A STUDY OF SEAL FRICITION IN HYDRAULIC SERVOMECHANISMS

by

Melvin S. Feldman, 1st Lieutenant, USAF
B.S., University of Pittsburgh, 1950

Eugene C. Wrieden, 1st Lieutenant, USAF
B.A.E., New York University, 1952

SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF MASTER OF SCIENCE at the MASSACHUSETTS INSTITUTE OF TECHNOLOGY 1954

Signature of Author ________________________________

Dept. of Aeronautical Engineering, May 24, 1954

Certified by ________________________________ Thesis Supervisor

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A STUDY OF SFAL FRICITION IN HYDRAULIC SERVOMECHANISMS

by

Melvin S. Feldman
Eugene C. Wrieden

Submitted to the Department of Aeronautical Engineering on May 24, 1954, in partial fulfillment of the requirements for the degree of Master of Science.

ABSTRACT

The friction forces encountered in a high performance hydraulic servomechanism may drastically affect the servo characteristics. The main sources of friction are the hydraulic seals used in the servo actuating cylinder.

In this thesis, the frictional characteristics of the two kinds of hydraulic actuator seals (i.e., piston and rod) were studied separately. Seals made of rubber, cast iron, and Teflon were tested. The static and kinetic friction force levels were investigated. The relative effects of the various seals on the dynamic response of the servo were also examined. Some of the other topics included are as follows: a summary of friction theory, a description of the test equipment and procedures, and a discussion of the methods used to evaluate the test data.

Of the seals tested, two-element Teflon rings, used in conjunction with an "O" ring back-up, had the lowest friction levels. The dynamic response tests showed that all of the test seals affected the servo characteristics to approximately the same degree. However, a high seal friction level will actually deteriorate the response time of the servo; and thus may seriously affect its dynamic characteristics.

Thesis Supervisor: Walter Wrigley, Associate Professor of Aeronautical Engineering
May 24, 1954

Professor Leicester F. Hamilton
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge, 39, Massachusetts

Dear Professor Hamilton:

In accordance with the regulations of the faculty, we hereby submit a thesis entitled *A Study of Seal Friction in Hydraulic Servomechanisms* in partial fulfillment of the requirements for the degree of Master of Science.

Melvin S. Feldman

Eugenio C. Wrieden
ACKNOWLEDGMENTS

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OBJECT

The object of this thesis is to determine the frictional characteristics of various pressure seals that can be used in hydraulic servomechanisms, and to determine the effect of these seals on the hydraulic servomechanism performance.
CHAPTER I

GENERAL INTRODUCTION

A. Background

Since the time of the first powered flight, commercial and military aircraft have increased rapidly in both size and maximum speed attainable. The forces required to move the control surfaces have increased proportionately until the pilot is no longer able to supply these forces himself. He must be aided by a "mechanical muscle", and this mechanical muscle most often is a hydraulic servomechanism.

The need for the hydraulic servo was also felt with the development of the autopilot. The electronic equipment could sense the corrections needed to properly position the control surfaces, but the hydraulic servo was needed to supply the forces required.

At this point one might ask, why use a hydraulic system to supply this actuating force? Pneumatic and electrical devices are widely used and time-tested sources of power. Since the operation of lighting and electronic equipment requires an electrical system, it would appear that electricity would provide the ideal power source. However, this is not true. The equipment required for electrical operation of devices which require considerable torque becomes both heavy and bulky. In operations which necessitate instantaneous stoppage of the driving mechanism, it is not possible to use an electric motor. The inertia of its armature is too high to permit instantaneous stoppage when the current is shut off.

Pneumatic power operation of the control surfaces, either by compressed air or vacuum, has the inherent disadvantages of uneven pressure control and a lag in operation due to compressibility of the air. In addition, air pressure systems present difficult maintenance problems and the hazard from the use of extremely high-pressure air storage bottles. Vacuum systems are incapable of supplying more than a minute quantity of power, unless an enormous actuating cylinder is used.

Hydraulic systems have moving parts of low inertia, and consequently, such systems are ideally suited for operations requiring instantaneous stoppage. Since the hydraulic fluid is nearly incom-
pressible, the system does not exhibit the lags found in pneumatic systems. Thus, it can be seen that hydraulic systems are characterized by a high degree of controllability which is not found in either pneumatic or electrical systems.

For military aircraft a hydraulic system offers one important disadvantage: any leakage in hydraulic lines, as a result of vibration or gun fire, results in complete failure of the system. However, the advantages of (1) light weight, (2) low inertia of moving parts, (3) controllability, (4) ability to supply almost unlimited force, and (5) ease of installation and maintenance far outweigh this one disadvantage.

B. Hydraulic Actuator

The component of the hydraulic servo which actually applies the force to the control surface is the actuator. A schematic diagram of an actuator is shown in Fig. 1-1. It consists of a piston which is free to move in a cylinder. The direction and velocity of the motion depend on the differential pressure which exists between the two faces of the piston. The end rods transmit the force to the linkage system which is connected to the control surface.

![Fig. 1-1. Typical actuator.](image)

Although $P_1$ and $P_2$ are equal when there is no piston motion, the value of this pressure may be several hundred pounds per square inch. In order to maintain this pressure in the system, without causing excess leakage of hydraulic fluid, it is necessary to have hydraulic seals at each rod bearing (A). A third seal is needed on the piston (B), to enable differential pressures to exist between the two piston faces.

To obtain a good seal, the rod and piston seals must exert forces normal to the direction of piston motion. These normal forces, in turn, cause frictional forces to be developed. At one time this frictional problem was not very acute, but as more exacting requirements were placed on the servo, the problem increased in importance.
In modern high-speed aircraft, small control surface deflections will produce large changes in aerodynamic moments due to the high dynamic pressures acting on the surfaces. Thus, accurate control of the aircraft requires a servo control system with a high degree of angular resolution. This means that the system must respond to small input signals. The magnitude of the smallest input signal which will actuate the system is, in turn, dependent on the frictional forces existing in the actuator. It is for this reason that, in recent years, frictional problems have taken on increased stature in hydraulic servo design.

C. Purpose of Thesis

In undertaking this thesis, the authors hoped to obtain a comparative measurement of the friction levels which are encountered when using various types of hydraulic seals. Force levels required to initiate piston motion, as well as maintain it were studied. The effect of various seals on the dynamic response of the servo was also investigated. The results obtained are intended to serve as an aid to future designers of high-performance hydraulic servos, enabling them to select the type of seal best suited to their particular needs.

D. Approach to the Problem

Another look at Fig. 1-1 will reveal an important difference between the piston seal and the rod seal. The side of the piston seal which is subjected to the higher pressure is dependent on the direction of piston motion. This is not true in the case of the rod seals. It was therefore considered desirable to study the effects of the piston and rod seals separately. This placed the first requirement on the test equipment.

The next requirement was that some means of running the moving part of the actuator at constant velocity must be provided in order to obtain force vs. velocity measurements. A third requirement was that the performance of the servo without a hydraulic seal must be evaluated. The test equipment used to satisfy these requirements is described in Chapter III.

Once the test equipment was devised, some means of evaluating the results, both quantitatively and qualitatively, had to be decided upon. Appendix B describes how the actual numerical values were obtained from the test data. In order to determine why certain effects were noticed, during the testing, a knowledge of the theory of friction was required. This is presented in Chapter II. The actual results are presented in Chapter IV, and the conclusions drawn from them are in Chapter V.
CHAPTER II

THEORY OF FRICTION

A. Introduction

When two solid bodies are in mutual contact, forces which tend to oppose relative motion of the two bodies are developed at the contact surface. These forces are called frictional forces, and this type of friction is sometimes called external friction to distinguish it from internal friction, e.g., liquid viscosity. Although external friction is certainly the first type that was observed by man, there still does not exist any generally accepted theory about its causes. Accordingly, a brief review of the research which has been done on the subject shall be presented.

B. Early Investigations of Friction

In 1699, a French engineer named Amontons published the result of his experimental investigation on the friction of unlubricated solids. In this paper, he stated the two basic laws of friction:

1. Frictional resistance is proportional to the load.
2. Frictional resistance is independent of the area of the sliding surfaces.

He also concluded that the frictional force was always equal to one-third of the normal load. To explain this result he assumed that the irregularities on the two contact surfaces interlocked, and the relative motion necessitated lifting the load from one interlocking position to another. This process resulted in a loss of energy which was observed as a frictional force. Amontons' conclusions were tested by many investigators, one of them being Euler. Euler published a paper in 1750 in which he agreed with Amontons' conclusion that all surfaces had a frictional coefficient of one-third.

Probably the most systematic work was done by Coulomb in 1785. He examined the influence of a large number of variables on the friction. Coulomb was the first investigator to make a clear distinction between static friction, the force required to start sliding, and kinetic friction, the force required to maintain sliding. In addition, he showed that kinetic friction could be appreciably lower than static friction, and stated that the kinetic friction was independent of the velocity of sliding. This last observation is sometimes quoted as the
third law of friction. Kinetic friction is sometimes called Coulomb friction because of his work.

Further work was carried out by many observers in the first half of the nineteenth century. In 1854, Hirn\(^4\) investigated the difference between lubricated and un lubricated solids. He observed that the effect of velocity, surface area, and load differed in the two cases.

All of these earlier workers agreed with Amontons' assumption that friction was caused by the interlocking of asperites or summits of the surface irregularities. In 1829, Rennie\(^5\) published a paper in which he suggested that a more general theory should take into account the bending and fracture of these asperites. Lubricating action was explained by the assumption that the irregularities in the surfaces were filled by the lubricant, and, at the same time, were made more "slippery" by some unknown action.

