DESIGN AND OPERATION OF A HIGH-SPEED TEST FACILITY
FOR ROTATING FACE SEALS

By

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DESIGN AND OPERATION OF A HIGH-SPEED TEST FACILITY

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CHAPTER I

INTRODUCTION

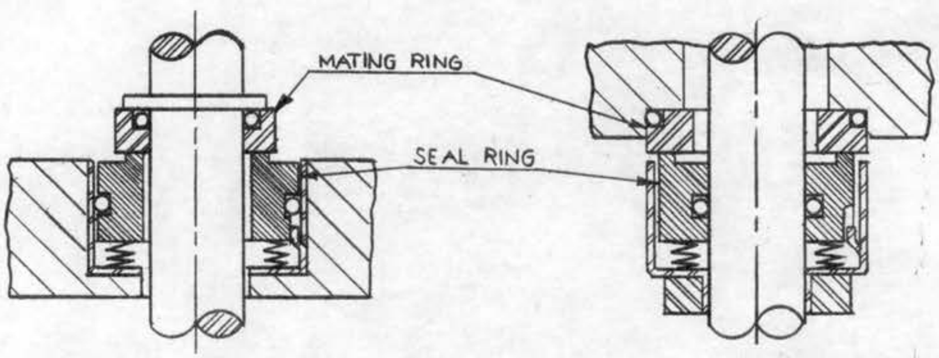
The demands made on present and proposed weapon systems or automated machines force the available power systems into a role of continually "catching up" with other areas of our technology. Fluid power, as a power system, is finding itself in this position of "catching up" in many instances, especially in applications involving high temperatures and high pressures.

In an effort to keep abreast of progress made in other areas, attention has been given to sealing problems of hydraulics and pneumatics. The approaches to this general problem have been and are sundry. Elaborate solutions, almost entirely experimental, have been proposed. The lack of a usable theoretical analysis and the unavailability of packing materials with properties suitable for extreme conditions have increased the difficulty of perfecting solutions to sealing problems.

Many of the problems incurred by high temperatures and high pressures are applicable to both static and dynamic sealing applications. Since the more severe demands are made on dynamic applications, the static sealing applications in general will not be further considered.

A dynamic fluid seal may be defined as a device that is used to prevent the passage of fluids or gases between parts of a mechanism having relative motion. One approach or method of effecting a dynamic fluid seal in a rotating application is to force a rotating ring against a stationary ring. The applied force required to effect the seal would be slightly greater than the axial force resulting from the pressure of the fluid being sealed. This type of sealing device is generally called a Mechanical Seal or Face Seal.

A Face Seal may be described by the particular way it is designed to effect a seal and the way it is installed. The two general classifications are rotary and stationary. The seal in Figure 1b is called a rotary seal since the seal ring rotates with the shaft (1).[1] The stationary seal is seen in Figure 1a. In figure 1b, the mating ring is fixed to the housing, and the sliding ring rotates with the shaft.



(a) Stationary Seal (b) Rotary Seal

Figure 1. Face Seal Types

Another classification of Face Seals is used along with the previously mentioned classification. Referring now to Figure 1, if fluid under pressure tried to flow across the face between the mating and seal ring in a direction which was radially inward toward the center of the shaft, such a seal would be classified as an Internal Seal. If the fluid tried to flow radially outward from the center of

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1. Note: ( ) refers to Selected Bibliography.

the shaft, the seal would be classified as an External Seal. The most desirable sealing device with regard to effecting a seal and to proper seal cooling would be an Internal Stationary Seal.

The design of Mechanical Seals, to the present, has been largely a trial and error process because a satisfactory theoretical analysis has not been made. When the several factors affecting a seal design are considered, it is not difficult to understand why a satisfactory theoretical analysis is not available. From this, one can see that experimental testing is presently the only approach to improving or modifying existing seal designs for new applications.

In the past few years, several attempts have been made through experimental testing to determine relationships which would allow the correlation of test data for any given situation. The sealing needs of high performance turbojet engines and of missile auxiliary power components have caused an increase in experimental testing.

In the realm of high-speed high-temperature shaft seals for turbojet engines, Taschenberg (2) designed and fabricated a seal test machine which would approach the simulation of actual seal running conditions. The test machine utilized a 50-horsepower variable speed gear motor with a speed range from 3000 rpm to 11,000 rpm to supply the estimated required power. The maximum surface speed of 30,000 feet per minute was achieved with the use of a belt and pulley arrangement. The facility incorporated an air system which would deliver air at 200 psi and at a temperature of 1000°F. An oil system was provided for bearing lubrication and for use in the test cavity in place of air. The design of this test machine was such that two identical sealing devices had to be installed in the machine in order to obtain test data for seal refinement.

Brenza et al (3) used a very simple test rig to obtain data at rubbing speeds of only about one-twentieth that which exists in an operating seal. From this data conclusions were drawn; however, the conclusions were only qualified estimates with respect to the actual seal surface speeds.

All of the so-called seal test machines are not necessarily designed such that a standard sealing device can be utilized for experimental testing. Johnson et al (4) designed a machine which would provide a superfinished surface at temperatures above 700°F moving at surface speeds of 10,000 feet per minute. Cylindrical rider specimens of selected material were held in contact with the moving surface, and pertinent data was recorded. This type of machine properly could be called a wear test machine. Brenza et al (3) used this type of machine for a preliminary study, and Pennington et al (5) discussed a high temperature wear test machine which was built to test the dry wear characteristics of materials at temperatures up to 1200°F.

Sibley et al (6) developed a high-speed high-temperature rubbing-wear apparatus which was more elaborate than the relatively simple wear test devices previously mentioned. This apparatus permitted experiments to be conducted at rubbing speeds of about 12,000 feet per minute and unit load pressures up to 20 psi. The seal specimens were subjected to high-temperature (1000°F to 1800°F) oxidizing or reducing gases. This apparatus utilized a cantilever arm fitted with strain gages for recording friction torque. The contacting surfaces were loaded by the use of pneumatic bellows.

Barnes and Ryder (7), in a progress report on rubbing seals, discussed a seal test rig which would operate in the general speed range of interest and would provide temperatures and face loads above those which most face seals would run. The test rig was designed to provide a fast, rough screening test for materials that might be used for rubbing seals. The actual surface speed which this rig would attain was 8850 feet per minute (15,000 rpm) while the minimum rotative speed was 4500 rpm. The primary drive mechanism was a 7.5 horsepower dynamometer with a gear speed-increaser. The seal test cavity was pressurized with dry or damp air. The pressure could be varied from zero to 100 psi. The net unit load on the seal faces could be varied from zero to 100 psi. Since this test rig was lacking in refinements and instrumentation, the results obtained were not precise but were indicative that many improvements are possible in existing sealing devices.

In view of the fact that experimental testing must be utilized, an endeavor was made to design a facility for testing high-speed rotating Mechanical Seals which are either currently in use or proposed replacements. The facility was constructed and put into operation in the Mechanical Engineering Laboratory of Oklahoma State University.

In order that one might understand this approach to the determination of the relationships in sealing problems, a detailed description of the design is presented as well as the operating procedures for the facility.

CHAPTER II

DESIGN PARAMETERS OF THE TEST FACILITY

The objectives of this endeavor were to design, construct, and put into operation a facility for testing high-speed rotating Face Seals. The facility was intended to be adaptable to a variety of Mechanical Seals which may come into use in aircraft applications. Specifically, the facility would accomodate stationary Face Seals of either the internal or external types. An effort was made to allow for future changes which would give the facility a greater range in speed as well as seal adaptability.

