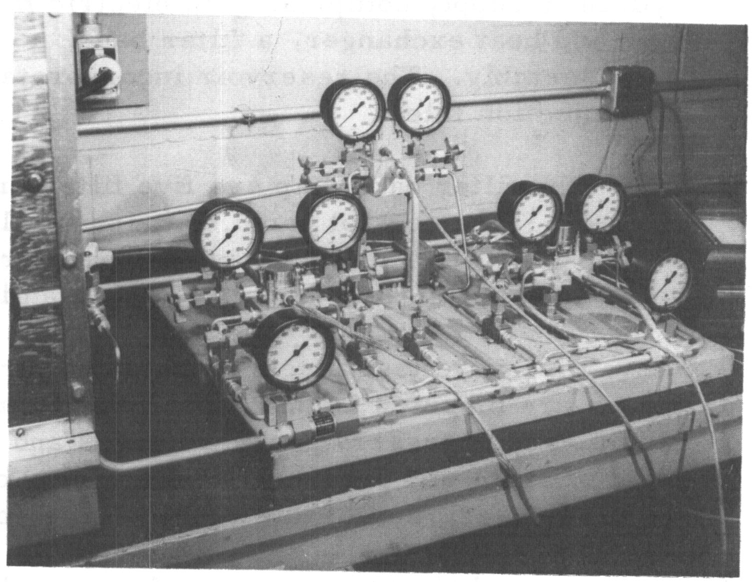

Figure 4. View of Contamination Test Stand Actuator Board

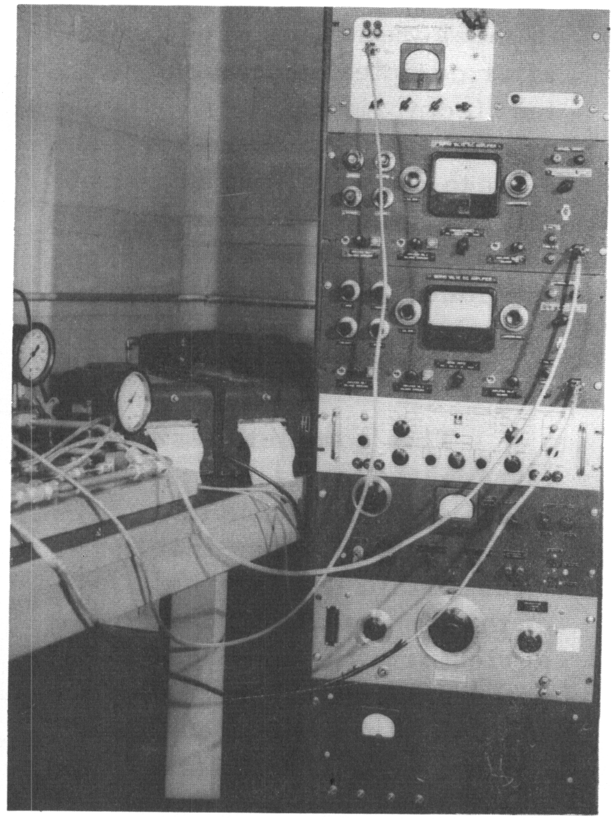

Figure 5. View of Contamination Test Stand Control Panel

The following is the general pattern of these tests. Three servo valves are operated in separate closed loop servo systems and drive unloaded actuators. Prefiltered oil is delivered to the servo valves from a single hydraulic supply. A sample of contaminant of a particular size range, say 10 to 20 microns, is then added to the system oil and the valves are operated with this amount for 10 minutes. If during this time, no malfunction has occurred, another sample of contaminant is added to the oil and the valves operated for another 10 minutes. This procedure is repeated until all three valves cease to operate satisfactorily. The amount of contaminant and time necessary to cause malfunction of each valve is recorded for subsequent analysis.

Valve malfunction is detected by monitoring Brush recordings of the differential current to each valve. Typical recordings are shown in Figure 6. Recording (a) shows a valve under normal operation; Recording (b) shows a valve that has been forced "hard over"; while Recording (c) shows a valve that has begun to oscillate.

In addition to the above, spool end pressures, actuator end pressures, and oil temperature are monitored and recorded during each run, that is, after each time contamination is added. Oil samples are extracted from calibrated traps located on the control panel once every other run. The oil is fed directly into a filter flask where the contaminants are separated. Oil viscosity is measured before and after completion of each test (comprising the number of runs necessary to cause failure of all three valves). The filter disks containing the contaminant are kept for later analysis using the "standard method".

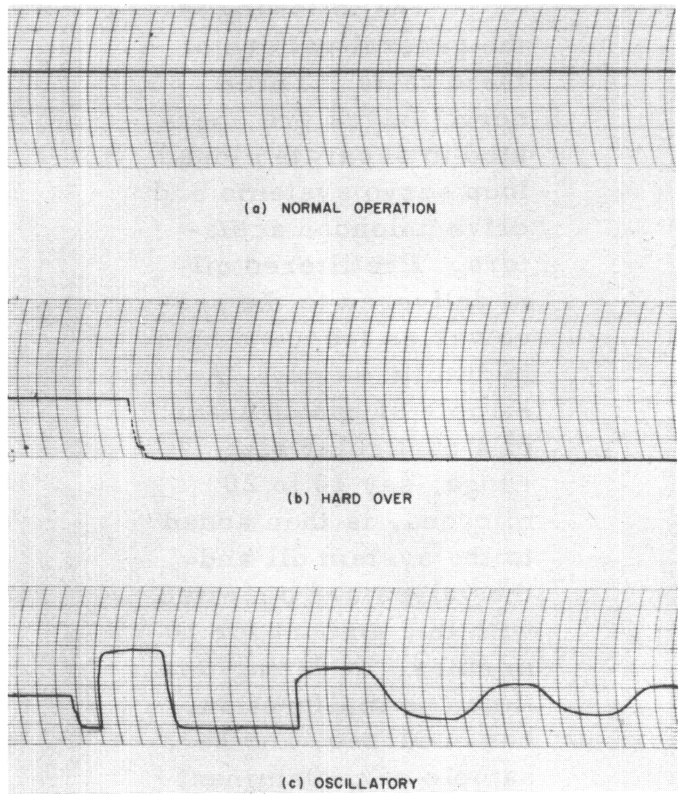

Figure 6. Error Signal as Monitored on a Brush Recorder (Depicting 3 Modes of Operation)

This complete test is repeated at least three times with each type and size range of contaminant to ascertain the validity of the test results. Each series of tests requires some preliminary preparation and testing to ascertain the original status of the valve and to insure clean oil. The tests of the valve include measurement of valve threshold, leakage at null and rated current, and no-load flow at plus and minus rated current.

The system oil is purified by flushing out the contaminated oil from the previous test and introducing oil, prefiltered through a 10 micron Cuno filter, into the system. The system oil is then run through the filter bank for a period of at least 2 hours or until a sample of the oil is clean. The fiberglass filter used in the filter bank is actually rated at 2 microns but allows occasional particles large as 7 microns to pass.

In addition to the above tests, samples of the contaminants have to be prepared and contamination measuring apparatus cleaned. This process is described in detail in Appendix II and follows the "standard method" proposed by the SAE A-6 committee, using the incidental light option. The sample sizes to be used depend on the valve, contaminant, and test conditions. Sample sizes of 10 milligrams per gallon were typical for Arizona road dust while 12.5 milligrams per gallon were typical for carbonyl iron in actual tests.

Following the contamination tests, a valve cleaning procedure is required to determine the cause of valve failure and to place the valve back in operation. The preliminary tests for the subsequent contamination tests serve as a post-mortem in establishing the status of the valves after they have been cleaned and placed back in an operating condition.

The cleansing procedure begins with the removal, cleaning, and replacement of filters. (The filters are removed one at a time for a "hard over" condition). The valve is then retested and if it fails to operate normally, the valve orifices are removed and cleaned. If upon replacement, the valve still fails to operate satisfactorily, the nozzles are cleaned. This is a progressive cleaning procedure intended to ascertain where the clogging occurs, although it is by no means always as straightforward as outlined.

The engineer conducting the test must use considerable discretion in following the proper sequence of steps. For instance, if the valve fails with spool end pressures high, the defect would not very likely be in the filters or orifice but in the nozzles. If the spool end pressures are low on the other hand, the prime difficulty is not likely at the nozzle. Oscillating conditions, generally, although not always, can be attributed to the nozzles.

Following the completion of the tests, particle counts are made of all the samples taken. The counts are then compared with the weight of contamination added and together provide useful data of the contamination tolerability of the valves tested.

4. Test Results

The major portion of the test program was spent in developing the set of test procedures listed above. Because of their developmental nature, most of the test results cannot be listed here, as they merely showed whether the particular method of approach used was satisfactory or unsatisfactory. For example, a number of weeks were spent in testing with glass beads. As indicated earlier, the test results showed that glass beads are unsuitable for testing because they quickly disintegrate and provide a lot of background clutter which make particle counting impossible. In addition, they expedited the wear of the Vickers pump so rapidly that it lasted only about 150 hours, and, during its last hours of operation, the pump could not supply 3000 psi.

In other tests, an effort was made to determine the proper size contamination sample rate and operating time per run, but the correlation obtained from particle counts was not completely satisfactory. The required size contamination sample rate was found to be a function of the type of valve and supply pressure. Valves using high quiescent flows from the first stage required lower size contamination sample rates than low quiescent flow valves. This is of course intuitively obvious, as the higher the flow, the sooner the upstream orifices will become clogged.

A figure of 10 minutes was selected for the operating time per run. This figure was based, for the most part, on the need for this amount of time in making measurements, recording data, and preparing test equipment.

Table II lists data that were obtained from a few tests of the Moog 9126, Cadillac Gage FC-200, and Bendix HR-9 (an old version). The Moog 9126 was used throughout the program for test development purposes.

TABLE II

OVER-ALL TEST RESULTS

Valve No.	Test No.	Contamination Type and Size (microns)	Total Contamination Added before Failure (mg/gal)	Total Operating Time before Failure (minutes)	Spool End Pressure Initial (psi)		Spool End Pressure Final (psi)		Test Pressure (psi)	Failure Symptom	Apparent Cause of Failure
					Left	Right	Left	Right			
Vi Moog 9126 Ser. No. 1	1	AC Dust 20-40	41.5	10					3000	Sluggish	Not the Filter
Vi Moog 9126 Ser. No. 1	2	AC Dust 20-40	10.5	5					3000	Sluggish	
Vi Moog 9126 Ser. No. 1	2a	AC Dust 20-40	21.5	10					3000	Sluggish	
V_7 Cad. Gage FC 200	3	AC Dust 20-40	10	4	1000	1000	100	100	3000	Sluggish	Filter
V_8 Moog 9126 Ser. No. 2	3	AC Dust 20-40	30	47	970	940	20	30	3000	Sluggish	Filter
V_9 Bendix Pac HR-9	3	AC Dust 20-40	30	47	380	340	100	0	3000	Hard Over	Filter Orifice

Table III provides a little more suitable data on six valves tested for Bendix Pacific for the purpose of comparing the contamination tolerability of three different first-stage designs. Valves I and IV have the standard wet coil design; II and V incorporate a baffle to discourage any flow from the first-stage drain to the torque motor; III and VI incorporate a positive diaphragm seal inhibiting the drain oil from entering the torque motor chamber.

Most of the data in Tables II and III are self-explanatory, but the portion pertaining to the causes of failure requires considerable elaboration. First, the heading "Apparent Failure" was used because it is not possible to predict with 100 percent certainty what the actual cause of failure was. The general area of failure may be diagnosed very accurately but the actual element of failure cannot.

There are several signs of failure. These are as follows:

(1) One filter or one upstream orifice clogs up, creating a "hard over" condition in which the spool moves to one side. One spool end pressure is very low, the other is higher.

(2) Two filters or two upstream orifices clog, creating low spool end pressure and little, if any, control.

(3) One nozzle clogs up, creating a hard over condition with higher than normal spool end pressures, one higher than the other (this is rare).

(4) Both nozzles clog up, causing high spool end pressures and resulting loss in control or hunting. In the latter case, the clogging may be intermittent, but repetitious.

(5) All first-stage orifices clog, and hunting results at low spool end pressures.

(6) The output spool sticks, but first stage's operation is satisfactory as indicated by normal control of spool end pressures. This has only occurred with carbonyl iron.

(7) First stage becomes unstable at a high frequency, a case of magnetic contamination at the torque motor pole pieces.

In addition, there are combinations of the above. No provisions were made for obtaining pressure readings between the upstream filter

TABLE III

BENDIX OVER-ALL TEST RESULTS

Valve No.	Test No.	Contamination Type and Size (microns)	Total Contamination Added before Failure (mg/gal)	Total Operating Time before Failure (minutes)	Spool End Pressure Initial (psi)		Spool End Pressure Final (psi)		Threshold ma at 25 psi		Viscosity (centistokes) for Given Temperature		Test Pressure (psi)	Failure Symptom	Apparent Cause of Failure
					Left	Right	Left	Right	Pre	Post	Initial	Final			
1	1	AC Dust 20-40	57.5	115	620	625	0	10			9.68 @129°F	8.4 @160°F	1500	Hard Over	Filter Clogged
1	2	AC Dust 20-40	65.0	35	625	700	50	25			9.3 @158°F	11.8 @120°F	1500	Sluggish	Filter Orifice Nozzle
2	1	AC Dust 20-40	57.5	115	520	540	0	45			9.68 @129°F	8.4 @160°F	1500	Hard Over	Filter
2	2	AC Dust 20-40	55.0	20	540	550	0/60	110/10			9.29 @130°F	8.45 @142°F	1500	Osc	Filter
3	1	AC Dust 20-40	67.5	125	600	610	100/150	150/100			9.68 @129°F	8.4 @160°F	1500	Osc	Filter
3	2	AC Dust 20-40	60.0	30	690	690	75	125			9.3 @158°F	11.8 @120°F	1500	Hard Over	Filter Orifice
4	3	AC Dust 20-40	50	64	620	620	0	25			10.9 @140°F	10.8 @135°F	1500	Hard Over	Filter Orifice
5	3	AC Dust	65	90	610	620	40	50			10.9 @140°F	10.8 @135°F	1500	Sluggish	Filter * Orifice
6	3	AC Dust 20-40	50	64	710	710	0	60			10.9 @140°F	10.8 @135°F	1500	Osc	Filter
1	4	AC Dust 10-20	80	95	630	680	45	75		0.03@ 100psi	9.84 @135°F	8.12 @155°F	1500	Hard Over	Filter Orifice
1	6	AC Dust 10-20	100	102	640	680	60	60	0.004	0.020	9.29 @130°F	8.45 @142°F	1500	Sluggish	Filter Orifice
2	4	AC Dust 10-20	80	95	575	570	15/85	85/15		0.03@ 100psi	9.84 @135°F	8.12 @155°F	1500	Osc	Filter

*Nozzle

TABLE III

BENDIX OVER-ALL TEST RESULTS (cont'd)

Valve No.	Test No.	Contamination Type and Size (microns)	Total Contamination Added before Failure (mg/gal)	Total Operating Time before Failure (minutes)	Spool End Pressure Initial (psi)		Spool End Pressure Final (psi)		Threshold ma at 25 psi		Viscosity (centistokes) for Given Temperature		Test Pressure (psi)	Failure Symptom	Apparent Cause of Failure
					Left	Right	Left	Right	Pre	Post	Initial	Final			
2	6	AC Dust 10-20	100	102	650	650	50	50	0.024	0.12	9.29 @130°F	8.45 @142°F	1500	Hard Over	Filter Orifice
3	4	AC Dust 10-20	20	20	740	720	730/50	50/730		0.02@ 100psi	9.84 @135°F	8.12 @155°F	1500	Osc	Filter Orifice
															Nozzle
3	6	AC Dust 10-20	80	82	620	620	40	70	0.020	0.080	9.29 @130°F	8.45 @142°F	1500	Hard Over	Filter
4	5	AC Dust 10-20	90	94	625	640	0	30	0.014	0.002	9.48 @136°F	7.38 @150°F	1500	Hard Over	Filter Orifice
5	5	AC Dust 10-20	100	104	640	640	70/100	100/60	0.004	0.010	9.48 @136°F	7.38 @150°F	1500	Osc	Filter
6	5	AC Dust 10-20	20	20	710	710	750/770	780/760	0.008	0.008	9.49 @136°F	7.38 @150°F	1500	Osc	Probably Nozzle
4	7	Type SF Carbonyl Iron 3μ Nom	37.5	32	700	675	670	660	0.007				1500	Stiction	2nd Stage
5	7	Type SF Carbonyl Iron 3μ Nom	37.5	32	650	650	420/850	750/370	0.010				1500	Osc Closed Loop	2nd Stage
6	7	Type SF Carbonyl Iron 3μ Nom	50	45	750	725	50	570	0.020				1500	Hard Over	2nd Stage

and upstream orifices as these are often integrally fabricated. In addition, special drilling and tapping would be required in cases where they were not so fabricated.

The apparent causes of failure, noted in Tables II and III, were based on the progressive dismantling and cleaning procedure described in the test procedures. Summarizing briefly, the filters were first removed, either together or one at a time, depending on the symptoms, and then cleaned and replaced. If the valve did not operate satisfactorily, the orifices were removed and cleaned. The final step, if the valve still did not function normally, was a nozzle cleaning. This always returned the valve back to an operating condition when the first stage was clogged.

The second stage difficulties were easy to diagnose. Spool sticking could be determined by varying the differential current and monitoring spool end pressures. Oscillations due to silting could be determined by opening the servo loop to see whether the oscillations disappeared. If they did, silting was very likely the cause of difficulty. Other components within the loop were also checked to make certain that they were not contributing to the valve oscillation.

One of the drawbacks of the above procedure is that once the filters are removed, the remainder of the hydraulic circuit in the first stage is exposed to any contaminant clinging to the side walls of the filter cavity. This contaminant can clog the upstream orifices on reassembly and one never knows for sure what contaminant was the original source of failure. In the analysis, it was generally concluded that if the failure occurred abruptly, it was caused by the upstream orifices clogging. The rate of filter clogging also provided a clue. This rate could be determined by monitoring the spool and pressures. The latter generally decreased as contamination was added, indicating buildup of contamination in the filters. Figures 7 and 8 show the fall-off of end pressures for two typical valves.

From the aforementioned tables it can be seen that the valves operating at 3000 psi failed with a much lower level of added contamination than did those operating at 1500 psi. The Cadillac Gage FC 200 failed at a contamination level of 10 mg/gal while the Moog 9126 and Bendix Pacific HR-9 failed with 30 mg/gal of contamination. The average amount of added contamination of an equivalent size that was required to clog the Bendix valves that were operated at 1500 psi was 58 mg/gal. The major reason for these differences in contamination susceptibility is related to their first stage quiescent flows.

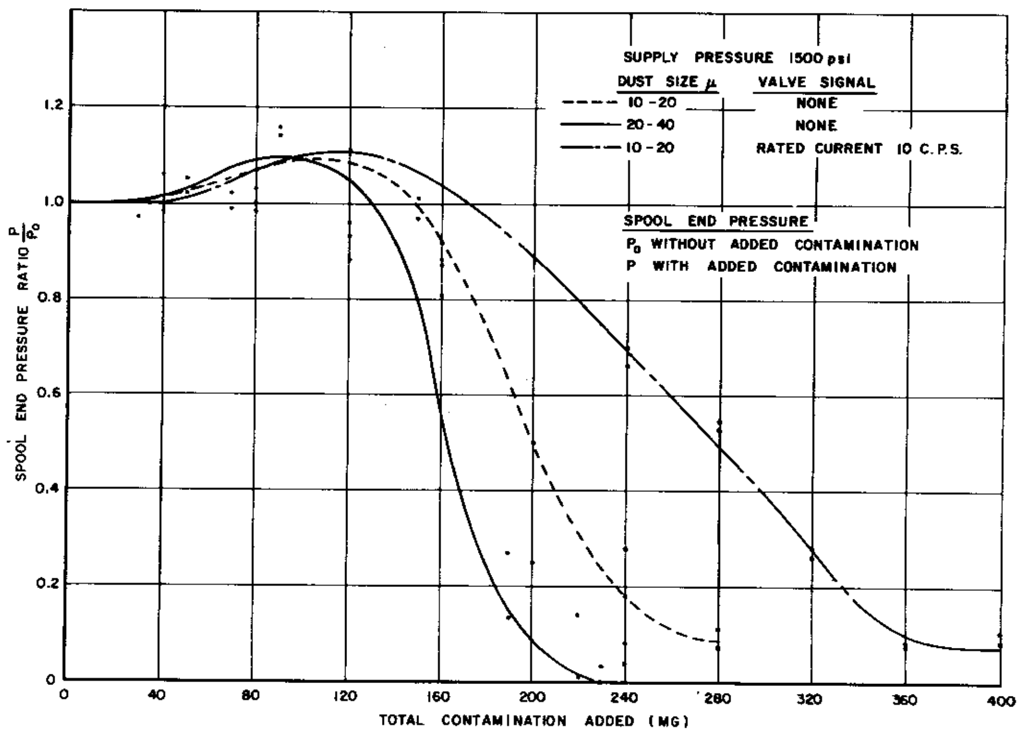

Figure 7. Spool End Pressure Ratio vs. Total Contamination Added

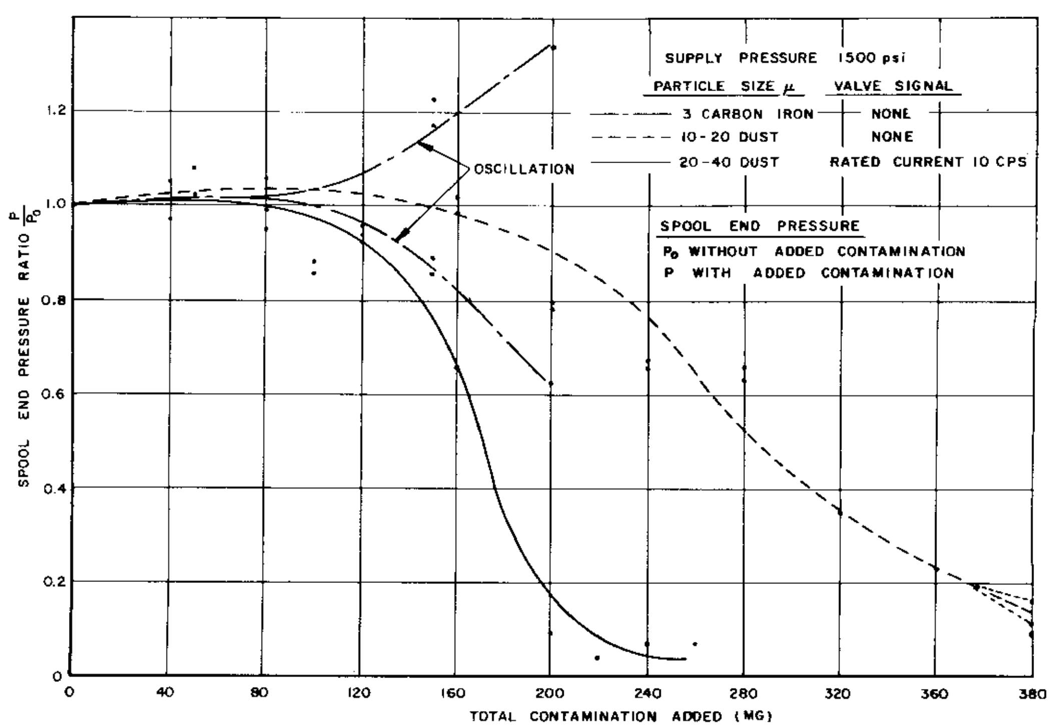

Figure 8. Spool End Pressure Ratio vs. Total Contamination Added

Moog 9126 Serial No. 2	0.064 gpm
Cadillac Gage FC 200	0.14 gpm
Bendix Pacific HR-9	0.068 gpm
Bendix Pacific HR-34 Mod.	0.035 gpm

It can be seen that the Cadillac Gage valve was exposed to approximately four times as much flow, and thus contamination, as the Bendix valves. This easily accounts for the apparent difference in contamination susceptibility due to pressure.

Aside from the Cadillac Gage valve failure, the lowest level of added contamination that caused malfunctioning was 20 mg/gal. This amounts to 0.533 gm/100 cc which is comparable to contamination weight specifications set up by other organizations, such as Boeing Airplane Company.

Table IV shows the particle counts that were made during the various tests, using the "standard method" with the incidental light option. It may be inferred from the data that the personnel making the particle counts have not fully perfected the technique. If this is so, they may have solace in the fact that they are not alone, as evidenced by the lack of agreement in the results of the SAE A-6 Committee investigation mentioned earlier. Actually, not too much reliance was placed on the particle counts during the latter stages of the test program.

A few comments are necessary with regard to this method. It appears that some standard method of checking the technique must be devised before the techniques can be accepted as a universal standard. For instance, much of the error in contamination counts can probably be attributed to initial cleanup of the glass used for obtaining samples. Standards should be set up to limit the maximum number of particles of various sizes that can be permitted in blanks which have passed prefiltered petroleum ether. In addition, some figure might be provided for the contamination level of the filter disk as it is removed from the manufacturer's container, based on the best that can be obtained under closely controlled test conditions. These data have presumably been obtained by certain experimenters, but it would help a great deal if some standard of comparison were established and published for people who are new to the field and do not have the time and funds for a research program to obtain the facts. A school for teaching these techniques may also be desirable.

TABLE IV

PARTICLE COUNTS

Test No.	Run No.	>40 (microns)	20-40 (microns)	< 20 (microns)	10-20 (microns)	< 10 (microns)
B-1	Blank	260	3,120	12,950		
	Clean	830	1,410	22,152		
	2	440	12,695	175,469		
	4	590	2,120	234,300		
B-2	Blank	1,380	14,952	23,643		
	Clean	575	5,190	38,297		
	2	Uncountable	Uncountable			
B-3	Blank	185	1,080	22,791		
	Clean	340	3,620	102,997		
	2	525	15,476	681,600		
	4	615	11,885	Uncountable		
B-4	Blank	31	260		1,030	2,030
	Clean	175	1,685		5,425	51,929
	1	520	740		10,230	138,000
	2	41	465		30,000	235,800
	4	4,777	8,141		50,660	348,000
B-5	Blank	79	89		356	24,060
	Clean	107	2,970		12,935	30,330
	1	153	147		8,845	230,253

It is believed that the method used to obtain contamination samples was adequate in that at no time was the sample exposed to the laboratory air. Of course, the disk is exposed to air during counting, a relatively long period of time. A good dust-free air-conditioned atmosphere in the counting area is necessary, especially if the contamination density is low.

5. Conclusions

The following conclusions can be made with regard to the actual test program:

(1) The test procedures devised need some revision to take into account the fact that the valve filters actually filter the system oil and reduce the contamination level. Therefore, even though contamination is added in steps, the actual contamination level does not necessarily increase but may actually decrease for certain contamination sizes. A discussion of this revision and a recommended procedure is described in the next subsection.

(2) The tests, as presently devised, are useful more as valve filter tests rather than valve design tests, because the valve malfunction in most cases results from filter clogging. These data, of course, are important to the system designer but might just as well be obtained by testing the filters alone.

(3) Better control of the contamination contributed by the system is required in these types of tests. The first pump that was used contributed nearly as much contamination as was added externally. The situation was much improved with the second pump but further improvement is required.

(4) Contaminants under 10 microns in size can be tolerated at very high levels. The valves operated acceptably when the contaminant in the size range actually formed several solid layers on the filter disk.

(5) Glass beads are not a suitable contaminant for testing electrohydraulic servo valves. Carbonyl iron might be used with isolated coil valves although their high density increases the problem of contamination control.

(6) The clogging tendency of the valve filters, and, therefore, the valve, is dependent to a great extent on the quiescent flow of the first stage and the particle size rating of the filters. A valve using higher quiescent flows for improved response should therefore employ filters with larger particle size rating if possible. It is always important that the filter particle size rating be no smaller than necessary to protect critical restrictions, so as to reduce clogging due to small particles.

(7) Small, hard contaminants, such as the normally rated 3 micron carbonyl iron quickly bind the output spools, posing a problem in duplicating actual system magnetic contaminants which are much softer.

6. Recommendations for Future Work

There are two major problems that must receive consideration before the next phase of this study can proceed. These concern the control of contamination density in the oil and the measurement of the contamination level.

As mentioned earlier, the tendency of the valve to filter the oil poses a problem in controlling the densities of contaminant for test. Here are several ways of attacking this problem.

One would be to remove the filters and test the valve without them. This is not convenient with most valves as the filters and upstream orifices form an integrated unit. In addition, this presents a very unrealistic test of the valve and does not provide information that can easily be interpreted in terms of system requirements.

Another approach would be to add contamination to the supply system and obtain a uniform distribution by circulating the oil with the filters and valves in the system. Then, the valve is placed in the circuit and operated for a short period of time so as not to clog the filters by more than 20 percent. The valve is then removed and cleaned, and the oil repurified. Contamination of a higher density is then added and the test repeated, etc. This procedure has the advantage that the valve always sees the desired contamination density. However, it involves a prohibitive amount of testing and cleaning time.

A third method would be to determine the characteristic contamination removal vs. contamination added curve for the filter and add sufficiently more contamination each run to maintain constant contamination density until the filters clog. However, such a characteristic would probably be difficult to predict and maintain.

A fourth approach is the one suggested for the next program. This involves testing with smaller amounts of contaminant than those required to cause filter clogging and conduct the tests for long periods of time, say 40 hours. The contamination level is monitored by particle count methods and maintained at the desired value by adding new contamination during the progress of the tests.

At the end of the 40 hour period, the test is repeated for a higher level but still not sufficiently high to cause filter clogging, etc. This method has the advantage of closely simulating the contamination levels in actual systems.

The following improvements to the present system and methods are also recommended:

(1) Replace the pump with a version which contributes less system contamination. A modified gear pump employing external gear drives is recommended.

(2) Reduce the noise level of the system by employing flexible couplings between the system and the pump. The gear pump would also be less of a noise generator.

(3) A softer and larger size group of carbonyl iron contaminant instead of the type used in the present phase.

(4) Continue to use the Arizona road dust as obtained from AC Spark Plug.

(5) Take samples before each valve rather than at the common inlet as was done in the present phase. The variation in contamination density due to variation in oil flow velocity through different tubes could then be observed.

7. Filter Capabilities

a. General

The need for micronic filters in military airborne systems incorporating electrohydraulic servo valves has stimulated a great deal of research and development by filter manufacturers in an effort to improve small particle filtration efficiencies and increase filter reliability under severe environmental conditions. The effort has been directed along the line of new designs, materials, and manufacturing techniques.

As a result of this work, a number of new filter designs have been placed on the market during the last few years, each intended to solve the micronic filtration problem. There is, however, considerable disagreement between manufacturers and users alike as to what constitutes the best design. An attempt to resolve these disagreements would be beyond the scope of this contract. However, an effort will be made to describe the various types of filters currently available, presenting some of their advantages and limitations, and to relate their capabilities with the contamination size requirements imposed by servo valves, as described earlier in this chapter. In addition, a brief discussion of existing and tentative filter specifications will be made.

There are three main methods of micronic filtration. They are depth filtration, surface filtration, and magnetic filtration.

b. Depth Type Filtration

This type of filtration is accomplished by absorption and by adsorption. Filtration by absorption is the entrapment of impurities due to the mechanical wetting and soaking action, while adsorption is a process of entrapment that involves a chemical attraction. The main disadvantage of this type of filtration is in the size of the filter required to remove any appreciable amount of contamination. Because of this disadvantage, the depth type of filter is not used in aircraft systems. It is used, however, on test stands and for industrial applications.

c. Surface Type Filtration

This type of micronic filtration is accomplished, as the name implies, by stopping the impurities at the surface of the filter due to the smaller dimensions of the flow passages. When the flow passages are controlled and maintained below a uniform size, no contamination larger than that size will get by the filter. However, because this type of filtration has only a single line of resistance, long particles with small diameters can pass and subsequently cause clogging due to the log jamming phenomenon discussed earlier. One of the main advantages of this type of filtration is the large capacity for accommodating accumulated contamination. This is due to the fact that the surface of the filter can be convoluted, increasing the effective area of the filter several-fold. It is interesting to note that the impurities caked onto the surface actually tend to reduce the minimum size of particles passed. Because of the large capacity per unit size characteristic of the surface type filter, it is used extensively for airborne service.

The surface type filters can be classified according to the type of elements used. The two main types are the paper element and the metallic element.

The "paper" elements are usually made up of a porous cellulous material impregnated with a phenolic resin. The impregnation is necessary to prevent destruction of the paper that would occur when the filter medium was soaked by the oil. At the same time, the resin also adds to the physical strength and stiffness of the element.

Another advantage of this type of filter is the relatively low cost of the element. This allows the element to be replaced when clogging occurs, thereby bypassing a cleaning process that would otherwise be necessary.

The disadvantages of paper filters are that they tend to disintegrate at temperatures of around 200°F and some users complain of a media migration problem (portions of the filter media found downstream of the filter) at lower temperatures. High temperature (up to 400°F) elements have been developed but have not been accepted by aircraft designers. Paper element filters are available rated at 5 microns, nominal. In other words, the manufacturers claim the filters will remove 90 percent of all particles greater than 5 microns.