C. Recent Developments

Lord Hayleigh questioned these earlier theories when he suggested that the difference between a polished surface and the surface of a fluid might not be very great. His view was confirmed by Biehly\(^6\), who published his classical work on polishing in 1921. Biehly showed that polishing and grinding were two essentially different processes.

At the turn of the nineteenth century, Ewing\(^7\) introduced the view that friction had its origin in surface forces and was due to molecular cohesion between the two solids. This theory received strong support from Sir William Hardy\(^8\) whose work on static friction was published in 1936. He considered that the friction could be explained in terms of the surface fields of the solids. The action of lubricants was explained by assuming that the lubricant caused a reduction of the molecular field of force at the surface of the solid. Hardy further contended that the reduction could be determined quantitatively.

In 1929, Tomlinson\(^9\) attempted to correlate the interaction between the molecules at the surface of two un lubricated solids in sliding contact. His theory was that the friction was due to the energy dissipated when the molecules were forced into each other's atomic fields and were then separated. He assumed that the molecular fields of force were approximately equal for all substances, and the area of molecular contact could be calculated from Hertz's\(^10\) equation for the elastic deformation of spherical surfaces. Although Tomlinson found approximate agreement between experimental and theoretical results, many of his experimental values for friction were in sharp disagreement with those found by other observers.

Another theory which has come into prominence in recent years is that pressure welding or adhesion takes place at the points of contact of the summits of the surface irregularities, and small metallic junctions are thus formed between the two surfaces. When the two surfaces are
moved relative to each other, these metallic junctions must be sheared. To a first approximation, the friction force may be interpreted as the force required for shearing these junctions. A number of observations tend to support this theory. One of them is the seizure of two clean metal surfaces when brought in contact. Another is the transfer of metal which occurs when two surfaces are in rubbing contact. The metal which is transferred from one surface to the other appears to be firmly bonded to the second surface. Among the chief contributers to this theory have been Ernst and Merchant, Holm, and Bowden. Bowden and Leben have reported the observation of a surface temperature flash which would appear to substantiate this theory in the case of metals.

However, welding cannot be considered as a generally present feature in friction, since it certainly does not exist in the case of non-metals. It is for this reason that the molecular attraction theory, first expounded by Ewing, has received added consideration. It is known that molecular forces only extend to a distance of about 10^{-7} centimeter. When two surfaces are brought together, however, the distances between the atoms of each surface may approach to within 2 or 3 Angstrom units. (1 Angstrom unit = 10^{-8} centimeter.) Therefore, the possibility does exist that the molecular forces do play an important part in frictional phenomena.

In 1952, I-Ming Feng suggested a modification of the welding theory. As in the earlier theories, he started at the points of actual contact of the two surfaces. In an earlier work, Bowden had measured the real area of contact by measuring the electrical conductivity of the two surfaces in contact. His results showed that this area varies with load, and in the case of flat steel surfaces it may be less than one 10,000th of the apparent area. Under these conditions, small loads will be sufficient to cause plastic deformation of the asperites. Due to this deformation, Feng concluded that the interface of the contact point is roughened, resulting in a mechanical interlocking effect. He stated that this effect is the primary cause of metal transfer and wear. Near the surface of the interface, the metal becomes severely strain-hardened. When the two surfaces are moved relative to each other, the combination of surface-hardening and mechanical interlock causes the asperites to be sheared off at their weakest points. This shearing is accompanied by a temperature flash.

On the basis of existing experimental data, it is not easy to decide which of these processes (molecular forces, pressure welding, or mechanical interlock) plays an essential part in the phenomena of friction. Feng's theory does explain a number of effects that have gone without explanation so far. However, further experimental work is needed to substantiate his theory.

D. Influence of Variables on Friction

Although the exact mechanism which is responsible for friction is not agreed upon, the influence of many variables on the friction has been observed by many of the workers previously mentioned. Accordingly, the
effects of the more important of these variables shall be discussed.

**Area of Contact and Pressure:** Amontons' first law of friction can be expressed mathematically as:

\[ F = \mu N \]  

(II-1)

where:

- \( F \) = the total frictional force acting between two bodies
- \( N \) = the total force acting normal to the boundary
- \( \mu \) = the proportionality factor, called the coefficient of external friction

The coefficient of friction is dependent only upon the nature of the two contacting surfaces.

Amontons' second law states that the total frictional force is also independent of the apparent area of the surfaces. In Bowden's work on the measurement of the true area of contact, he also observed that this area is almost independent of the size of the surface and is only influenced slightly by the shape and roughness of the surface. With this fact, the reason for this second basic law of friction becomes apparent. Both laws have been verified by many experimenters.

**Velocity of Sliding:** Coulomb stated that the coefficient of kinetic friction was independent of the sliding velocity. Many observers have shown this to be true for relatively low speeds. In his paper published in 1939, Tichvinsky verified Coulomb's conclusion for relative speeds of 60 to 600 cm/sec. However, the rule seems to break down for higher speeds. Stanton performed some experiments in which he showed that the coefficient of kinetic friction decreased quite markedly at the higher relative speeds. The speed at which this occurs seems to vary considerably with the materials considered. As yet, no one has given a satisfactory explanation of this effect.

The kinetic friction is usually much lower than the static friction, and this can be illustrated in the following manner. Suppose a block of weight \( W \) is resting on a horizontal surface. Assume that a horizontal force \( P \) is applied to the block as shown in Fig. 2-1. When \( P \) is zero, the friction force \( F \) is also zero. As \( P \) is increased, \( F \) also increases.

![Diagram](image)

*Fig. 2-1.* Friction force.
and is numerically equal to F. Finally as motion is impending, F reaches a maximum value. This is the value which is used to calculate the coefficient of static friction. When motion takes place, F decreases rapidly to the value of the kinetic frictional resistance. A sketch of the frictional force plotted against the applied force is shown in Fig. 2-2. By dividing the frictional force by the weight of the block W,

![Diagram](image)

**Fig. 2-2. Variation of frictional resistance.**  
(Reference 18)

the friction coefficient is obtained. After motion takes place, the block accelerates since there is a net force applied to it. Therefore, that portion of the curve to the right of the peak value of F represents the variation of friction coefficient with velocity. Note that the decrease from the static to the kinetic frictional coefficient is not instantaneous, but occurs over a finite velocity range.

"Stick-Slip" Effect: If one of the two sliding surfaces is not rigidly mounted, but has a certain degree of elastic freedom, a discontinuity of the motion is observed. Bowden called this effect the "stick-slip" process, and he performed some basic experiments17,18 to study it. The apparatus which he used is shown schematically in Fig. 2-3. If the

![Diagram](image)

**Fig. 2-3. Schematic diagram of apparatus used to illustrate stick-slip effect.**

lower surface is moved at a constant velocity v, an intermittent motion of the upper body will be observed. If this motion is registered on
moving picture film, pictures similar to that shown in Fig. 2-4 are obtained. This picture was drawn from an actual record obtained by Bowden. The ordinate was originally displacement of the upper body. This can be recalibrated in terms of the coefficient of friction as was done in Fig. 2-4. Since the force exerted by the spring is directly proportional to displacement, this force can then be divided by the force normal to the two surfaces to yield the coefficient of friction.

![Diagram](image)

Fig. 2-4. Coefficient of friction vs. time in a stick-slip process for bearing metal on steel.

It can be seen that the motion is intermittent, consisting of periods of steadily increasing pull on the upper body (AB) followed by a sudden decrease in force (BC). Bowden found that during the periods of increasing force the upper slider was moving with a velocity equal to that of the lower surface. This effect can be explained in terms of the static and kinetic friction coefficients. Between A and B the upper body is displaced and the restoring force increases until it reaches a maximum at B. At B, a rapid "slip" occurs, and due to the relative velocity of the two surfaces, the frictional force decreases. When the force has reached the value at C, the two bodies again "stick" and the cycle is repeated. Bowden showed that the value of \( \mu \) at B is the coefficient of static friction \( \mu_s \), while the value at the midpoint of the slip, BC, is the coefficient of kinetic friction \( \mu_k \).

From the preceding discussion it can be seen that this effect will not occur unless two conditions are present:

1. Kinetic friction is lower than static friction
2. One of the surfaces possesses a certain degree of elastic freedom.

The magnitude of the slip will therefore depend on the relative values of \( \mu_s \) and \( \mu_k \) and the rigidity with which the two surfaces are mounted.

**Lubrication:** The effects previously discussed were mainly concerned with un lubricated surfaces. When a small amount of lubricant is introduced between the two contacting surfaces, the frictional characteristics may be changed quite drastically. Just how much the characteristics will be affected depends on the type of lubrication which exists.
The most marked effects will be observed if hydrodynamic or perfect lubrication is present. In this form of lubrication, the two rubbing surfaces are separated by a lubricant layer of appreciable thickness. When this condition exists, the asperites of the two surfaces will no longer be in contact, and the friction resistance is due entirely to the viscosity of the lubricant.

Reynolds developed the classical theory of hydrodynamic lubrication. He showed that an oil film can carry a load if it is wedge shaped, and the load carrying capacity depends on the viscosity of the oil, the relative velocity of the two sliding surfaces between which the wedge is formed, and the geometry of the wedge. The wedge cannot exist if the relative velocity of the surfaces is zero, and consequently, it is not possible for hydrodynamic lubrication to exist between two surfaces at rest. The minimum film thickness for conditions of hydrodynamic lubrication may range from 0.001 to 0.0001 inch depending on load and speed. When this occurs the coefficient of friction may be of the order of 0.001. This often represents a reduction by a factor of several hundred from the values obtained for unlubricated surfaces.