The specific requirements which the machine met were as follows:

1. Test shaft speed range - 2500 rpm to 25,000 rpm
2. Test section fluid pressure range - zero to 500 psi
3. Test section flow rate - varying from zero to 1.5 gpm
4. Fluid temperature range to 500°F
5. Loading System pressure range up to 1000 psi.

Commercially available components were to be utilized in every instance where this was deemed feasible.

CHAPTER III

FACILITY DESIGN AND COMPONENT SELECTION

In general, the primary components of a seal testing machine would consist of a support for the stationary portion of a sealing device and a power source to drive the rotating portion of a sealing device. Other components of a test facility could be considered secondary. A detailed description of the design of each component is given.

Seal Test Section and Support

The Seal Test Section consisted of that portion of the Test Facility in which a sealing device could be mounted for testing. The Seal Test Section included a rotating member and a limited movement member.

The rotating member of the Seal Test Section was called the Sliding Ring Adapter. The Adapter was mounted directly on the high speed shaft. As can be seen in Figure 2, the Sliding Ring Adapter has two parts. The section which was mounted on the shaft need not be removed to change the sliding ring. The section which actually drives the sliding ring contains an O-ring groove which allows the installation of an O-ring. The O-ring was needed to effect a static seal between the sliding ring and the adapter.

The non-rotating or relatively stationary member of the Seal Test Section was called the Seal Test Head. This member provided the mounting for the stationary portion of a Face Seal, as can be seen in

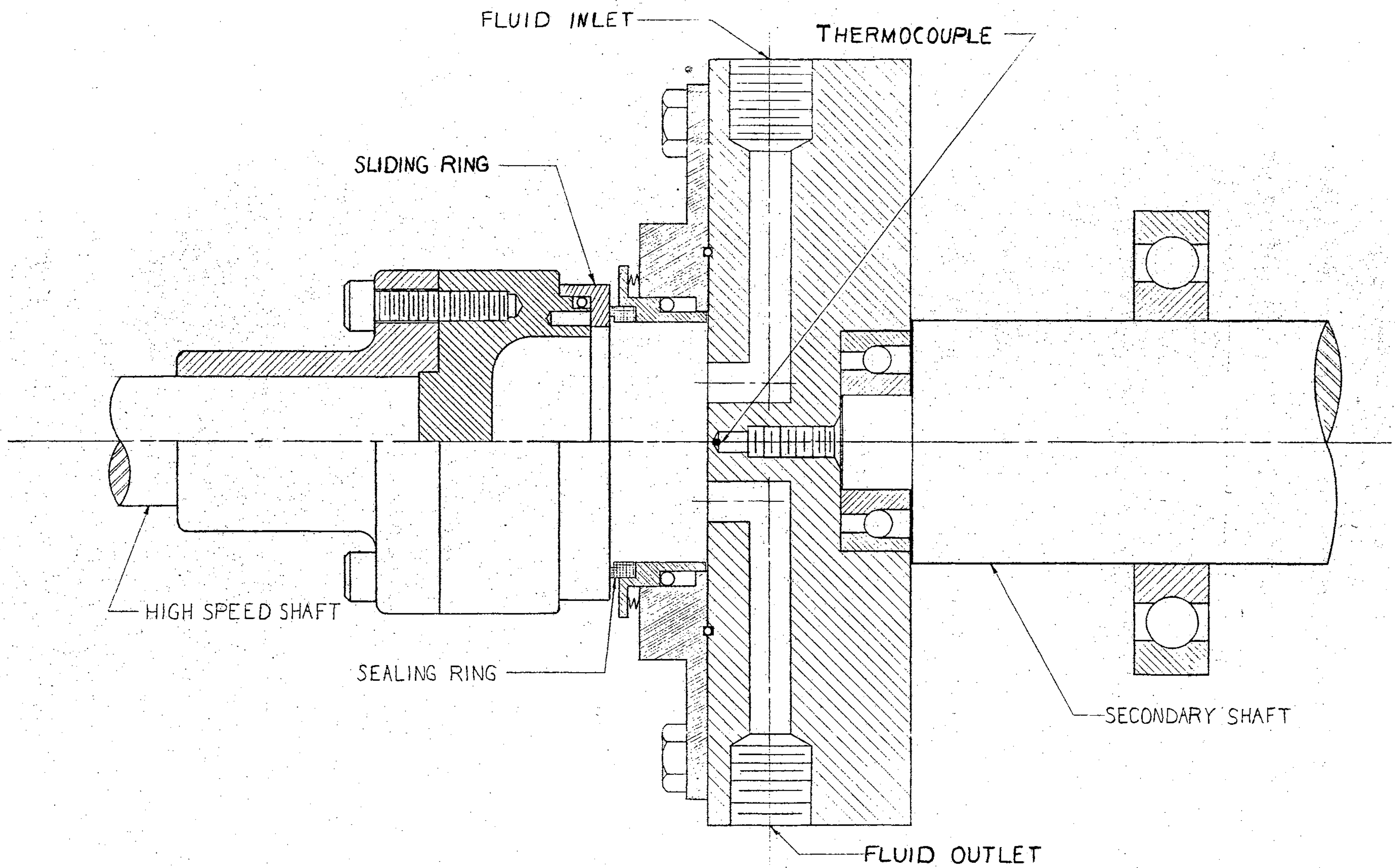

Figure 2. Layout of the Seal Test Section

Figure 2. The Seal Test Head had several functions other than providing a mounting for the sealing device. Since one of the parameters in seal testing is the friction horsepower, a method was needed to determine the power dissipated. Since the rotative speed would be an independent test variable, the torque required to maintain the stationary portion of the seal in equilibrium would provide a means for determining the friction horsepower. A simple means of measuring the friction torque was used. A torque arm with a weight pan at a known arm length was mounted on the back side of the Seal Test Head as seen in Figure 3.

The Seal Test Head was also used to provide passageways for the fluid used during a test. The fluid was to be both a lubricant and a heat transfer medium, as well as to provide a medium through which the Test Section could be pressurized.

In order that the torque measurements be as accurate as desired, a ball thrust bearing was mounted in the Seal Test Head. The thrust bearing was mounted on the Secondary Shaft (see Figure 2), which was in turn supported by a pair of readily available pillow block roller bearings. The pillow block bearings were not entirely necessary but were economically feasible and therefore desirable.

In anticipation of any leakage which might occur past the sealing surfaces of the Mechanical Seal during a test, a drip pan and shield arrangement was provided. The shield was made of ¼-inch transparent plastic and could be removed or replaced with ease. The drip pan and shield can be seen in Figure 4.