The metallic element filters are of the woven mesh type, sintered woven mesh type, and sintered powder type. Actually, this last type might be considered a category of depth filter except that it doesn't use absorption or adsorption, but instead utilizes entrapment resulting from pores that form a network of tortuous paths through the filter.

The wire mesh elements are generally made up of closely woven Type 304 Stainless Steel wire-woven in Germany. The filter rating is varied by changing the type of weave and/or the size of the wire. The screens are convoluted to increase the surface area. The weave gives the filter element the effect of depth. Wire mesh filters with an absolute rating of 15 microns are commercially available. In size, they are slightly smaller than equivalent paper filters. Among the advantages of this type of filter are high temperature capabilities, greater uniformity of pore size, minimum media migration, and greater filter differential pressure capability. The latter is particularly important as these filters will withstand a full supply pressure of 3000 psi, or even higher. The ductility of the wire allows for convoluting the sheets as mentioned before. Among the disadvantages are the high cost of the filter element, and a lack of rigidity such that the convoluting process tends to vary the pore size.

This last limitation is corrected in the sintered wire mesh type elements. This element is made up by sintering the woven wire, rolling to maintain the original size, and resintering. This process, of course, increases the cost but further improves pore size uniformity and the rigidity of the unsintered screen type elements. However, some manufacturers have found the sintered type to be less vibration resistant.

The sintered powder type filter elements are generally made up of either bronze or stainless steel powders. The method most generally used for forming the filter entails placing the metallic powder in a mold, applying a predetermined pressure, and then placing the mold in a sintering furnace where bonding takes place at the points of contact through a solid-state diffusion phenomena. To obtain uniformity and reproducibility of the final product, the particle size distribution of the metallic powder must be kept to within 2 percent. This is the type of filter most generally used within the servo valve. They are rated by their manufacturers in sizes as low as 3 microns, nominal.

Since filtration occurs due to the entrapment of particles in the tortuous pathways through the sintered powders, there is less likelihood of long thin (fibrous) particles penetrating the filter than in the wire-woven type. Other features of this type of filter are high corrosion resistance and high pressure capabilities. Disadvantages of the sintered powder elements are a tendency to slough particles (Media Migration) and according to some reports, poor sustained high temperature qualities.

d. Magnetic Filters

Magnetic filters have evolved from mere magnetic traps to filters that are integral combinations of micronic filters and magnetic filters. In essence, this combination is accomplished by using a metallic screen type micronic filter as a flux return path. Manufacturers claim filters efficiencies of as high as 90 percent on magnetic particles down to 1 micron in size. The main advantage of the magnetic filter is its ability to remove particles below 5 microns. The latter can cause silting and sticking problems at the second stage. In addition, they can cause first-stage malfunction when wet torque motor valves are used.

Dry torque motor servo valves, of course, eliminate the possibility of the magnetic particle in the oil changing the torque motor characteristics. For high temperature applications, however, the tendency is to still use the wet torque motor type servo valves to eliminate the effect of high temperature differentials or hot spots at critical valve components. The disadvantages of this type of filter are its higher cost and larger size for a given flow rating.

In closing, it is to be noted that the filters described here are not designed to be used to clean dirty oil but rather to protect a servo valve in a reasonably clean environment.

e. Filter Specifications

There are four filter specifications in various stages of development. They are MIL-F-5504A, MIL-F-5504B, MIL-F-25682, and MIL-F-8815.

MIL-F-5504A is obsolete. It specifies the use of Air Cleaner Test Dust. The filter is required to remove 98 percent of all particles whose two smallest dimensions are greater than 10 microns.

MIL-F-5504B is also obsolete although it was never officially released. It differs from MIL-F-5504A in that it specifies the use of glass beads as furnished by Fram Corporation. The filter is required to remove 95 percent of all beads greater than 10 microns.

MIL-F-25682 is the latest specification and is to supersede MIL-F-5504. The filter is required to remove all particles greater than 25 microns.

MIL-F-8815 is the proposed absolute 15 micron filter specification. The specification is written to cover filters that previously were termed 5 micron units. The filter is now required to remove all particles greater than 15 microns.

It is expected that the last two specifications will be the ones that will be officially adopted.

CHAPTER III

STUDY OF LIFE AND RELIABILITY OF ELECTROHYDRAULIC SERVO VALVES

A. Evaluation and Criteria

The present and predicted use of hydraulic flight control systems in aircraft and missiles dictates the need for valid information on the reliability of the components of such systems. Failure of these systems will, or may be expected to, result in loss of the vehicle together with its human passengers, if any. This information, for maximum usefulness, should be in quantitative form and be applicable to the environmental conditions in which the particular flight control system will be operating.

A review of literature on the subject of reliability indicates the majority of authors to be in agreement with the following definition: reliability is the probability that a part, component, or system will satisfactorily perform its intended function in a specified environment for a specified period of time. Reliability is, therefore, directly related to failure frequency and numerically is unity minus the time integral over the specified time, of failure frequency per initial number of units for the item being considered while it is operated in the specified environment. Once this information is known for each component of the system, suitable calculations can be made to predict reliability of the flight control system as a unit.

Although this ideal situation appears to be reasonably well defined, certain practical factors quite often make a statistically accurate determination of component and systems reliability difficult to obtain. Not the least of these problems is the decision as to when a failure has actually occurred. As an example, performance of an electrohydraulic servo valve is described by several interrelated characteristics, some or all of which may be of importance in the particular control system where the valve is being used.

Realistically, the specifications for this particular valve should spell out such limiting performance characteristics as are required and should allow wide tolerances in noncritical areas of performance. As such, a failure to meet specifications would constitute a valid failure. In a majority of situations, however, specifications are not so carefully written and the valve's characteristics must be carefully analyzed to determine the allowable variations of each in the particular operation at hand. Deterioration of valve performance such as to cause failure in one application may not constitute failure at all in another application.

Once practical failure criteria and operational environment have been determined, the realization of a workable reliability figure from test data can be attempted. Probably the most useful type of data from which reliability may be calculated is failure rate as a function of time. Failure rate information with a high level of confidence must be obtained from a large sample of components tested to failure under the specified environmental conditions. In most situations, this procedure is time-consuming and costly. Furthermore, any small change in one or more environmental factors may easily produce a major change in results.

An alternate and less costly test method may be used when component reliability over a limited time is the information desired. A small number of components can be tested over the desired period of time in the specified environment, and the probability of the component operating successfully for this period of time can be determined from test results. Level of confidence at which this probability can be stated is a function of the number of components tested, but, in general, satisfactory results can be achieved with 10 to 20 units. As is also true of the first test method mentioned, any change in specified environment will, however, require that new tests be made.

Component failure history, in general, may be expected to follow a fairly well-defined pattern as a function of time. A relatively large rush of early failures is to be expected due to difficulties built into the component in manufacture. Following these, a period of time will ensue during which failure frequency of components remaining in service will be fairly constant. Failures will occur randomly in time, causing the number of failures to be proportional to the number of operating components remaining in the sample. The third and last phase of this pattern constitutes those failures due to actual wearout of the component.

Because of the varied nature of failures occurring during the first phase of this pattern, it is difficult to give it any mathematical description. In some cases, particularly those where excellent quality control is maintained, early failures are so few as to be undetectable in the over-all failure pattern.

Solution of the differential equation describing failure history in the second phase results in the relationship

$$R = \epsilon^{-\frac{t_I - t_2}{T}} \tag{1}$$

where R is the reliability or probability of satisfactory operation of the component during the time interval $t_I - t_2$, and T is the mean time to failure during the second phase.

Failure frequency per component at the start of the test during phase 3 of this pattern will, in general, follow a Gaussian distribution about a mean "wearout" time to failure. An equation representing such a procedure is

$$Y = \frac{1}{\delta\sqrt{2\pi}} \; \epsilon^{-\frac{(t - T)^2}{2\delta^2}} \tag{2}$$

where Y is the failure frequency, T is mean time to failure, t is the elapsed operating time and δ the standard deviation of the distribution curve.

Figure 9 represents a possible composite history of failure rate per

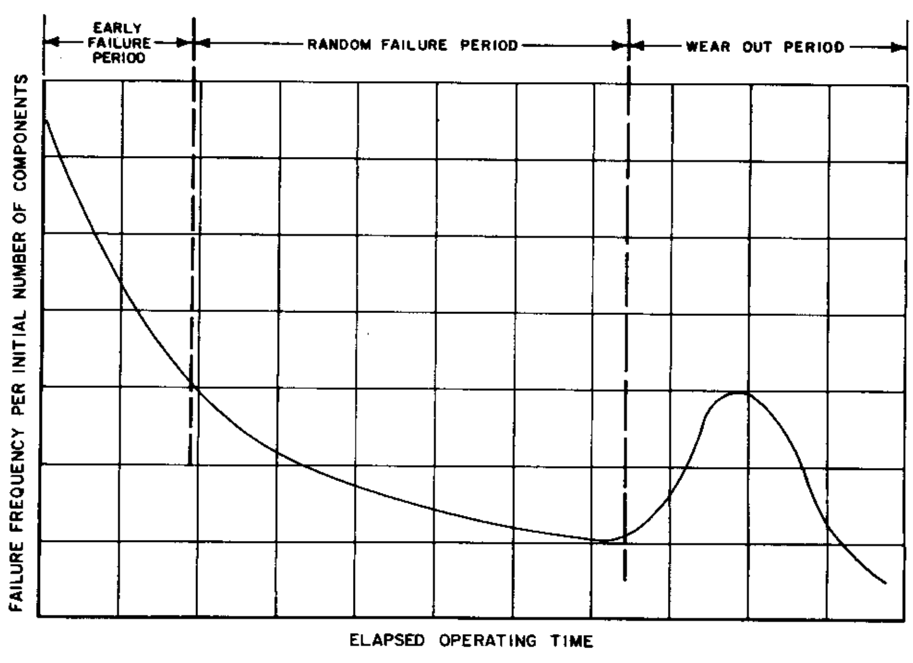

Figure 9. Idealized Composite Component Failure Frequency Distribution

component at the start of the test showing effect of the three phases of this pattern. Reliability of the unit being considered over any specific time interval, t_o to t, is unity minus the integral of this curve over that period of time. In actual practice many components will exhibit a failure pattern such that almost all failures are of the "wearout" type. Others, particularly those components that have a large number of parts or are subject to many inputs, may exhibit a uniform failure rate for a long period of time.

Presuming the accurate determination of component reliability, the mathematical determination of an accurate prediction of system reliability from this data has been the subject of many papers. A simple series combination of elements results in the reliability formula.

$$R_T = R_1 \cdot R_2 \cdot R_3 \cdot R_4 \cdot \cdot \cdot R_N \tag{3}$$

where R_T represents the system reliability and R_N represents reliability of the Nth component. For a redundant or parallel combination of elements,

$$R_T = 1 - F_1 \cdot F_2 \cdot F_3 \cdot \cdot \cdot F_N \tag{4}$$

where F_N represents the probability of failure of the Nth component. Where such systems are combined and include interaction between components and systems, the problem becomes much more complex and it is difficult to write universal formulae to fit all situations.

If no provision had been made in the development of a component for testing to failure or for determining percentage of failures during a specified time, failure data can be obtained in some cases from actual field experience. If, however, no organized plan is prearranged for the recording of such data, useful information is quite difficult to obtain. Further, if the component is subject to wide variations in environment, it will be essentially impossible to correlate failure history with any particular environmental stress. Hydraulic flight control system components, in particular the servo valve, have been subjected to little, if any, laboratory failure testing. Determination of valve reliability must therefore be attempted from such field data as are available, and presentation and discussion of these data will be made in a subsequent section of this report.

B. Aircraft Applications

Electrohydraulic servo valves are being used in two general classes of aircraft control systems, namely stability augmentation systems and autopilots. Although an electric motor drive has been employed in several autopilots, the lightweight and fast response of hydraulic systems have resulted in their use in most recent designs.

Historically, the aerodynamic development of high-speed aircraft has required increased consideration of means for oscillation damping in the yaw, pitch and roll axes. Servo systems when utilized for this purpose must provide continuous control; hence, use of electrohydraulic servo valves was a logical development. To date, the major use of servo valves in damper systems appears to be in the yaw axis. At least two companies, North American and Lockheed, have used servo valves in pitch dampers as well. No information was obtained on existence of any roll dampers as such in production aircraft. However, on those planes containing an autopilot, this mechanism serves the purpose of a roll damper.

Damper circuits are, in general, of the series type, such that the pilot cannot feel in his control stick any control surface motion produced by the damper. This action is usually obtained by appropriate mechanical coupling in the form of an extensible link into the power actuator. In one Northrop plane, however, yaw damping is obtained by use of a split rudder. The damper system drives one part of the rudder, the pilot having direct control of the other part.

Those portions of the control system directly associated with the autopilot are usually connected in parallel with the control stick. With this type of design, any motion of the control surfaces due to autopilot action is reflected back to the control stick. Mechanization of this parallel drive method has in a majority of planes been accomplished by direct mechanical coupling of the autopilot actuator to the stick.

Recent introduction of the dual input valve having both mechanical and electrical input has provided a second means of obtaining series control. As yet, this type of valve has not had universal acceptance. Convair is using it in production models of one of their planes but other manufacturers have either used it experimentally or not at all. Although use of this type of valve reduces the number of parts in the system, it does so at the expense of introducing a new relatively complex part. Aircraft engineers in general have not, therefore, been convinced that the advantages of dual input valves outweigh the disadvantages.

Whether used for stability or autopilot control, the servo valve and its associated electronic and hydraulic system have complete or partial control of the aircraft during its normal operation. Consideration must therefore be given to the consequences of failure in such a system. Information obtained during the survey phase of this contract was such as to indicate that to date all autopilot and stability augmentation systems have been designed so that the plane can still be manually controlled if the automatic systems were to fail.

For this condition to be met, however, requires that the plane be aerodynamically stable without automatic control. In a few of the recent aircraft this condition has only been marginally met. As an example, it is reported that failure of the yaw damper on one type of aircraft causes relatively severe oscillations. Although this plane theoretically can be flown to a landing in this condition, pilots have been reluctant to do so.

Another possibility exists if one of the automatic control systems should fail suddenly, particularly if a hard over position of the control surfaces was the result. All systems about which information was obtained have some form of limited authority for the damper, but if failure were sudden, a certain finite time would be required for the pilot to become aware of and take action to compensate for the new conditions. In this time interval the aircraft could possibly enter into a violent maneuver. In the present designs such a possibility has been prevented by the use of "G" limiters or by switching the damper out of the control loop at critical altitudes. In some aircraft, failure of the damper system results in its actuator being spring recentered.

Control system design engineers in the aircraft industry essentially agree that future high-speed aircraft will require continuous stability augmentation while in flight. Although this fact indicates complete dependence on the servo valve and its associated systems, the engineers concerned with such problems are not at present willing to design planes with this feature. Apparently, it is hoped that some alternate solution to the problem will be found in time.

C. Reliability Considerations

1. Oil Contamination

Because of the small orifices and relatively low forces present in some sections of an electrohydraulic servo valve, this component is highly susceptible to malfunction from suspended particles in the oil. Various means of filtration and innovations in valve design have been used to reduce the probability of failure to a low value. In addition, attempts have been made to establish a practical means of measuring contamination level and in turn to determine maximum safe levels for proper valve operation. This general subject is discussed in detail in the oil contamination section, Chapter II of this report, and reference is made to the material contained therein.

2. Oil Seals

A large number of "O"-ring seals are generally used in hydraulic control systems and these are normally quite effective. For minimum or negligible leakage, careful attention must be paid to O-ring groove and part mating surface dimensions and for this reason new systems may experience leakage. O-rings are subject to damage from age, high temperature, some synthetic fluids, and nuclear radiation, so the material used must be consistent with the type of service expected.

Dynamic O-ring seals are usually kept to a minimum but when used, the rings are subject to wear and must be periodically replaced. Experience has indicated that O-ring leakage can in the majority of cases be discovered by ground crew personnel before any serious trouble develops; however, the possibility of loss of oil while the plane is airborne is always present.

3. Pumps

Aircraft type pumps of necessity must be small and lightweight. This advantage, however, is obtained at the sacrifice of long life. The pump manufacturers have rated their products for a specific number of operating hours and have run life tests to prove the validity of their claims. Apparently little trouble has been experienced with pump failure during rated lifetime since no comments of this type were made by any aircraft manufacturers visited.

Tests made for the oil contamination phase of this contract showed evidence of two interesting pump life characteristics. Initial tests with a Vickers 3909 aircraft pump in a closed system resulted in the production of powdered yellow color nonmagnetic contaminant in the oil. Since the system was all steel except for brass parts in the pump, it was concluded that the contaminant was brass and was being produced by the pump. Later, additional contaminant in the form of glass beads and AC air cleaner test dust was added to the system. Under influence of these additives, the rate of production of brass contamination was greatly increased and the pump failed appreciably before the limit of its guarantee period.

4. Feedback Components

Several types of position feedback components are available for use in an electrohydraulic control system. If a dc system is employed, some form of linear or rotary potentiometer would be required. Experience with wire-wound pots in this service has in general been unsatisfactory due to wear on the contacts and wire. A high resistance plastic element potentiometer has recently been introduced which should eliminate the wire breakage problem. Very little experience has been built up on this component but the majority of comments indicated satisfactory operation.

To circumvent the wire-wound pot problem, several designs use ac feedback with a differential transformer as the feedback component. This system has proved satisfactory and appears to have good reliability.

5. Other Components

Certain other hydraulic components may be part of the control system loop, including an accumulator and one or more transfer valves, as well as the piping, filter, heat exchanger, fittings, and actuator. No information was obtained on the expected reliability of these components. The problem is apparently not severe, however, as no specific comments were received concerning their poor reliability. One exception to this statement exists in the form of O-ring or packing leakage on actuators and is an example of the difficulty in obtaining a good dynamic seal.

D. Valve Failure Data Analysis

Following a preliminary investigation of the field, it became apparent that only a limited amount of information was available on servo valve failure history. Although two sources of such data were found, no organized program existed for the obtaining and processing of same. As a result, such information as was obtained was uncoordinated and only of preliminary value.

Each manufacturer of servo valves may be expected to have a certain number of valves returned from every shipment. Reasons for these returns are many and varied and in many cases are not related to a valve failure as such. Two manufacturers cooperated by providing information on their returned valves, one to the extent of allowing his records to be examined by project personnel. Results from both companies were quite similar and have been edited for presentation in the following table. The percentage

Reasons for Failure	Failure Distribution		Percentage Return	
	Wet Motor (percent)	Dry Motor (percent)	Wet Motor	Dry Motor
1. Torque motor contaminated	38.2	0.0	3.62	0.00
2. First-stage openings plugged	1.7	2.2	0.16	0.07
3. O-ring leakage (external)	39.2	37.8	3.72	1.20
4. Defective coil	4.2	22.7	0.40	0.72
5. Excess spool friction	5.6	8.2	0.53	0.26
6. Null off center	2.7	2.2	0.27	0.07
7. Valve noisy	3.2	10.4	0.30	0.33
8. Damaged flapper	0.5	10.4	0.05	0.33
9. Miscellaneous	4.7	6.1	0.45	0.19

of valves returned because of failures on incoming inspection at the user's plant, during system inspection at the user's plant, or in the aircraft before transferral to the armed services, varied between 3.2 and 9.5 percent. Reasons for returned valves in situations where a failure during inspection or operation was indicated, together with approximate percentage in each category, are shown in the above tabulation. In all cases, the valves were of the two-stage type with double nozzle flapper first stage and spring restrained spool second stage.

Reference to Item 1 of the above tabulation shows the apparent advantage in improved reliability of the dry torque motor valve. Since the percentage of the latter type of valve which was returned was less than one-half that of the percentage of wet torque motor valves returned, this advantage is further confirmed.

Of the remaining failure mechanisms, O-ring failure is the most prevalent. Although many such failures result only in slow leakage and are not catastrophic, improved O-ring performance would be desirable.

It is interesting to note that considerable effort is now being spent on a study of the servo valve's first-stage susceptibility to contaminated oil. If the above tabulation is any indication, only a relatively small amount of trouble may be expected from this source when oil such as is to be found in most valve users' test stands and newly finished equipment is employed.

A second source of valve failure information was the aircraft and autopilot manufacturers. Although it was initially hoped that a large amount of data would be obtained from this source, only one company made any effort to correlate what meager records they had. As a result, the data collected consist essentially of those facts recited from the background and experience of the engineers interviewed.

A total of five aircraft manufacturing companies were visited on this phase of the contract. One had no comments relative to use of servo valves in airplanes. A second was experiencing considerable difficulty with the problem of keeping the hydraulic oil free of contamination. Sintered bronze filters were being used but a change to sintered woven steel filters was contemplated. As a result of the dirty oil, servo valve life was quite short, in many instances less than 100 hours. Source or type of contamination being produced had not been determined but work was being done along these lines.

The third company reported no significant difficulty with servo valves and expected the valves to last as long as the plane. Service life of the plane was expected to be approximately 2000 hours. A fourth company expected and was obtaining a service life of 500 hours from servo valves. Over a period of two years, they had 29 valve failures in 1200 installations. This company operates all valves for at least 25 hours before installing them in a plane.

The fifth company had not used servo valves in a variety of applications but had installed one valve in the yaw damper in each of 840 planes. Out of these installations, only five valve failures have occurred. There have also been 83 failures due to actuator leakage, 30 feedback potentiometer failures, and 2 failures because of contaminated oil. Maximum flight time on any one valve has been 250 hours.

All of the valves discussed above were of the conventional single electrical input type, although as far as the users experience is concerned, no attempt was made to distinguish between products of the various valve manufacturers. As mentioned earlier, only one production application of the dual input type of valve is believed to exist at present. The manufacturer supplying valves for this application reported that no valves had been returned for failures which had occurred after the initial 10 hours of operation. Approximately 1300 of these valves had been shipped at the time the information was obtained.

E. Means of Improving Life and Reliability

Analysis of the valve's operation and experimental evidence indicates that high contamination levels in the oil tend to cause rapid malfunction of the valve. Although adequate filtration apparently reduces this problem to one of small magnitude in most systems, an improvement in the valve's ability to operate with dirty oil would definitely improve system reliability. As explained in Chapter I of this report, two vulnerable locations are the nozzle-flapper orifice and the second-stage spool. Several means have been proposed to reduce the contamination sensitivity of the double nozzle flapper first-stage, spring centered second-stage valve. A discussion of the majority of these means follows:

(1) Dual First Stage

Use of a dual, parallel first stage can theoretically increase valve reliability in accordance with the expression for redundant elements, provided that means are available for switching from the disabled element to the remaining good element at the appropriate time. Such a switching circuit would of necessity introduce some complexity of its own. A simpler but somewhat less reliable arrangement would be to use two first stages including nozzles and flappers directly in parallel. Both stages would operate together normally. If one nozzle or orifice should plug and disable one stage, continued operation of the parallel stage should provide sufficiently acceptable control until such time as the aircraft could be landed and repairs made.

Although this approach would improve reliability over that of a single first-stage valve, the improvement may not be as great as expected. Reason for this statement lies in the fact that operation of the two first stages will not be independent. Both will be subject to the same contaminated oil and even if separate filters are used preceding each stage, both may be expected to clog in a similar manner. It is probable, therefore, that the time interval between disablement of one stage and disablement of the parallel stage would usually be relatively small.

(2) Larger Torque Motor

Use of a larger torque motor permits application of greater first-stage pressures and in turn results in greater forces applied to the spool. From the standpoint of improved reliability and reduction of coulomb friction effects on the spool, a large torque motor is desirable. It does, however, result in a larger and heavier valve which for some applications would be prohibitive.

(3) Time Modulated First Stage

The application of a time modulated signal to a properly designed torque motor and flapper, instead of the conventional proportional signal, appears to have some advantages from the standpoint of improved reliability in both the first and second stage. Although such has not been proved, it is believed that the periodic flapper motion from one nozzle to the other will tend to flush particles of a larger size through these orifices than would occur with the nominally centered flapper of a conventional valve.

Since, except for the transit time, the flapper is hard against one nozzle or the other at all times, pressure supplied to move the spool is always maximum. Also, because of flapper motion, an effective dither action is produced within the valve. Susceptibility of this valve to spool stiction is therefore appreciably less than a valve with the conventional proportional first stage. Reliability of the system is not, however, improved to the extent inferred, since the electronic circuitry associated with this valve is complicated by the necessary addition of the amplitude modulated multivibrator. Also, the valve contains no spool springs or means of internal feedback and must be operated closed loop with proper compensating networks in the feedback path to maintain stability.

(4) Jet Pipe First Stage

Recent developments have included a valve using a jet pipe first stage. From the standpoint of its ability to operate with contaminated oil, this valve appears to have some relative advantages. These are (a) a single large first-stage orifice, (b) balance first-stage oil passages with only one filter, and (c) relatively high pressures available to the spool. Although the diameter of jet used is smaller than a conventional valve nozzle, it is approximately five to seven times as large as the minimum dimension of the cylindrical nozzle-flapper orifice. One filter in the supply to the jet is required rather than one in the supply to each nozzle as in the conventional valve, which fact should improve first-stage balance.

(5) Modifications

Certain modifications can be applied to a conventional valve which appear to offer advantages in improved reliability from oil contamination effects. Larger spool forces can be obtained by use of a large diameter spool. This action alone would result in greater flow forces and larger spool mass, but by use of slotting and drilled-out spool ends, both items could be reduced to their former values.

Spool forces could also be increased by use of flow force compensation, but such action can be taken only at the expense of modifying the valve's static and dynamic response. Another possible modification would be the use of nozzles with reduced diameter. This action may be expected to produce greater differential pressure on the spool for a given amount of differential current. It will also, however, affect the no-load time constant and have other secondary influences which may be detrimental to valve performance. Most manufacturers use a nozzle diameter of approximately 0.02 inch which has apparently proved optimum.

In addition to effects of contaminated oil on the servo valve, failure of other components in the control loop must also be considered in the overall goal of improved reliability. As mentioned earlier, failure of wire-wound sliding contact feedback potentiometers has presented a problem in electrical feedback to the amplifier. This problem may be circumvented by elimination of the electrical feedback and substitution of direct mechanical feedback from the actuator to the torque motor armature. At least one manufacturer has such a unit available and it appears to offer some improvement in system reliability.

Another philosophy which has been used in an attempt to improve control system reliability involves the use of complete or partially complete dual systems. If the necessary added weight and increased space can be tolerated, this scheme improves reliability in accordance with the formula for redundant elements, as presented in Section A of this report.

In the past, choice of which of two parallel systems was being used was at the direction of the pilot. With present and future high performance aircraft this cannot be done and automatic means must be provided for sensing failure in one system and rapidly switching to the other. One autopilot manufacturer has proposed a situation where three parallel systems are used. The output of each is sensed individually and the aircraft automatically placed under the control of the two which most nearly agree.

Each of the several schemes described above should in one or more ways improve reliability of aircraft hydraulic flight control systems. In turn, each has certain disadvantages which must be weighed against the improvements. At present not a great deal of accurate information is known about the reliability of an electrohydraulic servo valve and other system components, particularly as a function of the various environments they must encounter. A combination of experience and reasoning has led engineers to those areas of valve design where improvements could be made which should increase reliability. This effort needs to be supplemented by a large amount of statistical information; for optimum design of future aircraft control systems, considerable work must be done in this direction.

F. Life Test

1. Life Test Philsophy

Failure of a part due to wear occurs as a function of time and environment with a Gaussian distribution about some mean life time under steady-state environmental conditions. Since a servo valve is composed of several parts, a mean wearout time for each part may be expected, although the first part to wear out will of course cause the valve to fail.

In a general view, one may consider time as one of the several concurrent environmental factors to which a valve is subjected. As such, any test procedure designed to measure life characteristics would logically result in a relationship between life and intensity of the particular characteristic including time being considered. As indicated above, this relationship should be in the form of a Gaussian distribution.

From a practical viewpoint, except for time, a majority of the environmental characteristics will reach and not exceed some maximum value of intensity. Using these maximum values, a test could be run which would provide failure or "wearout" information as a function of time under maximum intensity environmental conditions. To be of value, this test would of necessity have to be conducted under conditions known to exist in the specific application of interest. Deviation from these conditions could materially change the results.

2. Wear Measurement Methods

Several methods were considered for determining spool and sleeve wear, and the resultant effect of this wear on valve performance. These may be grouped into three categories: (1) radioisotope techniques, (2) direct physical techniques, and (3) valve performance measurements.

a. Radioisotope Techniques

One means of determining the amount of total wear on either the spool and/or its mating sleeve involves the irradiation of these parts by neutrons in a reactor. The parts would then be placed back in the valve and the valve operated in a closed system with small oil volume. Amount of wear would be determined by a count of the radioactive isotopes in the oil after specified periods of valve operation.

This method has the advantage that a wear rate may be determined after a relatively small number of hours of valve operation. For proper use, however, these data must be related to a wear pattern and, in turn, to a relationship between wear and valve life for the particular valve being tested.

Disadvantages of the method include the uncertainty introduced by the assumption that all radioactive wear particles will be evenly distributed in the oil, since particle traps could exist in the system piping and cause error in the resultant measurements. Another disadvantage is the fact that this method does not determine the location of wear but is useful only in measuring total amount of wear. In addition, since at least one part of the valve will be radioactive, certain minimum handling precautions must be observed.

b. Direct Physical Methods

Direct physical methods include both optical and mechanical measurement techniques. Diametral clearances in a majority of servo valves are of the order of 0.0001 to 0.0002 inch and tolerances of lineal dimensions at the orifice edges are of an equivalent order of magnitude. For wear measurements to be significant, it was felt that accuracies at least as good as 10 percent of the clearances and tolerance dimensions would be required. This accuracy requirement eliminates a majority of the conventionally employed optical and mechanical wear measurement methods. In addition, the cylindrical shape of the spool and the importance of ascertaining wear on the square land edges further complicates the measurement task. After a limited investigation involving microscopic examination, optical comparators, and devices for measuring surface roughness, it was concluded that within the economic limits of the contract, no satisfactory optical or mechanical devices were available for wear measurements on the spool and/or sleeve.

c. Valve Performance Measurements

Since the effects of wear on valve performance rather than the wear pattern itself are of direct interest, measurement of deviation in performance from initial results as a function of valve use should provide the desired data directly. Of the several performance criteria normally measured, quiescent flow or leakage appears to be most closely linked with the effects of spool and sleeve wear. In certain applications, particularly those employing small capacity power supplies, minimum leakage is of primary importance.

3. Test Procedure

As an example of this latter type of test, the following procedure was devised to obtain information on servo valve life as demonstrated by spool wear.

The valve was manifolded in the conventional manner and the output ports connected directly together. Provision was included, however, for removing this short circuit connection and blocking the ports so that valve leakage or quiescent flow could be measured.

Environmental conditions were maintained within ±5 percent of the following nominal values:

(1)	Supply pressure	3000 psi
(2)	Oil temperature	125°F
(3)	Ambient temperature	72°F

Oil used in the test system was MIL-O-5606 and filtration employed consisted of (1) a magnetic filter, (2) a 25 micron screen, (3) a nominal 10 micron paper filter, and (4) a nominal 5 micron paper filter. A microscopic analysis of the oil using procedures presently being considered by the A6 committee of the SAE resulted in a particle count per 100 milliliters of oil of 1200 particles greater than 25 microns in size and 16,000 particles between 5 and 25 microns. Particles less than 5 microns were not counted but the number was much higher than for either of the other size ranges.

The actual test procedure consisted of an initial measurement of valve quiescent flow. Following this measurement, a sine wave signal of 30 percent rated differential current amplitude and a frequency of 4 cps was applied to the torque motor. Stated environment was maintained and the process stopped after approximately 400,000, 700,000, and 1,000,000 cycles of operation. Each time quiescent flow of the valve was remeasured under identical conditions, the premise being that any significant effects of wear would be evidenced by increased leakage.

4. Evaluation of Test Results

The valve employed in these tests was a Moog Model 21-100 which employs the conventional two-stage nozzle flapper design with a dry torque motor. Results of quiescent flow measurements in accordance with the above procedure are shown on Figure 10. As indicated, the initial flow was approximately 25 percent less than the flow measured at 400,000 cycles. Subsequent measurements at 700,000 and 1,000,000 cycles showed negligible change in quiescent flow.