Unfortunately, it is often impossible to obtain hydrodynamic lubrication, particularly at low speeds and high loads. When this occurs, a condition known as thin film lubrication may exist. This is the case when the film thickness becomes so small that, in addition to the viscosity of the lubricant, its physical and chemical interaction with the solid surfaces becomes important. The classical theory of lubrication is no longer applicable, and modifications have to be applied. Any lubricated machine element that operates under starting and stopping conditions must pass through this region of lubrication. In spite of this fact, very little has been done in the way of research on thin film lubrication.

When the two surfaces are separated by lubricant films of molecular dimensions, a condition known as boundary lubrication exists. In this region, the asperites of the two surfaces will penetrate the lubricant layer so that solid to solid contact is present. The frictional force is then dependent on two things: the force required to shear the junctions of the asperites, and the force required to shear the lubricant film. In the region of boundary lubrication it has been found that vegetable oils are best suited for use as lubricants. However, for conditions of hydrodynamic lubrication, mineral oils have been found superior.

E. Frictional Characteristics of Non-Metals

In this thesis, in addition to cast iron piston rings, various types of Teflon and rubber hydraulic seals were also tested. Consequently, a short discussion of the frictional properties of these two non-metals will be presented.

Shooter and Thomas performed some frictional studies on various plastics. Over the range of loads used (1 to 4 kg.) they found that
Amontons' laws were obeyed. Of the four plastics tested (Teflon, polythene, polystyrene, and perspex), Teflon was found to have the best frictional characteristics. The friction of Teflon sliding on Teflon and Teflon sliding on steel was found to be comparable with that of ice on ice. The friction was unaffected by lubricants. Up to a temperature of almost 300°C, the frictional and mechanical properties of Teflon were found to be unchanged.

Both, Driscoll, and Holt investigated the frictional properties of rubber. They also found that Amontons' law is obeyed over a fairly wide range of loads. The friction of rubber on glass and steel surfaces was found to be higher on smooth surfaces than on rough. The friction was found at first to fall with increasing speed. However, at higher velocities, the friction was found to increase. At sliding velocities of about 10 cm/sec they obtained friction coefficients as high as 4 for rubber sliding on steel. Other investigators have found frictional coefficients of the order of one for a variety of solids sliding on rubber. This wide range of experimental values can be explained by the fact that standardization of surface cleanliness is very difficult with such a material.
CHAPTER III

TEST EQUIPMENT AND PROCEDURES

A. Introduction

The requirements which were placed on the test equipment were discussed in Chapter I. The first requirement was that the equipment must separate the effects caused by the piston seal from those caused by the rod seals. Since the tests of the piston seals were performed first, the equipment and procedures employed in these tests will be discussed in the first part of the chapter. Any changes which were made for the rod seal tests will be contained in the last portion of the discussion.

B. Piston Seal Tests: Equipment

Seals Tested: A total of nine different piston seals were tested. They were as follows:

1. Wide "O" ring
2. Narrow "O" ring
3. Wide "O" ring with Teflon back-up rings.
4. Narrow cast iron piston ring
5. Wide cast iron piston ring
6. Three-piece cast iron piston ring
7. Teflon "O" ring
8. Narrow Teflon piston ring
9. Wide Teflon piston ring

The engineering data for these seals is contained in Appendix C. The manufacturer's recommendations for groove dimensions were used for all seals. However, the friction forces measured for the narrow "O" ring were found to be quite high. Therefore a tenth test was performed using this "O" ring in a deeper and narrower groove.

Servo: A positional servo was used in the tests of the piston seals. The block diagram is shown in Fig. 3-1. Ramp voltages were generated (See Appendix B) and fed into the servo. It was thought that a better measurement of the static or break-free friction could be obtained if a gradually increasing signal, rather than a step, was used to drive the servo. It was also anticipated that the ramp inputs would drive the piston at constant velocity, and thus kinetic frictional forces for various velocities could be determined.
Fig. 3-1. Positional servo.

In testing the various types of Teflon and cast iron seals the stick-slip effect, discussed in Chapter II, was exhibited. In response to a ramp input, the piston was seen to move in a series of small steps. (See Fig. B-2, Appendix B.) In order to minimize this effect the basic positional servo was modified somewhat by increasing the feedback sensitivity. The results of this modification will be discussed in Chapter IV.

Actuator: The actuator used in this portion of the testing was a modified spool valve with a one inch diameter cylinder. A schematic diagram of the actuator is shown in Fig. 3-2. The spool was replaced by a piston and a spacer which moved with the piston when a differential pressure was applied. Since the piston had no end rods, the friction which was present was caused solely by the piston and its seal. The inside surface of the cylinder was honed and had an 8μ in. RMS surface finish.

The spacer served two purposes. First, since the actuator cylinder had originally seen service as a spool valve, several fluid ports
were located in the center of the cylinder. The spacer limited the piston travel so that the piston seals would not be caught in these ports. The spacer also served to keep the total piston travel within the range of the positional pick-off device used. The total travel of the piston-spacer combination was 0.389 inches.

Actually, several interchangeable pistons were used in the tests. Each piston had two different size grooves, one at each end, to accommodate the various seals.

**Positional Pick-off Device:** A "Linearsyn" differential transformer was used to obtain positional data from the piston. This device is shown schematically in Fig. 3-3. The voltage across each of the two secondary windings varies linearly with the position of the core. When the core is centered between the two windings, the voltage across each winding will be the same. Since the output voltage is equal to the difference between these two voltages, this position of the core will be the null position.

The core was connected to the piston by means of a brass rod. Both the rod and the core were housed in an aluminum cylinder which opened directly into the actuator. The three windings were contained in a cylindrical package which encircled the rod and core assembly. The device could be nulled for any arbitrary position of the piston by moving the windings relative to the core. A set screw fixed the position of the windings once a null position had been chosen.

**Control Valve:** The control valve was an electrohydraulic valve made by the Moog Valve Company. (Model number V5). It was sufficiently fast so that its dynamics did not affect the dynamics of the servo.

**Pressure Measurement:** In order to measure the forces acting on the piston, some means had to be devised to measure the differential pressures acting across the piston. Since it was anticipated that the pressure levels would be quite low, a relatively sensitive device was called for.
Accordingly, the Components Group of the Tracking Control Project designed the pressure gage shown schematically in Fig. 3-4.

![Diagram](image)

**Fig. 3-4. Pressure gage.**

The spring was soldered securely to the closed end of the brass bellows. The other end of the spring and bellows assembly was fastened to a base. The pressure which existed on one of the piston faces was applied to the inside of the bellows, while that which existed on the other piston face was applied to the outside of the bellows. Since the spring was soldered to the bellows, it was either compressed or elongated until the spring force balanced the pressure force on the bellows. The displacement of the closed end of the bellows was measured by means of a "Linearsyn" as shown in Fig. 3-4.

As was mentioned previously, the piston was to be driven at constant velocity by feeding ramp voltages into the positional servo. It was therefore anticipated that the pressure variations would not occur very rapidly. The step response of the pressure gage was observed, and the response time was on the order of several milliseconds. Thus, the gage could be expected to accurately follow the pressure variations in the system.

During the starting up and shutting down processes some very high differential pressures were built up across the actuator. Since the safe limit for the gage was 50 psi it was found necessary to incorporate a safety shut-down device to protect the gage from these excessive pressures. This device is described in Appendix D. The calibration curve for the gage is shown in Appendix A.

The hydraulic portion of the test equipment is shown in Fig. 3-5. At the left are two solenoid valves (one is almost hidden), which were part of the pressure gage safety device. An oil filter can be seen behind the two valves. Next to the valves is the actuator. The 'Linearsyn' can be seen projecting from the end. The pressure gage is located next to the actuator, and another 'Linearsyn' can be seen attached to the closest end. The control valve can be seen behind the actuator.
Fig. 3-5  Hydraulic portion of piston seal test equipment.
Three pistons and some of the seals tested are located in the right foreground.

**Phasing Networks:** Since a 400 cycle carrier was used throughout the system, it was necessary to adjust the phase of the signals obtained from the two Linearsyns. The phasing networks used are shown in Fig. 3-6. Adjustment of the phasing would, of course, alter the null, and therefore a null adjustment was also included as shown.

![Phasing network diagram](image)

**Fig. 3-6. Phasing network.**

**Velocity Measurement:** By differentiating the positional signal obtained from the actuator "Linearsyn," the piston velocity could be obtained. The system which was used as a differentiator is shown in Fig. 3-7. The positional signal was fed into an amplifier which, in turn, supplied current to a motor. A tachometer was connected to the output shaft of the motor. Thus, the tachometer output voltage was proportional to motor speed. The motor also drove an "Autosyn" through a gear train. The "Autosyn" output was fed, through a phasing network, back into the amplifier. To provide damping, the tachometer output was also fed back into the amplifier.

![Differentiator diagram](image)

**Fig. 3-7. Differentiator.**
Only one phase of the three-phase "Autosyn" stator was used. Therefore, the "Autosyn" output was proportional to the sine of the angle through which the "Autosyn" had rotated. If this rotation is limited to angles less than 10 degrees, the output of the "Autosyn" is then approximately proportional to the angle of motor rotation.

When a constant positional signal is fed into the system, the motor will drive the "Autosyn" until the "Autosyn" output matches the input signal. After this initial transient is over, the motor will not move unless there is a rate of change of positional voltage. If this rate of change of voltage is constant, the motor will be driven at constant speed, and the tachometer output will therefore be constant.

Since the velocities encountered were small, the output of the positional pick-off was multiplied by a factor of twenty before being fed into the differentiator. When this was done, the output of the differentiator was extremely noisy. Accordingly, it became necessary to filter the multiplied signal before using it as an input signal to the differentiator.