After the Test Section was designed, consideration was given to a

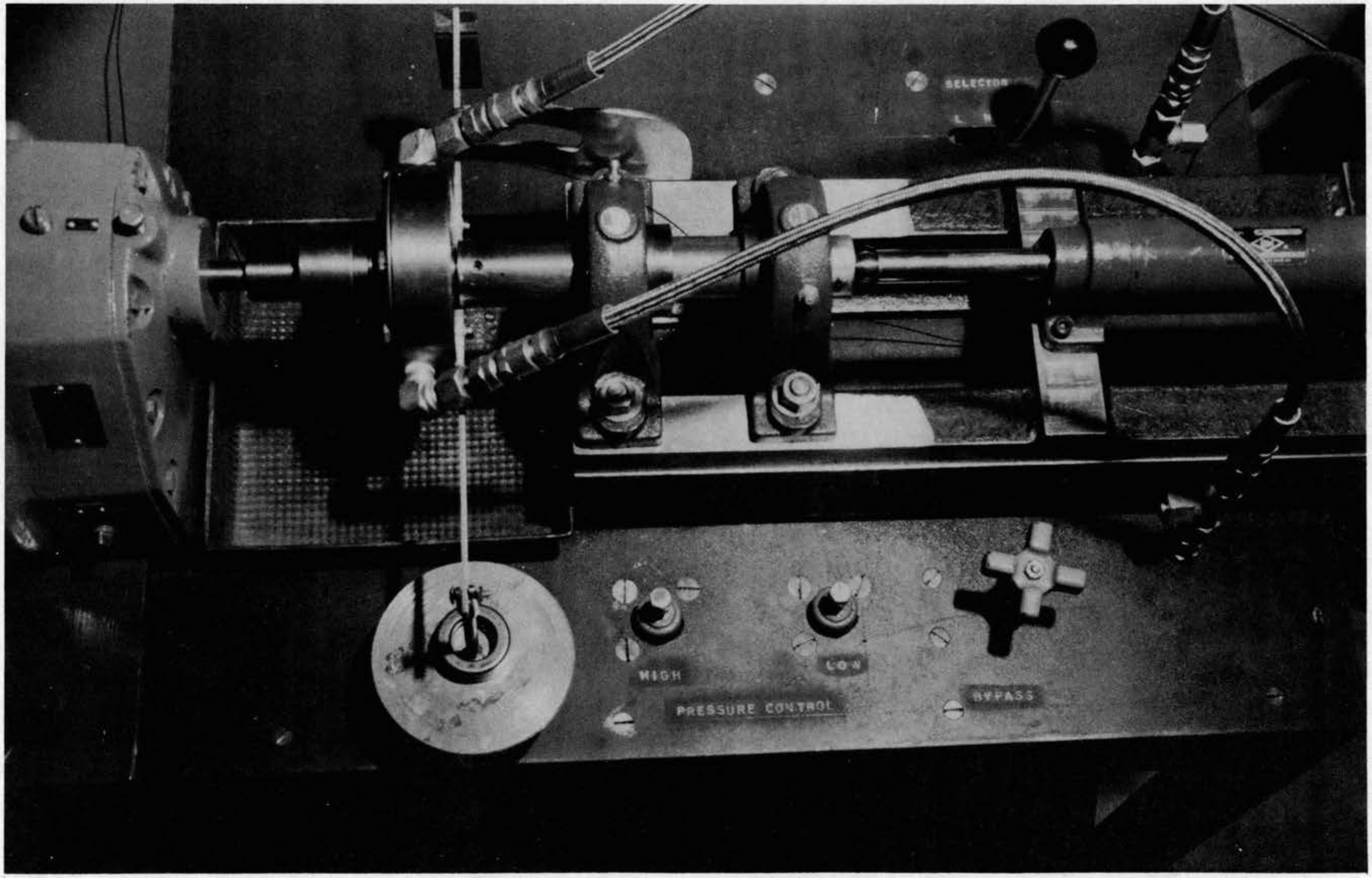

Figure 3. Plan View of the Test Stand

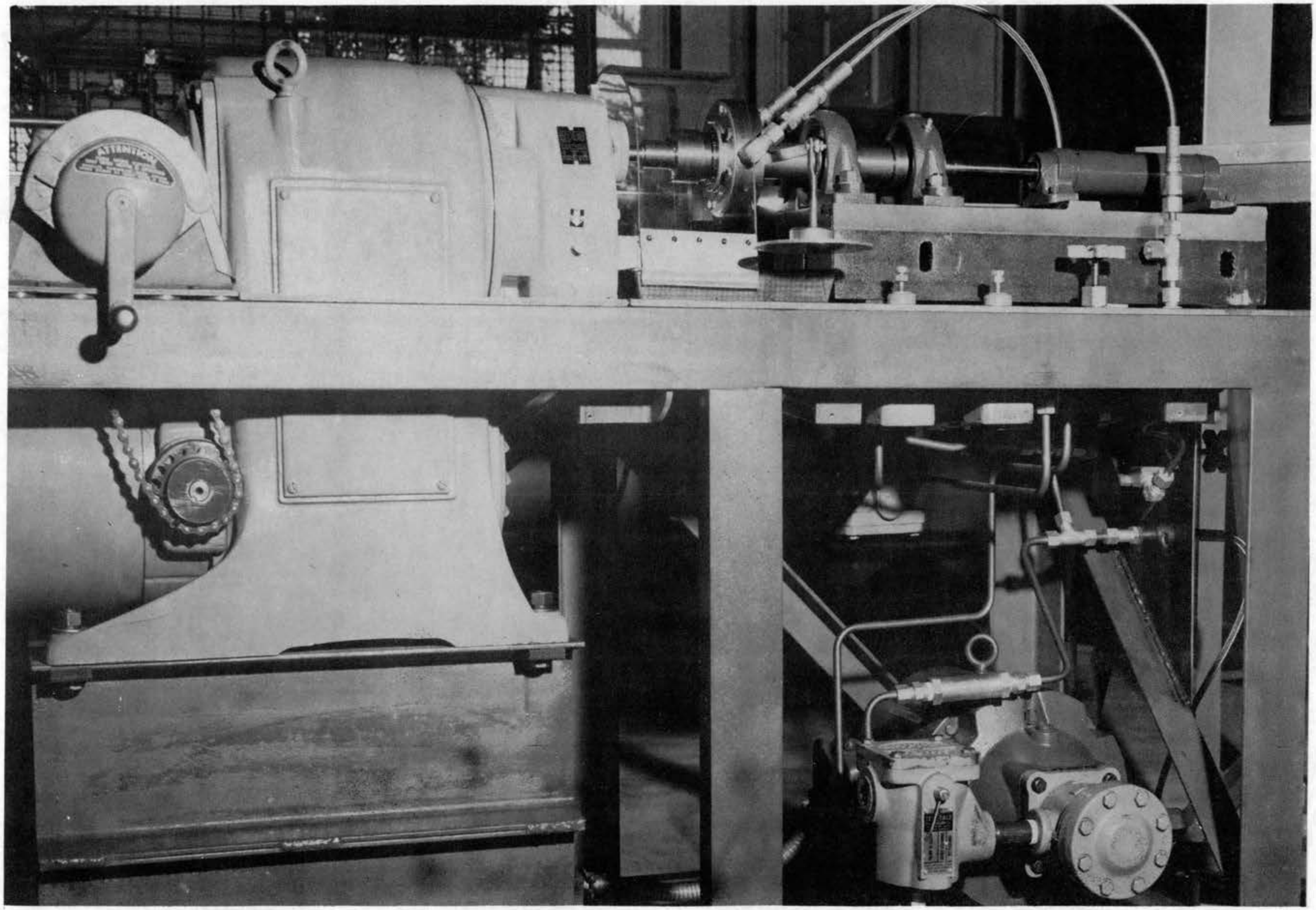

Figure 4. Elevation View of the Test Stand

suitable support or framework which would house the necessary equipment and provide a level rigid setting for the Seal Test Section. Layouts were made to establish the relative location of each component of the Test Facility. The layouts were used to establish the exact dimensions which were used in the fabrication of the stand. Since a rigid frame was considered a necessity, weight was not given any consideration in the final design of the Test Stand. The frame was fabricated of structural steel with all joints welded. The finished Test Stand was placed upon felt pads to prevent any tendency of it to move laterally.