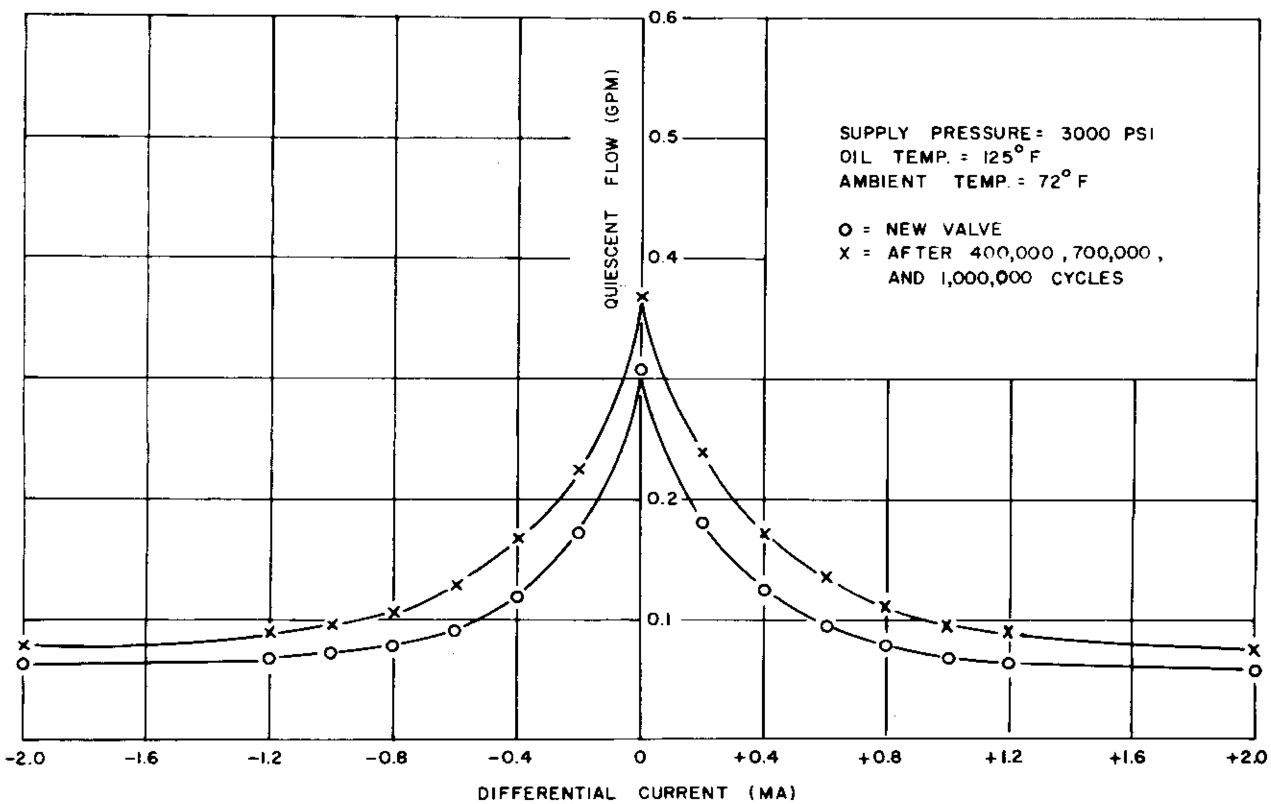

Figure 10. Quiescent Flow vs. Differential Current for Moog 21-100 Valve

Since only one unit was tested, the statistical value of the results is negligible and no conclusions can be drawn relative to the effects of spool and sleeve wear. However, the procedure employed seems satisfactory and given a sufficiently large sample, it is expected that significant conclusions could be reached. Any change in environment would require that new tests be run.

CHAPTER IV

EFFECTS OF NUCLEAR RADIATION ON ELECTROHYDRAULIC SERVO VALVES AND ASSOCIATED CONTROL COMPONENTS

A. Introduction

One of the probable requirements of flight control systems in future aircraft is that they be capable of subsisting and operating in nuclear-radiation environments. Two sources of nuclear radiation are anticipated. One of these is the nuclear powerplant in the aircraft itself, and the second is radiation emanating from nuclear bomb blasts. Both of these are apt to subject the airplane to relatively large amounts of nuclear radiation, although at much different dose rates. For instance, the bomb blast may subject the aircraft to a particular dose in a matter of seconds that takes several hundred hours to accrue as a result of radiation from the powerplant. Thus, not only the dosage but also dose rate may be an important consideration in component design and selection.

There are two different approaches to this problem. One of these is to provide sufficient shielding to reduce the radiation level to an acceptable level while the second is to develop components that can withstand sufficiently large amounts of radiation. The high weight penalty of the former precludes its general use except for protecting personnel within the vehicle. Thus, the major emphasis must be placed on developing materials and components which will withstand the radiation levels expected.

Little information is presently available on the radiation levels expected, as many of the airframe designs are still in a preliminary stage. Data which are available are classified and cannot be listed here. However, sufficient published information is available to permit the conclusion that material which absorbs an equivalent dosage of 10^8 rads probably will have considerable utility in airborne application. By an equivalent dosage is meant the amount of gamma radiation and neutron flux, singularly or in combination, that can cause the same amount of damage in a material. Such a term is an approximation at best, but is particularly useful for the dosages in the range listed above, where damage is the result of excitation and ionization, whether it be caused initially by neutron flux or gamma radiation.

It was the purpose of this task to investigate the problems associated with operating electrohydraulic servo valves and associated equipment in a nuclear environment and to test a suitable unit under gamma radiation.

The associated equipment of primary interest were the actuator to which the valve is manifolded, and the feedback sensor connected to the actuator.

The program had the following objectives:

(1) From a literature survey and visits to cognizant organizations:

(a) Determine the nature and dose rate of the radiation expected from the two sources, nuclear powerplants and bomb blasts.

(b) Ascertain what materials and fluids are best suited for airborne hydraulic equipment operating under nuclear radiation, and determine what hydraulic components were commercially available that could be used for the test program.

(2) Design, select, and fabricate an assembly consisting of a servo valve, actuator, and feedback sensor which appears suitable for operating in airborne nuclear environment.

(3) Formulate a test program for evaluating the capabilities of the above unit under gamma radiation and perform the necessary testing in the Cook Electric Company Cobalt facility.

The following sections describe the results of this work.

B. The Materials and Design Problem

1. Anticipated Radiation and Temperature Levels

As of the time of this writing, no Air Force specifications had been established on the nature and magnitude of radiation levels of nuclear aircraft. Some data are available on anticipated levels but this information is quite speculative in that final aircraft and propulsion systems designs have not been fixed. For the above reason, and because such data would require classification of this report, they will not be lised here. However, this information was used as a yardstick in evaluating and selecting components for this investigation. As indicated in the Introduction, published data indicate that an equivalent dosage of 10^8 rads is a good figure for determining the suitability of materials for general aircraft use.

Although the statement of the problem for this task did not include any reference to interreaction of the nuclear and temperature environments, it became quickly evident from the initial investigations that the temperature problem, which is briefly discussed in TR 55-29, Part V, could not easily be separated from the nuclear-radiation problem as both would exist simultaneously in a nuclear airplane.

A study of the interreactions of these two environments must eventually be made if satisfactory component designs are to be accomplished. Thus, while no effort was spent in this study investigating these interreactions, components were selected that could withstand high temperatures as well as high nuclear-radiation levels for probable use in future investigations. The temperature range considered was 400° to 1000°F.

2. Servo Valve-Actuator-Feedback Sensor Design Problem

Figure 11 is a schematic of a typical valve-actuator assembly, illustrating critical points insofar as nuclear radiation is concerned. In general, these are also the critical points when high temperature applications are considered.

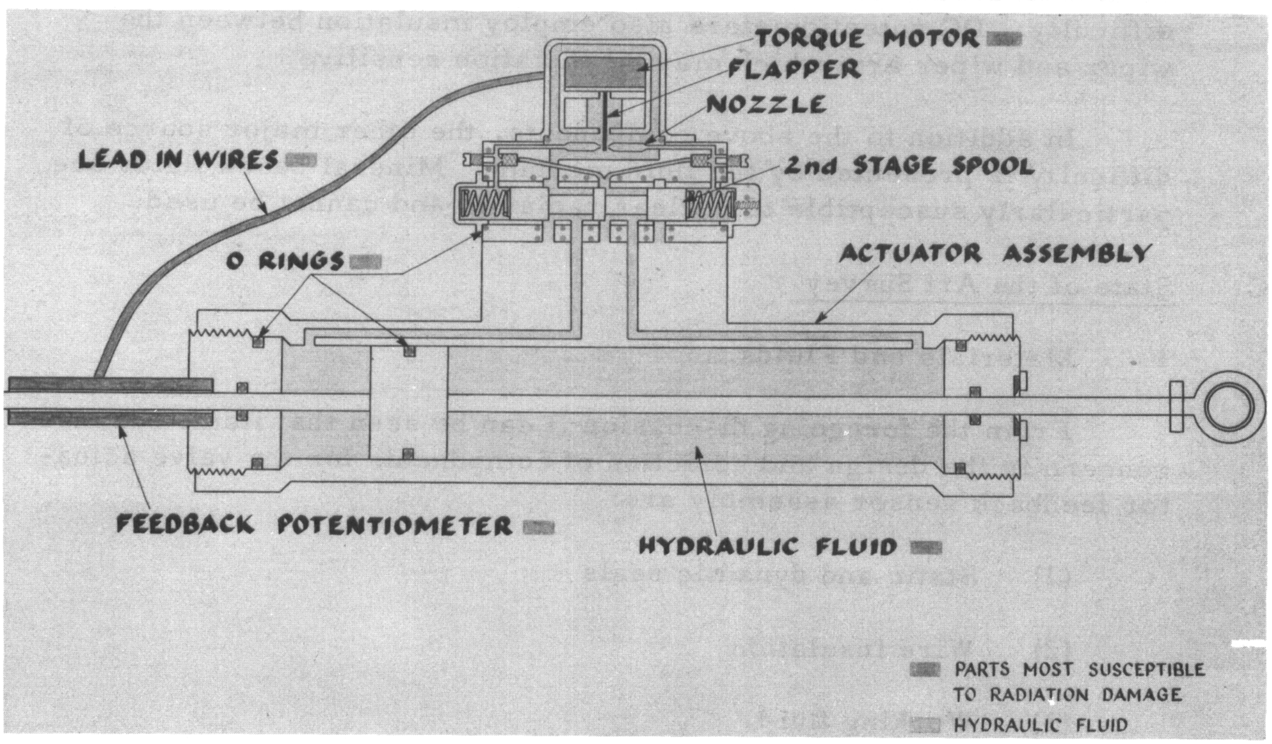

Figure 11. Parts of Valve-Actuator Assembly Most Susceptible to Radiation Damage

Little difficulty is expected with metallic materials from the radiation dosages under consideration. This should be the case whether the parts be built of steel, aluminum, beryllium, copper, etc.

Concerning the electrohydraulic servo valve, difficult problem areas are static O-rings used throughout the valve for sealing purposes and on the manifold between valve and actuator. No dynamic seals are employed in servo valves.

Another possible trouble source in the servo valve is the torque motor, which requires insulation around coil and lead-in wires.

The main problem areas of the actuator, besides the manifold sealing problem already mentioned, center around the dynamic seals, both on the piston and on the output shaft. Although the sketch shows a double ended actuator, the type most commonly used employs only one shaft, thus requiring only one shaft seal.

The feedback sensor may be ac or dc excited. In either case, insulation around coil wires and lead-in wires presents the main difficulty. DC potentiometers also employ insulation between the wiper and wiper arm which may be radiation sensitive.

In addition to the above components, the other major source of difficulty is presented by the fluid medium. Mineral based fluids are particularly susceptible to nuclear radiation and cannot be used.

C. State of the Art Survey

1. Materials and Fluids

From the foregoing discussion it can be seen that items of most concern in the design and selection of components for the valve actuator feedback sensor assembly are:

(1) Static and dynamic seals

(2) Wire insulation

(3) Working fluid.

The remainder of the elements are metallic in nature and should not pose a serious problem for the radiation dosages under consideration.

Various seal materials and elastomers have been irradiated in both reactors and gamma facilities during the last few years, and for the most part, being organic in nature, they have demonstrated a very low resistance to nuclear radiation. However, two materials, Viton A and Buna N, have shown a considerable resistance to radiation, having 25 percent radiation damage levels of 1×10^8 rads or better. (In this case the 25 percent threshold level indicates the point at which any one of the following properties of the material deteriorates beyond military specifications: stress relaxation, elongation, tensile strength and shore durometer hardness). Viton A, however, has a much higher resistance to high temperature of the two and is therefore receiving greatest attention at present.

Other high temperature materials such as teflon and the silicones have demonstrated a considerable weakness to nuclear radiation. Table V lists approximate temperature and radiation damage levels for some of the best known seal materials.

The type of material best suited for a particular application depends on the design of the seal. Viton A and Buna N appear best suited for O-rings while glass teflon might be used for backup rings, if needed. Packing type seals used for dynamic sealing may require ceramic asbestos, metallic asbestos, or pliable metallic materials. Of course, stainless steel O-rings and reed type sealing elements are available for static and dynamic sealing, respectively, but it does not appear that these have reached the state of development to satisfy all leakage requirements.

Various types of wire insulation are under study and investigation for nuclear application. Among those found suitable for the present study were aluminum and magnesium oxide, polyethylene, and glass without boron. The oxides, in addition, have the advantage of having very high temperature capabilities. Other insulating materials such as mica, porcelain, and quartz have high radiation and high temperature resistances, but are difficult to acquire in a form suitable for wire insulation. Conventional electrical insulating materials have been found unsatisfactory.

Most of the fluids presently in use in hydraulic systems are unsatisfactory for radiation environments. Mineral based oils, in particular, break down quickly under radiation. The very high temperature silicones are not much better.

The best radiation resistance has been offered by MIL-O-8200, a silicate ester which has been operated successfully at dosages

TABLE V

TEMPERATURE AND RADIATION LIMITS FOR VARIOUS SEAL MATERIALS

Seal Material	Maximum Temperature (^{o}F)	25% Threshold Limit (rads)
Viton A	550	1×10^{8}
Buna N	300	2×10^{8}
Teflon	450	4.5×10^{5}
Silicon 550	480	7.5×10^{6}
Neoprene	200	3×10^{7}
Polyethylene	210	1×10^{8}
Glass-teflon	550	1×10^{9}
Ceramic Asbestos	600	2×10^{9}

exceeding 3×10^8 rads. OS-45, another silicate ester, has not proved as satisfactory although it is superior to mineral based oils by a large margin.

2. Components

Very little is presently available in the form of components developed specially for nuclear-radiation application. Most of the present effort is concerned with irradiating standard components and fluids under static and dynamic conditions to determine their radiation limits. This process is called screening and has provided some interesting data. For instance, it was found that certain materials and oils behave more satisfactorily when operated dynamically than when operated statically. For oils this may be explained by noting that an increase in viscosity caused by polymerization during irradiation is counteracted by a decrease in viscosity due to working of the oil.

Convair has tested in their Fort Worth reactor a simple hydraulic circuit using a Vickers pump which discharged oil through a fixed orifice. The only pump modifications consisted of the replacement of the standard organic seal inside the pump with one made of Buna N. OS-45 oil manufactured by Monsanto Chemical Company was used. The system received an integrated dose of 2×10^7 rep and operated normally throughout the test. Visual inspection upon completion of the test indicated no abnormal wear or formation of varnishes or sludges on the internal pump components. A slight increase in viscosity, flash point, panel coking, and acidity were found.

Convair also has tested a group of standard differential transformers to an approximate dosage of 4×10^7 r gamma radiation and 1×10^{15} nvt of neutrons in the energy range of 2.3 mev with no detectable deterioration in performance. No special insulation was used.

Lockheed Marietta Division has irradiated a flight actuator and servo valve in the Cobalt 60 facility of Cook Electric Company to a dosage of 5×10^7 rs. This equipment was fabricated from standard parts and operated at an oil temperature of 200°F. A Cadillac Gage FC-2 servo valve was used in combination with a F 104 yaw damper actuator. MIL-O-8200 oil was supplied by a New York Air Brake pump driven by a General Electric 3 hp induction motor. A piston type accumulator was used. The oil was filtered by a 10 micron Aircraft Porous Media filter.

Viton A O-rings were used in the pump, actuator and accumulator. The accumulator backup rings were made of a Viton A asbestos compound. Woven glass and polyethylene were employed for electrical insulation.

The system was operated dynamically and excited by a random noise generator. Some of the pertinent test results were:

(1) The rate of gassing of the oil was low.

(2) Teflon backup rings used on the actuator were still intact but brittle.

(3) The Viton A material was pliable at the end of the test but developed a compression set.

(4) The accumulator lost its charge due to coke buildup on the internal surfaces. (This unit is perhaps one of the components most difficult to design for nuclear application.)

(5) The viscosity of the oil showed little change.

Lockheed plans to irradiate the same system in their new nuclear reactor. A 400°F test in the reactor is also planned.

The Nuclear Section of the Flight Control Laboratory at Wright Air Development Center is presently conducting a screening program of control components and instrumentation used in military aircraft. However, no valves or actuators, other than the ones described herein, have been irradiated as yet. The irradiation of the differential transformers discussed earlier is part of that program.

D. Experimental Program

1. Plan of Attack

The experimental program can be divided into four parts: selection of components to be tested, design and fabrication of the test stand, formulation of the test procedures, and finally the conducting of tests in the Cobalt 60 gamma facility.

The equipment to be irradiated included an electrohydraulic servo valve, a linear displacement actuator, and a feedback sensor. This equipment was to be operated as part of a closed loop servo with the electronics located outside the radiation cell.

The actuator was to be loaded by a spring-mass combination of a suitable size to create large load pressures in the valve when excited by sinusoidal inputs. In addition, it was necessary that the mass oil resonant frequency be sufficiently low so that compressibility variations due to oil gassing and changes in valve characteristics could be detected. Since the load was also subjected to radiation, it was important that it employ no materials of low nuclear-radiation resistance.

A hydraulic supply, to be located outside the radiation cell, was also required. This unit was to have a capability of providing flows of 3 gpm at 3000 psi and should have automatic shutoff capability in the event of leakage or other system failure to permit unattended testing.

The components to be tested were to be operated dynamically with varying inputs simulating actual operation in an airplane. Both static and dynamic performance checks were to be made periodically to detect any deterioration of components during the progress of the irradiation of the equipment under test.

2. Design and Selection of Components

Because of economic limitations of this program, no development of new components could be considered. Instead, the design of the assembly had to be based on existing component designs, although material substitution and minor redesign could be permitted where necessary. Fortunately, a sufficiently large number of advanced designs have been developed for high temperature applications which could be applied to this problem with only the need for minor material substitutions. Generally speaking, components and materials that have a high temperature resistance are also suitable for nuclear application. The following discussion briefly describes the components selected and alternatives that were considered.

a. Electrohydraulic Servo Valve

As discussed earlier, the standard servo valve employs a number of O-rings which are used as static seals. For nuclear-radiation application these must either be eliminated or made of radiation resistant materials. The complete elimination of the O-rings involves a major redesign in most cases and therefore was not specified for the valves selected. Special radiation resistance materials were specified for both O-rings and coil insulation.

Three valves were considered for this program, the Cadillac Gage FC-2 and FC-200, and the Moog high temperature valve. A special stainless steel version of the FC-2 valve had been operated successfully for several hours at temperatures in excess of 500°F, using Viton A seals. However, its high price and long delivery time eliminated it from further consideration.

An FC-200 valve was ordered but canceled by mutual agreement when it became clear that Cadillac Gage could not deliver the valve in time for tests. Viton A and glass had been specified for the O-rings and wire insulation, respectively. The coil was to have been wound with silver wire on an American Lava alsimag No. 196 (magnesium silicate) coil form. The use of plastic or organic materials was not allowed.

The valve ordered from Moog (see Figure 12) is a version of the 400°F high temperature valve and was delivered for testing. All O-rings are made of Viton A. The torque motor coil is constructed of copper wire coated with boron-free ceramic and silicon insulation.

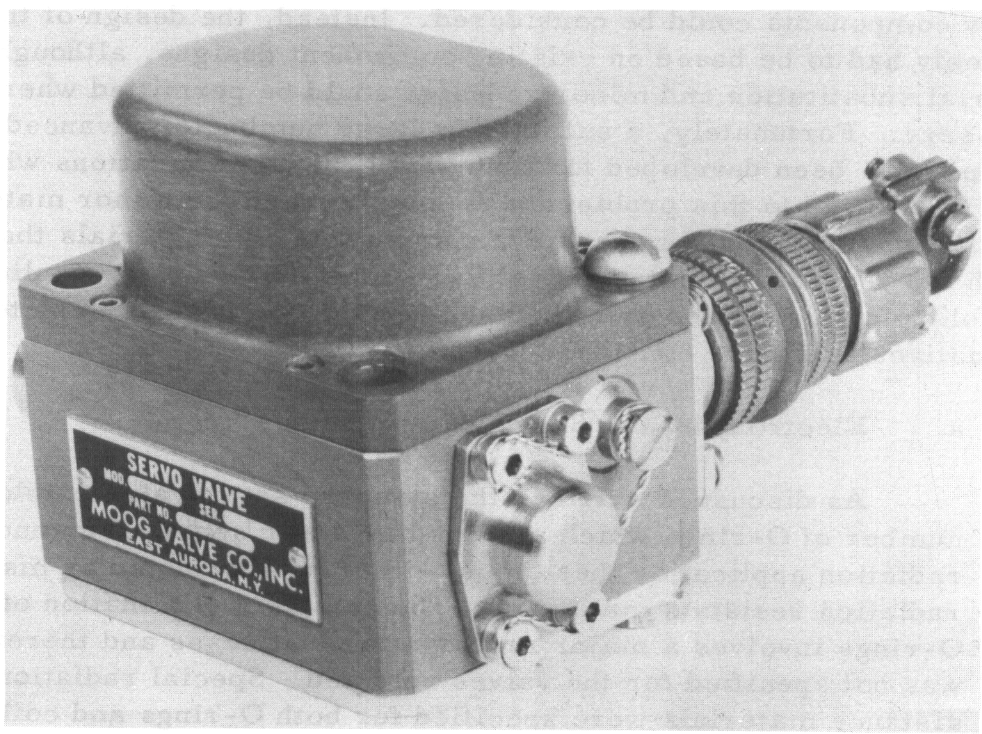

Figure 12. Moog Valve for a Nuclear Environment

b. Actuator

The main problem in selecting hydraulic actuator designs for a high temperature nuclear environment revolves around the rod end seals. The piston seals are not a serious problem since the leakage across them is internal. Metallic piston rings, which reduce leakage across the piston sufficiently to eliminate dynamic response difficulties, are the best choice as they are highly radiation resistant and can withstand high temperatures.

In most actuator designs the static seals, except for those on the manifold, can be eliminated. Metallic O-rings or Viton A O-rings may be used for the manifold.

The rod seals must provide positive sealing without excessive friction and must have reasonably good life characteristics. (In this case life may be defined as the time period elapsed until the leakage becomes excessive).

A number of actuator designs were investigated from the standpoint of the rod sealing problem. Among these were:

(1) The standard aircraft actuator, in which the organic O-rings used as rod seals were replaced by Viton A or Buna N substitutes.

(2) An industrial design that employs a conventional stuffing box with a packing gland, preferably of the spring loaded type, to drop the entire cylinder pressure. The sealing material is metallic.

(3) An actuator employing the new reed type seals such as those manufactured by Fulton Sylphon Division.

(4) A two-stage pressure drop design in which piston rings are employed to drop the cylinder pressure to drain pressure, and then a bellows, the ends of which are fastened by leakproof fits to the housing and shaft, to seal the drain pressure.

(5) A second two-stage pressure drop design employing a high pressure drop bushing to drop the cylinder pressure to drain pressure, and then a stuffing box to seal the drain pressure.

All of the above designs would probably provide satisfactory operation for a reasonably long time period. However, it appears that the two-stage pressure drop seal typified by (4) and (5), although somewhat more complex than the others, offers the most from the standpoint of reliability and long life. Design (1) is a stopgap type at most, as the Viton A or Buna N seals should deteriorate considerably more rapidly when used as dynamic seals rather than static seals. Design (2) poses a friction problem and, in addition, requires frequent servicing. Design (3) requires extremely smooth shaft finishes and has not received sufficient service experience to warrant consideration here.

The two-stage pressure drop design has the advantage that if the drain seal became defective, the control actuator is not disabled. In addition, small leaks can usually be detected before complete seal rupture occurs, thus permitting the replacement of the seal before failure. Design (5) appears superior to (4) in that bellows are not yet available that will provide the necessary long strokes without exposing large areas to drain pressure. These create steady-state forces on the actuator shaft which can be balanced out for normal operating drain pressures in single ended actuators by adjusting the cylinder to shaft diameter ratio. Pressure surges, however, upset this balance and subject the actuator to large unbalancing forces.

The stuffing box does not present this problem, and, in addition, operates with relatively small friction loads at the low drain pressures.

An actuator incorporating the features of the latter design is manufactured by Aeroproducts Operations, Allison Division, General Motors (see Figure 13) and was purchased for this program. The actuator design has been employed in an aircraft turbine engine exhaust nozzle control at ambient temperatures up to 1000°F.

The all-metallic seal design utilizes self-energizing metal O-rings for the static seals and metal chevron seals composed of stainless steel ribbon, asbestos fibers, and mica flakes for the low pressure seal. The actuator is rated for 3000 psi, has a 1-1/2 inch bore, a 2 inch stroke and a 7/8 inch rod. Piston sealing is accomplished by a single, 2 piece cast iron piston ring. Piston rod and head are chrome plated.

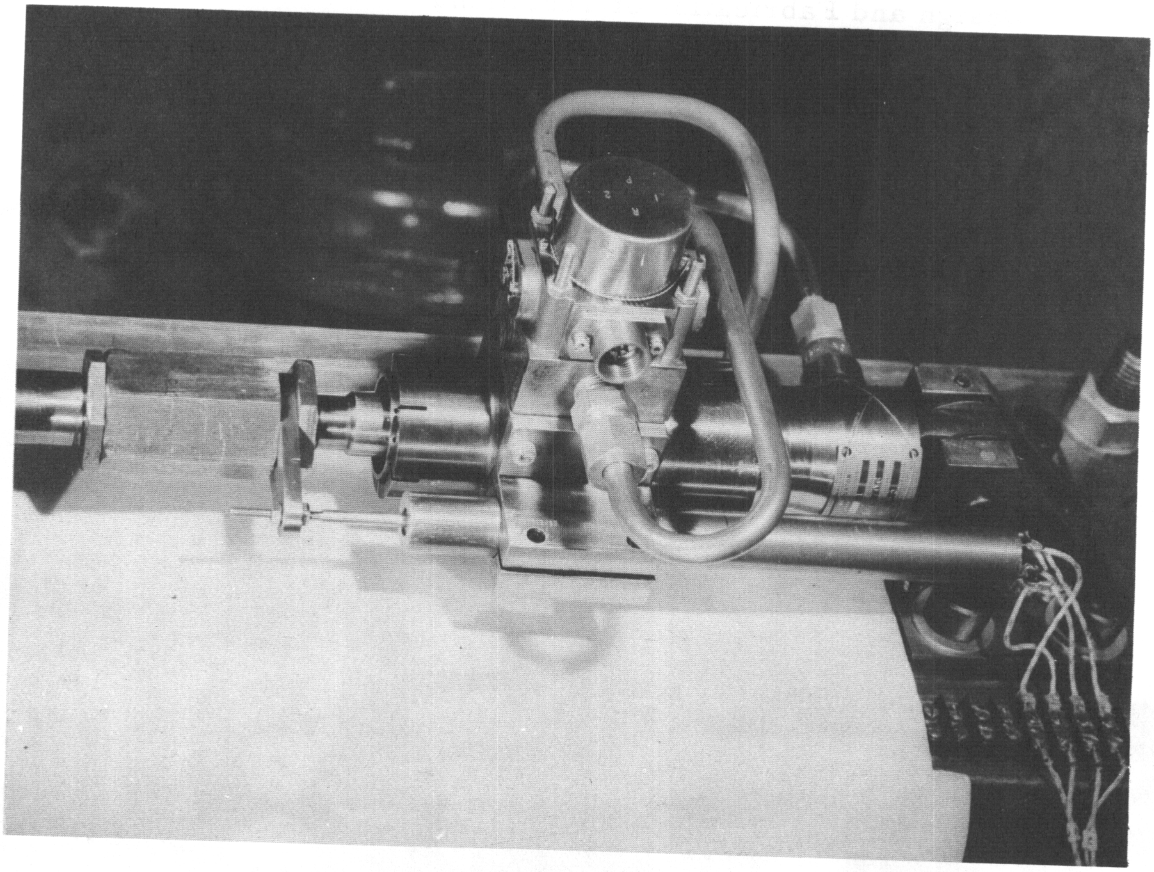

Figure 13. Aeroproducts Operation, Allison Division, General Motors Hydraulic Actuator

c. Feedback Sensor

A linear motion ac pickoff was chosen for this application to obviate problems that might arise due to potentiometer wear during tests. A unit employing anodized aluminum magnet wire was purchased from Schaevitz Engineering. They also provided price quotations on units using glass insulation, but these were rejected when it was found that the glass was not completely free of boron.

Other organizations were contacted with regard to linear motion pickoffs, but no other quotations were received

3. Design and Fabrication of Test Stand

a. Hydraulic Supply

The hydraulic supply is a complete packaged unit and is separate from the load setup. It consists of a 5 horsepower motor driving a Vickers Aircraft type hydraulic pump with flow capability of 3 gallons per minute at 3000 psi pressure (see Figure 14). The reservoir has a capacity of 1-1/4 gallons.

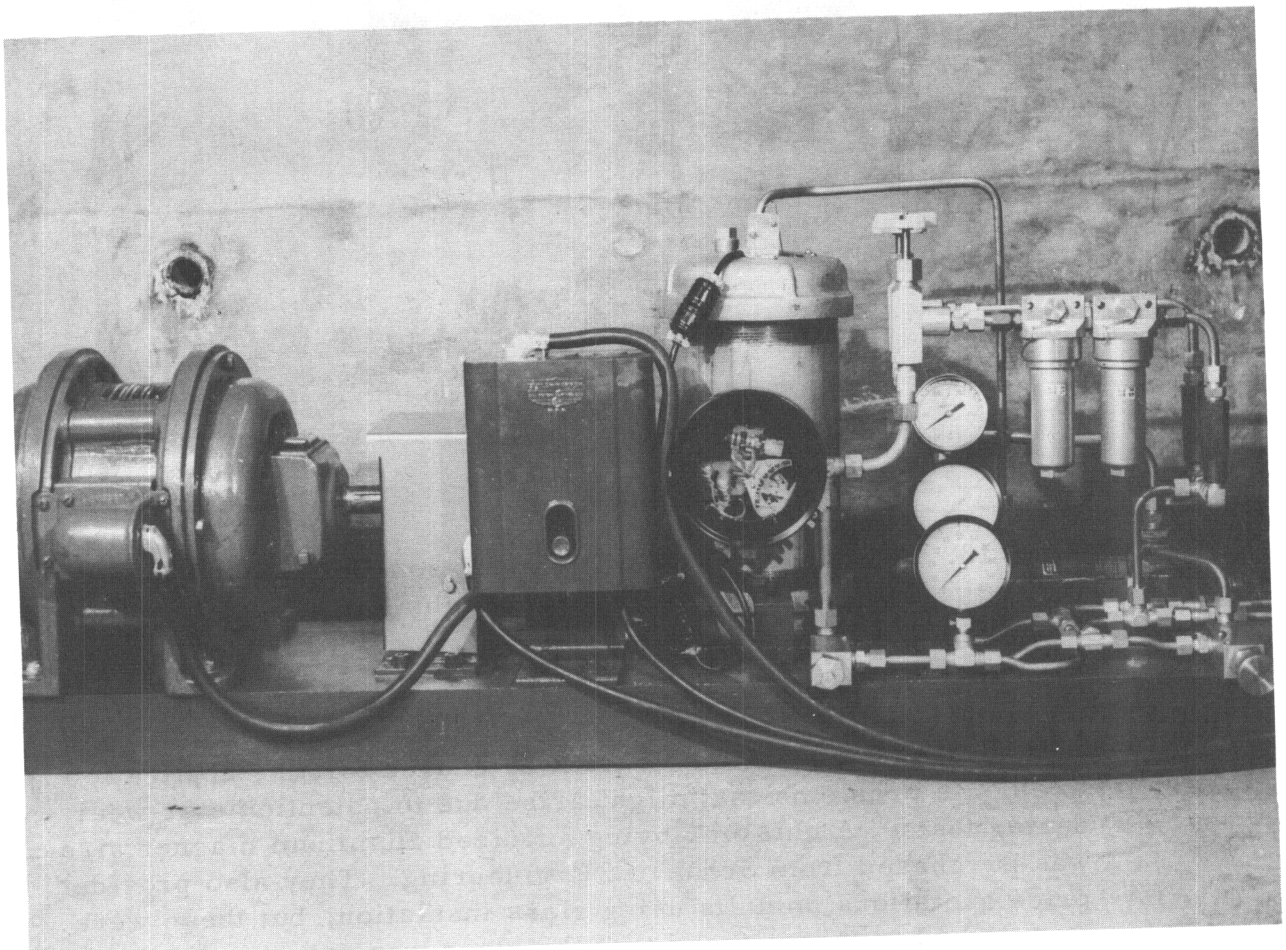

Figure 14. Hydraulic Supply Assembly

The system oil is continuously filtered with 2-5 micron and 10 micron metallic filters manufactured by Purolator Products, Incorporated. Provision is made for cleaning the filters without the necessity of a shutdown. The pump, motor, heat exchanger, reservoir, relief valve, filters, thermometer and pressure gages are all mounted on a 4 foot long 12 inch channel.

This assembly will be located outside of the test cell and it will not be subjected to radiation.