The effects which were to be studied were anticipated to be relatively slow. For this reason the lag in the differentiator was not expected to cause any large errors to be introduced in the velocity measurements. As the testing progressed, however, some high frequency effects were observed. Therefore, an accurate picture of the velocity variation could not be obtained when these effects were present.

The filtering of the input signal to the differentiator resulted in an additional lag in the velocity indication. However, this lag was negligible in comparison with the differentiator lag.

Recording of Data: A four-channel Sanborn recorder was used to record position, pressure, velocity and input to the servo. Only the pressure and velocity data were actually evaluated. The recording of position and servo input was solely an aid to testing. Some sample recordings are shown in Appendix B.

C. Piston Seal Tests: Procedure

Break-free Friction: Before the testing was begun, the actuator "Linearsyn" was adjusted so that its null occurred somewhere near the center of the total piston travel. When this was done, the maximum positive and negative values of the servo input were checked to insure that the piston would not be driven into the stops at either end of the cylinder.

The piston was then run at constant velocity and stopped. It was then started in the same direction and stopped again. This was repeated several times, the velocity and direction of travel being held constant. The direction of motion was then reversed and the starting and stopping cycles repeated. Several runs were then taken in which the direction of motion was reversed after each piston motion. By moving the piston
in this manner, the effects of past history of motion could be determined.

**Kinetic Friction:** The piston was run at various velocities in one direction. The direction of motion was then reversed, and the piston was again run at various velocities. As was mentioned previously, the stick-slip effect was observed in the tests of the cast iron and Teflon piston rings. This effect could be minimized by doubling the feedback sensitivity of the servo. Accordingly, two varying velocity runs were performed for these seals, each with a different feedback sensitivity.

**D. Rod Seal Tests: Equipment**

**Seals Tested:** A total of six different types of rod seals were tested. They were as follows:

1. Wide "O" ring
2. Narrow "O" ring
3. Wide "O" ring with Teflon back-up rings
4. Teflon "O" rings
5. Wide Teflon ring
6. Narrow Teflon ring

The engineering data for these seals is contained in Appendix C. No tests were performed with any cast iron rings, since they were not commercially available in the form of rod seals.

**Servo:** The positional servo used in the piston seal tests was modified slightly. First, the feedback sensitivity was increased and this value was maintained throughout the tests. The second modification was the insertion of a potentiometer ahead of the amplifier. The reason for this second modification will be explained in the section on test procedure.

**Actuator:** The actuator was manufactured by the Hydraulic Controls Company from the specifications of the Components Group of the Tracking Control Project. A schematic diagram of the actuator is shown in Fig. 3-8. The rod had a diameter of three-quarters of an inch, and had

![Diagram](image.png)

**Fig. 3-8.** Schematic diagram of actuator used in rod seal tests.
a 4M in RMS surface finish. One of the rod bearings had a lapped seal as indicated in the diagram. The other bearing contained the test seal. A "Linearsyn" was used, as before, to obtain a positional signal from the actuator. Since the friction level of the lapped seal was extremely low, the force required to move the rod was essentially the force necessary to overcome the friction of the test seal.

Additional Changes in Equipment: In the piston seal tests, the operating pressure level of the system was kept constant. However, in the rod seal tests changes in the operating pressure level could be expected to influence the results obtained, since such changes would change the differential pressure existing across the seal. Therefore, a needle valve was placed at each side of the control valve to permit variation of the operating pressure level.

An analysis of the data obtained from the piston seal tests showed that the range of velocities covered was quite low. As was stated previously, the positional signal was multiplied by a factor of twenty before being used as the differentiator input. To enable the differentiator to follow higher velocities, this multiplication factor was reduced to six for the rod seal tests.

The position of the solenoid valves used in the pressure gage safety device was also changed as described in Appendix D. With the new arrangement, the system could be operated with the pressure gage out of the system.

The hydraulic portion of the test equipment is shown in Fig. 3-9. The control valve, with the two needle valves used to vary the operating pressure level, is at the left of the picture. On the right, the pressure gage and two solenoid valves can be seen. The actuator is in the center. Two of the interchangeable rod bearings are seen in the right foreground.

E. Rod Seal Tests: Procedure

Step Response: In order to obtain a measure of the effects of the various test seals on the servo dynamics, the step response of the servo was observed on an oscilloscope. The procedure used was as follows. First, the pressure gage was taken out of the system. Then, the operating pressure level was set at the maximum value (425 psi) by closing both of the by-pass valves. Step inputs were next fed into the servo. The magnitude of the step was adjusted to a value which would cause the fluid flow through the control valve to approach a maximum. When this point of maximum fluid flow is reached, the response time of the servo will increase. With the aid of the oscilloscope, the amplitude of the step was adjusted to a value which was slightly below the value which caused the maximum flow.

The gain of the system was then adjusted until the observed reponse indicated the system had a 0.7 damping ratio. The sign of the input was then reversed, and the process repeated.
Fig. 3-9 Hydraulic portion of rod seal test equipment.
The two gain settings which were thus obtained were averaged, and the average value used for the tests of static and kinetic frictional forces. By following this procedure the damping ratio and the flow through the valve were maintained at essentially constant values for the seals tested.

**Break-free Friction:** It was found that a better indication of the break-free force could be obtained by driving the servo with step inputs. The rod was driven several times in one direction. Then the direction of motion was reversed, and the cycle repeated. Finally, several runs were taken in which the direction of motion was reversed after each rod movement. During these tests, rod position and velocity, differential pressure, and the error signal were recorded on the four-channel Sanborn recorder.

**Kinetic Friction:** Several ramp voltages were fed into the servo to drive the rod at various velocities in one direction. The direction of motion was then reversed, and the process was repeated. The Sanborn recorder was used to record rod position and velocity, differential pressure, and input signal instead of error signal. As in the piston seal tests, only the data obtained on the velocity and pressure was actually evaluated.

In the tests of the three Teflon seals, the servo was unstable when the pressure gauge was placed in the system. For this reason, only the ramp responses were recorded for these seals.

**Recording of Step Responses:** Photographs were taken of the rod position and the differential pressure in response to a step input. This was done for the wide "O" ring and the narrow Teflon rings. The system was operated at maximum operating pressure level during these tests. Since the pressures encountered in this test were above the maximum safe pressure to which the differential pressure gauge could be subjected, two Statham gages were used to obtain a pressure signal for these photographs. Also, these gages had a much lower compliance such that they would not influence the dynamics of the test system.
Fig. 4-1. Force vs velocity curves for piston seals
CHAPTER IV

RESULTS

A. Piston Seals

A tabulation of the measured average static frictional forces is contained in Table 4-1. The sign convention used is purely arbitrary. The column headings refer to the direction of piston motion. For

<table>
<thead>
<tr>
<th>Type of Seal</th>
<th>Friction Force in Pounds</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>+ after +</td>
</tr>
<tr>
<td>Wide &quot;O&quot; ring</td>
<td></td>
</tr>
<tr>
<td>(a) Plain</td>
<td>+ 1.3</td>
</tr>
<tr>
<td>(b) With Teflon</td>
<td>+ 1.7</td>
</tr>
<tr>
<td>back-up rings</td>
<td></td>
</tr>
<tr>
<td>Narrow &quot;O&quot; ring</td>
<td></td>
</tr>
<tr>
<td>(a) Recommended groove</td>
<td>+ 5.9</td>
</tr>
<tr>
<td>(b) Deep groove</td>
<td>+ 6.6</td>
</tr>
<tr>
<td>Wide cast iron ring</td>
<td>+ 2.7</td>
</tr>
<tr>
<td>Narrow cast iron ring</td>
<td>+ 3.2</td>
</tr>
<tr>
<td>Wide Teflon ring</td>
<td>+ 0.4</td>
</tr>
<tr>
<td>Narrow Teflon ring</td>
<td>+ 1.1</td>
</tr>
</tbody>
</table>

* Two-element rings with "O" ring back-up. (See Appendix C)

Example: + after - means the piston was moved in a positive direction after being moved in a negative direction.

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The three-piece cast iron ring is not included in the table because the pressure required to move it was above 50 psi. Since the safe limit for the pressure gage was 50 psi, no values could be measured. The Teflon "O" ring is also not included in this table. As was mentioned previously, the servo was unstable with this seal installed. However, the oscillating force was found to vary between approximately 47 pounds. This should be a good indication of the magnitude of the static frictional forces.

The force vs. velocity curves are plotted in Fig. 4-1. For the reasons cited above, the three-piece cast iron ring and the Teflon "O" ring are not included. The narrow "O" ring is also omitted since the force levels were quite high, and its inclusion would have resulted in a crowding of the other curves. Since there was no great difference between the force levels for the wide and narrow cast iron rings, only the results for the wide ring are shown to avoid crowding of the curves.

Static Friction: The various "O" rings exhibited a marked hysteresis effect. This can be explained by considering what happens to the "O" ring during piston motion. First, the "O" ring will have a tendency to roll in the groove during piston motion. As a result of this rolling, a small amount of energy will be stored in the "O" ring. If the piston is subsequently moved in the direction opposite to that which caused the roll, the energy stored in the "O" ring will produce a force which will aid the piston break-out. On the other hand, if the piston is moved successively in the same direction, the force produced by the "O" ring will hinder the piston motion; consequently, the break-free force will be dependent on the past history of piston motion.

Another effect which may be exhibited is the extrusion of the "O" ring into the clearance gap away from the pressure as shown in Fig. 4-2(a).

![Fig. 4-2. Extrusion of "O" ring.](image)

This extrusion will cause an increase in frictional force, in addition to shortening the "O" ring life. To prevent this effect, back-up rings are sometimes used as shown in Fig. 4-2(b).