Drive Mechanism

There were various ways of obtaining the speeds and power necessary for the success of the Test Facility. Consideration was given to steam and gas turbines, various mechanical mechanisms, hydraulic systems, and electrical motors with speed control, either by electronic or mechanical means. The drive chosen was a constant-speed electric motor in conjunction with a mechanically variable speed drive mechanism including a speed increaser. The output shaft at the speed increaser had a minimum speed of approximately 2500 rpm and a maximum speed of approximately 25,000 rpm. A slight variation of the maximum and minimum speeds occurred due to the stop settings. The maximum allowable thrust loads which could be placed on the output shaft of the Varidrive were 60 pounds in and 40 pounds out. These allowable loads narrowed the range of operations from that originally considered as desirable. The drive unit was mounted such that the Sliding Ring

Adapter would be in its operating position when mounted on the drive unit output shaft, as can be seen in Figure 4.

The power requirements for the drive unit were estimated from the horsepower equation derived by Lewis (8). This formula for the power absorbed by a Face Seal was based on an assumed constant pressure distribution between the Sliding and Mating rings, and is as follows:

$$HP = \frac{(5.289 \times 10^{-6})\ n\ f\ L\ (D^3 - d^3)}{(D^2 - d^2)}$$

where D = outside seal diameter, inches

d = inside seal diameter, inches

n = angular velocity, rpm

L = total face load, pounds

f = coefficient of friction

The total face load on a seal depends upon the design of the seal as well as the pressure to be sealed. Lewis pointed out that a controversy existed over the relationship of the face load and operating pressure. In view of this, the largest face load possible under a given set of conditions was desirable for estimating the friction horsepower. The largest total face load would be approximated by the following equation:

$$L = \pi/4\ (D^2 - d^2)p + K$$

where

p = fluid pressure

K = spring force

After the total face load, L, had been estimated, calculations were made for the friction horsepower. The most severe operating condition would probably be at the maximum rotating speed and at the maximum fluid pressure. The maximum speed used in the calculation was 30,000 rpm and the maximum pressure used was 500 psi. The estimated friction horsepower of a 2-inch inside diameter Face Seal, operating under the above pressure and speed, would be approximately 6.25 horsepower, assuming the coefficient of friction to be 0.01.

Since the conditions under which the 6.25 horsepower would be developed were extremely severe, a five-horsepower unit which could be overloaded to 7.5 horsepower was chosen for the drive mechanism of the Test Facility.

Hydraulic System

Since a Face Seal normally operates under some pressure, a certain amount of heat would be generated. For long life, the heat generated should be kept to a minimum; therefore lubrication would be desirable. A hydraulic system was designed for the Test Facility which would facilitate the simulation of various operating conditions involving temperatures and pressures.

Primarily, the system was to provide for a circulation of fluid. The circulation was desirable for two reasons, the first of which was to control the temperature in the Seal Test Head. If there were no fluid circulation, the temperature of the fluid in the test cavity would continue to rise until some equilibrium condition was reached, and therefore, some maximum temperature would become evident. Even though this would be a normal condition for many applications of Face Seals, a deviation was allowed here for the sake of testing. The circulation of fluids at different temperatures would provide for many different maximum temperatures of the Seal Test Head. The second reason for circulation was to provide a continuing supply of fluid for lubrication.

The first problem encountered in the design of the system was the selection of the rate of fluid flow through the Test Section. The first estimation was five gpm. This estimation was first selected in order that estimations be made regarding the heat transfer requirements of the system. The system tubing size chosen was 3/8 of an inch outside diameter for ease of fabrication and maintenance. Since the resistance to a flow of five gpm through 3/8-inch tubing incurred a large pressure drop, a compromise resulted in the reduction of the originally selected flow rate. The final design provided for a maximum rate of flow of 1.5 gpm, and a bleed valve to allow a variation of the flow through the test cavity which would result in a lower cavity pressure. The pump selected was included in a commercially available motor-pump unit. The pump was a constant displacement balanced vane-type pump. Figure 4 shows the location of the motor-pump in the Test Stand.

Another problem was the question regarding the pressurization of the test cavity. Since no part of the system would be required to do any work, the only pressure rise would be the result of line resistance. It was necessary, therefore, to use a loading device. The first proposed design called for the use of a counterbalance valve. Further investigation revealed that a directly operated variable setting relief valve could be utilized. Since a relief valve having the desired pressure range of from 25 psi to 500 psi was not commercially available, two relief valves and a manually operated detent open-center selector valve were utilized. As can be seen in Figure 5, the selector valve provided one position in which the only pressure incurred was line resistance, one position for low pressure (25 to 125 psi) operation, and one position for high pressure (125 to 500 psi) operation. The pressure control valves were mounted such that the setting could be easily varied. Physically, they were mounted with just the control screw protruding through the table top near the front edge of the Test Stand. The selector valve was mounted such that only the control handle protruded through the table top. The selector valve was mounted near the back side of the Test Stand. These controls can be seen in Figure 3.

The system was provided with two filters, as seen in Figure 5. The filter which immediately follows the hydraulic pump was to filter all the fluid passing through the pump. The filter through which all fluid passing through the test cavity must flow was to protect the selector valve and the two relief valves from particles which may enter the system at the test cavity, particularly bits of carbon if a carbon ring should fail during a test. The filters selected for use had a maximum operating pressure of 750 psi. They were of such a design which allowed internal bypassing of fluid in the event that the filtering elements became clogged. Another very desirable feature of the filters was the indicator which would show in a glance the approximate state of cleanliness of the filtering element. The two filters were both mounted for ease of maintenance. One was mounted using the hydraulic pump for support, and the other was mounted using the selector valve for support.