For safety reasons and to allow the system to operate unattended, an extra relief valve which serves as a safety valve is used and set for 50 psi above the operating pressure. In addition, a reservoir low level and high temperature electrical switch wired in series with the motor starter is incorporated to shut the system down in the event of leakage or high oil temperature. MIL-O-8200 fluid was selected as the hydraulic oil.

b. Load Setup

The load for the actuator consists of a 50 pound mass restrained by two centering springs (see Figure 15). The mass

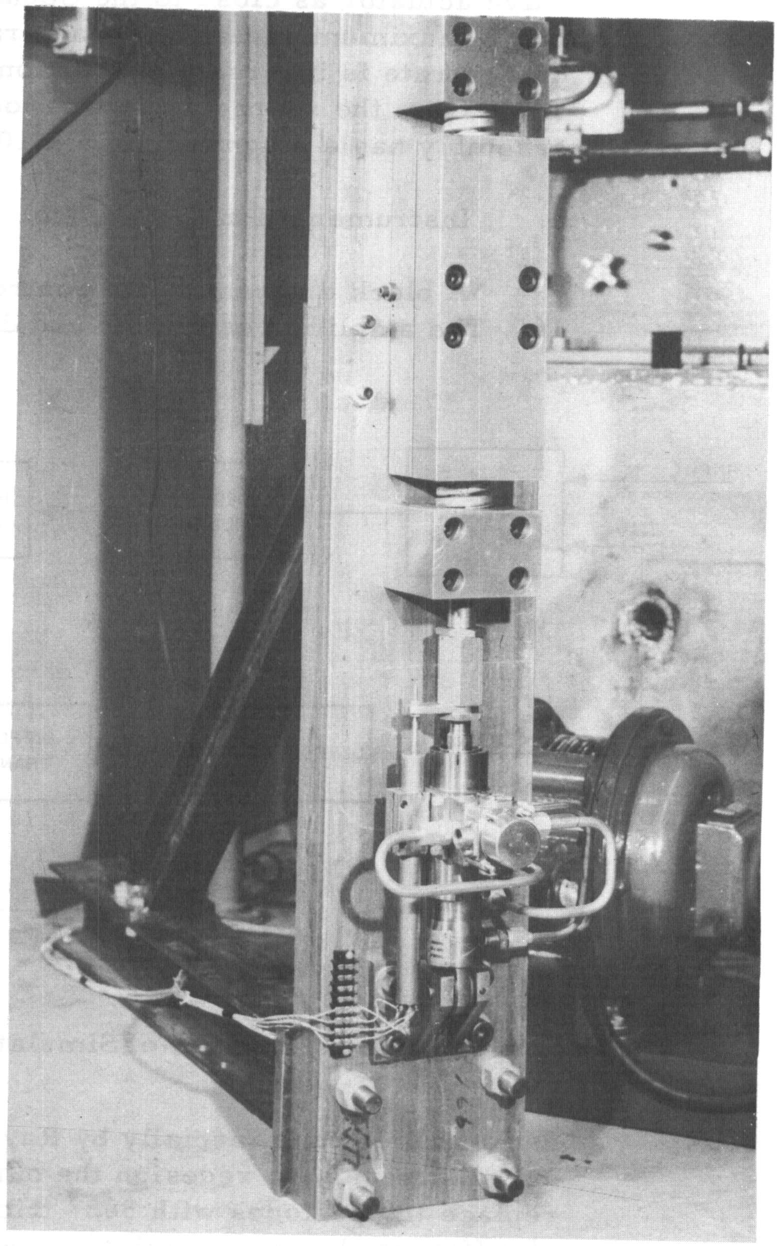

Figure 15. Dynamic Spring Mass Load Assembly

is clamped to a 1 inch shaft that extends through the center of each coil spring. The shaft then is supported and guided at both ends by linear ball bearings. The shaft is secured to the actuator by means of a coupling. Both the actuator and load assembly are bolted to a 4-1/2 foot long 8 inch aluminum channel. The channel is mounted vertically and is supported by two beams. It is mounted in this manner to permit placement of the valve actuator as close to the Cobalt 60 source as possible so that a maximum radiation dosage rate may be obtained. The dosage rate is inversely proportional to the square of the distance from the source and at the location of the valve actuator assembly has a magnitude of 5×10^5 rs.

c. Instrumentation and Control

A block diagram of the control system is shown in Figure 16. The amplifier oscillator and demodulator are part of a

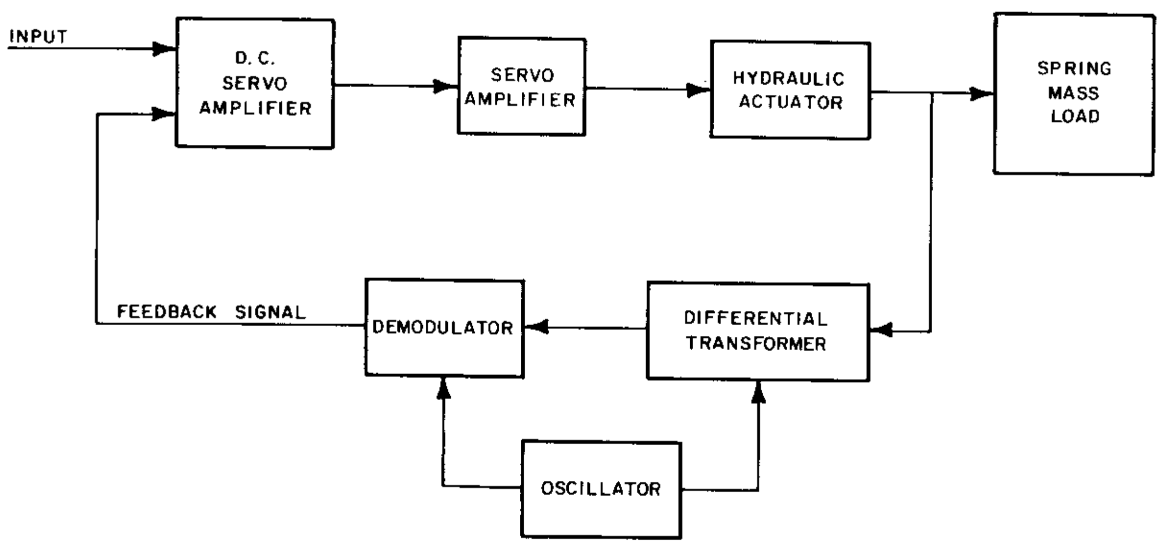

Figure 16. Block Diagram of Simulated Control System

package sold commercially by Raymond Atchley. It was necessary, however, to redesign the output stage of the amplifier and replace the pentodes with 5687 triodes to permit measurement of differential current with the use of cathode resistors. The output tubes supply approximately 100 ma current to the valve. The entire electronic unit is located outside the test cell.

Electronic instrumentation provided included a Hewlett-Packard oscillator for providing sinusoidal and square wave inputs, a two-channel brush recorder, Hewlett-Packard ac voltmeters for measuring input and output signal amplitudes, and a Serviscope to indicate relative phase of the system output with respect to input. One bourdon type pressure gage was provided for measuring the pressure on one side of the piston in the actuator. This was done by bringing one capillary tube from the special manifold block on which the servo valve was mounted out through one of the openings provided in the test cell.

A thermometer for measuring the return oil temperature was located outside the cell along with pressure gages for measuring the supply pressure.

Dosimetry was provided for the experiment by Inland Testing Laboratories and consisted of a graphite wall carbon dioxide flow ion chamber. The walls of the chamber are specially ground to accommodate the Cobalt 60 gamma-ray peak energies of 1.17 and 1.33 Mev. The instrument was calibrated at the University of Chicago's Argon Cancer Hospital. The Cobalt 60 source is described in Appendix III.

4. Experimental Test Procedures

The valve-actuator-feedback sensor assembly will be irradiated under a dose rate of 5×10^5 rs for a period of 200 hours, providing a total dosage of 1×10^8 rs. During the entire course of the test, the valve will be operated under a supply pressure of 3000 psi and, except during performance checks, will be excited by a 5 cps sinusoidal signal of 10 percent rated amplitude.

The frequency and amplitude are maintained low to minimize wear. It is anticipated that future tests will be conducted with a random noise generator to further reduce the wear problem.

The tests of necessity have to be conducted over a 24 hour period. Therefore, during a period of 16 hours, the test setup will be run unattended, except for the presence of the Facility technician. During the period of time, the input and output signals will be recorded every 15 minutes for a period of 30 seconds on the two-channel brush recorder. A timer will be used to actuate the recorder. As indicated earlier, provision will be provided for automatically shutting down the test setup in the event of leakage or high oil temperature.

Performance tests will be conducted on the valve and control system prior to irradiation, during irradiation, and following the completion of the tests. These are described below.

a. Preirradiation to Post Irradiation Tests

The following is a list of tests which will be performed both prior to and after irradiation. The test procedures for those tests referenced by an asterisk are described in the WADC Technical Report TR-5529, Part II.

(1) Electrohydraulic Servo Valve*

(a) DC resistance of torque motor coils

(b) Quiescent flow test

(c) Load differential pressure vs. differential current at zero load flow

(d) Load flow vs. differential current test

(e) Dynamic no-load frequency response test.

(2) Actuator

(a) The static Coloumb friction of the actuator will be measured employing a pressure control valve and two pressure gages located on each side of the actuator. Measurements will be made with the actuator at center and at differential pressures of 1000, 2000, and 3000 psi.

(b) Leakage across the piston will be measured at differential pressures of 1000, 2000, and 3000 psi with the output shaft blocked at center.

(3) Feedback Sensor (Differential Transformer)

(a) The resistance across each coil and to the housing will be measured.

(b) A plot of the output voltage vs. lineage displacement of the iron core will be made for the rated supply voltage and frequency.

(4) Hydraulic Oil

The viscosity and specific gravity of the oil will be measured at a temperature of 150°F.

5. Irradiation Tests

Once every day the following tests will be performed on the valve-actuator assembly:

(1) Frequency Response Under Load

In this test, the amplitude ratio and phase shift of the control loop will be measured, with special emphasis placed on determining the resonant frequency accurately. The test will be conducted with an oil temperature of 150°F ± 20°F

(2) Spool end pressures with zero input signal will be monitored and recorded twice a day, at intervals of approximately 7 hours.

(3) The valve null will be recorded twice a day as in (2) above.

(4) The viscosity of the oil will be measured once a day at 150°F.

E. Test Results

1. Test Conditions

The radiation dose rate actually employed in these tests was essentially equivalent to that stated in Section D above, although because of space requirements in the Cobalt test cell, certain changes were made. The valve torque motor was subjected to a dose rate of 3.5×10^7 ergs/gram-hr in carbon and the Schaevitz differential transformer employed as a feedback sensor subjected to 1.65×10^7 ergs/gram-hr. Total radiation time was 281.6 hours, which resulted in a total radiation of 9.8×10^9 ergs/gram for the torque motor and 4.6×10^9 ergs/gram for the differential transformer.

Total amount of oil employed in the system was 1.84 gallons. Temperature of this oil was maintained at 150°F ± 2 degrees and supply pressure maintained at 2500 psi.

2. Servo Valve

Because the special Moog valve for nuclear service was not delivered in time for testing under gamma radiation, discussions among Air Force and Cook Research Laboratories personnel resulted in the decision to use a conventional valve for the initial tests. A Cadillac Gage FC-200 valve with no modifications was therefore manifolded to the General Motors actuator and subjected to the radiation described above.

3. General Observations

The FC-200 valve and associated servo system was operated continuously over the entire test period. No noticeable degradation in performance was observed until shortly before the test was concluded, at which time the spool end pressure dropped rapidly and system response became so sluggish that satisfactory control of the valve could not be maintained. No leakage was observed at the O-ring seals although some external leakage from the actuator developed during the test. This latter difficulty was not believed to be related to the gamma radiation since metal seals were employed in the actuator.

After completion of the radiation exposure and before rerunning performance tests on the servo valve, it was disassembled and examined. Although all the O-rings had apparently been providing a satisfactory seal, each had taken a permanent set such that their cross sections were more nearly rectangular than circular. A picture of these O-rings is shown in Figure 17. These rings were not reused but it is doubtful that they would again provide a satisfactory seal.

Visual examination of the valve's one filter showed some evidence of particle clogging although the effect did not seem serious. There was also some clear thin jelly-like material present, and it is entirely possible that this material could have had a clogging action on the filter. The adaptor for measuring

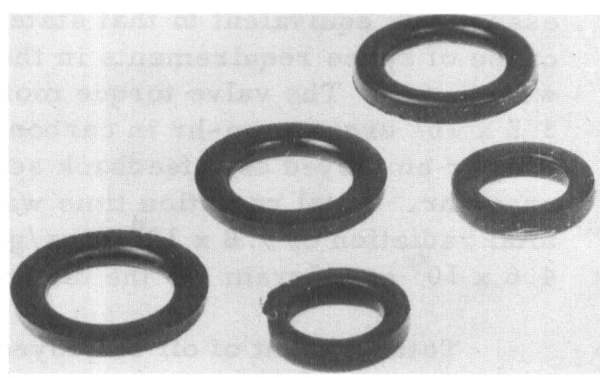

Figure 17. Compression Set of Static O-rings after Gamma Irradiation

spool end pressure also had an accumulation of the jelly material. This is shown in Figure 18.

Upon removal of the torque motor cap, a pungent smell was immediately noted. Visual evidence of change was observed on the nylon coil form which was originally white in color and after irradiation was brown. This form was also slightly rubbery after irradiation whereas before it was rigid. The teflon wire insulation had the same discoloration.

Oil which had leaked from the actuator and collected on the servo valve was found after irradiation to have congealed into a thick jelly. Although no such material was found internally in the hydraulic system, the material found on the valve's filter may have been early evidence of similar action. Since the oil within the hydraulic system was continually in motion, it did not receive as high a value of radiation as that oil which was on the valve's surface and continually exposed to the maximum dose rate.

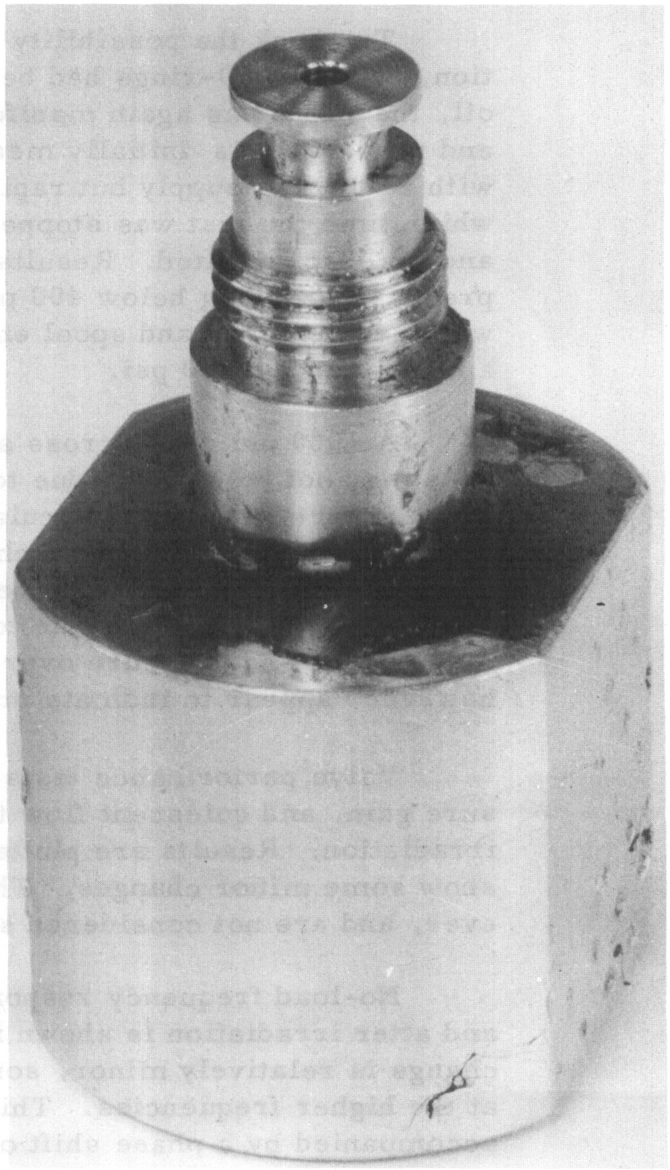

Figure 18. Adapter for Measuring Spool End Pressure, Showing Jelly Material which Formed from Oil after Gamma Irradiation

4. Preirradiation to Post-irradiation Tests

a. FC-200 Servo Valve

To check the possibility of filter clogging during irradiation, after new O-rings had been installed, using the irradiated oil, the valve was again manifolded to the test actuator. Spool end pressure was initially measured at approximately 1000 psi with a 3000 psi supply but rapidly decreased to 400 psi, at which time the test was stopped. The filter was then cleaned and the test repeated. Results were similar with spool end pressure dropping below 400 psi within 5 minutes. The filter was then removed and spool end pressure was measured at approximately 2000 psi.

A 1000 psi drop across a filter is not normal but in this case may not be entirely due to clogging. The valve manufacturer reports that the particular valve tested for this report was one of a large group which contained filters that were defective when received from the vendor. A replacement filter was received but arrived too late to be used in the tests. The drop-off in spool-end pressure over a short period of time does, however, appear to indicate progressive clogging of the filter.

Valve performance tests in the form of flow gain, pressure gain, and quiescent flow tests were run before and after irradiation. Results are plotted in Figures 19, 20, and 21, and show some minor changes. These are of small magnitude, however, and are not considered significant.

No-load frequency response of the FC-200 valve before and after irradiation is shown in Figure 22. Although the change is relatively minor, some reduction in gain was observed at the higher frequencies. This gain reduction was not, however, accompanied by a phase shift of any significant magnitude.

b. Actuator

Coulomb friction of the actuator was approximated by measuring the pressure required to just start it moving in each direction from a stopped position and using this information to calculate force applied. Before irradiation, motion toward the rod end required an average force of 305 pounds. Motion away from the rod end required 315 pounds. After irradiation, motion

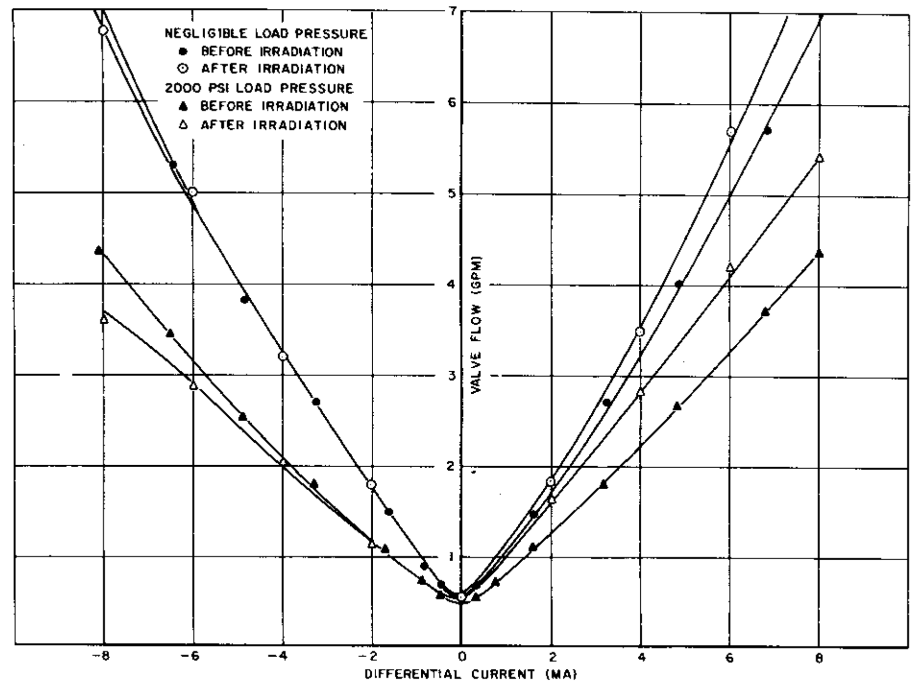

Figure 19. Flow Gain of FC-200 Valve Before and After Gamma Irradiation

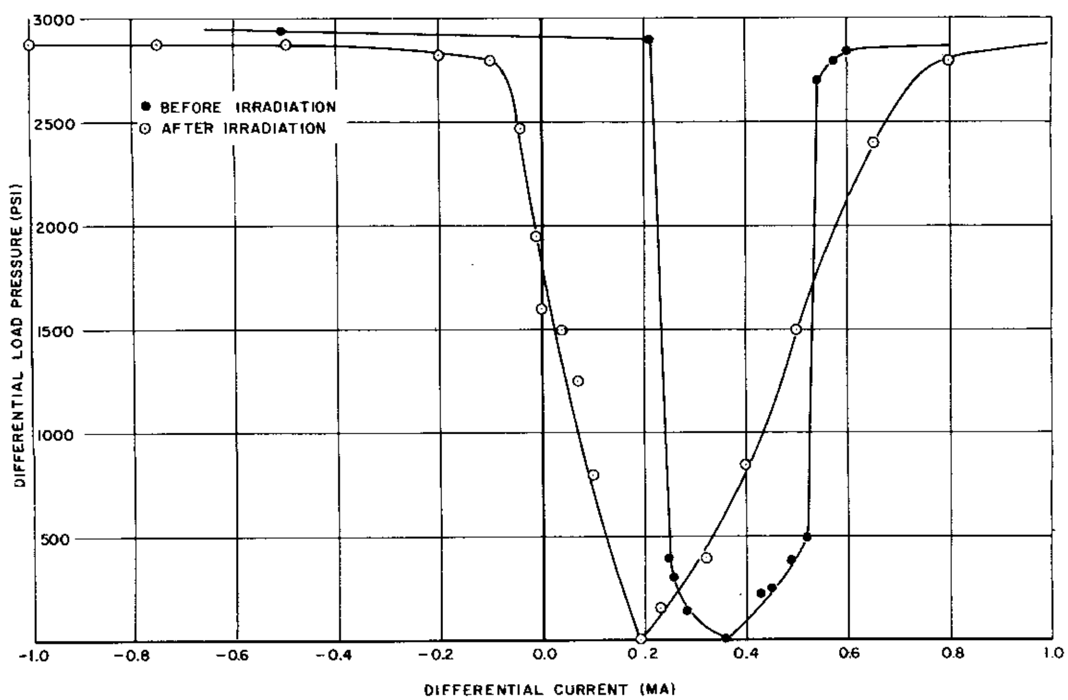

Figure 20. Pressure Gain of FC-200 Valve Before and After Gamma Irradiation

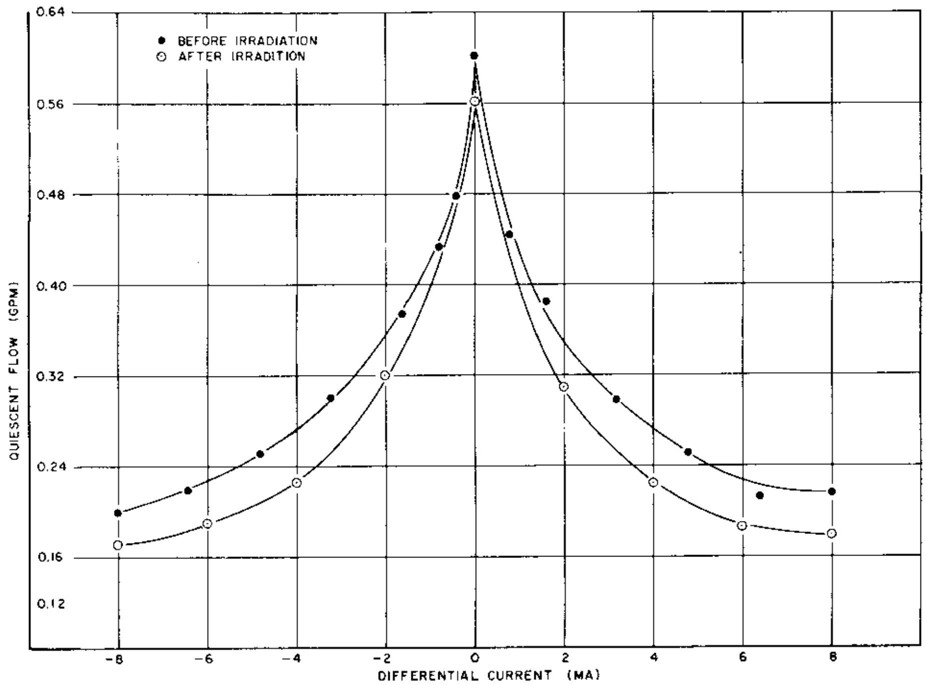

Figure 21. Quiescent Flow of FC-200 Valve Before and After Irradiation

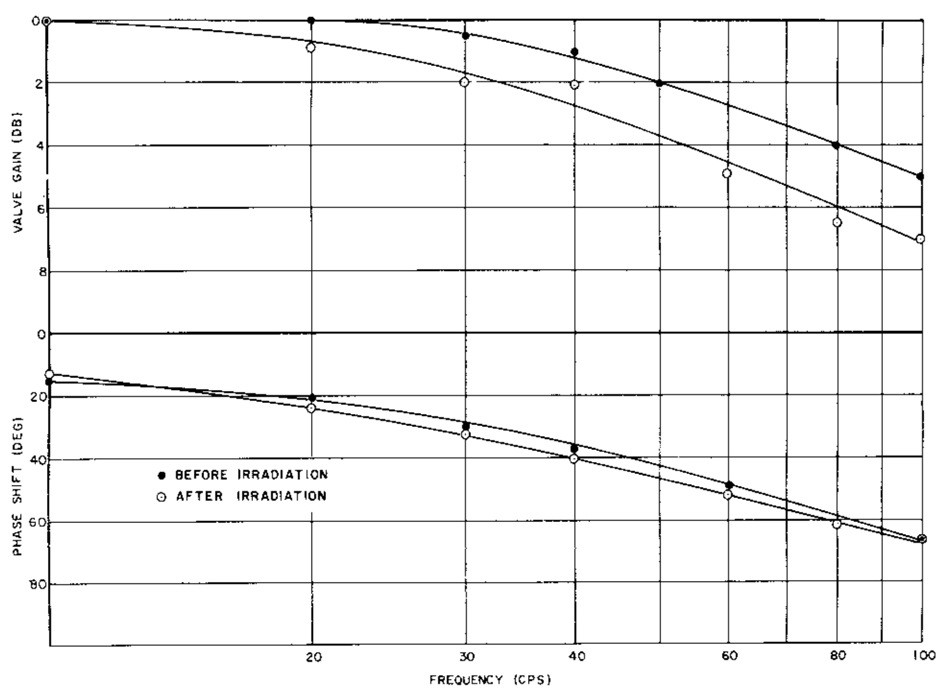

Figure 22. No-Load Frequency Response of FC-200 Valve Before and After Gamma Irradiation

toward the rod end required an average of only 129 pounds and motion away from the rod end 112 pounds.

Leakage across the actuator showed no significant change after irradiation. Values measured included the leakage in both the valve and the actuator and were 0.39 gpm at 3000 psi, 0.34 gpm at 2000 psi, and 0.28 gpm at 1000 psi.

c. Feedback Sensor (Schaevitz Differential Transformer)

Measurement of output voltage vs. core displacement showed no change in the calibration after irradiation.

5. Irradiation Tests

a. Frequency Response

Frequency response of the test system under load during the irradiation test in terms of gain is shown on Figure 23 and

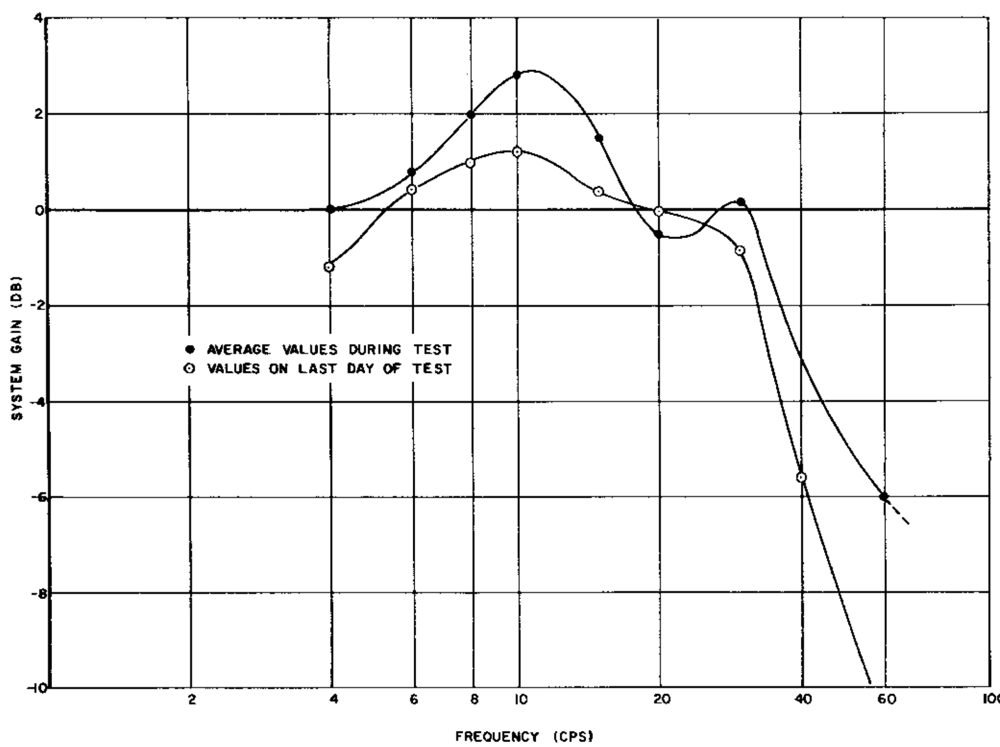

Figure 23. Frequency Response (Gain) of Test System Before and After Irradiation

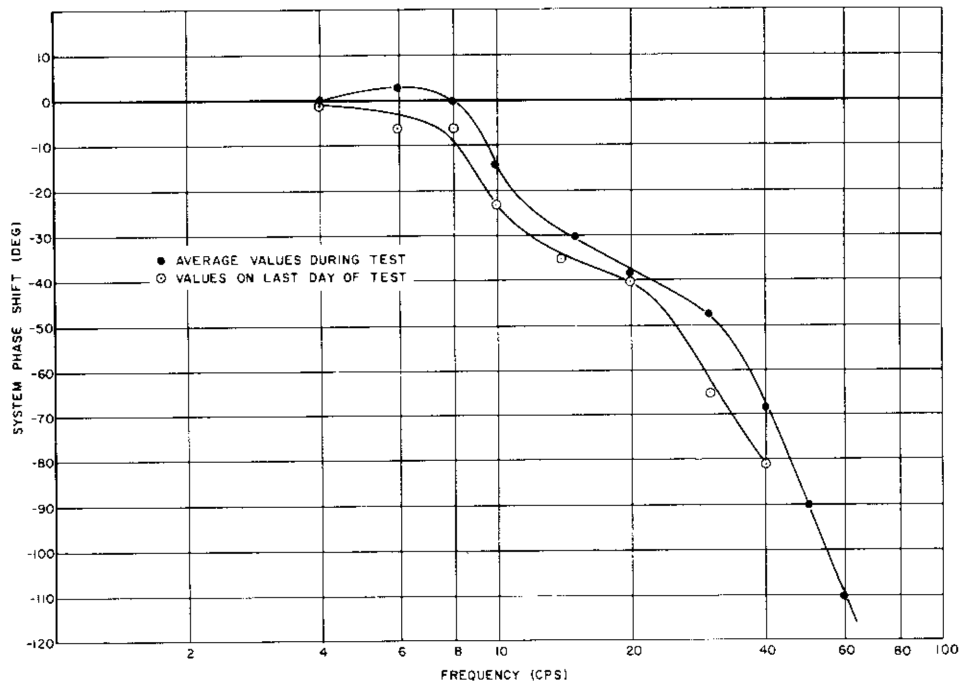

Figure 24. Frequency Response (Phase) of Test System Before and After Irradiation

in terms of phase on Figure 24. Response of this system was within a narrow band close to the lines drawn through the solid dots for the first 250 hours of the test. The response described by the lines through the open dots was obtained approximately 3 hours before the test was stopped and indicates a significant degradation in performance at that time. As was the result in the no-load tests on the valve, more change developed in system gain than in system phase. No change in oil-mass resonance due to gassing or change in oil bulk modulus could be detected.

b. Spool End Pressure

Capillary tubing was used as a means of bringing spool end pressure out of the test chamber. As a result, no dynamic pressure data could be obtained. In addition, because of equipment limitations, pressure readings were taken at only one spool end.

Spool end pressure remained relatively constant at approximately 1000 psi during all but the last day of the test. In the last few hours preceding halting of the test, this pressure decreased gradually. In the last hour, it decreased from 800 psi to 400 psi at which time control of the load was lost and the test halted.

c. Valve Null

Differential current required to null the valve was measured once during each day of the test. Oil temperature remained at 150°F ± 2°. Although the measured current varied from 0.35 ma to 0.50 ma, the variation was essentially random and could not be correlated with radiation effects.

d. Viscosity of Oil

Viscosity of the Oronite 8200 fluid at 100°F was measured once each day. A graph of these data is plotted on Figure 25. The viscosity, as indicated, decreased from an initial value of 25.65 c.s. to 20.83 c.s.

6. Conclusions

Study of the data obtained in this test strongly indicates the primary reason for failure to be clogging of the one filter in the supply line of the valve. Reason for this clogging was not, however, as clearly demonstrated but was believed due to a combination of normal contamination in the oil, plus the slight congealing of oil as found on the surface of the filter. Although oil viscosity decreased slightly, it appears that some localized jelling had developed and that this action was instrumental in causing system failure.