When using the Teflon back-up rings in conjunction with the wide "O" rings, a slight increase in frictional force was measured. This would seem to indicate that extrusion did not occur in the case of the wide "O"
ring. This contention is supported by the following figures published in the Precision Rubber Products Corporation Handbook: for an "O" ring with a Shore durometer hardness of 70, the maximum recommended clearance on a side is 0.010 inch for differential pressures up to 250 psi. The clearance space between the piston and cylinder of the test actuator was well below this figure, and extrusion would therefore not be expected to occur.

In all cases, the narrow rings had a higher friction level than the corresponding wide rings. This was especially noticeable in the case of the wide and narrow "O" rings. At first glance, these results would appear to be the reverse of the expected behavior. The normal force acting between the seal and the cylinder wall is dependent on the pressure exerted by the seal and the area of contact. Since the wider seals have a larger area of contact, the normal force, and hence the friction force, should be greater for these seals. A closer observation will reveal why this is not the case.

The pressure exerted by the seal on the cylinder wall is dependent upon two things: the differential pressure across the seal, and the initial squeeze on the seal. Since the differential pressures never reached very high values, the pressure is mainly dependent on the initial squeeze. Seal squeeze is defined in Fig. 4-3.

![Diagram of Seal and Piston](image)

\[
\% \text{ Squeeze} = \frac{B}{A} \times 100\%
\]

**Fig. 4-3. Seal squeeze.**

In order to obtain a good seal, the narrow "O" ring must be subjected to a greater squeeze than the wide "O" ring. This is due to the fact that the narrow "O" ring has a smaller cross-sectional area, and hence, a smaller contact area. An examination of the groove dimensions (Appendix C) will show that the wide "O" ring had about a 12% squeeze while the narrow "O" ring was subjected to about a 19% squeeze. These are the values recommended by the manufacturer.

Because of the high friction levels obtained with the narrow "O" ring, another test was performed. The squeeze was reduced to 10% by increasing the depth of the groove. This did not have the expected
effect, for the friction level increased slightly. A look at Fig. 4-4 will explain why this effect was observed. In both cases, since the differential pressures are approximately equal, the deformed "O" ring will take the same shape. However, in the case of the deep groove, the "O" ring will have a clearance space at the bottom of the groove. Therefore, the pressure will act upon the undersurface of the seal, and this will result in a larger normal force between the contacting surface of the seal and cylinder.

In the cases of the wide and narrow two-element Teflon and cast iron seals, the preceding discussion is also applicable. A wide "O" ring was used behind the wide seals, and a narrow "O" ring behind the narrow seals. This was done to provide the necessary resilience of the seals. The same percentage squeeze (approximately 12% for the wide "O" ring and 19% for the narrow one) was maintained for the "O" rings used in conjunction with these piston rings.

Kinetic Friction: The forces measured for motion in the negative direction are higher than those obtained for positive motion. It was unfortunate that the authors' time schedules did not permit a more thorough analysis of the test results before the testing was completed. If time had permitted, an investigation of this directional effect could have been performed. On the basis of the tests which were performed, it is impossible to determine the cause of this variation of running friction with direction.

In spite of the above limitation some important results can be gathered from the curves in Fig. 4-1. The curves for the wide "O" ring (plain and with Teflon back-up rings) all show an increase in force with increasing velocity. On the other hand, the curves for the Teflon and cast iron rings exhibit an initial decrease in force. As was discussed in Chapter III, the stick-slip effect is only observed for materials whose static friction is greater than the kinetic friction. The stick-slip effect was not observed in any of the "O" ring tests, and the reason for
this is readily seen from the plotted results as described above.

It was also observed that the stick-slip effect was more pronounced for the cast-iron seals. The reason for this is again brought out by the curves. The curves for the cast iron rings show a much sharper initial decrease in friction force than those for the Teflon rings.

Another observation was that the stick-slip effect was more pronounced in the direction of positive motion. The curves for the Teflon and cast iron rings in Fig. 4-1 reveal that there is a sharper initial decrease of friction force for motion in the positive direction; consequently, the experimental observations appear to be explained.

As was mentioned previously, a thorough investigation of the causes of these directional effects could not be performed because of time limitations. However, in one test, the actuator was reversed with respect to the valve. When this was done, the directional effects of the stick-slip phenomenon were decreased somewhat. It was still slightly more pronounced for motion in the positive direction.

The intensity of the stick-slip effect is also dependent on the degree of elastic freedom existing between the two contacting surfaces. In the discussion in Chapter III, a spring was used to represent this elastic freedom. In a hydraulic servo, the gain of the system is analogous to the spring constant of the simple apparatus described in Chapter III. An increase in the gain will effectively increase the amount of elastic restraint, and the intensity of the stick-slip effect should be decreased. This result was verified in the course of the testing.

With the exception of the wide Teflon rings, all of the seals exhibited an increase of force with velocity at higher velocities. This would indicate that some viscous frictional forces were also being measured. For some reason, the wide Teflon rings showed a steadily decreasing force vs. velocity characteristic. Perhaps if the velocity had been increased further, these curves would have also exhibited a rise in friction force.

3. Rod Seals

A tabulation of the measured average static frictional forces is contained in Table 4-2. As in the case of the piston seals, the sign convention used is purely arbitrary, and the column headings refer to direction of rod motion.
TABLE 4-2

Static frictional forces for rod seals.

<table>
<thead>
<tr>
<th>Type of Seal</th>
<th>Friction Force in Pounds</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>+ after +</td>
</tr>
<tr>
<td></td>
<td>- after -</td>
</tr>
<tr>
<td></td>
<td>+ after -</td>
</tr>
<tr>
<td></td>
<td>- after +</td>
</tr>
<tr>
<td>Wide &quot;O&quot; ring</td>
<td></td>
</tr>
<tr>
<td>(a) Plain</td>
<td>+ 9.2</td>
</tr>
<tr>
<td>(b) With Teflon back-up rings</td>
<td>+ 8.9</td>
</tr>
<tr>
<td></td>
<td>- 7.7</td>
</tr>
<tr>
<td></td>
<td>+ 3.5</td>
</tr>
<tr>
<td></td>
<td>- 2.3</td>
</tr>
<tr>
<td>Narrow &quot;O&quot; ring (Deep groove)</td>
<td>+ 5.9</td>
</tr>
<tr>
<td></td>
<td>- 6.3</td>
</tr>
<tr>
<td></td>
<td>+ 2.1</td>
</tr>
<tr>
<td></td>
<td>+ 1.4</td>
</tr>
<tr>
<td>Leakage Occurred</td>
<td></td>
</tr>
<tr>
<td>Wide Teflon ring*</td>
<td></td>
</tr>
<tr>
<td>(a) Pressure level = 425 psi</td>
<td>+ 9.5</td>
</tr>
<tr>
<td></td>
<td>- 8.6</td>
</tr>
<tr>
<td>(b) Pressure level = 200 psi</td>
<td>+ 8.6</td>
</tr>
<tr>
<td></td>
<td>- 7.4</td>
</tr>
<tr>
<td>Narrow Teflon ring*</td>
<td>+ 4.5</td>
</tr>
<tr>
<td></td>
<td>- 4.6</td>
</tr>
<tr>
<td>Teflon &quot;O&quot; ring</td>
<td>- 10.1</td>
</tr>
<tr>
<td></td>
<td>- 10.5</td>
</tr>
<tr>
<td></td>
<td>+ 9.5</td>
</tr>
<tr>
<td></td>
<td>- 8.4</td>
</tr>
</tbody>
</table>

* Two-element rings with "O" ring back-up. (See Appendix C)

The force vs. velocity curves are plotted in Fig. 4-5. The narrow "O" ring and the Teflon "O" ring were not plotted since both of these seals leaked. Therefore, the values obtained for these seals would not give a true picture of the friction levels when compared with the other seals.

Static Friction: The hysteresis effect observed for the "O" rings in the piston seal tests was again exhibited, to a lesser degree, for the rod seals. Due to the higher differential pressures existing across the rod seals, the rolling of the seal was probably limited. A comparison of the results for the plain "O" ring and the same "O" ring with Teflon back-up rings indicates that some extrusion of the seal into the clearance space did occur, since the use of the Teflon rings caused a decrease in friction level.

The Teflon "O" ring was also seen to exhibit a very slight hysteresis. The effect is much less marked in this case since the Teflon is much less resilient than the rubber; consequently, the amount of twisting is less in the case of the teflon seal. Because of this lack of resiliency, a good seal could not be obtained with the Teflon "O" ring, and some leakage occurred.

The narrow "O" ring was only tested with a deep groove. For this case, a large amount of leakage occurred, and the operating pressure could only be brought up to 275 psi. Therefore, the values obtained for
Fig. 4-5. Force vs. velocity curves for rod seals
this seal were greatly influenced by the lubrication which was present.

In these tests, the wide Teflon rings exhibited a higher frictional level than the corresponding narrow seals. As was explained in the discussion of the piston seals, the pressure exerted by the seal on the surface against which it rubs will depend on the initial squeeze and the differential pressure across the seal. In the case of the piston seals, the squeeze was the more important factor. However, since the differential pressure was about 425 psi for the rod seals, this factor is more important for these seals. Consequently, the increased squeeze on the "O" ring used behind the narrow seals does not influence the value of the normal force as much as in the case of the piston seals.

A reduction of the operating pressure level caused a decrease in friction force which is to be expected.

The circumferential length of the contacting surface was different for the piston and rod seals. By dividing the average magnitude of the break-free force by the length of the contacting surface, the results tabulated in Table 4-3 are obtained. The hysteresis effects have been neglected, i.e., only those values obtained from successive movements in the same direction were used.