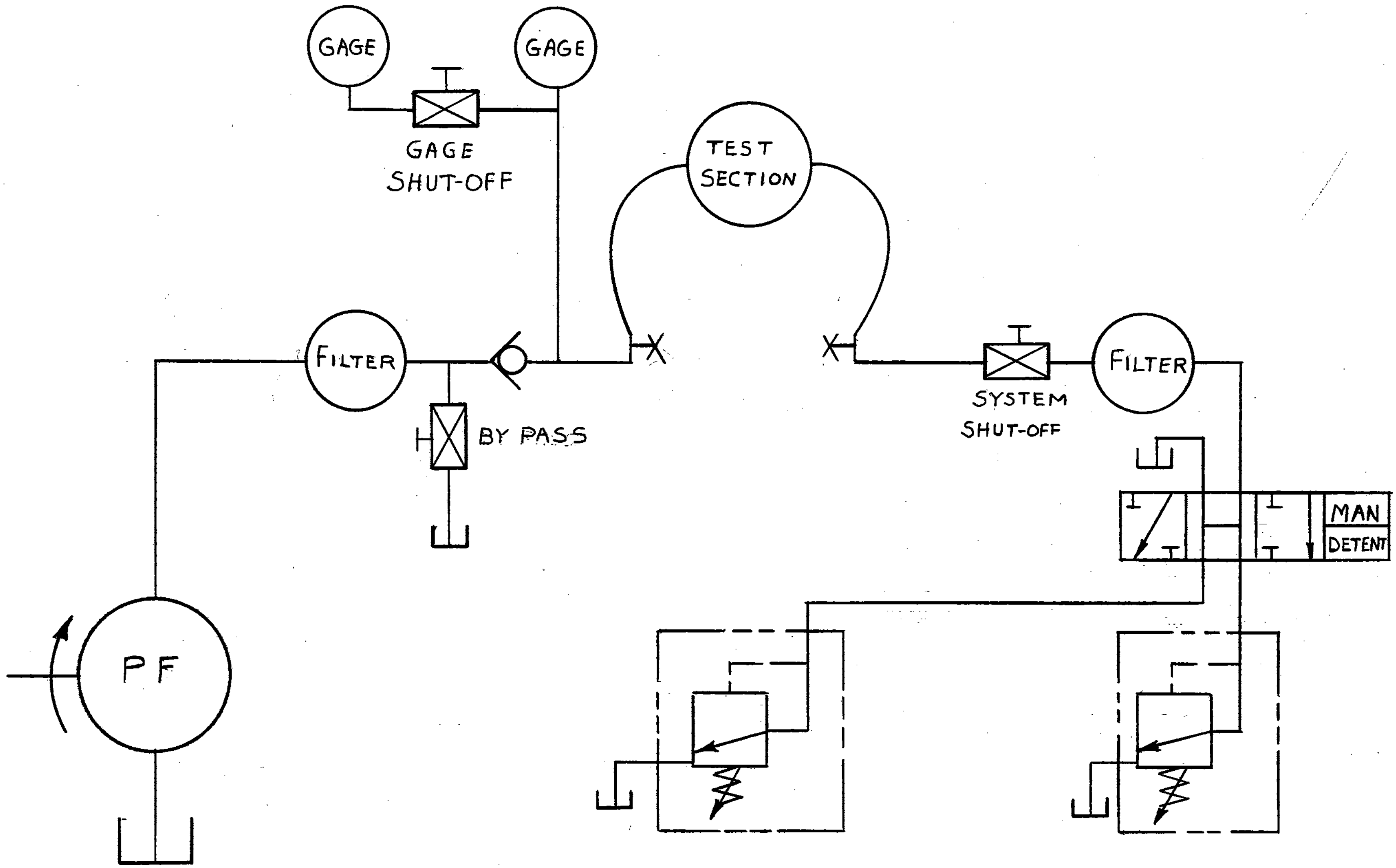

Figure 5. Diagram of the Hydraulic System

The system reservoir was designed and constructed to meet the particular needs of the system. In order that high temperature fluids be available for tests, two immersion heaters were placed in the reservoir. The reservoir was physically placed on the frame of the Instrument Stand. This location was finally chosen for ease of maintenance and to meet space requirements as well as being a compromise with the use of the immersion heaters.

A provision was made in the design of the system which would allow static tests to be run on the Test Facility. A check of Figure 5 would reveal the location of a check valve just upstream of the pressure tap and a shut-off valve just downstream of the Test Section. Two tees were placed on the line, one upstream and one just downstream of the Test Section. One leg of each of the two tees was plugged. By closing the shut-off valve and pressurizing the closed section of the system with air or other gases through one of the tees, the Facility could be adapted for testing Face Seals under dry running conditions.

A hydraulic loading system was used to put a load on the Face Seal during a test. The system was composed of an integral reservoir and hand pump and a hydraulic cylinder. Estimations regarding the maximum force required at any time resulted in the use of a 1½-inch bore cylinder and a hand pump which would provide nominally 1000 psi fluid pressure. The hand pump reservoir was physically mounted on the Instrument Stand. The actuating cylinder was mounted such that the cylinder rod end would press against the end of the Secondary Shaft opposite to the end on which the Seal Test Head was mounted.

Electrical System

The electrical system, the diagram of which is seen in Figure 6, was designed to provide a certain amount of safety for the operator of the Facility. A fifteen-horsepower magnetic controller was used to control the electrical supply. Two starter stations were used to control the magnetic controller. One of the starter stations or "Panic Buttons" was located on the Test Stand as far removed from the exposed high speed rotating parts as feasible. The other starter station was located in the center of the Instrument Stand panel. A red light was mounted next to the starter stations to indicate when the magnetic controller was "on."

The Varidrive and the Motor-pump were provided with automatic controllers for the protection of each unit. The starter stations for these units were located on the right end of the Instrument Stand panel.

The two immersion heaters were each provided with manually operated single-phase circuit breakers for the protection of each of the heating elements. These control boxes were located on the Instrument Stand next to the reservoir as seen in Figure 7.

Instrumentation

The objective of the instrumentation used was to provide the Test Facility with simple but effective means of obtaining the desired test data. As in the case of the previously mentioned torque measurements, a simple arm and weight arrangement would provide the desired data with sufficient accuracy.