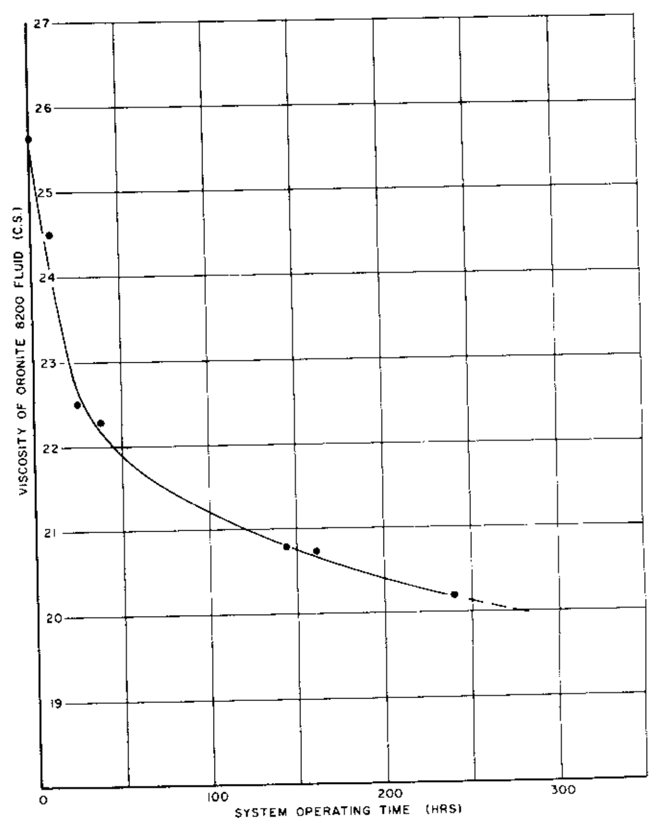

Figure 25. Viscosity of Oronite 8200 Fluid During Test System Irradiation Test

As was expected, no deleterious effects on the actuator traceable to radiation were observed. Had conventional O-ring seals been employed, however, these would have taken a set as did those in the valve. Further, it is highly improbable that O-rings used for dynamic seals would have operated satisfactorily under the radiation environment. Decrease in coulomb friction force is believed due to normal wear.

Radiation effects on the valve itself were as noted previously, principally on the O-rings. Some change was also produced on the torque motor coil form but with a radiation level no higher than that employed in this test, the effects should not be serious.

CHAPTER V

EVALUATION TESTS OF THREE NEW SERVO VALVES

A. Purpose

These valve evaluation tests represent a continuation of the valve tests reported in WADC Technical Report TR 55-29, Parts 1, 2, and 4. The purpose of this phase of the present project was to evaluate valves incorporating new and promising design concepts.

B. Valves Selected

The following valves were selected for evaluation:

(1) Cadillac Gage FC-200

(2) Raymond Atchley 410

(3) Airesearch Mfg. Division Time Dwell

A picture of all three valves is shown in Figure 26. A description of each valve and reasons for its selection are given below.

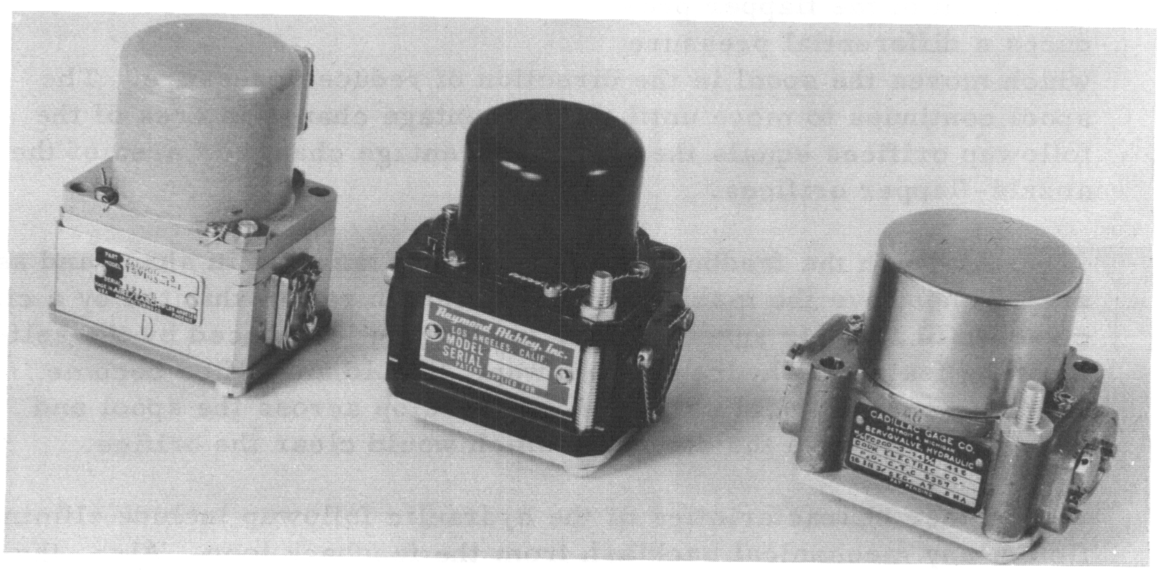

Figure 26. View of Valves Tested

1. Cadillac Gage FC-200

The Cadillac Gage FC-200 (Figure 27) is a two-stage flow control valve incorporating hydraulic followup and was selected because of this unique method of spool position feedback. The valve uses a dual nozzle-flapper first stage of dry coil construction. Output differential pressure from this stage is used to drive a four-way second-stage spool which is not spring restrained.

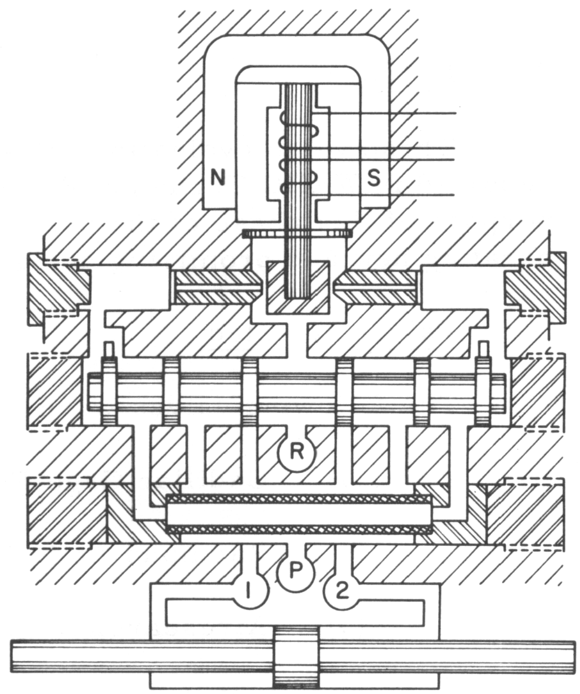

Figure 27. Schematic of Cadillac Gage FC-200 Valve

Hydraulic followup is accomplished by an extra slotted land on each end of the spool which with the mating part in the sleeve form orifices in series with the fluid supply to each nozzle. These variable orifices are equivalent to the fixed upstream orifices in a conventional nozzle-flapper first stage. In operation, displacement of the flapper produces a differential pressure which moves the spool in the direction of reduced pressure. The spool continues to move until the percentage change in area of the followup orifices equals the initial percentage change in area of the nozzle-flapper orifices.

Although the feedback orifices are rectangular in shape and as such do not have the maximum area to length ratio exhibited by a circular orifice, their susceptibility to clogging is reduced by the self-cleaning feature of hydraulic followup. Should an orifice become clogged, a differential pressure will develop across the spool and cause it to move in the direction which would clear the orifice.

Other characteristics of the hydraulic followup include elimination of any mechanical backlash from the feedback loop. Also, the torque motor itself is not included in the closed loop, so any changes in motor performance will appear directly in the valve output.

The valve contains a single sintered metal filter and is reportedly capable of operation in fluids from -65°F to +500°F and ambient temperatures of -65°F to +1000°F. No external null adjustment is provided.

2. Raymond Atchley 410

The Raymond Atchley 410 (Figure 28) is a two-stage flow control valve incorporating a jet pipe first stage with mechanical feedback. It was chosen for test because of the departure from a conventional nozzle-flapper type of valve.

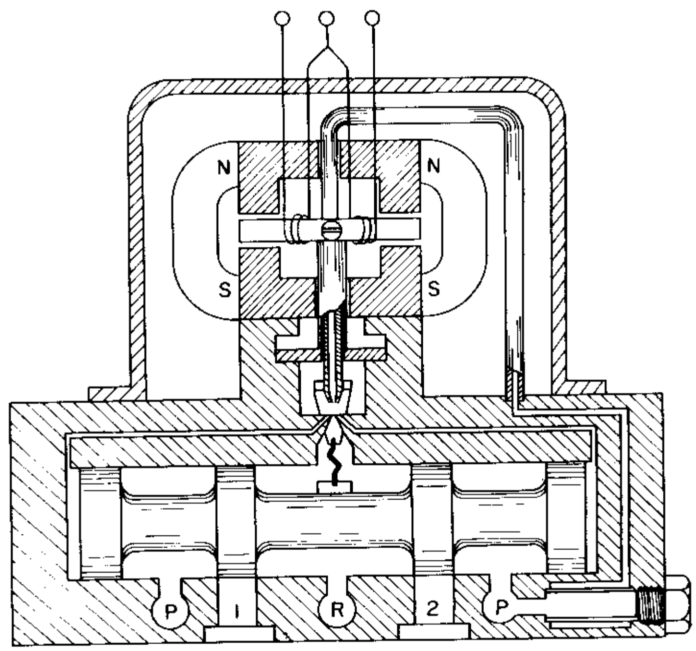

Figure 28. Schematic of Raymond Atchley 410 Valve

In operation, an electrical signal applied to the torque motor causes deflection of an annular torque tube which surrounds the flexible pipe supplying oil to the jet forming orifice. Deflection of the jet in turn causes a differential pressure to exist in the two receiver pipes which supply fluid to the ends of the spool. Motion of the spool in the direction of reduced pressure results and continues until the force applied to the torque tube through the feedback spring equals the initial force produced by the torque motor.

Where supply pressure exceeds 1500 psi, the pressure upstream of the jet is reduced to this value by an appropriate orifice. The valve uses no filtration other than a 200 micron screen and contains no external null adjustment. The torque motor is isolated from the fluid and is of dry coil construction.

3. Airesearch Manufacturing Division Time Dwell

The time dwell servo valve of the Garrett Corp., Airesearch Manufacturing Division (Figure 29) is a two-stage valve using a dual nozzle-flapper first-stage and a second-stage spool without spring restraint. Operation of the valve is unique in that a square wave type of input is applied to the torque motor. Proportional control is obtained by a variation in relative lengths of the positive and negative pulses of this input signal.

An electronic driver unit in the form of an amplitude modulated multivibrator must be used with this valve. The error signal is applied to this driver, which in turn produces the time varying square wave input signal of relatively fixed amplitude and frequency for the torque motor. With this input signal, except for transit time, the flapper is hard over against one nozzle or the other at all times. The difference in relative length of nozzle-flapper contact or dwell time controls the rate of spool motion and in turn the rate of change of valve flow.

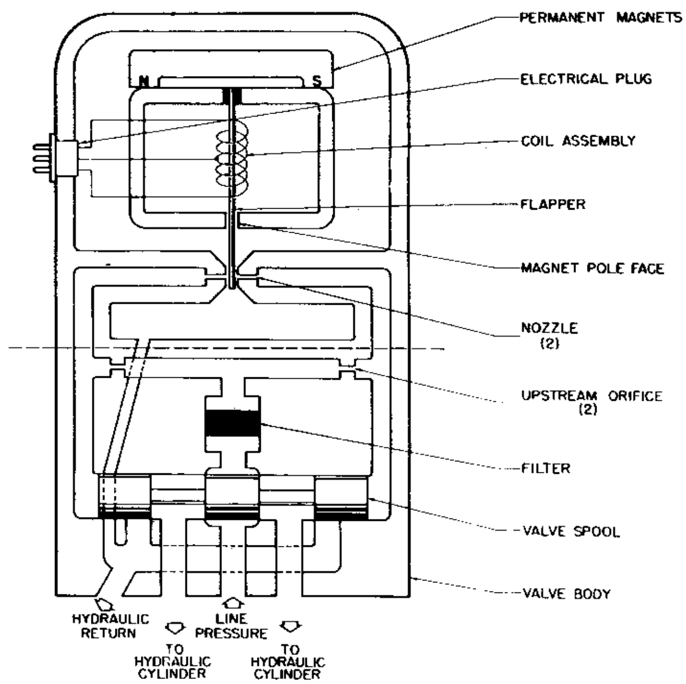

Figure 29. Schematic of Airesearch TSV103-1-1 Time Dwell Valve

Ideally, this valve may be considered as a perfect integrator. Because of Bernoulli forces on the spool, a better approximation is to consider the valve's no-load transfer function to be in the form of a first-order lag with a large time constant. Because there is neither spring restraint on the spool nor internal feedback from the spool, the valve itself tends to be marginally stable and must be employed in a closed loop system with rate compensation. This fact also prohibits use of several of the standard static tests in evaluating valve performance.

Since the nozzle-flapper orifices are on the average full open 50 percent of the time, they should be relatively free of clogging. Also, full supply pressure is available to the spool to overcome any stiction effects. Silting effects on the spools are reduced because of the effective dither action produced by the square wave input. This action also reduces the required axial spool tolerances and should tend to reduce the difficulty of valve manufacture.

C. Discussion of Tests

The following is a list of tests conducted on the Cadillac Gage FC-200 and Atchley 410 valves:

(1) Torque motor dc resistance

(2) Hysteresis

(3) Quiescent flow

(4) Load differential pressure vs. differential current at zero load flow (pressure gain)

(5) Short-circuited load pressure vs. differential current

(6) Load flow vs. differential current with load pressure as a parameter (flow gain)

(7) Load flow vs. load differential pressure (pressure-flow curves)

(8) Null shift vs. supply pressure

(9) Null shift vs. temperature

(10) Dynamic no-load frequency response.

In general, the test procedures used in the evaluation of these valves were essentially those described in WADC Technical Report TR 55-29, Part 2. Because no external null adjustment was available, a modification was necessary, however, for tests (3) through (7). In each case a differential current of positive polarity slightly greater than rated value was applied to the torque motor (+9 ma for each of the two valves tested). The desired value of current was then reached by decreasing the input from this maximum positive amount. This procedure eliminated any hysteresis effects and produced a smooth curve in the tests where results were plotted as a function of differential current.

In tests (8) and (9), the procedure also was varied from that previously described in an effort to eliminate hysteresis effects and to improve repeatability of results. No change was made in the piping setup; however, in both tests an initial 9 ma was applied to the valve before each reading and the actuator driven over to one end of its travel by this means. Current was then slowly decreased to and beyond zero until the actuator just started to move. This point was then considered to be one side of the null deadband under the particular conditions of the test. To determine the other side of the deadband, the actuator was initially driven to the other end of its travel and the process repeated in reverse polarity.

Because of its unique method of operation, only a few of the standard evaluation tests could be employed on the Airesearch Division, Time Dwell valve. Those tests run on this valve were:

(1) Load acceleration vs. differential current (open loop acceleration gain)

(2) Quiescent flow

(3) Dynamic no-load frequency response.

Test (1) is a new test and was designed as a means of evaluating valve gain. Detailed procedure for this test is described in Appendix V. Test (2) involves a total leakage measurement at valve null, with the loop closed, and with rated signal applied and the loop open. By total leakage is meant both first and second stage flow and drain with the load ports blocked. Test (3) (see Appendix V) is a version of the standard no-load frequency response, except that the error signal is measured at the input grid of the mixer unbalancer (see Figure 30) and the acceleration output is measured. The amplifier was supplied by Airesearch Manufacturing Division.

D. Summary of Test Results

Results of the tests on the Cadillac Gage and Raymond Atchley valves are presented in graphical form in Appendix IV, consisting of Figures 37 through 62, and the data contained in these curves are summarized in Tables VI, VII, and VIII of this chapter. Average values are presented in these tables since most of the curves are unsymmetrical and nonlinear, and in some cases multiple valued. Where greater accuracy is required, the graphical presentation should be utilized.

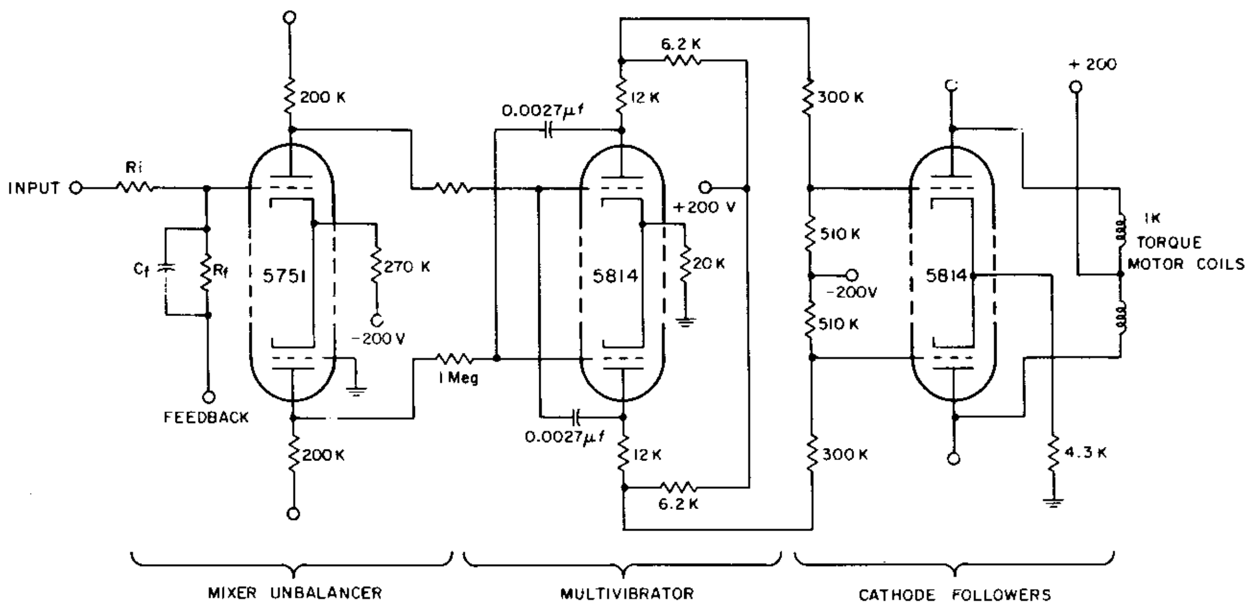

Figure 30. Time Modulated Servo Valve Amplifier-Driver

TABLE VI

STATIC TEST RESULTS

	Rated Flow (gpm)		Mfrs. Rated Current (ma)	Min/Max Load Pressure psi	Range of Null Shift vs. Temp (ma)		Range of Null Shift vs. Pressure (ma)		Percent Hysteresis 3000 psi	
	2000 psi	3000 psi			Dither	No Dither	Dither	No Dither	Dither	No Dither
Cadillac FC-200	3.5	3.7	8	2750	1.1	1.8	0.52	0.47	2.3	2.3
Raymond Atchley 410	4.3	4.8	8	2600	0.05	0.12	0	0	2.3	3.1

	Flow Gain GPM/MA						Pressure Gain psi/ma		Leakage (gpm)			
	3000 psi			2000 psi					First Stage		Total	
	No. Load	1000 psi Load	2000 psi Load	No Load	1000 psi Load	2000 psi Load	3000 psi	2000 psi	3000 psi	2000 psi	3000 psi	2000 psi
Cadillac FC-200	0.87	0.69	0.50	0.72	0.51	0.41	5,000	9,000	0.20	0.16	0.60	0.47
Raymond Atchley 410	1.0	0.85	0.58	0.80	0.58	0.45	170,000	70,000	0.095	0.08	0.19	0.14

TABLE VII

DYNAMIC TEST RESULT - NO-LOAD

Servo Valve Model and Mfr.	NO-LOAD FREQUENCY RESPONSE								
	20 Percent Rated Diff Current			60 Percent Rated Diff Current			100 Percent Rated Diff Current		
	Freq. at -3 db (cps)	Freq. at 90° (cps)	Phase Shift at -3 db (degrees)	Freq. at -3 db (cps)	Freq. at 90° (cps)	Phase Shift at -3 db (degrees)	Freq. at -3 db (cps)	Freq. at 90° (cps)	Phase Shift at -3 db (degrees)
Cadillac FC-200	68	>100	52	72	>100	58	150	>100	>100
Raymond Atchley 410	84	>100	75	150	>100	75	158	>100	>75

TABLE VIII

TORQUE MOTOR COIL RESISTANCE (OHMS)

Cadillac Gage FC-200		Raymond Atchley 410-298		Airesearch TS V 103-1-1	
Coil No. 1	Coil No. 2	Coil No. 1	Coil No. 2	Coil No. 1	Coil No. 2
996	1026	1178	1184	1008	1002

Table VI summarizes the static hydraulic tests and Table VII presents the dynamic no-load test results. Torque motor ac resistance values are listed in Table VIII.

The following is a brief description of the column headings in Table VI where such headings are not self-explanatory.

1. Rated Flow

This represents the flow at rated current and supply pressures of 2000 psi and 3000 psi, with a load pressure drop of two-thirds supply pressure. These are the flows and pressures at which the valves are normally rated. The data were taken from the flow vs. differential current curves.

2. Maximum/Minimum Load Pressure

This was obtained from the short-circuited load pressure vs. differential current curves. It is the value of pressure at the pressure null (antinull).

3. Range of Null Shift vs. Temperature

This represents the total extent of the null shift over the temperature range of $100^{o}F$ to $200^{o}F$ to $100^{o}F$.

4. Range of Null Shift vs. Pressure

This represents the total extent of the null shift over the pressure range of 3000 psi to 1500 psi to 3000 psi.

5. Percent Hysteresis

This represents the ratio of the envelope width along the differential current axis to the total rated current.

6. Flow Gain

This represents the average slope of the flow vs. differential current curves at a point in between the null zone and flow saturation zone.

7. Pressure Gain

This is the slope of the load pressure vs. differential current (at zero load flow) curves. The slope is taken in the vicinity of minimum differential load pressure.

8. Leakage, First Stage

This represents the asymptotes parallel to the differential current axis at the extremities of quiescent flow vs. differential current curves.

9. Leakage, Total

This represents leakage at the null point of the quiescent flow vs. differential current curves. It is a composite of the first-stage quiescent flow and the second-stage leakage.

E. Analysis of the Test Results

In this section each valve will be analyzed from the standpoint of the static test results, the dynamic test results, and general operating characteristics.

1. Cadillac Gage FC-200

a. Static Test Results

The pressure-flow curves of this valve are essentially parabolic and indicate that it is not gain compensated. Linearity of the valve output flow is good; however, with 3000 psi supply pressure, there is a positive change in slope of the flow gain curve at rated differential current. Pressure gain is relatively low, being 5000 psi/ma at 3000 psi supply pressure. In addition, the valve at times exhibited sticking around null as evidenced by the threshold in other of the pressure gain curves. Both first stage and total leakage of this valve were relatively high, necessitated, according to the manufacturer, by the dynamic requirements. First stage leakage was 0.20 gpm with 3000 psi supply pressure or 5.7 percent of rated flow. Total leakage was approximately 17 percent of rated flow. Hysteresis of this valve was essentially 2.3 percent with or without dither.

Null shifts due to both temperature and pressure changes were relatively high. Over the range of temperature used, the maximum null shift was 23 percent of rated current with the majority of shift occurring while increasing temperature between $170^{\circ}F$ and $180^{\circ}F$. Pressure changes between 1500 psi and 3000 psi caused a shift in null equal to 6 percent of rated current. Addition of dither acted to reduce the magnitude of both types of null shift. The large temperature null shift may be due to dimensional changes in the variable upstream orifice.

b. Dynamic Test Results

This valve exhibited relatively good no-load dynamic response with the 3 db point occurring at 70 cps with inputs of 20 percent and 60 percent rated differential current. At 100 percent differential current the 3 db point occurred at 130 cps. Phase shift at 100 cps for all inputs was approximately 70 degrees.

c. General Operation

Frequency response of this valve is better than many valves of comparable rating but it is subject to somewhat excessive temperature null shift. The valve tested exhibited a sonic resonance near null, rendering pressure gain and leakage tests at this point difficult. The valve operated smoothly elsewhere.

2. Raymond Atchley 410

a. Static Test Results

The pressure-flow curves of this valve are parabolic and show no evidence of gain compensation. Pressure gain is relatively high being equal to 170,000 psi/ma at 3000 psi supply pressure. Reference to the curve of flow gain for this valve shows good linearity up to 50 percent of rated differential current. At higher values some dropoff in flow gain was noted.

Hysteresis in this valve showed some improvement with application of dither but was not excessive in either case. With no dither, hysteresis was 3.1 percent of rated current and 2.3 percent with dither. These values are lower than those obtained from curves submitted with the valve. Leakage was quite low, being 0.19 gpm or 4 percent of rated flow with 3000 psi supply pressure. First-stage leakage was only 2.2 percent of rated flow.

Relatively small values of null shift for this valve were observed as a function of temperature variation. Maximum shift over the range of temperatures used was 1.5 percent of rated differential current. No shift of null due to variation of supply pressure could be measured.

b. Dynamic Test Results

This valve exhibited a relatively good frequency response, having negligible amplitude attenuation below 100 cps, while phase shift at 100 cps was approximately 75 degrees. The 3 db point occurred at 170 cps. This response, however, appeared to be sensitive to magnitude of the input signal with the widest bandwidth occurring with the larger inputs.

c. General Operation

This valve also exhibited a resonance near null, but otherwise performed satisfactorily. Curves for the pressure null shift tests were not included because of the negligible changes which resulted.

3. Airesearch Manufacturing Division, Time Dwell Valve

As indicated in the test procedures section, three tests were performed on the time-modulated valve. This constitutes merely a preliminary test of the valve as a detailed experimental and analytical study of the time-modulated valve operating alone, and in the system will be made as part of the work to be carried on under the next contract.

The first test conducted on the valve was an acceleration gain test in which the output acceleration was measured as a function of the input current. The test was conducted using a sinusoidal input and measuring the output acceleration amplitude via an accelerometer as the input amplitude was increased. The ratio of acceleration to input signal is shown by the curve of Figure 31. Theoretically, the curve should be a straight line, but flow forces reduce the gain for inputs away from the null. The final saturation is the result of flow limiting in the first stage.

Tests such as the above are difficult to conduct as the switching action is sensed by the accelerometer and a very noisy waveform results. For this test it is possible to select the proper frequency so that the resulting noise is a minimum.

The quiescent flow of the valve at null was 0.146 gpm, while at positive and negative rated current the quiescent flow was 0.150 gpm and 0.190 gpm, respectively. This leakage is not abnormally high.

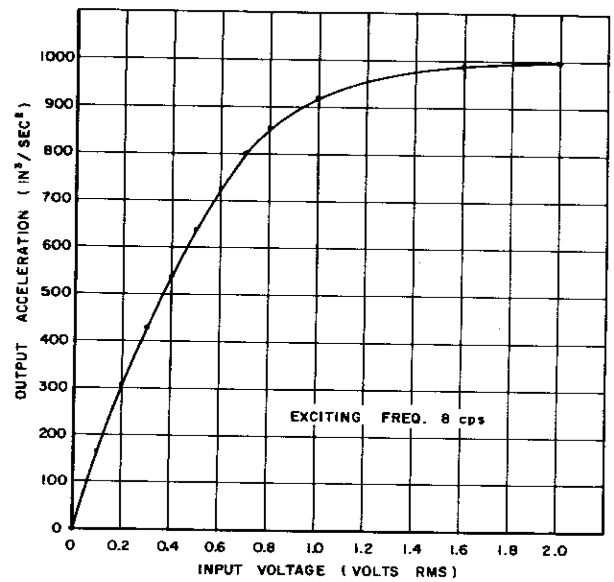

Figure 31. Acceleration Gain of Airesearch Valve

The final test conducted on the valve was the no-load frequency response test. The results using the accelerometer were not satisfactory because of poor waveform and the high noise level at most frequencies. Data taken by means of the velocity pickoff indicated that the amplitude was falling off approximately 6 db per octave from 7 cps to 25 cps, after which the waveform became too distorted for useful measurements. Phase data could not be measured accurately because of the poor waveform problem.

As indicated earlier, a complete analytical and experimental investigation of the valve will be made under the next contract. An attempt will be made to increase the switching frequency somewhat to improve experimental measurements and also to reduce the chattering of the valve. It is highly probable that in some application this chattering could expedite O-ring failure and cause other difficulties such as exciting resonances.

The servo had a tendency to drift at a low frequency during the progress of the tests, an undesirable condition. The actual cause of drift was not determined and may not have been caused by the valve.

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APPENDIX I

DETAILED DESCRIPTION OF TEST STAND

The test stand used for the oil contamination study investigation (see Figures 2, 3, and 4) was constructed specifically for that study. It consists of a control panel, an electric motor-driven pump, reservoir, heat exchanger, filters, and valve test assembly. The test stand has provision for testing three servo valves concurrently.

The control panel contains the following items:

(1) General Electric three-phase motor controller, which provides off-on control of the motor used as the pump prime mover

(2) A Denison relief valve (designated (V_1) on the hydraulic schematic), provides any operating pressure up to 3000 psi

(3) A 1/2 inch Marsh needle valve (V_2) is used as a bypass to adjust system pressure

(4) A Marsh 0-5000 pressure gauge (G_1) is used to monitor the system pressure

(5) A Marsh valve (V_3), is used as a snubber in conjunction with gage G_1

(6) A 1/2 inch Marsh needle valve (V_4) is used in conjunction with V_5 in withdrawing oil samples to determine contamination level at the input to the servo valves under test

(7) A 1/2 inch Marsh needle valve (V_5) is used in conjunction with V_4.

(8) A 1/2 inch Marsh needle valve (V_6) is used to control flow to the filter array

(9) A 1/2 inch Marsh needle valve (V_7) is used to control flow to the actuator section

(10) A 0-5000 psi Marsh pressure gage (G_2) is used to monitor the drain pressure

(11) A 1/4 inch Marsh needle valve (V_8) is used as a snubber for (G_2)

(12) A 1/2 inch Marsh needle valve (V_9) is used in conjunction with V_{10} to withdraw oil samples for determining viscosity

(13) A 1/2 inch Marsh needle valve V_{10} is used in conjunction with (V_9)

(14) A Weston thermometer (T_1) is used to monitor input temperature to the actuator section

(15) A Weston thermometer (T_2) is used to monitor the temperature of the oil returning from the actuator section.

The electric motor used to drive the pump is a 10 hp, 440 volt, 3 phase, 3530 rpm motor, GE 4254A1.

The pump in the system is a Vickers constant displacement unit No. PF 3909-25.

The reservoir is a 2-1/2 gallon battery jar. The jar cover houses a small electric motor-driven agitator and has a capped opening where the contamination is introduced into the system.

The filter bank is parallel with the valve test assembly. It consists of five filters connected in series. These are, progressing from the first to the last:

(1) (F_1) Purolator, 10 micron, paper element type

(2) (F_2) Purolator, 5 micron, paper element type

(3) (F_3) Aircraft Porous Media, 2 micron, fibrous glass filter

(4) (F_4) Aircraft Porous Media, 10 micron, sintered stainless steel wire-woven type

(5) (F_5) Aircraft Porous Media, 5 micron, sintered stainless steel wire-woven type.

The filters are used for cleaning the contamination from the oil before and after the tests.

The system contains a Ross heat exchanger, used to maintain a constant system temperature.

The valve test assembly contains three test actuators to which the valves are manifolded. Spool end pressures are tapped off through the manifolds along with load pressures. The spool end pressures of each valve are individually monitored with Helicoid gages, G_3, G_4, G_5, G_6, G_7, and G_8. The load pressures are all monitored by Helicoid gages, G_9 and G_{10}, which are connected to the individual actuators through Marsh needle valves, V_{17}, V_{18}, V_{19}, V_{20}, V_{21}, and V_{22}. The Marsh needle valve (V_{23}) is used for circulating the oil around the valve assembly.

Mechanically coupled to each of the three actuators is a position feedback transducer. Two of the feedback transducers are Bourn's Align-O-Pots, 50K ohms, Model No. 156, linear motion potentiometers. The other positional pickoff is a Shaevitz LVDT 1000 S-L.

Three control amplifiers are used. One is an Atchley amplifier modified by Cook Research Laboratories (see Figure 32). This amplifier has a 5000 cycle oscillator and a demodulator for use with the Shaevitz type position pickoff. The modifications consist essentially of changing the output stages from pentodes to triodes, allowing the differential current to be measured in the cathode circuit. This allows continual monitoring of the differential current on a brush type recorder with either three or four wire torque motors.

The other two amplifiers (Figure 33) are three-stage amplifiers designed and built by Cook Research Laboratories. The first two stages are Philbrick operational amplifiers k2-w and k2-x. The output stage is a push-pull dc amplifier manufactured by Engineered Electronics Company. An additional stage of amplification is used to convert the balanced output of the push-pull amplifier to an unbalanced output for use in monitoring the differential current from an unbalanced detector.