**TABLE 4-3**

Comparison of the friction force per inch for the piston and rod seals.

<table>
<thead>
<tr>
<th>Type of seal</th>
<th>Friction Force per Inch of Seal Length</th>
<th>Relative force ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Piston</td>
<td>Rod</td>
</tr>
<tr>
<td>Wide &quot;O&quot; ring</td>
<td>0.41</td>
<td>3.6</td>
</tr>
<tr>
<td>(a) Plain</td>
<td>0.51</td>
<td>3.4</td>
</tr>
<tr>
<td>(b) Teflon back-up</td>
<td>0.14</td>
<td>3.8</td>
</tr>
<tr>
<td>Teflon ring</td>
<td>0.32</td>
<td>1.9</td>
</tr>
<tr>
<td>Teflon &quot;O&quot; ring</td>
<td>2.2*</td>
<td>4.4</td>
</tr>
</tbody>
</table>

* Approximate value because of instability of servo.

The value of the relative force ratio of the rod seal force per inch to the piston seal force per inch for the wide Teflon ring is much higher than for the other seals. This would seem to indicate that some other factor entered into the tests of these seals. The Teflon "O" ring has a lower value for the relative force ratio than the other seals. This is due to the leakage which occurred in the rod seal tests, and thus introduced lubrication effects.
To the best of the authors' knowledge, the two-element Teflon rings used in conjunction with the rubber "O" ring back-up have never before been used as seals. This unprecedented use was the suggestion of K.D. Garnjost of the Instrumentation Laboratory.

Kinetic Friction: Some interesting directional effects were also observed in the rod seal tests. A look at Fig. 4-5 will reveal that all of the force-velocity characteristics exhibit an increase in force, at the higher velocities, for motion in the negative direction. Direction of motion was defined as shown in Fig. 4-6. When the motion was in the negative direction, a thin film of hydraulic fluid would adhere to the rod surface. This would provide lubrication between the test seal and the rod surface, and would also give rise to some viscous effects. The oil film would be at least partially wiped off of the rod surface by the test seal, and the lubricating action would not be as pronounced for motion in the positive direction. Thus, the viscous effects would only be noticeable in the negative direction, and would cause the force-velocity characteristics to rise with increasing velocity.

Evidently the wiping action was not as effective for the "O" ring with the Teflon back-up rings, since the friction force rises for motion in either direction. This decrease in the effectiveness of the wiping action can be attributed to the increase in the depth of groove and consequent decrease in initial "O" ring squeeze. (Appendix C). This was done to insure clearance for the Teflon rings.

The stick-slip effect could not be observed for any of the Teflon seals since the servo became unstable with the pressure gage in the system. This instability can be attributed to the fact that the pressure gage had a large compliance.

Unlike the piston seal tests, the stick-slip effect was observed for the wide "O" ring when it was moved in a positive direction. The reason for this is apparent from Fig. 4-5. The force-velocity curve shows a decrease of force for increased velocity in the positive direction.
For motion in the negative direction, a slight initial decrease in force occurs, but this is followed by a rapid increase in force. The stick-slip effect would therefore not be expected to be as noticeable for negative motion.

**Step Response:** As described in Chapter III, a potentiometer was inserted ahead of the amplifier in the servo used for the rod seal tests. The pressure gage was removed from the system and the two Statham gages were installed in its place; thus, the test servo had the lowest compliance attainable with this test set-up. The settings of this potentiometer which gave a damping ratio of 0.7 are tabulated in Table 4-4. These values give an indication of the relative loop gain. The operating pressure level was 425 psi, and the response time was 0.02 seconds. The narrow "O" ring is not included since the leakage which occurred did not permit the pressure level to reach 425 psi.

**TABLE 4-4**

Potentiometer settings (relative loop gain) for a damping ratio of 0.7.

<table>
<thead>
<tr>
<th>Type of Seal</th>
<th>Direction of Motion</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>+</td>
<td>-</td>
</tr>
<tr>
<td>Wide &quot;O&quot; ring</td>
<td>0.463</td>
<td>0.396</td>
</tr>
<tr>
<td>(a) Plain</td>
<td>0.520</td>
<td>0.420</td>
</tr>
<tr>
<td>(b) Teflon back-up rings</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Wide Teflon ring</td>
<td>0.460</td>
<td>0.396</td>
</tr>
<tr>
<td>Narrow Teflon ring</td>
<td>0.450</td>
<td>0.382</td>
</tr>
<tr>
<td>Teflon &quot;O&quot; ring</td>
<td>0.489</td>
<td>0.402</td>
</tr>
</tbody>
</table>

It can be seen that the potentiometer settings were dependent on the direction of motion. A test was run for the wide Teflon rings with the actuator reversed with respect to the control valve. When this was done, it became necessary to reverse the sign of the error signal for stability. Thus, a positive value of the input signal would still cause a positive motion of the rod. (The sign convention for the direction of rod motion was unchanged.) However, the sign of the control valve input current was reversed; consequently, the spool of the control valve moved in the opposite direction for a positive input signal.

The subsequent testing of the reversed actuator showed a similar directional effect, but the higher gain setting occurred for the negative direction of motion. This would eliminate the actuator from consideration as a cause for this effect.
Another test was run using a different control valve. The same directional effects were observed; consequently, these effects cannot be attributed to any asymmetry of the control valve.

The average values for the gain setting were used in the static and kinetic frictional tests. Only a small difference in gain setting is observed for the various seals. This difference could very well be accounted for by the observer's accuracy in determining the exact setting for which a damping ratio of 0.7 was attained. Certainly the variations in relative gain settings do not indicate that any one seal is preferable from a dynamic response consideration.

Some sample step responses are shown in Appendix B. It can readily be seen that the pressure variation during the steady-state condition is more rapid for the narrow Teflon rings than for the wide "O" ring. In both cases, however, the pressure goes through positive as well as negative values during steady state conditions. This variation causes the rod to oscillate slightly about the zero error position.

The use of a 0.7 damping ratio resulted in an initial overshoot of the rod. When this occurs, an error signal is built up; however, this error signal is reversed in sign from the initial error signal caused by the application of the step input. Thus, as the rod passes through the zero error position, the pressure would be expected to decrease. Finally, the pressure would be applied to the opposite end of the rod in order to bring it back to the steady state position.

The above effect is exhibited very clearly by the narrow Teflon rings. The pressure is seen to lag behind the error signal. This is due to the fact that a finite amount of fluid must flow through the control valve before the pressure is built up.

The response for the "O" ring exhibits a very erratic variation of pressure. As a result, the pressure correction for the overshoot is not clearly shown in the sample response. This behavior is no doubt due to the hysteresis effect which occurs for the "O" rings.

As was mentioned previously, the variations in the relative loop gain do not indicate a superiority of any one of the seals tested. However, it should be noted that a high friction level will affect the response of the servo to small error signals. The reason for this is as follows: the build up of a pressure difference across the actuator requires a finite amount of fluid flow through the control valve. The rate of fluid flow will be small for a small error signal; thus, if a large pressure level is required for piston break-out, the response time of the servo will be increased. Consequently, the type of seal chosen will affect the dynamic response of the servo, although this was not clearly shown by the experimental results.
CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

A. Techniques of Friction Measurement

A positional servo afforded the best control of piston motion. The best technique in measuring the break-free forces was to feed small amplitude steps into the servo. In order to determine the force-velocity characteristics of the seals, ramp voltages proved to be the most desirable servo input.

B. Piston Seals

The two-element Teflon piston rings exhibited the lowest static and kinetic frictional levels. The break-free force for the wide Teflon rings was approximately half that for the narrow rings. This difference can be attributed to the fact that a wider, more resilient "O" ring was used in conjunction with the wide Teflon rings. Although these rings exhibit a slight stick-slip effect, this can be minimized by decreasing the compliance of the servo.

The break-free friction was also quite low for the wide "O" ring and the same ring with Teflon back-up rings. However, the force required to move the piston is dependent on the past history of motion, and this characteristic is highly undesirable.

The two-element cast iron rings showed reasonably low static frictional levels. There was a sharp initial decrease in friction with velocity, and this, coupled with a rising force-velocity characteristic for the wide "O" ring, led to comparable kinetic frictional forces for the cast iron and wide "O" ring seals. Again, the stick-slip effect exhibited by the cast iron seals can be minimized by decreasing the compliance of the servo.

The narrow "O" ring had a very high friction level. It also exhibited the hysteresis effect observed in the wide "O" ring tests. A decrease in initial squeeze of the seal caused a slight increase in friction.

The Teflon "O" ring also had a very high friction level, and in addition caused instability of the servo. Furthermore, the groove dimensions must be maintained to very close tolerances, and the piston must
be made in two parts to permit installation of the seal.

The friction level of the three-piece cast iron ring was extremely high, and was beyond the range of the pressure gage.

Consequently, from the standpoint of low friction levels, the wide Teflon piston rings appear to be the best seals tested.

C. Rod Seals

The narrow Teflon rings exhibited the lowest static and kinetic frictional levels for the rod seals tested. The wide Teflon rings, wide "O" rings (plain and with Teflon back-up rings), and the Teflon "O" ring had frictional levels which were about twice that of the narrow Teflon rings.

The narrow "O" ring was tested only with less than the recommended initial squeeze. A large amount of leakage occurred. The hysteresis effects were again observed, to a lesser degree than in the piston seals, for all the "O" rings.

Some leakage was also observed for the Teflon "O" ring. The close tolerances which must be maintained on the groove and rod dimensions make this a difficult seal to install.

Care must be taken in designing a servo which utilizes Teflon rod seals. If the servo loop is not tight enough, i.e., if there is too large a capacity in parallel with the control valve and actuator, the servo may be unstable.