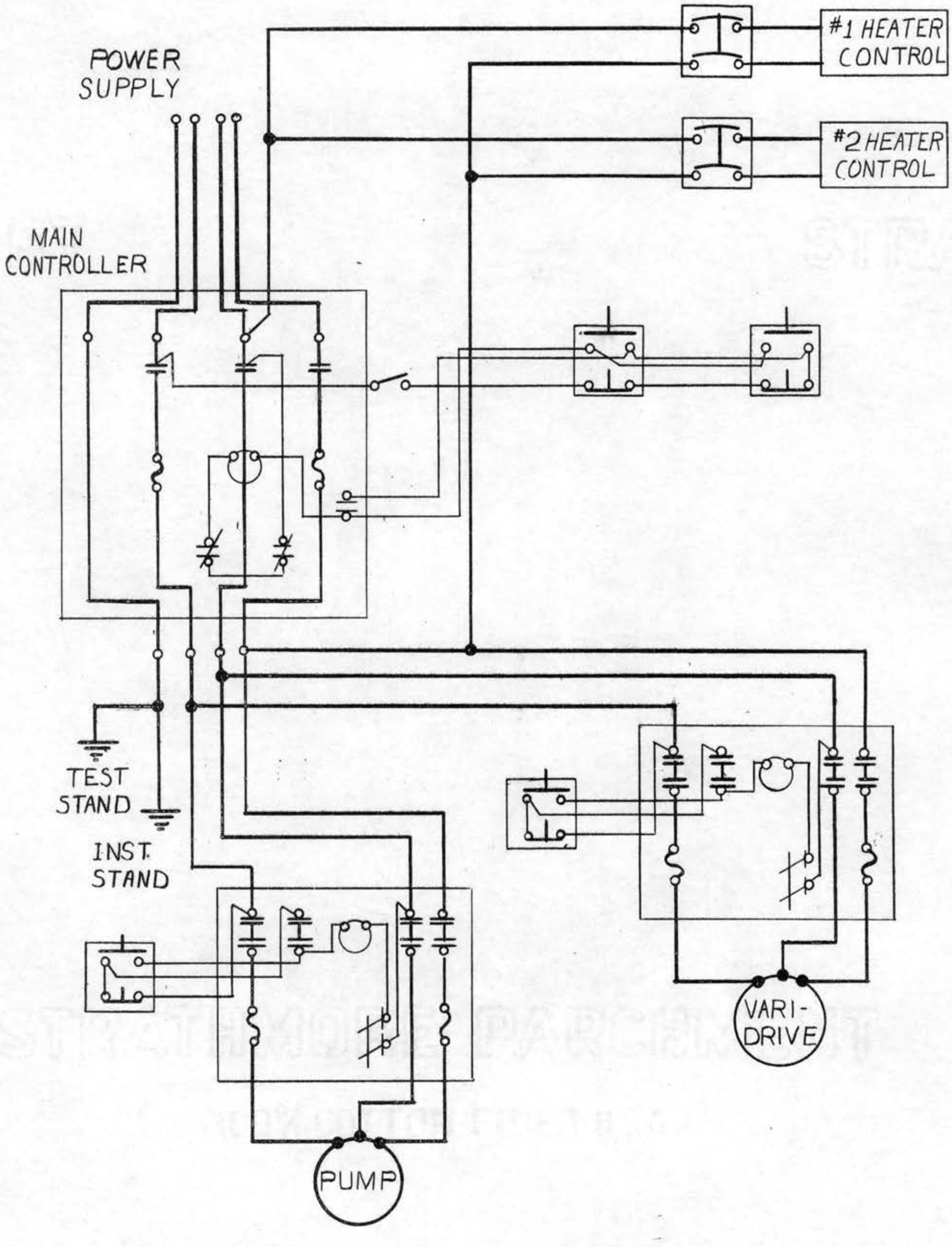

Figure 6. Diagram of the Electrical System

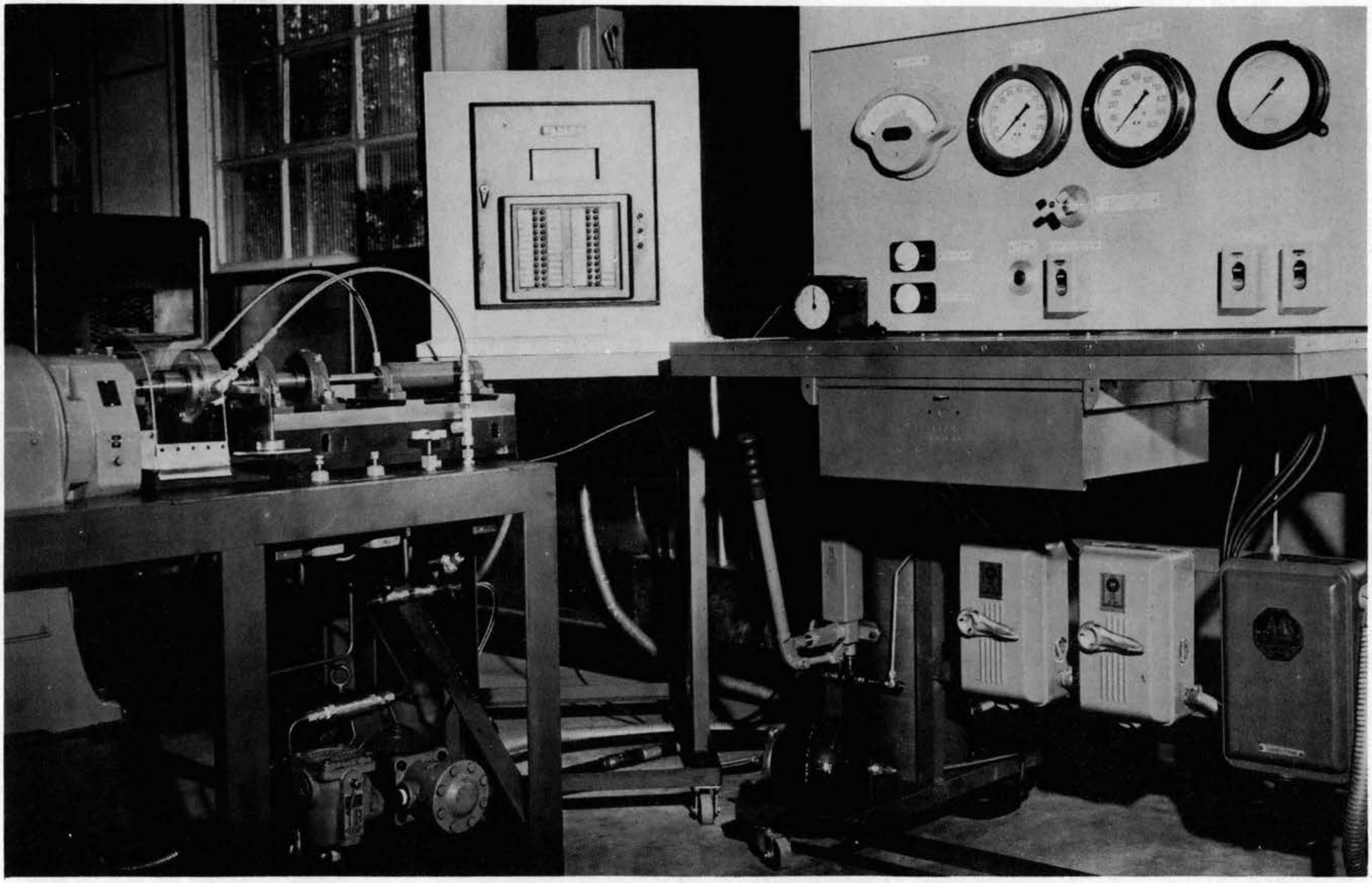

Figure 7. The Test Facility

The hydraulic pressure of the circulating system was determined with the use of a pair of bourdon tube pressure test gages. Two gages were used in order that accurate readings could be obtained throughout a range from zero to 500 psi. As can be seen in Figure 8, the low-pressure gage (0 to 300 psi) was protected from pressures above its range with a shut-off valve.

The speed of the output shaft on the variable speed power unit was measured with an electric tachometer, which was supplied with the power unit. A calibration was made of the tachometer since a check of the speed indicated that the tachometer was slow in the high speed end of the speed range.

An indication of the temperature at various points throughout the Test Facility was considered desirable. Of prime importance was the temperature of the fluid in the test cavity, since this temperature would be the average temperature at which the Face Seal would be required to effect a seal. The method used for obtaining the fluid temperature was that of inserting an iron-constantan thermocouple in the Test Head in such a position that the temperature indicated on the recorder would approximate the fluid temperature. The temperature recorder used was a 24-channel indicator with a range 0 to 1200°F calibrated for use with iron-constantan thermocouples. Figure 7 shows the temperature indicator and its stand.

A thermocouple was inserted in the lubricating oil of the Speed Increaser on the Varidrive Unit. This was done because the speed increaser was one of a relatively new design, and there were some misapprehensions about its operation in an extended testing program.