In conjunction with the test stand, the following equipment is used for sampling the oil to determine contamination level:

(1) A vacuum pump, Millipore No. XX 60 000 00, is used to draw the oil sample and the petroleum ether wash through the millipore filter

(2) A millipore filter holder, No. XX 10 047 02, to hold the millipore filters

(3) Three filter flasks, one for the filtered oil, one for use in the vacuum system as a trap to prevent air lock, and one to hold the raw petroleum ether (see Figure 34).

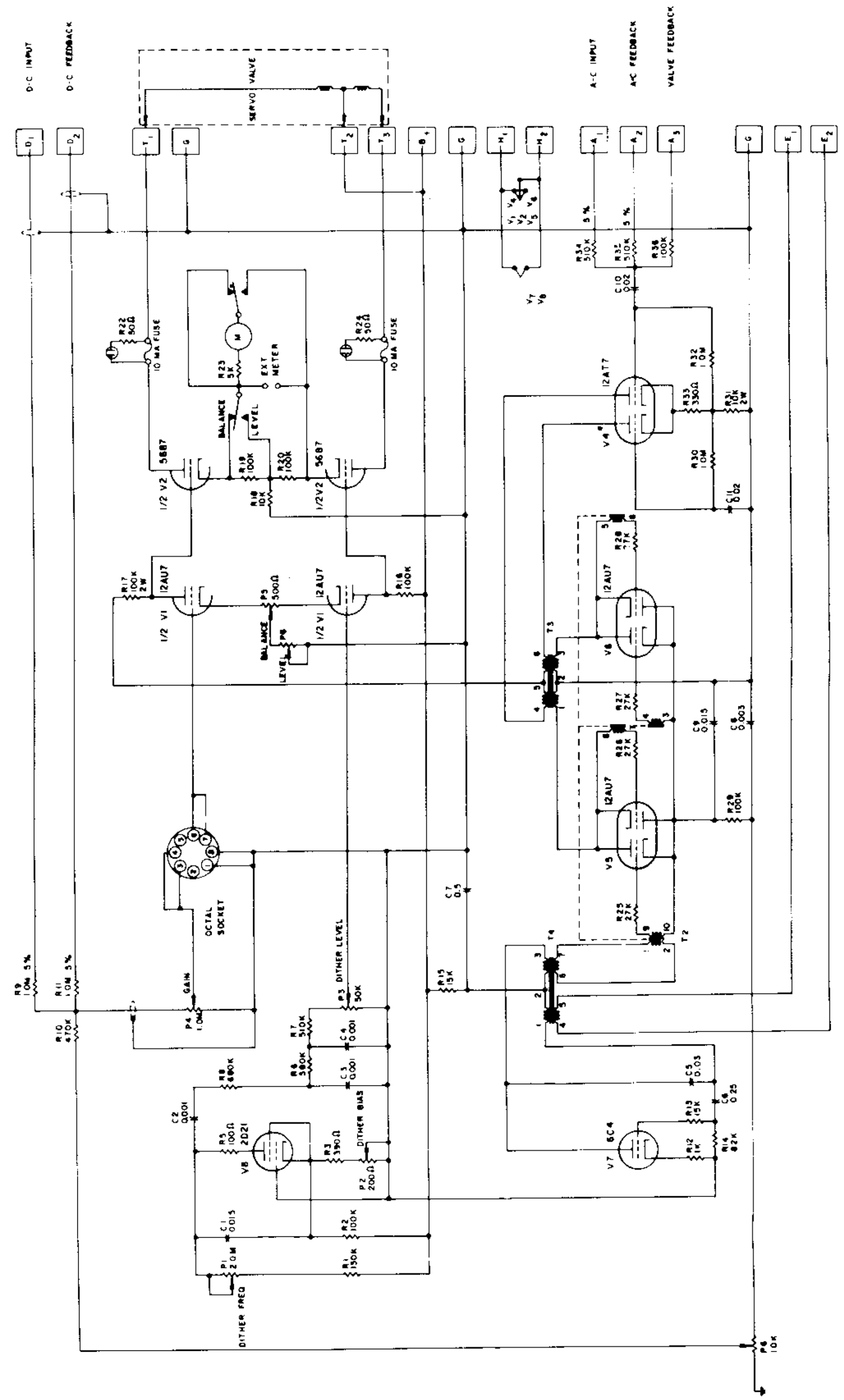

Figure 32. Schematic of Modified Atchley Amplifier

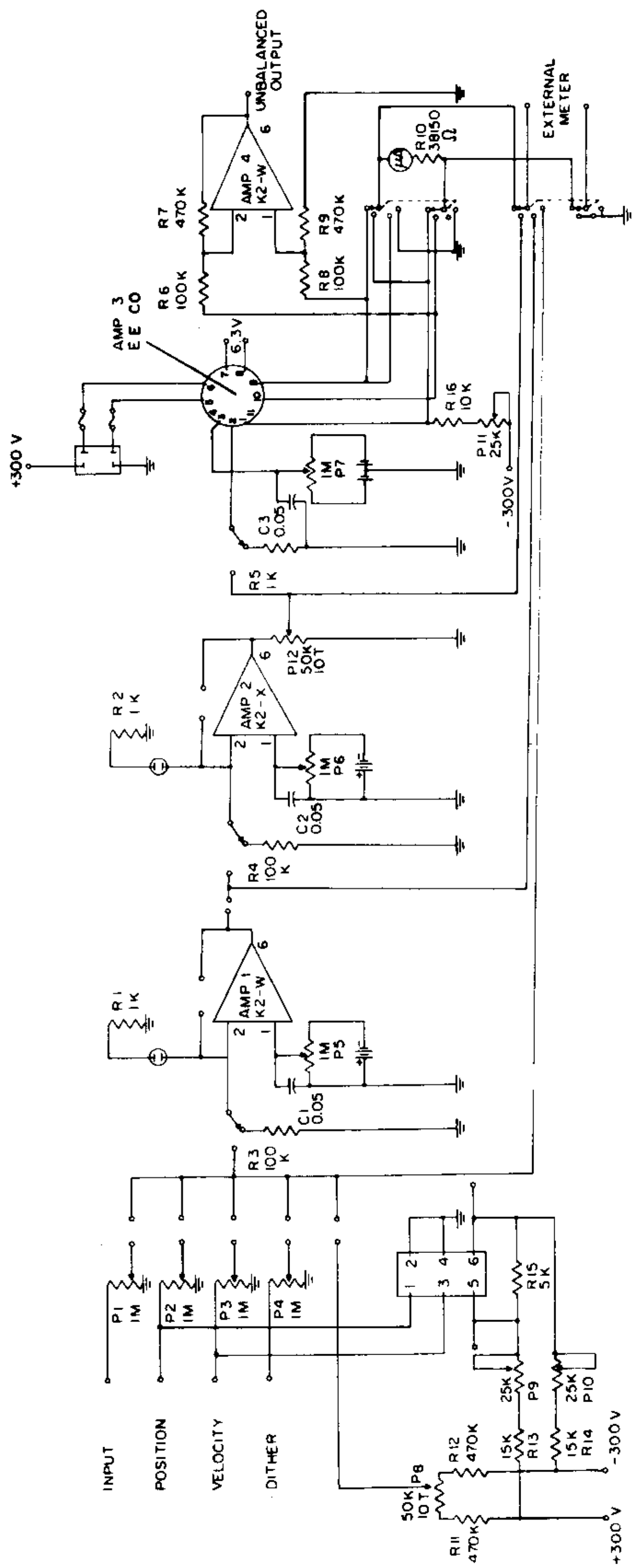

Figure 33. Schematic of Cook Research Laboratories Amplifier

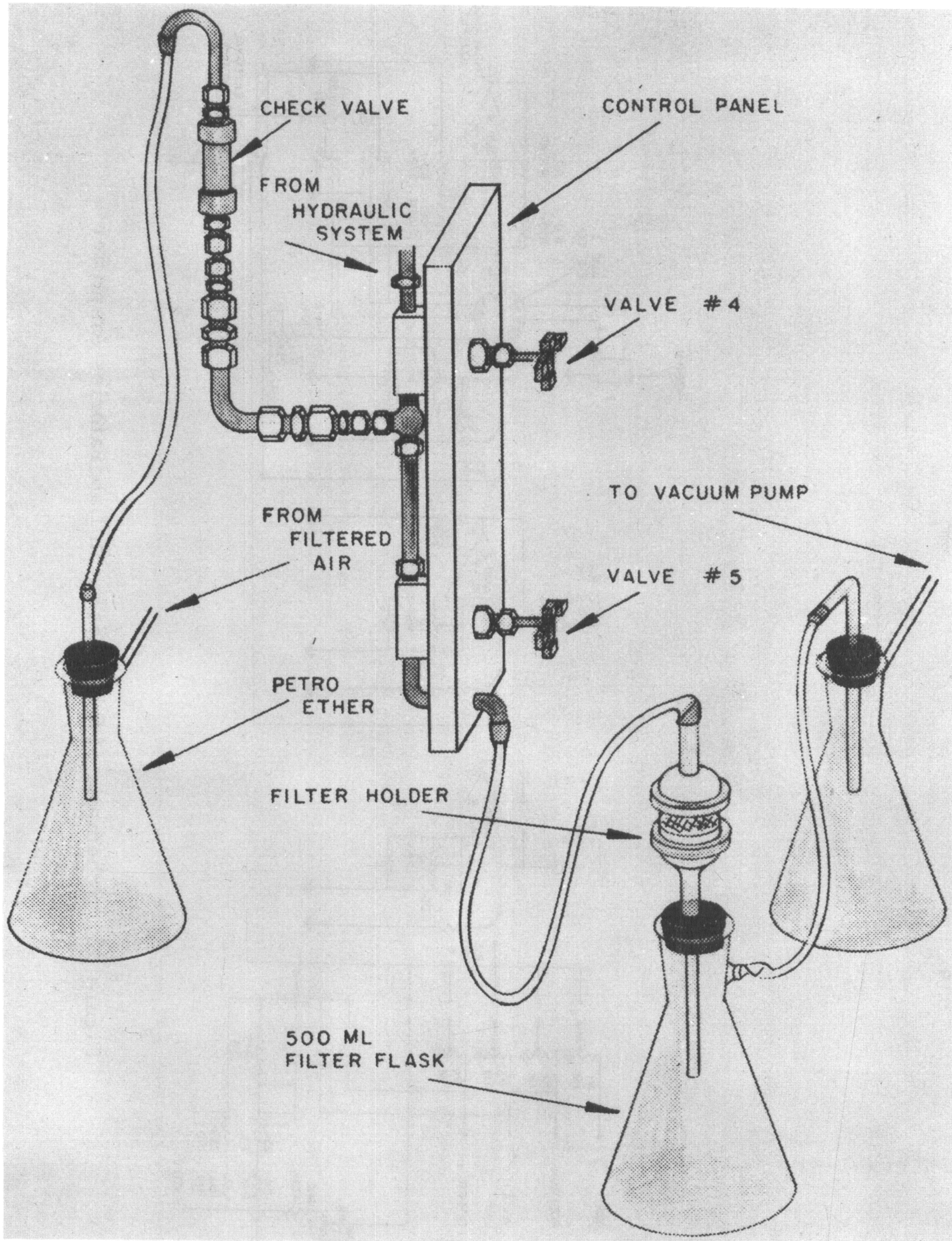

Figure 34. Sampling Apparatus

APPENDIX II

CONTAMINATION TEST PROCEDURES

1.0 APPARATUS

1.1 Sampling Apparatus

1.1.1 Filter Flasks

1.1.2 Filter Holder

1.1.3 Filter Funnel

1.1.4 Woulf Bottle

1.1.5 Tygon Tubing

1.1.6 Glass Tubing

1.1.7 Glass Stopcock

1.1.8 Forceps

1.1.9 Vacuum Pump

1.1.10 0.047 mm Millipore Filter Disks (HA)

1.1.11 Petri Dishes

1.1.12 Microscope (Binocular) 80x wide field

1.1.13 Microscope Lamp

1.1.14 Aluminum Foil

1.1.15 Rubber Bulb

1.1.16 Aerosol Standard Filter Holder

1.1.17 Reagents - Distilled Water, Isopropyl Alcohol, and Petroleum Ether

1.2 Electronic Equipment

1.2.1 Servo valve amplifiers with suitable feedback provisions; three required (see Figures 32 and 33)

1.2.2 Feedback Transducers

1.2.3 Brush Recorders and Amplifiers

1.2.4 Power Supplies

1.3 Hydraulic Equipment

1.3.1 Test Stand (see schematic Figure 2 and Figures 3 and 4)

2.0 GENERAL INSTRUCTIONS

2.1 White nylon coats should be worn by personnel performing cleaning operation to preclude excessive fiber contamination.

2.2 Cleaning and particle count must be performed in air-conditioned dust free environment where ingress and egress of personnel are kept to a minimum

3.0 CLEANING OF GLASSWARE AND TUBING

3.1 Cleaning Method

Each item of filtration apparatus that contacts sampled oil before filtration or the petroleum ether used to wash down the oil will be cleaned before each test by the following method,

3.1.1 Wash thoroughly in a solution of detergent and hot water. Rinse twice in hot tap water (soft).

3.1.2 Rinse twice with Millipore filtered distilled water.

3.1.3 Rinse twice with Millipore filtered isopropyl alcohol to remove the water.

3.1.4 Rinse twice with Millipore filtered petroleum ether to remove the alcohol.

3.1.5 Wrap all cleaned equipment in aluminum foil until assembled in the hydraulic laboratory.

4.0 FILTRATION AND PREPARATION OF REAGENTS

4.1 Clean all apparatus, as follows, if filtered reagents are not available.

4.1.1 Wash with detergent and water

4.1.2 Rinse twice with hot (soft)tap water

4.1.3 Assemble apparatus (filter holder, funnel filter flasks, clamps, and vacuum pump) with type HA Millipore filter disk in place

4.1.4 Filter 100-200 ml of distilled water into the filter flask. Remove funnel assembly and rinse the filter flask with the filtrate.

4.1.5 Repeat 4.1.4 three times

4.1.6 Filter the desired volume of distilled water

4.1.7 Rinse all storage bottles with 150 ml of filtered reagent

4.1.8 Repeat 4.1.7

4.1.9 Repeat 4.1.4 through 4.1.8 with isopropyl alcohol

4.1.10 Repeat 4.1.4 through 4.1.8 with petroleum ether

4.1.11 Fill large filter flask with filtered petroleum ether for use in the hydraulic laboratory

4.1.12 Filter 100 ml blank samples through Millipore. Analyze to ascertain that cleanliness level is as desired.

5.0 CLEANING HYDRAULIC OIL

5.1 Replace and clean filter elements and clean filter housings.

5.2 Fill the reservoir with hydraulic oil MIL-0-5606, prefiltered through a 10 micron Cuno filter, or equivalent.

5.3 Circulate the oil through the system and filter array until an oil sample indicates clean oil.

6.0 FINAL TEST PREPARATIONS

6.1 Static Servo Valve Tests

6.1.1 Mount the servo valve on the static stand

6.1.2 Set oil temperature to 150 degrees ± 15 degrees

6.1.3 Set oil pressure to valves rated operating pressure

6.1.4 Put positive rated current into the torque motor and record the flow as indicated on the flowmeter with no-load pressure differential equal to zero.

6.1.5 Repeat 6.1.4 with negative rated current

6.1.6 Block the load parts

6.1.7 Record the flow at rated current

6.1.8 Record the flow at valve null (approximately zero differential current). Valve null is where the blocked load flow is maximum.

6.1.9 Record the differential current required to obtain a load differential pressure of 25 psi

6.1.10 Closeoff oil pressure

6.1.11 Remove valve from static test stand and mount on contamination test stand

6.2 Assemble sampling apparatus to the test stand with Millipore filter in the filter holder and petroleum ether flasks connected to the check valve (see Figure 26)

6.3 Sample - Blank

6.3.1 Turn on vacuum pump

6.3.2 Open V_5

6.3.3 Pump rubber bulb

6.3.4 Open glass stopcock. Allow 150 ml of petroleum ether to go through the filter.

6.3.5 Close V_5

6.3.6 Remove filter holder assembly

6.3.7 Replace filter holder assembly with straight glass tube

6.3.8 Place used filter in Petri dish

6.3.9 Open V_5 and apply vacuum as you close V_5. Then close vacuum pump.

6.3.10 Replace glass tube with an assembled filter holder

6.4 Sample - Clean Oil

6.4.1 Turn on Hydraulic System

6.4.2 With V_1 adjust pressure to 500 psi

6.4.3 Open V_4 - Close V_4 after it has been open for 20 seconds. 75 cc of oil are trapped between V_4 and V_5.

6.4.4 Open V_5 - Oil will start to flow

6.4.5 When oil has filled the top of the filter holder turn on the vacuum pump, oil will then fill the tygon tube.

6.4.6 Pump rubber bulb

6.4.7 Open glass stopcock and allow 75cc of petroleum ether into the oil that is being filtered.

6.4.8 When the oil and petroleum ether mixture is down to top of filter holder once again, allow another 75cc of petroleum ether into the system.

6.4.9 Close V_5

6.4.10 When filter has dried, remove filter holder assembly and replace with glass tube.

6.4.11 Place used filter in Petri dish

6.4.12 Wash system down with 150 cc of petroleum ether by opening V_5

6.4.13 Replace glass tube with another complete filter assembly

6.5 Make particle count analysis on the samples taken

7.0 CONTAMINATION TEST

7.1 Turn on electronic equipment. Allow 1/2 hour warmup time and adjust.

7.2 Close V_6, open V_7

7.3 Turn on the Hydraulic System

7.4 Adjust system pressure with V_1

7.5 Turn on agitator

7.6 Add contamination

7.6.1 Open reservoir contamination fitting

7.6.2 Lower plastic contaminant container into the reservoir on the end of a nylon string moving up and down until all the contamination is rinsed into the oil.

7.6.3 Close the reservoir

7.7 Open V_{10} and Crack V_9. Take an oil sample for viscosity measurement.

7.8 Recording

7.8.1 Record amount of contamination added

7.8.2 Record temperature at input and output

7.8.3 Record spool end pressures

7.9 Take oil sample by opening V_4 for 20 seconds and following procedure of 6.3.4 to 6.3.13

7.10 Apply a temporary command signal to each valve, note the recorders

7.11 More recordings

7.11.1 Record the load pressures

7.11.2 After 9 minutes record the spool end pressures again

7.12 Repeat 7.5, 7.8, 7.10, 7.11 for all runs; repeat 7.7 for last run; repeat 7.9 for all odd number runs, 2nd run and last run or run during which failure occurs.

8.0 <u>DETERMINATION OF APPARENT CAUSE OF CONTAMINATION FAILURE</u>

8.1 Remove valves that have failed from the test stand

8.2 Circulate test stand oil through the filter array to clean the oil

8.3 If the symptom of failure was a hard-over condition and lower spool end pressure on one side, remove the filter on the side that indicates the lower spool end pressure.

8.4 For all other symptoms of failure, remove both filters.

8.5 Clean filters by washing in a solution of Kelite or equivalent industrial detergent.

8.6 Rinse well with tap water

8.7 Reverse flush with filtered compressed air

8.8 Replace cleaned filters

8.9 Mount valves on contamination test stand and check to see if control has been regained.

8.10 If control has been regained, the apparent cause of failure was filter clogging; if not, proceed to 8.11.

8.11 Repeat 8.3 through 8.10 for the orifices

8.12 If control still has not been regained, clean nozzle by flushing with compressed air.

8.13 Remount on contamination stand and determine if control has been regained.

8.13.1 When control is regained, it is assumed that the component of the valve most recently cleansed was the salient cause of the malfunction.

9.0 POST CONTAMINATION STATIC SERVO VALVE TESTS

9.1 Completely repeat 6.1

10.0 PARTICLE COUNT

10.1 Set microscope for magnification of 80x. Use oblique incidental light.

10.2 Set Petri dish on adjustable stage and remove cover.

10.3 Scan filter to determine approximate particle count in the following size ranges:

Less than 10 microns

10 - 20 microns

20 - 40 microns

Over 40 microns

10.4 Based on the previous approximation the following counts are required:

Condition (1) 0 - 100 particles on all the grid squares on the filter

Condition (2) 100 - 1000 particles on 20 randomly chosen grid squares

Condition (3) 1000 - 2000 particles on 10 randomly chosen grid squares

Condition (4) Over 2000 particles within 10 randomly chosen whipple disk area.

10.5 To arrive at the total statistical count, the following multiplication factors are required for the conditions listed above:

Condition (1) x 1

Condition (2) x 5

Condition (3) x 10

Condition (4) x 42.6*.

The test is now completed.

*The 42.6 is based on the particular microscope configuration used. It is the factor that makes the area actually counted equal to the entire filter area.

APPENDIX III

COBALT 60 GAMMA IRRADIATION SOURCE

The Cobalt 60 source at Inland Testing Laboratories is a pure gamma irradiation facility presently operating at a level of 62,100 curies. The source is comprised of 7000 cobalt cylinders, each measuring 1 inch long with a diameter of 1/4 inch. Each of the cylinders is individually encapsulated in 1/16 inch aluminum to provide a final slug 1-1/8 inches long and 3/8 inch in diameter. The over-all configuration of the source is cylindrical, and provides over 7 square feet of outside surface area for irradiation of test specimens. The core formed by the cylinder provides an additional 3 cubic feet of area which can be used to subject test specimens to more intense radiation. The source is normally kept in a subterranean storage chamber. An automatic elevator is provided to raise the source to access level.

The irradiation cell is a 16 foot concrete cube. The front wall and cell door are constructed of steel shells filled with high density (200 lb/cu ft) magnetite concrete. The three remaining walls are 68 inches thick, and the roof is 54 inches thick. The cell door is mounted on tracks and power-operated. The cell is fully shielded to permit safe operation, even with the source raised to the height of the access holes.

A view of the interior of the chamber is provided by an access window, which consists of a 36 inch heavy density, nonbrowning lead glass window, plus an 11 inch thick slab of standard glass on the exterior. A remotely operated overhead crane and a set of Argonne Type 8 master-slave manipulators enable movement of objects within the chamber. The manipulators have side indexing, which permits coverage of a larger working area than with conventional arms. Figures 35 and 36 illustrate some of the characteristics of the gamma facility.

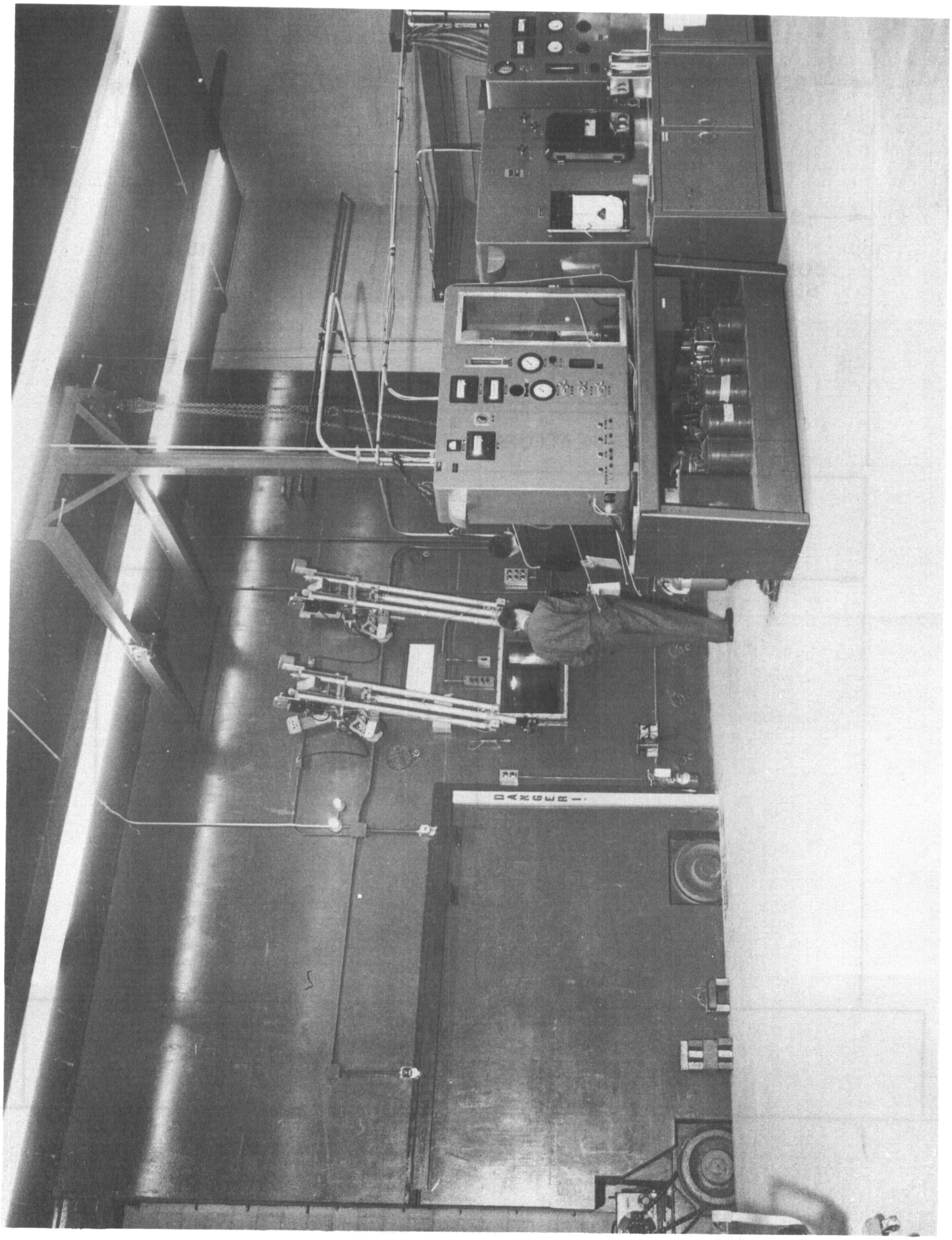

Figure 35. The World's Largest Cobalt 60 Source

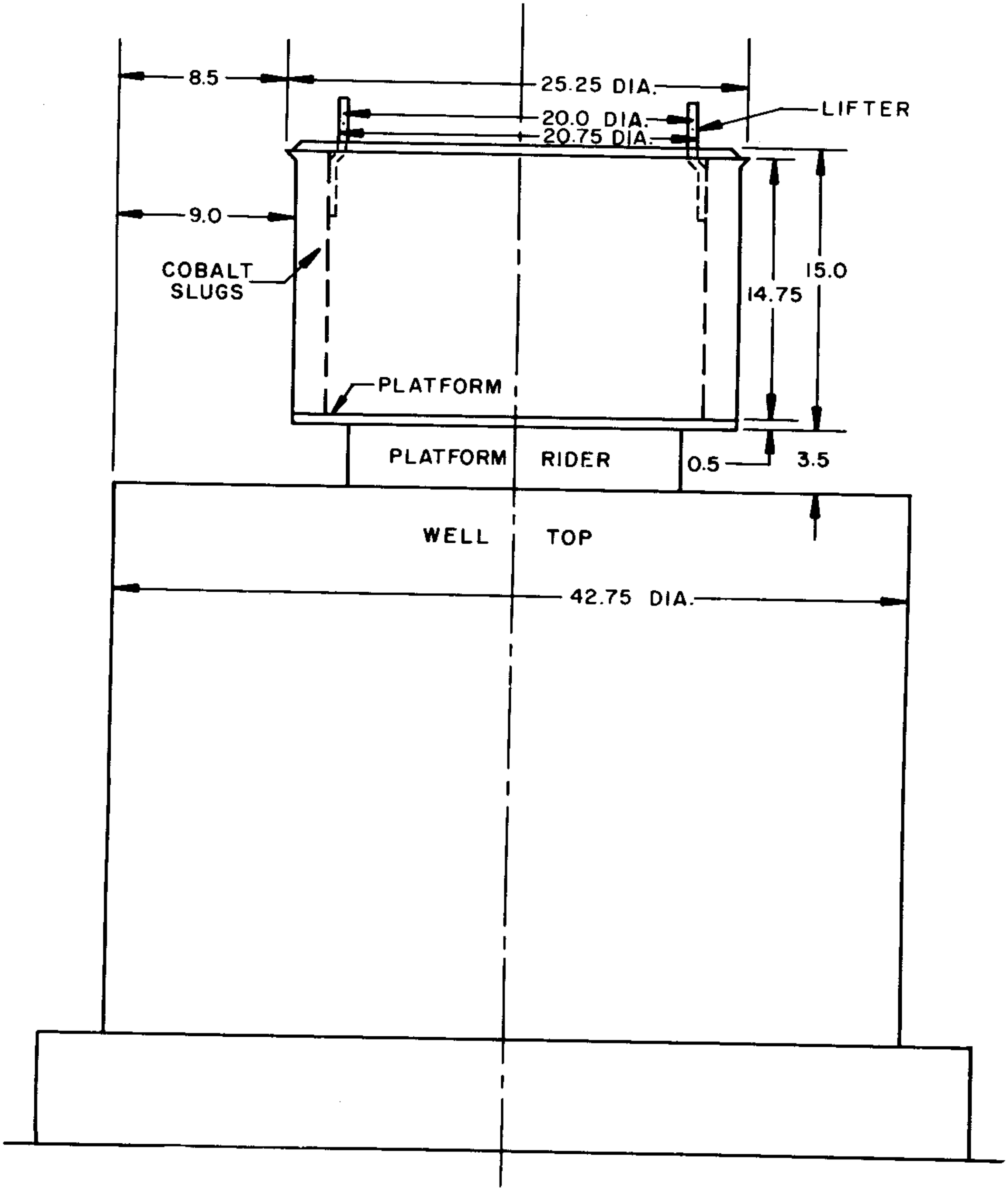

Figure 36. Dimension of Well and Cobalt Source

APPENDIX IV

NEW SERVO VALVE TEST RESULTS

The following figures, 37 through 62, show result of performance tests as conducted on the Cadillac Gage FC-200 valve and Raymond Atchley 410 servo valve. Results of these tests are summarized in Chapter V, Section D of the report.