All of the seals affected the dynamic response of the test servo to approximately the same degree; but it should be noted that a high level of static friction will strongly affect the angle resolving qualities of the servo. Consequently, the narrow Teflon seals gave the best results from a purely frictional standpoint. However, care must be taken in designing a servo using these seals since they may cause instability.

D. Recommendations

Many of the frictional effects observed showed a marked dependence on the direction of motion of the piston or rod. Some of these effects could be explained in the light of the experimental work which was done, but a few could not be accounted for. Therefore, further investigations of these directional effects are considered necessary.

The experimental results brought out the dependence of the force-velocity characteristics on the kind of seal tested, i.e., rod or piston seal. However, there were some indications that the servo design also influenced the kinetic friction characteristics; consequently, an investigation to determine the effects of varying the damping ratio and the response time on the force-velocity characteristics of the seals would also be desirable.

Finally, a more thorough investigation of the servo design necessary to prevent instability when using Teflon seals is also needed.
APPENDIX A

CALIBRATION OF EQUIPMENT

1. Differentiator

The differentiator used to obtain velocity measurements was calibrated in the following manner: a ramp voltage was used as the input to the differentiator. This input and the differentiator output were recorded on a Sanborn recorder. The record thus obtained for the differentiator sensitivity used in the piston seal tests is shown in Fig. A-1. With this procedure, the sensitivity of the differentiator, $S_D$, could be determined in volts per volt per second.

$$ S_D[\dot{e}; e] = \frac{e_D(\text{out})}{e_D(\text{in})} = 8.20 \text{ volts/volt/sec.} $$

In order to determine the differentiator sensitivity in terms of volts per inch per second, the sensitivity of the actuator "Linearsyn" had to be measured. This is done by dividing the change in voltage output, $\Delta e_L$, by the corresponding displacement of the core, $\Delta X_c$.

$$ S_L[\dot{d}; e] = \frac{\Delta e_L}{\Delta X_c} = 3.73 \text{ volts/inch} $$

The sensitivity of the "Linearsyn" for a velocity input and a voltage rate output is numerically equal to the above sensitivity.

$$ S_L[\dot{d}; e] = 3.73 \text{ volts/sec.} $$

Thus, the sensitivity of the differentiator in volts per inch per second is:

$$ S_D[\dot{d}; e] = S_D[\dot{e}; e] \cdot S_L[\dot{d}; e] = 31.4 \text{ volts/inch/sec.} $$

The sensitivity of the differentiator was decreased for the rod seal tests. The procedure described above was again used to determine the new sensitivity. The values of the sensitivities were as follows:

$$ S_D[e; e] = 2.50 \text{ volts/volt/sec.} $$
Fig. A - 1
CALIBRATION OF VELOCITY MEASURING INSTRUMENT
Fig. A-2. Differential Pressure Gage Calibration
\[ S_L[\dot{d}; e] = 5.68 \text{ volts/sec.} \]
\[ S_D[\dot{d}; e] = 14.2 \text{ volts/inch/sec.} \]

In order to check the accuracy of the velocity measurements, the recorded positional signal was calibrated for several of the rod seal kinetic friction tests. As was mentioned in Chapter III, only the pressure and velocity signals were usually calibrated. In the kinetic friction tests, the positional signal was approximately a ramp function, i.e., the piston moved at constant velocity; therefore, the velocity of the piston could be determined directly from the positional signal. Allowing for errors in reading the records, it is estimated that the velocity measurements were, in most cases, accurate to within 5% for reasonably slow piston movements. When the input signal was a step, the velocity measurements were very much in error due to the time lag of the differentiator.

2. Pressure Gage

The calibration of the pressure gage was accomplished with the system shown in Fig. A-3.

![Diagram](image)

Fig. A-3. System for calibration of pressure gage.

The pressure applied to one side of the pressure gage could be varied through the use of the two valves. The other side of the pressure gage was open to the atmosphere; thus, the calibrating gage measured the differential pressure existing across the pressure gage. The calibrating gage was a United States gage, number 10857. It had a range of 0 to 60 psi.

The calibration curve for the pressure gage is shown in Fig. A-2. The range of values from +45 psi to -45 psi was traversed several times in the calibration run. This was done to determine the repeatability of the readings. The maximum error observed was about 2\% of the full-scale reading.
APPENDIX B

EVALUATION OF TEST DATA

1. Piston Seal Tests

Some sample Sanborn recordings are shown in Figs. B-1, B-2, B-3, B-4.

A portion of a break-free friction run for the wide "0" ring is shown in Fig. B-1. The velocity signal was used to determine when piston motion first occurred; consequently, the value of the pressure at this point would determine the break-out force. Since these observed effects were relatively slow, the differentiator lag would be expected to introduce a negligible error into the results. The forces measured in this way had a certain spread; therefore, the values obtained from the records were averaged to obtain the results tabulated in Chapter IV.

The dependence of break-free force on past history of motion is clearly illustrated in Fig. B-1. This record is typical of all the "0" ring tests.

A sample static friction run for the wide cast iron piston rings is shown in Fig. B-2. The first indication of velocity can be seen to coincide with the occurrence of a pressure peak; therefore the value of this pressure peak was taken as a measure of the break-free force. This procedure was followed for all of the two-element cast iron and Teflon rings.

The stick-slip effect is clearly illustrated by Fig. B-2. As mentioned in Chapter II, Bowden showed that the average value of the varying force was equal to the kinetic friction force. Hence, the average value of pressure during piston motion was taken as a measure of the kinetic friction force. The velocity was also averaged during the stick-slip to obtain the numerical values used in Chapter IV.

The instability of the servo, which occurred with the installation of the Teflon "0" ring, is illustrated in Figs. B-3 and B-4. The record in Fig. B-3 is for the same gain settings used to obtain the experimental results for the other seals. In Fig. B-4, the loop gain was increased and the oscillations are seen to decrease in magnitude.
Fig. B-1
BREAK FREE FRICTION TEST FOR "O" RING NO. AN62278-15
Fig. 8-2
BREAK-FREE FRICTION TEST OF WIDE CAST IRON RINGS.
Fig. B-3
BREAK-FREE FRICTION TEST OF TEFON "O" RING
(SENSITIVITY OF FEEDBACK LOOP = 1)
Fig. 8-4
VARYING VELOCITY TEST OF TEFLOM "O" RING
(SENSITIVITY OF FEEDBACK LOOP = 2; INPUT TO AMPLIFIER MULTIPLIED BY 5)
2. **Rod Seal Tests**

A different technique was used to measure the break-free friction in the rod seal tests. As was mentioned in Chapter III, small amplitude step voltages were used to drive the servo. During these tests, the velocity indication could not be used to indicate when piston motion first occurred, since the differentiator lag would introduce an appreciable error.

Instead of recording the servo input, the error signal was recorded. A line was drawn from the leading edge of the step in the position signal to the leading edge of the step in the error signal. The intersection of this line with the pressure signal was the break-free pressure. Since the servo response time was about 0.02 seconds, a negligible time lag existed between the error signal and the rod position.

The techniques used to obtain force-velocity measurements were the same as those used in the piston seal tests.

Some sample step responses are shown in Figs. B-5, B-6, B-7, and B-8. These plots were drawn from photographs which were taken of the responses as seen on an oscilloscope. A comparison of the responses to small and large amplitude step voltages reveals the increase in response time which occurs when the step amplitude is increased beyond the capabilities of the control valve.

The reader is referred to Chapter IV for a more thorough discussion of the step response.
Fig. B-5. System response to a small step input using a wide "O" ring pressure seal.
Fig. B-6. System response to a large step input using a wide "0" ring pressure seal
Fig. 3-7. System response to a small step input using narrow Teflon rings as pressure seals.
Fig. B-8. System response to a large step input using narrow Teflon rings as pressure seals.
APPENDIX C

ENGINEERING DATA FOR SEALS TESTED

In the specifications for the piston and rod seals, the groove dimensions will be given in terms of the code letters shown in Fig. C-1.

![Groove dimensions](image)

A = Groove length
B = Gland width

Fig. C-1. Groove dimensions.

For the "O" rings (both rubber and Teflon), the dimensions shown in Fig. C-2 will be given.

![O ring dimensions](image)

C = I.D. of "O" ring
D = Cross-sectional diameter of "O" ring.

Fig. C-2. "O" ring dimensions.
The dimensions which will be given for the two-element Teflon and cast-iron rings are shown in Fig. C-3.

![Diagram of Teflon and cast-iron rings](image)

- **A** = Groove length
- **B** = Gland depth
- **C** = Thickness of two-element rings

**Fig. C-3.** Two-element ring dimensions.

1. **Piston Seals**

   **Wide "O" Ring:** A number AN-6227-B-15 "O" ring was used. The compound used in this ring has a Shore durometer hardness of 70. The dimensions of the wide "O" ring and its groove are as follows:

   \[
   A = 0.155 \pm 0.005 \\
   B = 0.123 \pm 0.002 \\
   C = 0.734 \pm 0.006 \\
   D = 0.139 \pm 0.004
   \]

   These groove dimensions are those recommended by the manufacturer for no ring roll.

   **Wide "O" Ring with Teflon Back-up Rings:** A number AN-6227-B-15 "O" ring was again used. A section through the seal is shown in Fig. C-4. Two number AN-6246 spiral Teflon back-up rings were used with the "O"

   \[
   B = \text{Gland width} \\
   C = \text{Specified width of Teflon back-up ring} \\
   F = \text{Thickness of two back-up rings}
   \]

**Fig. C-4.** "O" ring with Teflon back-up rings.
ring. The pertinent dimensions are as follows:

\[ \begin{align*}
A &= 0.261 \pm 0.002 \\
B &= 0.123 \pm 0.002 \\
C &= 0.734 \pm 0.006 \\
D &= 0.139 \pm 0.004 \\
F &= 0.104 \pm 0.002 \text{ (measured on Teflon rings)} \\
&= 0.100 \text{ to } 0.116 \text{ (specified)} \\
G &= 0.120 \text{ to } 0.123
\end{align*} \]

Note that the gland width was increased slightly to prevent any radial squeeze of the back-up rings.