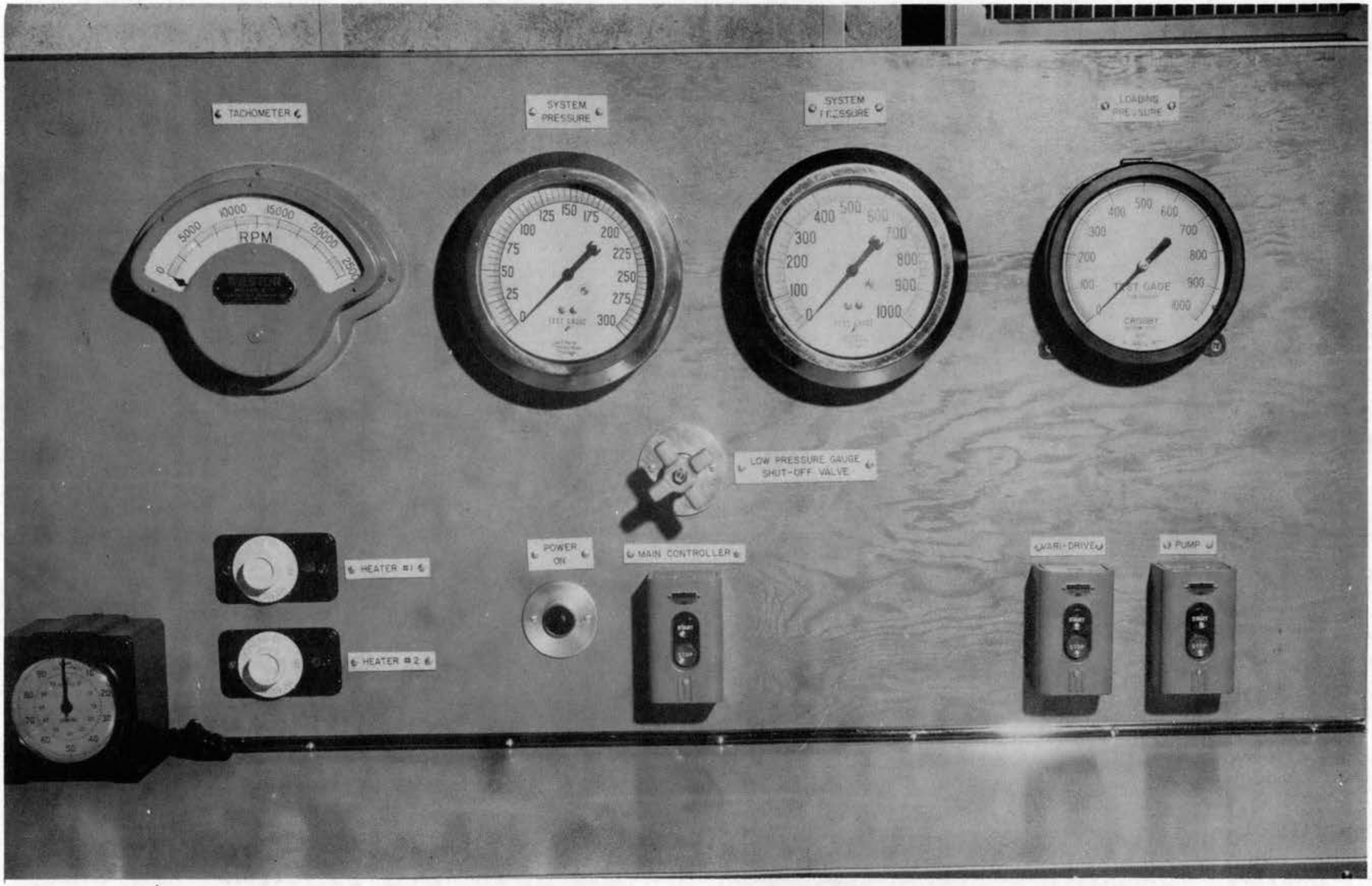

Figure 8. Elevation View of the Instrument Panel

CHAPTER IV

OPERATIONAL PROCEDURE

Before starting any portion of the Test Facility, a visual check was required. In the case that highly inflammable fluids were to be used during the test, a fire extinguisher was placed within easy reach for the sake of safety. The visual check included inspecting the system reservoir for the proper fluid level, checking the hydraulic lines for leakage, and seeing that the temperature indicator power supply plug was properly connected. The temperature indicator was then turned on by opening the front of the case and setting the switch to "on." Next, the selector valve handle was set in the neutral position, the system shut-off valve on the right end panel was fully opened, and the bypass valve on the table top was fully opened. The two pressure control valve screws were turned out such that the valve springs would allow the minimum pressure setting on each pressure control valve. The low pressure gage shut-off valve on the Instrument Stand panel was fully opened or fully closed, depending upon what pressures the particular test run would involve. If the hydraulic pressure approached 300 psi, the shut-off valve was closed.

The Test Facility was now ready to start. In order to start, the main power box on the wall was turned to "on." The small safety switch on the main magnetic controller was switched to "on," and then a main starter station either on the Test Stand or the Instrument

Stand panel would energize the main magnetic controller. The Varidrive and the hydraulic pump were both then ready to start.

The next step in starting the Test Facility was to load the Seal Test Section. This was done by first fully closing the valve on the loading system hand pump, and then operating the hand pump until the preselected loading system pressure was observed on the loading system pressure gage. At this point, a check was made to see that the drip pan and shield were properly positioned with regard to the Test Section. The shield was placed such that a small amount of clearance existed between the shield and the Seal Test Head as well as between the flexible lines and the torque arm. The hydraulic pump was then started.

After starting the hydraulic pump, which was started by depressing the "start" button on the Instrument Stand panel, the Varidrive was likewise started. The desired hydraulic pressure was then built up. If the desired hydraulic pressure was less than 25 psi, the selector valve was left in the neutral position and the bypass valve was slowly closed until the desired pressure was attained. If the bypass valve was completely closed before the desired pressure was reached, the system shut-off valve was slowly closed until the desired pressure was reached. If the desired operating pressure was in a range from 25 psi to 125 psi, the selector valve was shifted to "L" and the bypass valve was then slowly closed. After the bypass valve was closed, the screw on the pressure control valve marked "Low" was slowly turned in until the desired pressure was attained. When the desired operating pressure was above 125 psi, the selector valve was shifted to "H" and the bypass valve was then slowly closed. The screw on the pressure control valve marked "High" was then slowly turned in, until the desired operating pressure was reached.

Once the desired system operating pressure was reached, the shaft speed at the Seal Test Section was increased to the desired operating speed. At this point, if the test run was to be at higher than ambient fluid temperatures, the immersion heaters were turned on and set on the desired reservoir fluid temperatures.

Once the desired running conditions of hydraulic system pressure, loading system pressure, fluid temperature, and shaft speed at the Seal Test Sections were reached, the test was considered underway. The data to be recorded included the following:

1. Personnel operating the Test Facility
2. Date of test run
3. Loading system pressure, psi
4. Hydraulic system pressure, psi
5. Test Section shaft speed, rpm
6. Time of recording, hour and minute
7. Temperature of Seal Test Head, °F
8. Temperature of ambient air, °F
9. Temperature of fluid in reservoir, °F
10. Temperature of oil in Varidrive speed increaser, °F
11. Weight on torque arm balance pan, lbs.
12. Fluid leakage rate, drops per minute.

Note: Items 6 through 12 were recorded beginning at ten-minute intervals unless items 3, 4, or 5 were changed or the Test Seal failed.

There were two methods used for stopping the Test Facility: an emergency stop and a normal stop. The emergency stop was used only when the operator deemed such action necessary. If, during the operation of the Test Facility, the loading system pressure was lost allowing system pressure to pour fluid out through the Face Seal, an emergency stop was necessary.

A normal stop was considered as having several steps. If the immersion heaters were in use, they were to be turned off. The speed of the Varidrive was then reduced to the low speed stop; then the Varidrive was stopped by depressing the station "stop" button. After the Varidrive had ceased rotating, the hydraulic system pressure was reduced to zero by opening the bypass valve (and system shut-off valve, if used) and then placing the selector valve in the neutral position. After the system pressure was reduced to zero, the hydraulic pump was stopped. A short time was allowed to elapse before the loading system pressure was released. This was to allow all residual pressures to be reduced to negligible values.

After the loading system pressure was released, the loading cylinder was retracted to allow for the inspection and/or the removal of the test seal. If the Face Seal showed no appreciable wear, the test was continued the following working day.

At the conclusion of a day's run, the Facility was secured. This included shutting off all power, wiping up any excess fluid which had collected during the run, and covering the Seal Test Section with a cloth.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

The following conclusions and recommendations are presented for the consideration of those who may continue to work with fluid sealing problems in general and particularly with the Seal Test Facility.