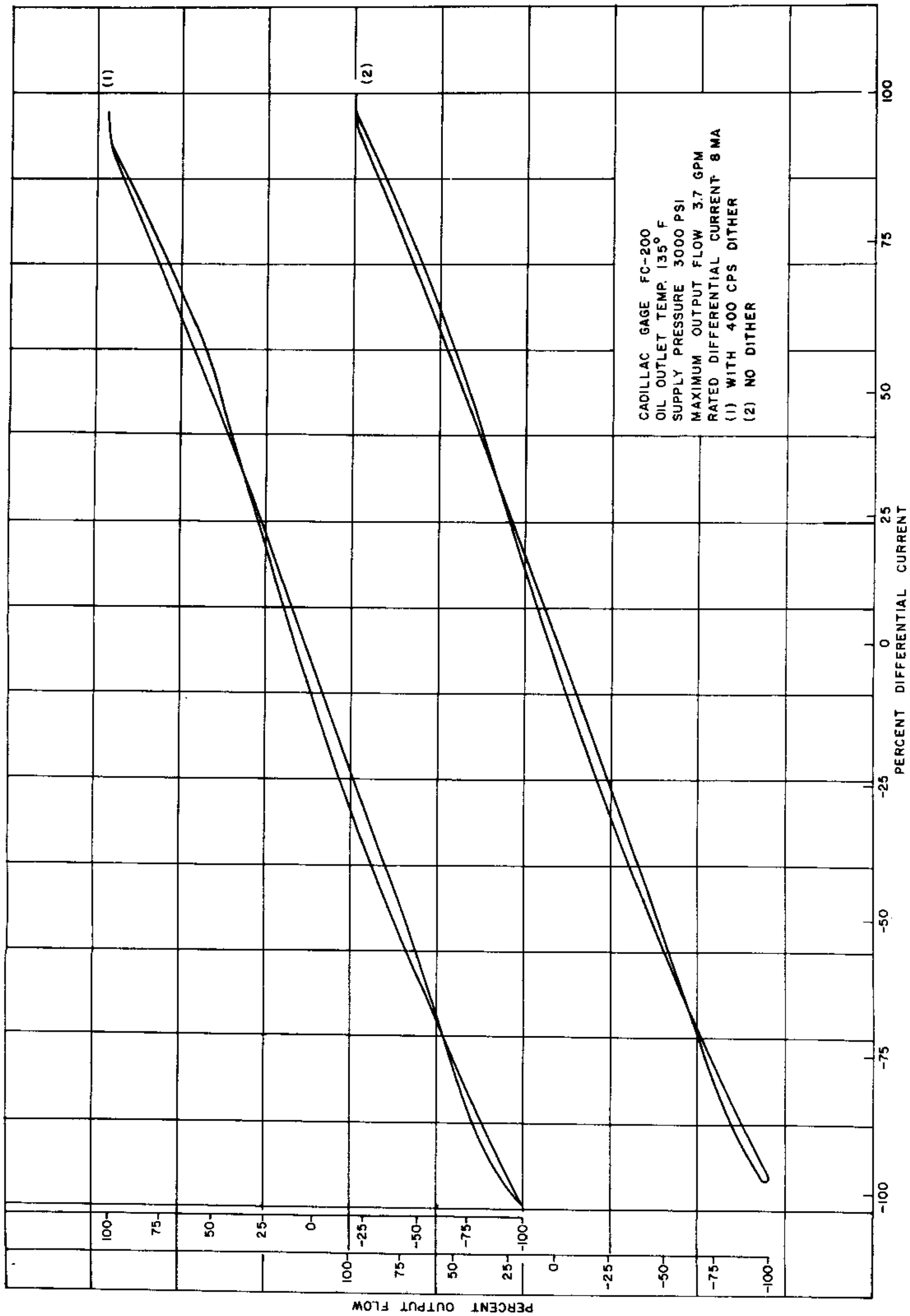

Figure 37. Cadillac Gage FC-200 Hysteresis-Rated Current

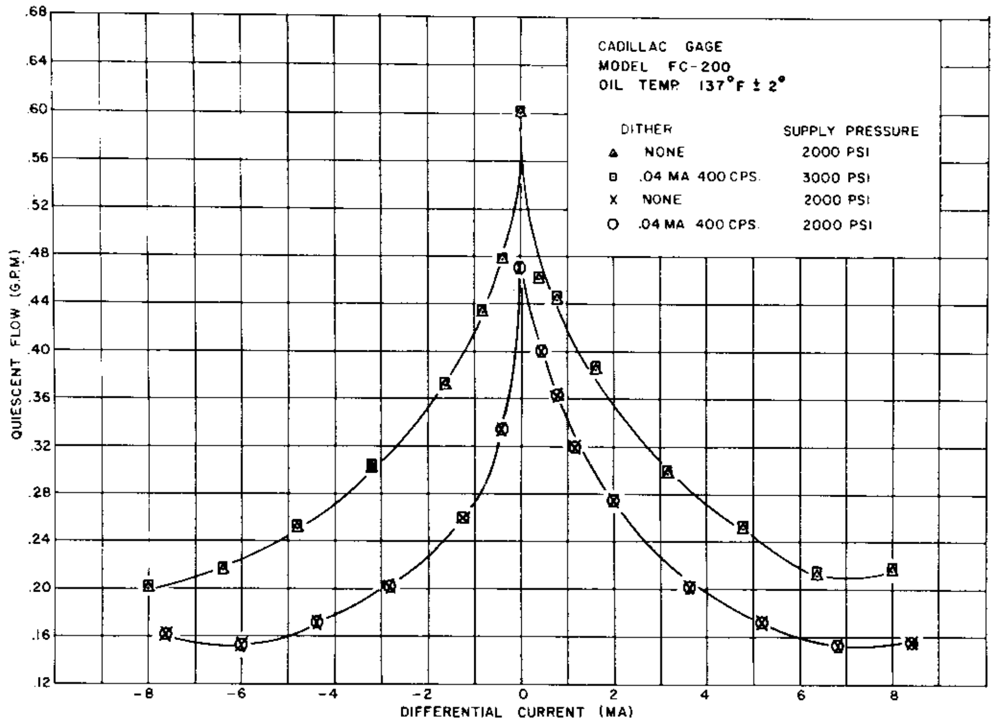

Figure 38. Cadillac Gage FC-200 Quiescent Flow vs. Differential Current

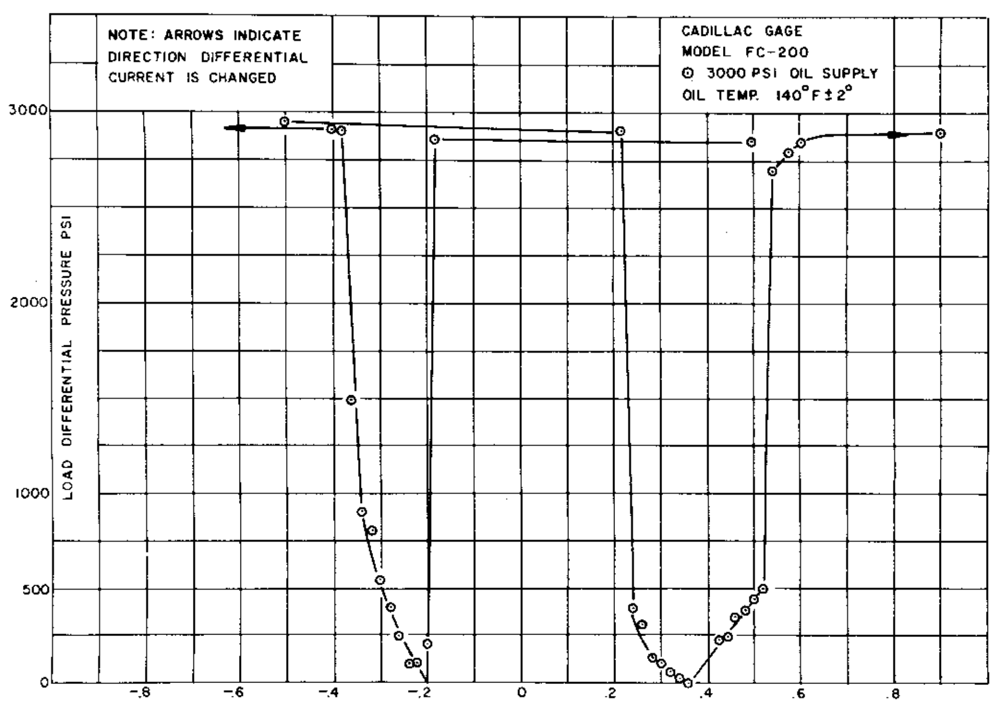

Figure 39. Cadillac Gage FC-200 Load Pressure vs. Differential Current at Zero Load Flow, 3000 psi Supply Pressure

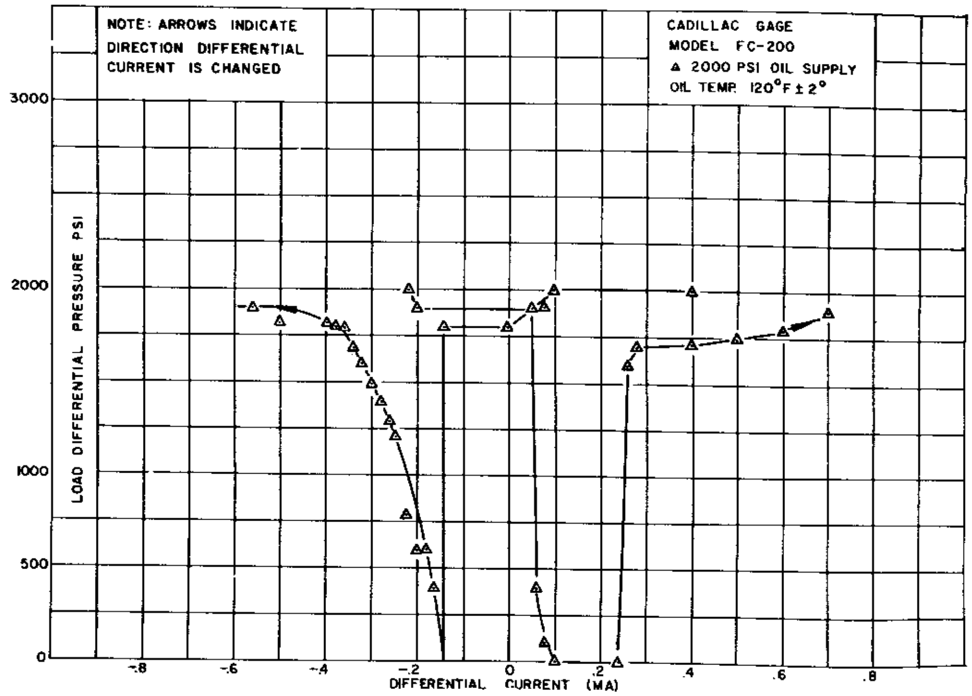

Figure 40. Cadillac Gage FC-200 Load Pressure vs. Differential Current at Zero Load Flow, 2000 psi Supply Pressure

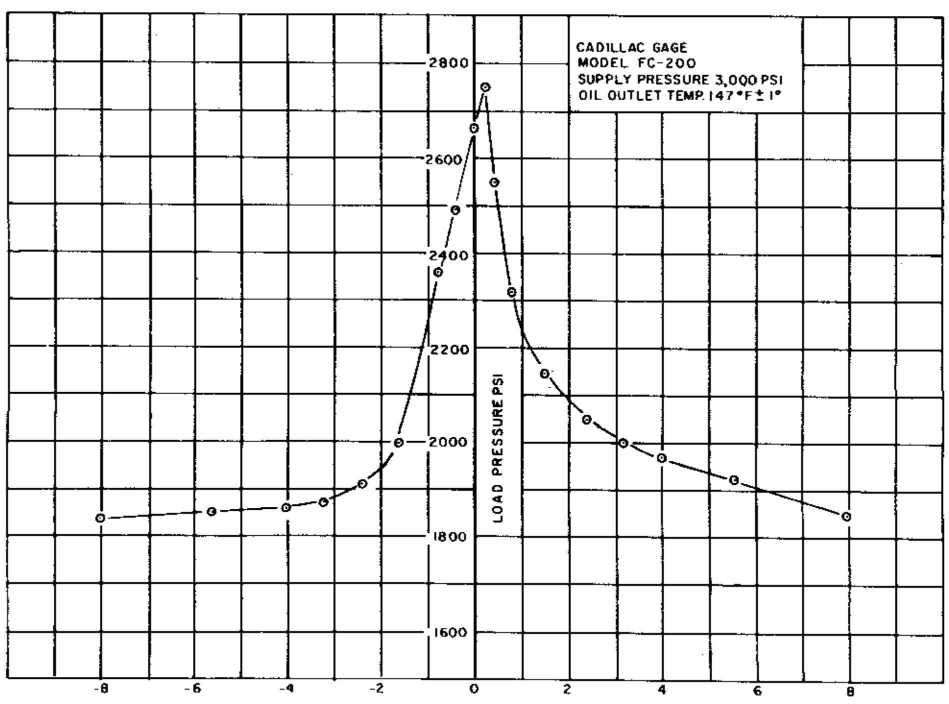

Figure 41. Cadillac Gage FC-200 Short Circuited Load Pressure vs. Differential Current

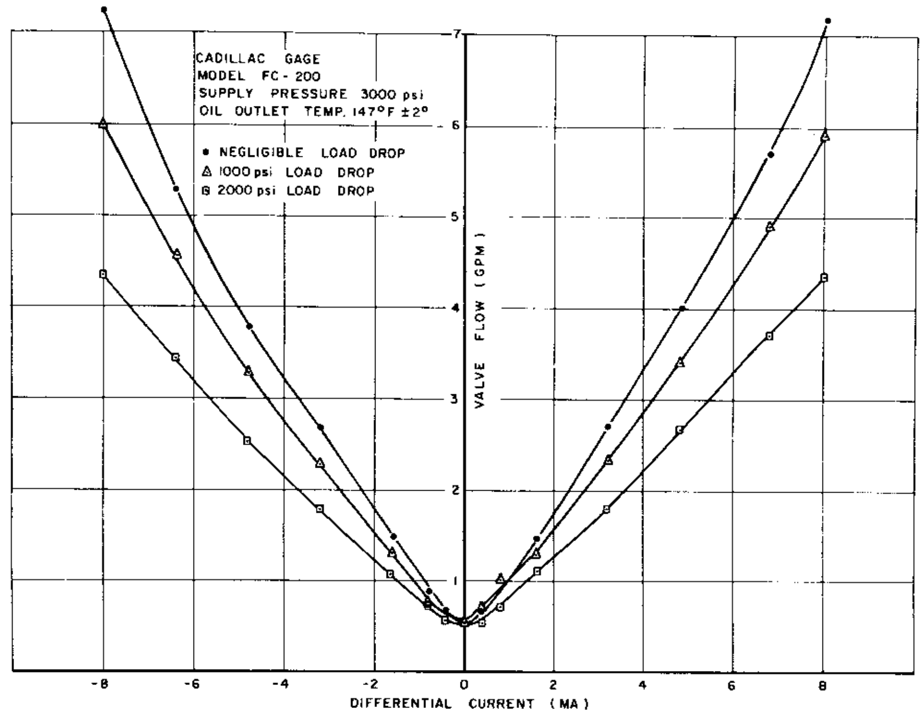

Figure 42. Cadillac Gage FC-200 Valve Flow vs. Differential Current 3000 psi Supply Pressure

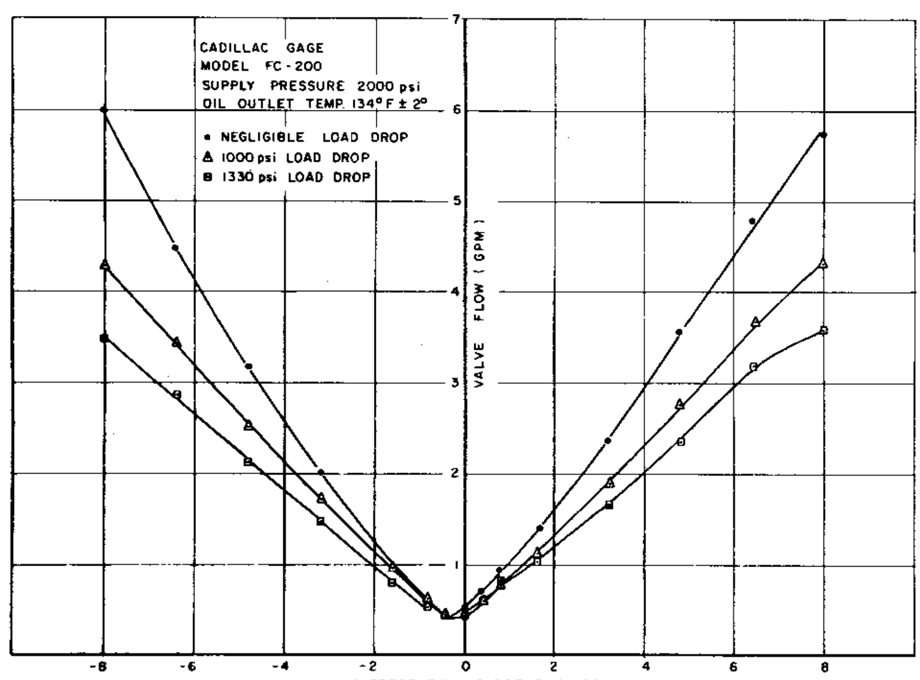

Figure 43. Cadillac Gage FC-200 Valve Flow vs. Differential Current 2000 psi Supply Pressure

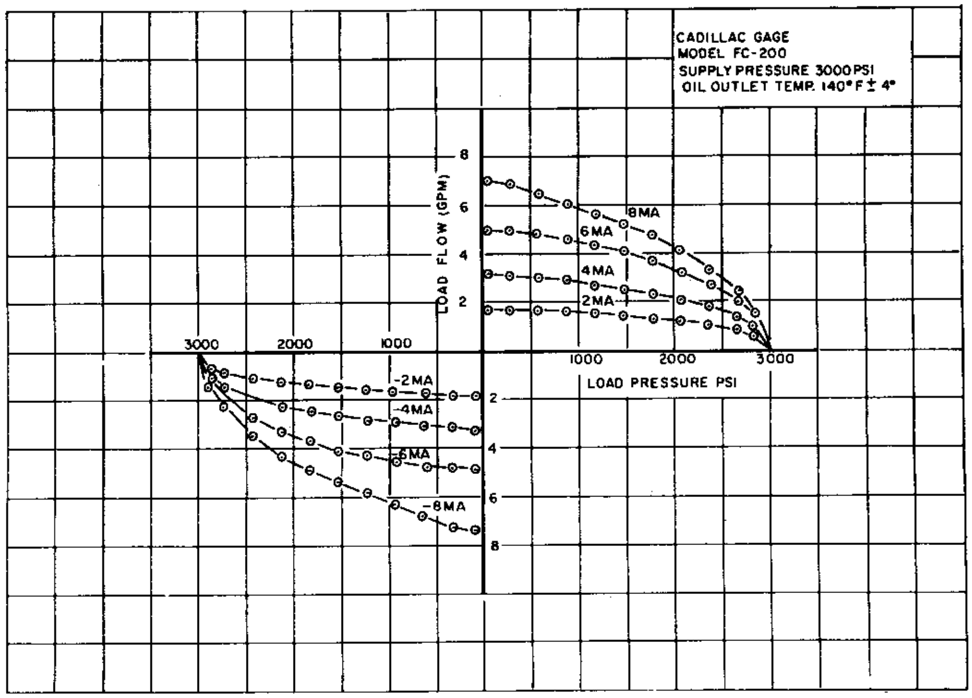

Figure 44. Cadillac Gage FC-200 Load Flow vs. Load Pressure 3000 psi Supply Pressure

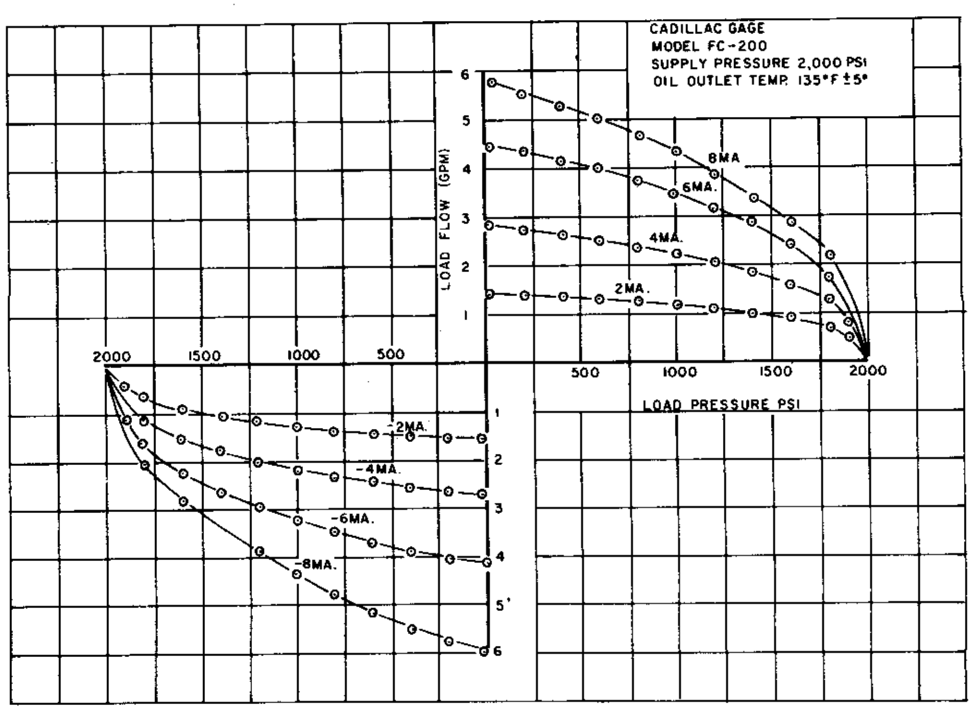

Figure 45. Cadillac Gage FC-200 Load Flow vs. Load Pressure 2000 psi Supply Pressure