**Narrow "O" Ring:** A Precision Rubber Products Corporation catalogue number 914-8 "O" ring was used. The dimensions are as follows:

\[ \begin{align*}
A &= 0.090 \pm 0.005 \\
B &= 0.057 \pm 0.002 \\
C &= 0.864 \pm 0.006 \\
D &= 0.070 \pm 0.003
\end{align*} \]

The groove dimensions are those recommended by the manufacturer for no ring roll. When a high friction level was observed, the groove dimensions were changed to the following:

\[ \begin{align*}
A &= 0.078 \pm 0.003 \\
B &= 0.063 \pm 0.002
\end{align*} \]

These dimensions are for a decreased squeeze and no ring roll.

**Wide Cast Iron Piston Ring:** This was a two-element ring similar to that shown in Fig. C-3. The dimensions are as follows:

\[ \begin{align*}
A &= 0.115 \pm 0.005 \\
B &= 0.140 \pm 0.002 \\
E &= 0.045 \text{ to } 0.050
\end{align*} \]
A number AN-6227-B-13 "O" ring was used to provide the back-up seal. The dimensions of the "O" ring are:

\[ C = 0.674 \pm 0.005 \]
\[ D = 0.103 \pm 0.003 \]

**Narrow Cast Iron Piston Ring:** The dimensions of this two element ring are as follows:

\[ A = 0.090 \pm 0.005 \]
\[ B = 0.107 \pm 0.002 \]
\[ E = 0.045 \text{ to } 0.050 \]

A FRP catalogue number 914-6 "O" ring was used with this seal. Its dimensions are:

\[ C = 0.739 \pm 0.005 \]
\[ D = 0.070 \pm 0.003 \]

**Three Piece Cast Iron Ring:** This three-element ring was essentially a wide two-element cast iron ring which used a third cast iron ring instead of an "O" ring back-up. The groove dimensions are:

\[ A = 0.1250 \pm 0.0005 \]
\[ B = 0.120 \pm 0.010 \]

**Teflon Piston Rings:** The dimensions for the side and narrow two-element Teflon rings were the same as those for the wide and narrow cast iron piston rings. In this application, the squeeze of the back-up "O" ring provides the radial sealing force since the Teflon rings do not have the necessary inherent stiffness of cast iron rings.

**Teflon "O" Ring:** A MS 29513 - 210 "O" ring was used. This had the same specified dimensions as the wide rubber "O" ring number AN-6227-B-15. The measured cross-sectional diameter of a sample lot of these Teflon "O" rings was 0.1395 \( \pm 0.0010 \). Unlike the rubber "O" rings, the Teflon "O" rings cannot be stretched; consequently, the piston had to be designed in two pieces to permit installation. (See Fig. C-5)
Fig. C-5. Piston design for Teflon "O" ring.

The dimensions of the groove are as follows:

\[ A = 0.141 \pm 0.001 \]
\[ B = 0.133 \pm 0.001 \]

2. **Rod Seals**

The same letter code will be used to specify the pertinent dimensions of the rod seals.

**Wide "O" Ring:** A number AN-6227-B-15 "O" ring was again used. The seal and groove dimensions for no ring roll have been given previously in Section 1.

**Wide "O" Ring With Teflon Back-up Rings:** A number AN-6227-B-15 "O" ring was used with two number MS-28782 spiral Teflon back-up rings. The groove dimensions are as follows:

\[ A = 0.255 \pm 0.005 \]
\[ B = 0.125 \pm 0.002 \]

**Narrow "O" Ring:** A FRP catalogue number 914-6 "O" ring was used. The dimensions associated with this seal and its groove are as follows:

\[ A = 0.078 \pm 0.005 \]
\[ B = 0.062 \pm 0.002 \]
\[ C = 0.739 \pm 0.005 \]
\[ D = 0.070 \pm 0.003 \]
These dimensions are for less than the recommended minimum squeeze and no ring roll.

**Wide Teflon Ring:** This two-element ring was similar, in principle, to the two-element piston ring. However, with the rod seals, the inner surface of the seal was in contact with the rod; consequently, the "O" ring used in conjunction with this seal fitted over the outside circumference of the ring. The pertinent dimensions for this seal are as follows:

\[
\begin{align*}
A &= \ 0.155 \pm 0.005 \\
B &= \ 0.153 \pm 0.002 \\
E &= \ 0.030 \pm 0.002
\end{align*}
\]

A number AN-6227-B-16 "O" ring was used with this seal. Its dimensions are:

\[
\begin{align*}
C &= \ 0.796 \pm 0.006 \\
D &= \ 0.139 \pm 0.004
\end{align*}
\]

**Narrow Teflon Ring:** The groove and seal dimensions for this two-element ring are:

\[
\begin{align*}
A &= \ 0.090 \pm 0.005 \\
B &= \ 0.093 \pm 0.002 \\
E &= \ 0.030 \pm 0.002
\end{align*}
\]

A catalogue number 914-7 "O" ring was used with this seal. The "O" ring dimensions are as follows:

\[
\begin{align*}
C &= \ 0.801 \pm 0.006 \\
D &= \ 0.070 \pm 0.003
\end{align*}
\]

**Teflon "O" Ring:** A NS 29513-210 "O" ring was also used as a rod seal. The dimensions of the groove are as follows:

\[
\begin{align*}
A &= \ 0.141 \pm 0.002 \\
B &= \ 0.133 \pm 0.001
\end{align*}
\]

It was necessary to design a two-piece bearing to permit installation of the seal.
APPENDIX D

PROBLEMS ENCOUNTERED IN THE PRESSURE MEASUREMENTS

Originally, the Components Group of the Tracking Control Project designed a pressure gage which utilized a copper diaphragm. The differential pressure which was to be measured acted on opposite faces of the diaphragm. The resulting deflection of the diaphragm was measured with a "Linearsyn" differential transformer.

However, a calibration of this gage revealed the existence of a large hysteresis effect, and the readings could not be duplicated with any accuracy. An inspection of the gage showed that the edges of the diaphragm had not been clamped tightly enough. Due to difficulties encountered in the heat treatment of the diaphragm, it was also somewhat warped. The combination of these two effects had caused the diaphragm to buckle at the center. This buckling had, in turn, given rise to an "oil-can" effect, i.e., the diaphragm would snap suddenly from one side to the other with very slight pressure changes.

Through improved techniques of heat treating and clamping of the edges, the hysteresis effect was reduced and the oil-can effect was eliminated. However, the errors involved were still too large, and this gage was finally abandoned. It is unfortunate that this gage was not usable since the compliance would have been lower than for the spring-bellows gage which was used. As was mentioned in Chapter IV, this compliance led to an instability of the servo in the Teflon rod seal tests.

In spite of the fact that great pains were taken to avoid exceeding the safe limit of 50 psi for the spring-bellows gage, it was found to be quite easy to build up high pressure transients during starting and stopping. Several "fool-proof" techniques were devised to avoid damaging the bellows. The gage was by-passed during these starting and stopping pressure transients. In the course of the testing, however, several large pressure build-ups occurred, and consequently, several pressure gages were damaged. It was fortunate that the calibration curves for all the gages were the same. This was due to the fact that only the bellows had to be replaced, and since the deflection of the bellows-spring combination was dependent almost entirely on the spring stiffness, the calibration remained unchanged.
Fig. D-1. Safety shut-down relay for pressure gage.
The safety device which was finally used was designed by the Components Group. It utilized the relay shown in Fig. D-1. The relay was opened when the "Linearsyn" on the pressure gage had an output of approximately one volt. This corresponded to a differential pressure of about 40 to 45 psi. The relay, in turn, energized two solenoid valves.

The following arrangement was used in the piston seal tests. A normally-closed solenoid valve was placed in the pressure line which supplied the control valve. Another normally-open solenoid valve was used to by-pass the pressure gage. Thus, when the relay was activated, the pressure to the system was cut off, and the pressure gage was by-passed.

For the rod seal tests, the normally-open valve was again used to by-pass the pressure gage. However, the normally-closed valve was used to isolate the gage and permit operation of the system without the gage.

This method of isolating the pressure gage proved very helpful in the initial application of pressure to the system; for if large pressure transients occurred, the system would still operate without danger to the pressure gage.
APPENDIX E

GENERATION OF SERVO INPUTS

The ramp voltages that were used as inputs to the positional servo were generated by integrating step voltages of various magnitudes. The system which was used for this purpose is shown in Fig. E-1. This

![Diagram of servo system]

Fig. E-1. Integrator.

piece of equipment had been used in previous test work carried out in the hydraulic laboratory at M.I.T.

For a given input voltage, the amplifier will produce a constant current which will, in turn, drive the motor at constant speed. Therefore, the tachometer will generate a constant voltage. This voltage is fed back to provide damping. The motor is also connected through a gear train, to the wiper of a potentiometer. Thus, the output of the potentiometer is proportional to the angle through which the motor has rotated, and if the motor is driven at constant speed, this output will be a ramp function. The caging switch can be used to drive the potentiometer back to its null position.

A step generator was also available in the hydraulics lab. Both
the operating level and the amplitude of the step could be varied.

The ramp function was used to obtain force-velocity data (i.e., kinetic friction); the step function was used to obtain static friction data.
APPENDIX F

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