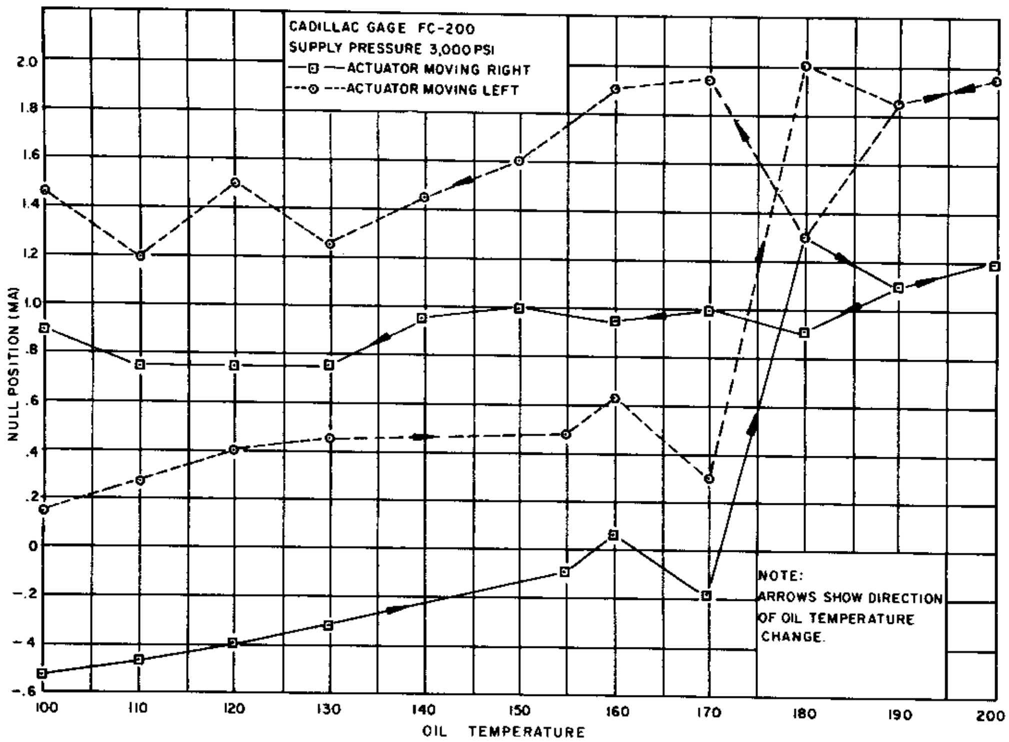

Figure 46. Cadillac Gage FC-200 Null Shift vs. Oil Temperature - No Dither

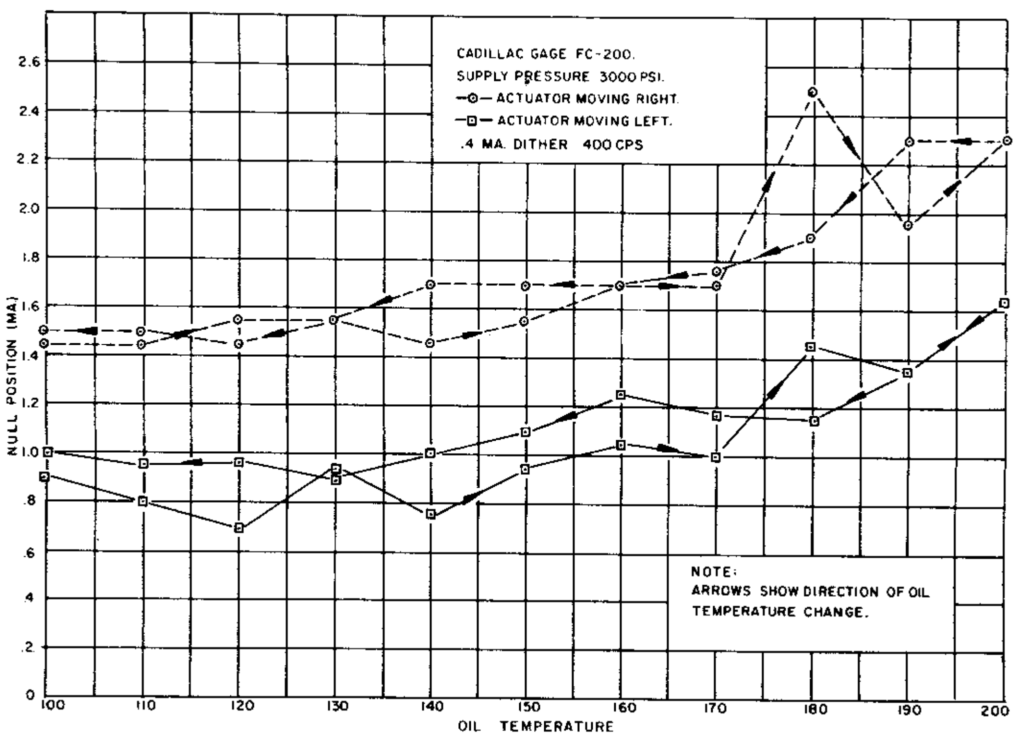

Figure 47. Cadillac Gage FC-200 Null Shift vs. Oil Temperature - Dither

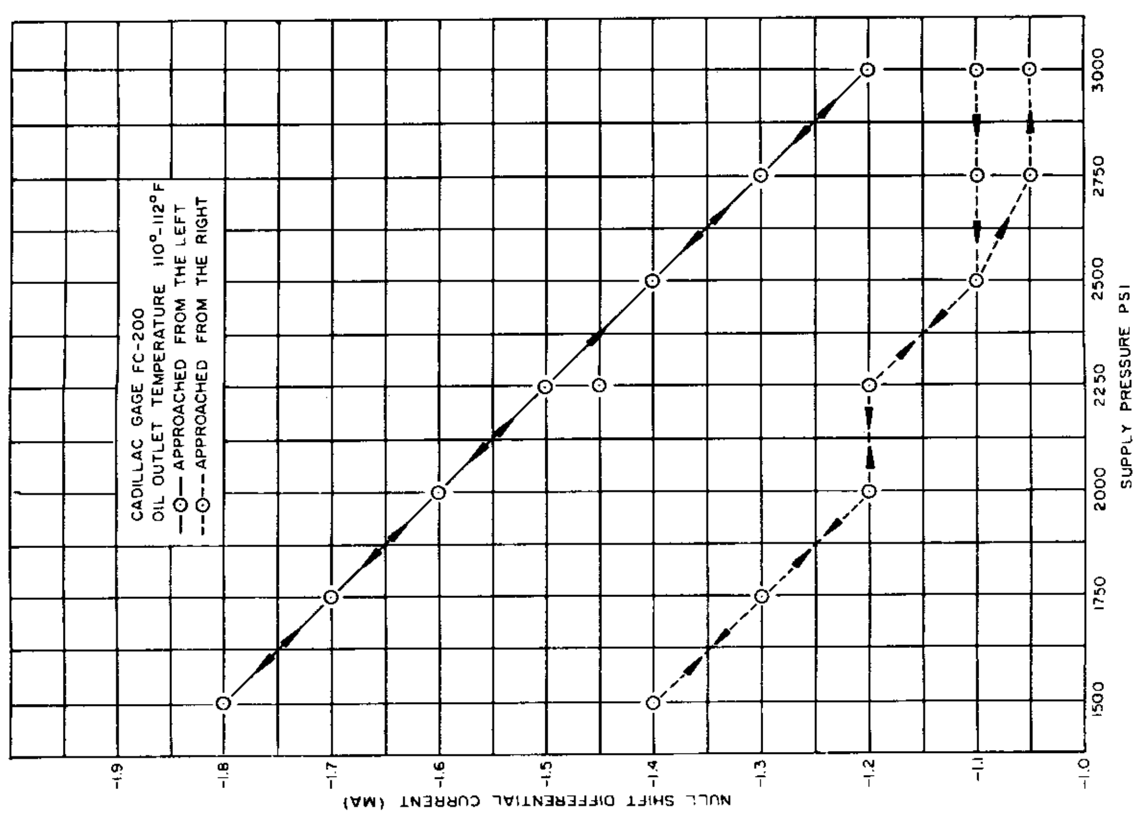

Figure 48. Cadillac Gage FC-200 Null Shift vs. Supply Pressure - No Dither

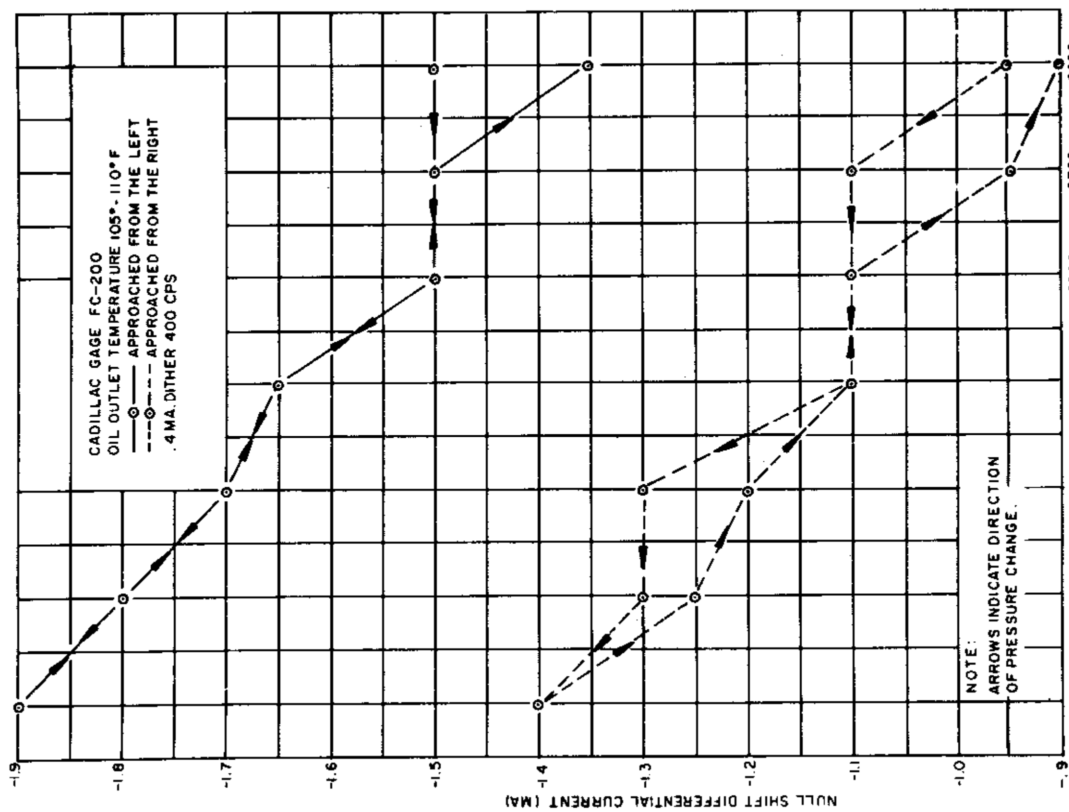

Figure 49. Cadillac Gage FC-200 Null Shift vs. Supply Pressure - Dither

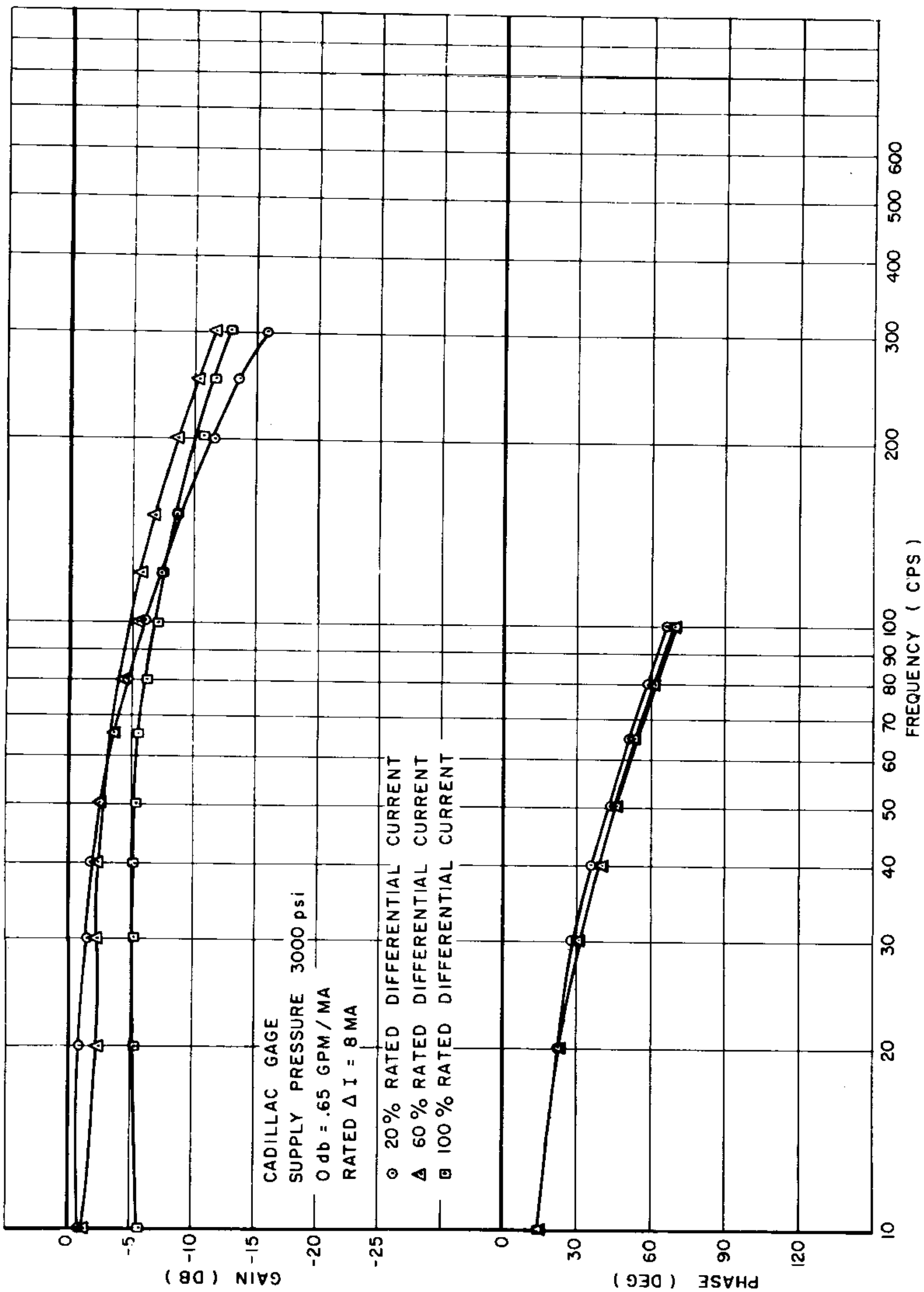

Figure 50. Cadillac Gage FC-200 No-Load Frequency Response

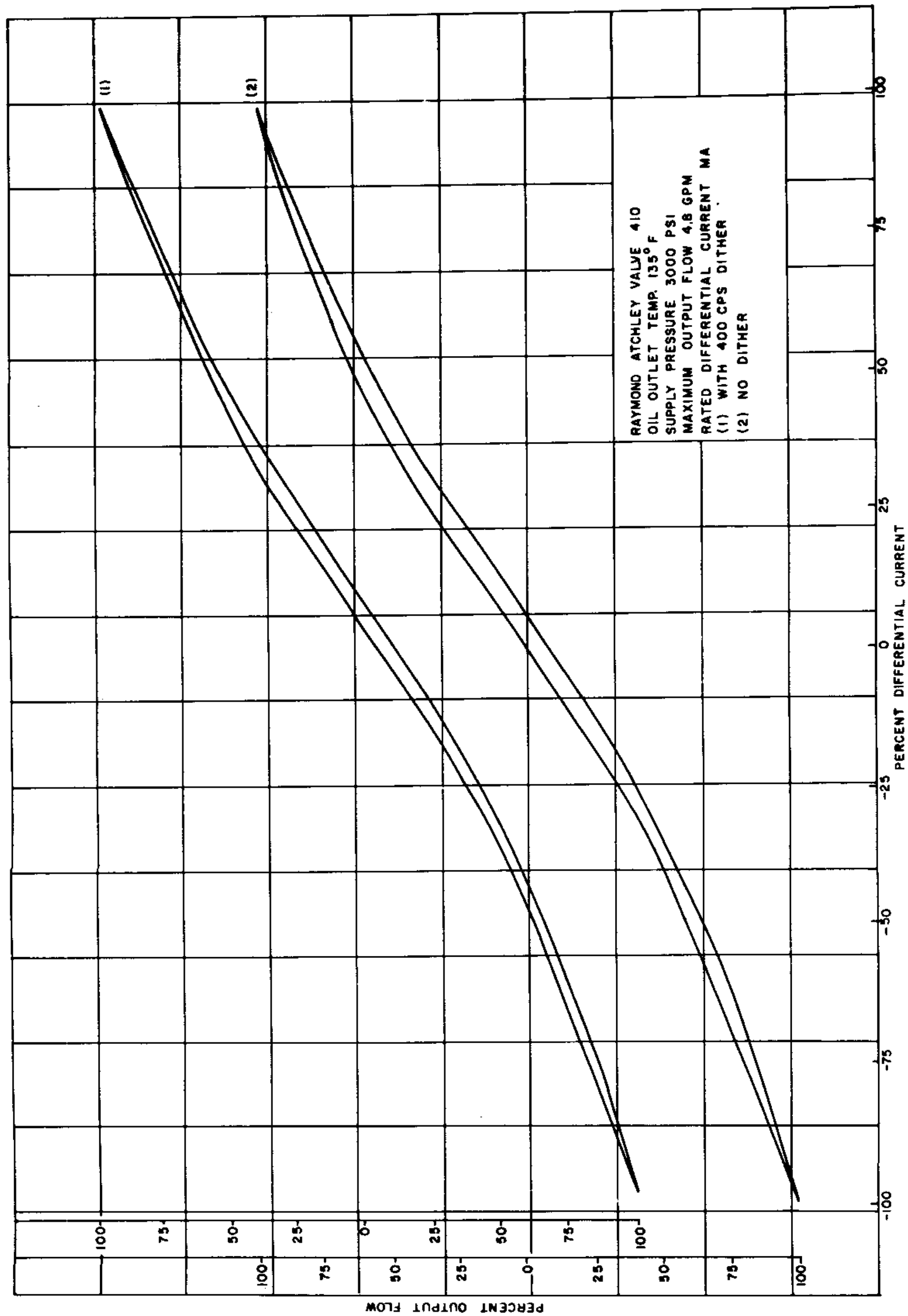

Figure 51. Raymond Atchley 410-298 Hysteresis vs. Rated Current

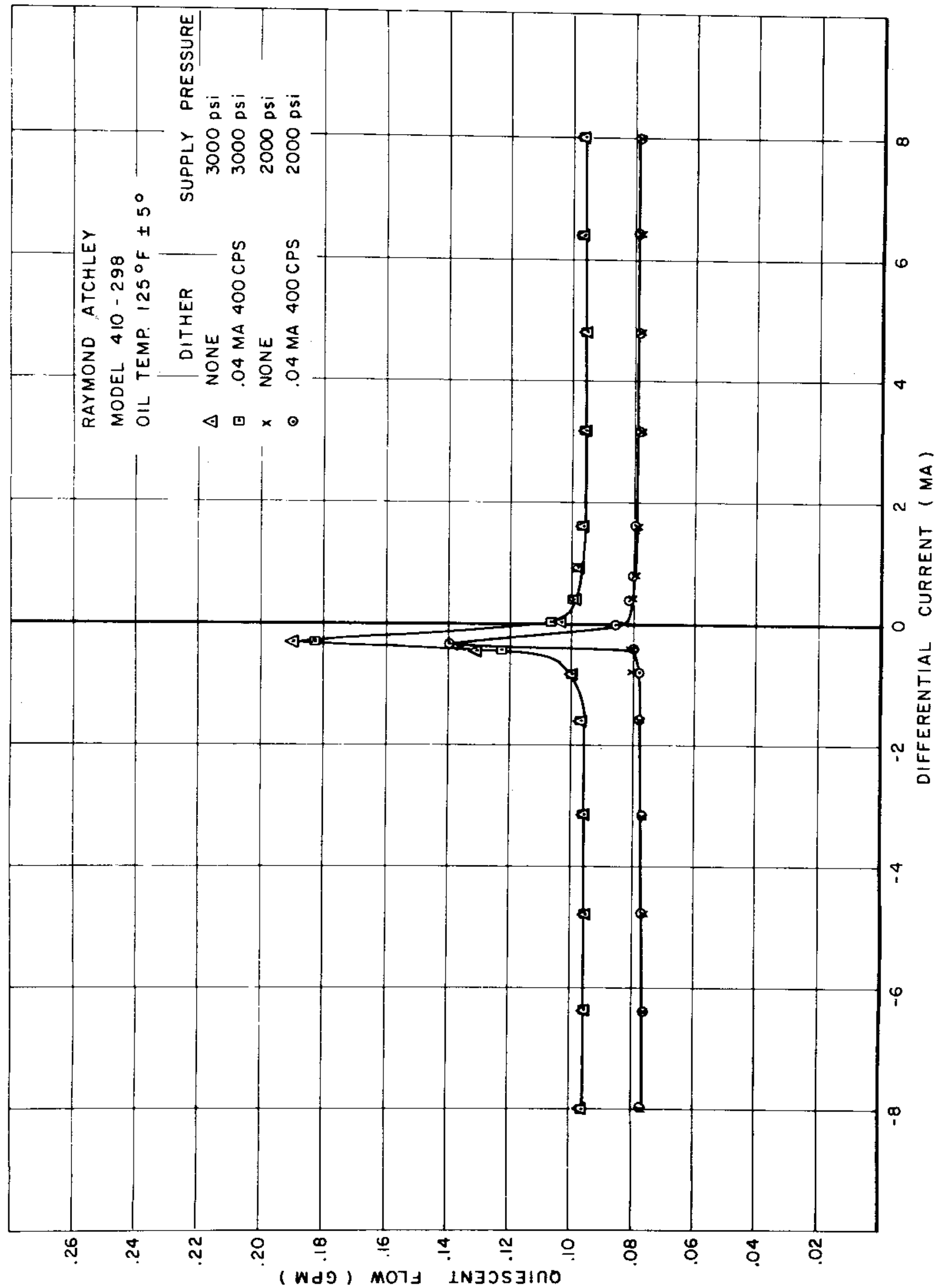

Figure 52. Raymond Atchley 410-298 Quiescent Flow vs. Differential Current

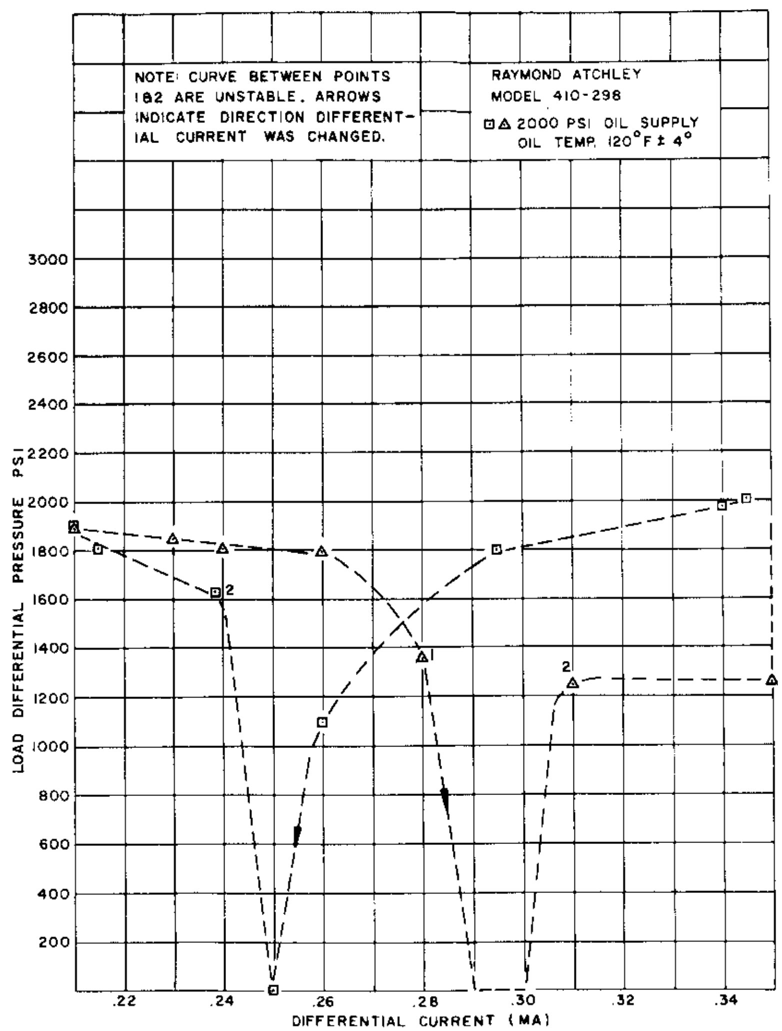

Figure 53. Raymond Atchley 410-298 Load Pressure vs. Differential Current at Zero Load Flow 2000 psi Supply Pressure

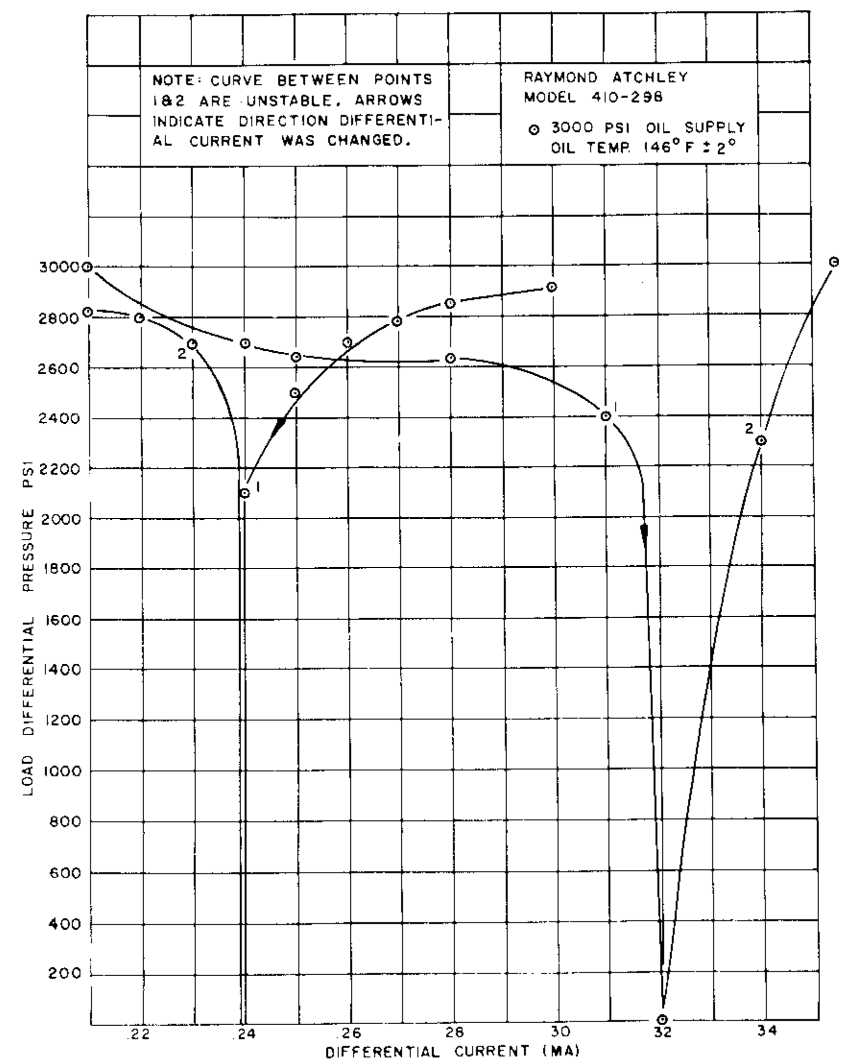

Figure 54. Raymond Atchley 410-298 Load Pressure vs. Differential Current at Zero Load Flow 3000 psi Supply Pressure

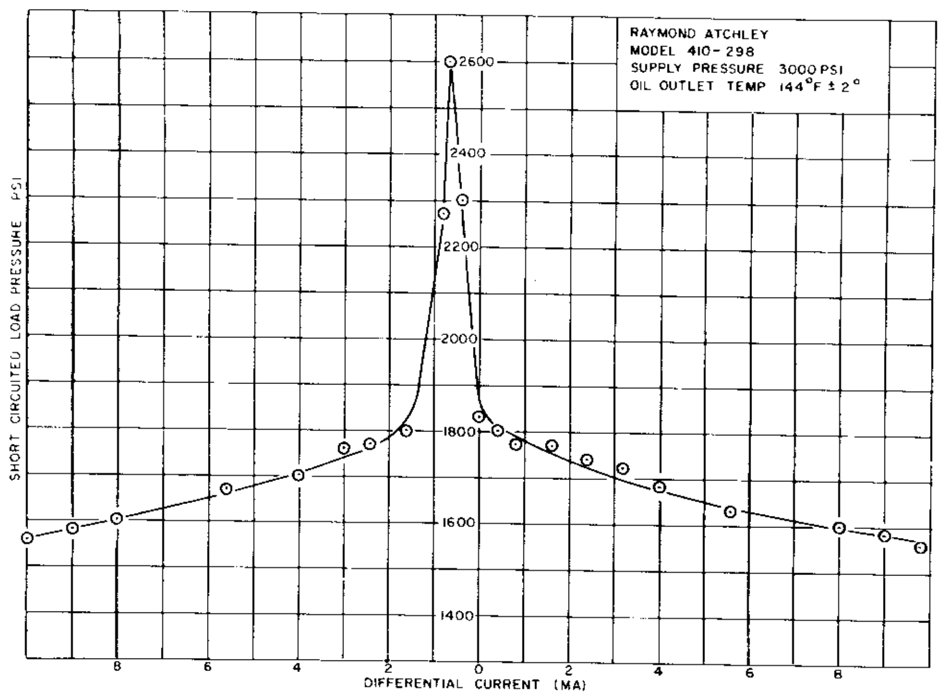

Figure 55. Raymond Atchley 410-298 Short Circuited Load Pressure vs. Differential Current

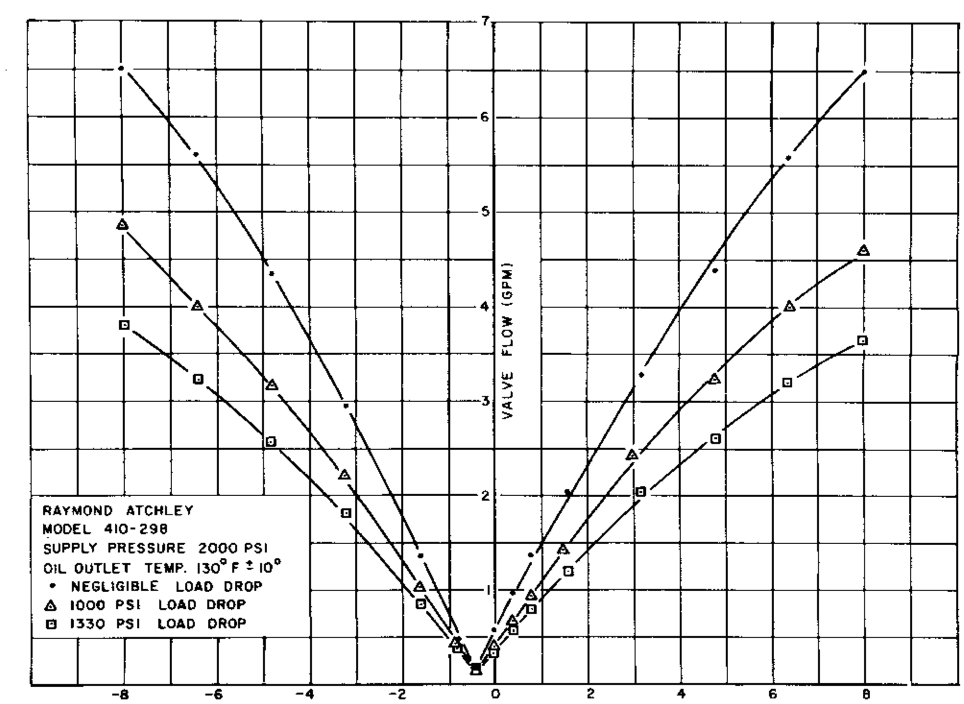

Figure 56. Raymond Atchley 410-298 Valve Flow vs. Differential Current 3000 psi Supply Pressure

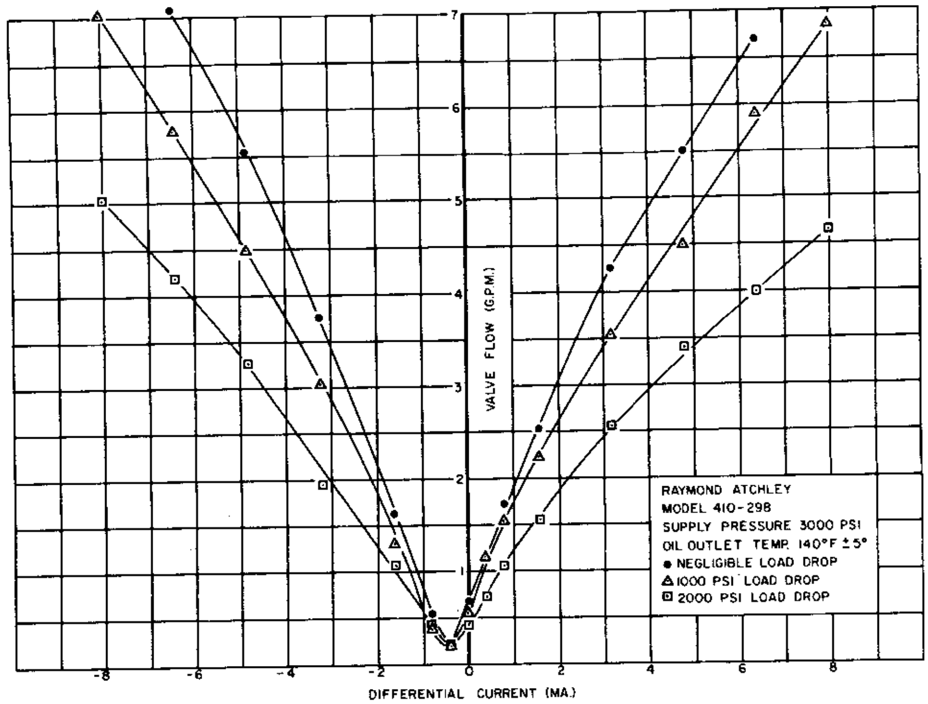

Figure 57. Raymond Atchley 410-298 Valve Flow vs. Differential Current 2000 psi Supply Pressure

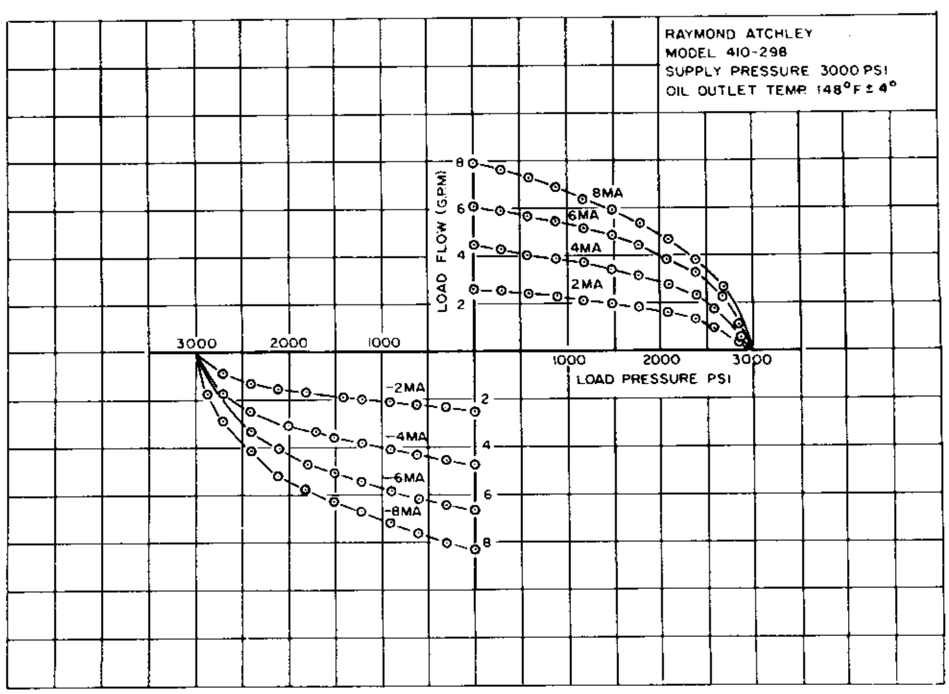

Figure 58. Raymond Atchley 410-298 Load Flow vs. Load Pressure 3000 psi Supply Pressure

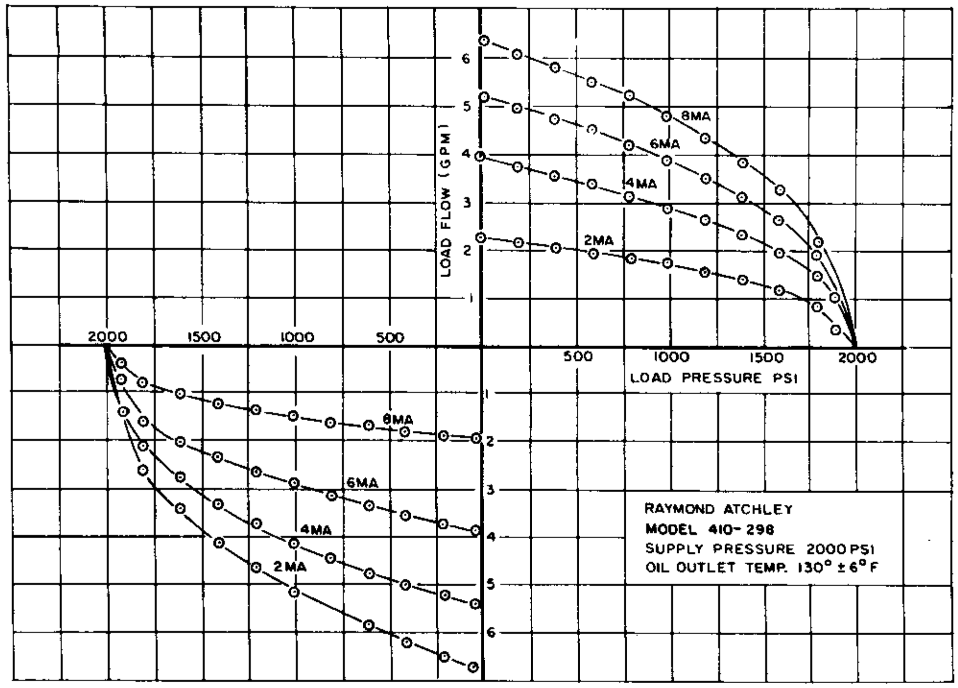

Figure 59. Raymond Atchley 410-298 Load Flow vs. Load Pressure 2000 psi Supply Pressure

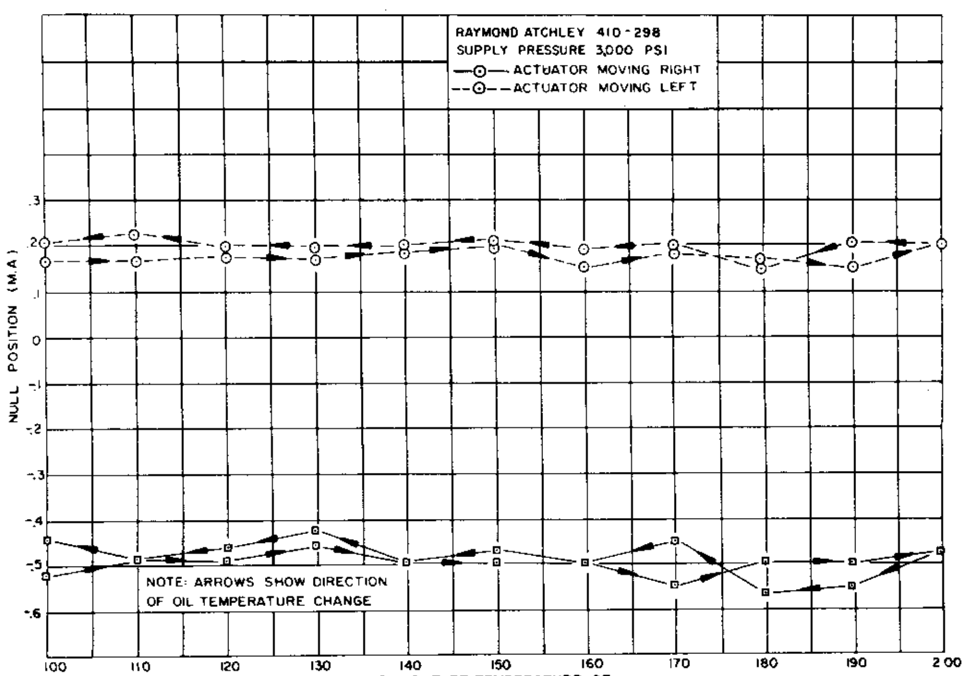

Figure 60. Raymond Atchley 410-298 Null Shift vs. Oil Temperature - No Dither

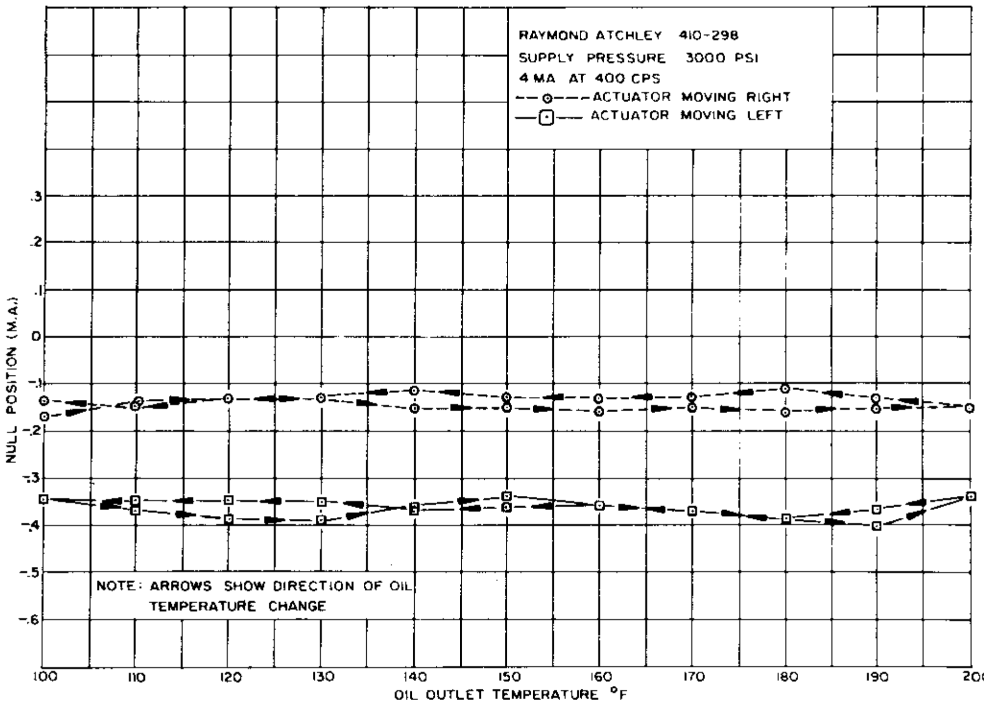

Figure 61. Raymond Atchley 410-298 Null Shift vs. Oil Temperature - Dither

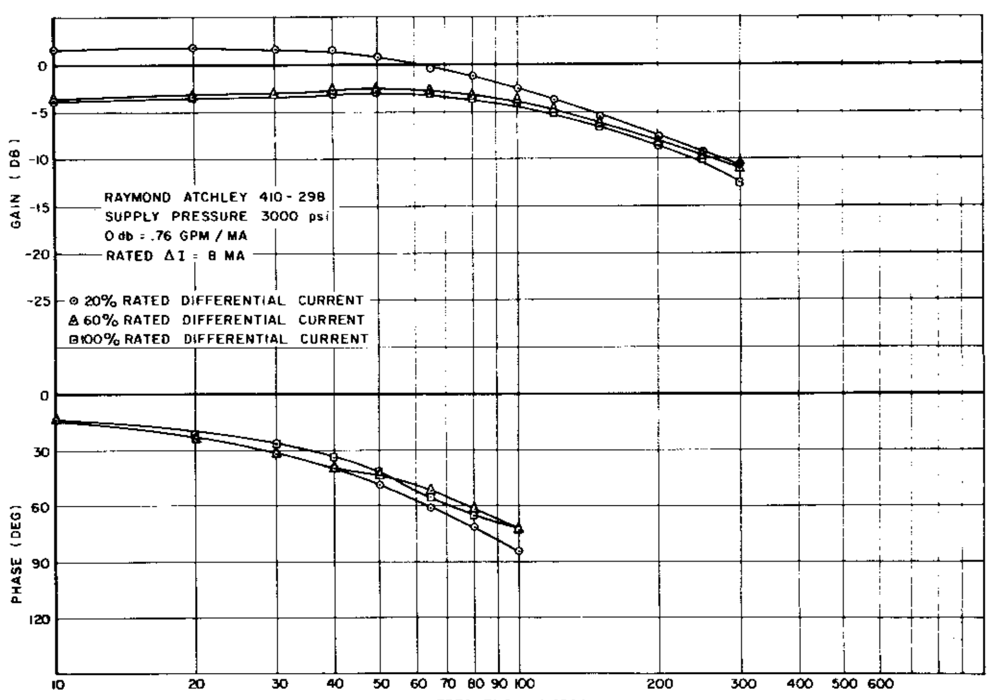

Figure 62. Raymond Atchley 410-298 No-Load Frequency Response

APPENDIX V

TIME-MODULATED VALVE TEST PROCEDURES

A. Measurement of Acceleration Gain

For this test the valve was manifolded to a small low friction actuator which had no-load attached. A small accelerometer was affixed to the actuator shaft and a 5 kc carrier signal applied to the accelerometer.

Gain was observed by applying a low frequency (8 cps) signal of measured magnitude to the valve driver amplifier and recording the magnitude of resulting acceleration. The input signal was varied in several steps between 0 and rated differential current and actuator acceleration measured at each value of input.

Instrumentation required for this test, in addition to the accelerometer, included a low frequency oscillator to supply a signal to the valve driver amplifier, a phase sensitive demodulator and amplifier, a vacuum tube voltmeter, and a low frequency oscilloscope.

To permit stable operation of the valve actuator system, it was necessary to include a position feedback loop with appropriate lead compensation.

B. Measurement of Frequency Response

For this test the valve was manifolded to the no-load actuator in a manner similar to that described under A, above. Identical instrumentation including the accelerometer was also employed.

Frequency response was observed by applying a signal of fixed measured magnitude to the driver amplifier and recording the magnitude of resulting acceleration. Frequency of the input signal was varied over the range of interest and output acceleration observed at each point. This information was then used to plot a curve of valve output or spool acceleration vs. frequency. Previous tests showed no attenuation in actuator response below 2000 cps, so over the frequency range of the test, actuator acceleration was directly proportional to spool velocity.