The flow rate and pressure ranges were both sufficient to permit the testing of Face Seals now in use or proposed for use in presently operational aircraft systems and components.

The speed range of the Varidrive was sufficient for most high-speed Face Seals. In the speed range from 10,000 to 15,000 rpm, two critical speeds were evident, and therefore no tests were attempted in this range.

The immersion heaters which were placed in the main system reservoir would be sufficient to meet the needs of any particular test, since almost any non-corrosive fluid used would have a maximum safe operating temperature below 500°F.

There are some characteristics of the Test Facility which are less than desirable; therefore, the following recommendations are submitted for consideration:

1. Since there is a definite limitation on the thrust loads which can be placed on the output shaft of the speed increaser, this limitation could be removed by removing the 14-inch spacer on

which the Varidrive was placed, and mounting the Varidrive directly on the Test Stand base frame. This could also reduce some of the vibration which the spacer seemed to augment. By dropping the Varidrive to the base frame, an Auxiliary Shaft with the necessary high speed radial and thrust bearings would need to be used along with a belt or gear drive driven by the Varidrive. The use of the auxiliary shaft could also allow a further increase in the maximum speed to something around 50,000 rpm.

2. If the Varidrive were lowered to the base frame, the problem of alignment could be eased by mounting both the Secondary Shaft and the previously proposed Auxiliary Shaft on a single piece of 1-inch plate steel. This plate would be mounted directly on the table top of the Test Stand.

3. The main system reservoir was provided with heating elements, but no provision was made for cooling the fluid other than by natural convection. Since much closer control of the fluid temperature was desirable, a cooling system would be expedient.

4. The recording of test data on extended tests was tedious and time-consuming, especially the recording of various temperatures, pressures, and the torque measurements. Both the temperatures and the pressures could be fed into a recording device which would produce recorded instantaneous data, plotted against time. This would be desirable because of the rapidity of events during seal deterioration.

5. In order to feed the torque measurements into a recording device as well as to obtain a more accurate picture of the torque

variations during a run, a pneumatic controller could be incorporated in a torque measuring device as a replacement for the pan and weights. The use of the pneumatic controller would require a relatively constant supply of low pressure air.

Another method of measuring the torque could be that of incorporating a pair of strain gages on a cantilever propped end beam. This measurement then could be continuously fed into a recording device.

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APPENDIX A

SAMPLE CALCULATIONS

The power requirements for the Drive Mechanism were estimated with the use of the Lewis Horsepower Equation as follows:

$$HP = \frac{(5.283 \times 10^{-6})\ n\ f\ L\ (D^3 - d^3)}{(D^2 - d^2)}$$

where

D = outside seal diameter, inches

d = inside seal diameter, inches

n = angular velocity, rpm

L = total face load, pounds

f = coefficient of friction.

The total face load was estimated by the following equation:

$$L = \pi/4\ (D^2 - d^2)\ p + k$$

where

p = fluid pressure

k = spring force.

The assumed values were as follows:

f = 0.01

n = 30,000 rpm

p = 500 psi

k = 5 pounds

$D = 2\ 5/8$ inches Dimensions of a standard

$d = 2$ inches Gits Unit Seal

Then

$$L = (.7854)(500)\left[\left(2\ \frac{5}{8}\right)^2 - (2)^2\right] + 5 = 1145 \text{ pounds}$$

and

$$fHP = \frac{(5.289 \times 10^{-6})(30{,}000)(0.01)(1145)\left[\left(2\ \frac{5}{8}\right)^3 - (2)^3\right]}{\left[\left(2\ \frac{5}{8}\right)^2 - (2)^2\right]} = 6.26 \text{ HP}$$

APPENDIX B

EQUIPMENT AND INSTRUMENTS

Equipment

1. Magnetic Controller: Allen Bradley, 220 V, 3 phase, 60 cycle, 15 hp.

2. Magnetic Starter: Cutler-Hammer 9586H6170 - 220 V, 3 phase, 60 cycle, NEMA, 1 enclosure - reset in cover, 5 hp, 17.5 amp heaters.

3. Motor: Varidrive Syncrogear Motor range maximum, rating 23-10 Spec 5.95 increaser gear ratio assembly 5-C arranged to provide minimum speed of 2500 rpm and maximum of 25,000 rpm, 5 hp, frame No. 23-215-22Y, 3 phase, 60 cycle, 4 poles, 220/440 V, VEUGHY.

4. Tachometer Generator: Elinco Ball Bearing Model BSJX-1071, AC Permanent Magnet Type, 1 phase, 1.0 volts/100 rpm, 12 poles, Serial No. 304782.

5. Magnetic Starters: Cutler-Hammer 958646120 - 220 V, 3 phase, 60 cycle, NEMA, 1 enclosure, reset in cover, 2 hp, 6.5 amp heaters.

6. Motorpump Unit: Vickers Hydraulic including balanced vane pump VK 105-A-10 Coupling MPCI-A2, Electric motor - Reliance Duty Master 2 hp, 1200 rpm, frame 213, 220/440 volt, 3 phase, 60 cycle, NEMA Standard Floor Type, open, 1155 rpm, 6.4/3.2 amps, identification No. Y133107A1.

7. Filter (2 each): Hydraulic, Rosaen Tell-Tale 5-F-149-M-3.

8. Needle Valve (3 each): Marsh FFG-3/8 1900.

9. Check Valve: Republic 453-1/2 S.

10. Selector Valve: 4 way valve, Dukes L01775-3DE.

11. Pressure Control Valve: Vickers RT-03-Z1-10, directly operated, Internal drain, range 25 to 125 psi. Rated capacity 8 gpm.

12. Pressure Control Valve: Vickers RT-03-131-10, directly operated, Internal drain, range 125 to 500 psi. Rated capacity 8 gpm.

13. Reservoir: Manufactured by O.S.U. Mechanical Engineering Laboratory, Size 8.875 inch O.D. by 28 inches, approximate capacity 6 gallons.

14. Hand pump: Hydraulic, Pine #785, with 3/4" piston and integral reservoir (70 cu. in.), range 0-1000 psi.

15. Actuating cylinder: hydraulic, Bendix-Westinghouse Model FCH, 1½" bore, 4" stroke, plain end piston rod, serial No. W1593.

16. Ball Bearing Pillow Block (2 each): SKF Part No. SY-108.

17. Super Precision Ball Bearing: SKF Part No. 7202 CTC/C78.

18. Circuit Breaker (2 each): Westinghouse De-Ion Circuit Breaker, 15 amps, 250 volts.

19. Immersion Heater (2 each): New Jersey Therm-X-Red, Type SSTR-243, rated at 3,000 watts at 230 volts, overall length 24", heated length 18", stainless steel outer sheath made of type 304, remote standard thermostat rated at maximum 500°F, copper sensing bulb and capillary tubing.

20. Push Button Station (4 each): Cutler-Hammer 10250H1956 - 220 V.

Instruments

1. Temperature Indicator: Brown Electronik Potentiometer Pyrometer, Model No. 156X63P48, Range 0 to 1200°F, Serial No. 541488, 110-125 volts, .45 amps.

2. Tachometer: Weston Indicating Meter, Model 273 R, 7.3 inch, Range 0-25,000 rpm.

3. Pressure Gage: J. P. Marsh Test, Range 0-300 psi, 1 psi increments.

4. Pressure Gage: J. P. Marsh Test, Range 0-1000 psi, 5 psi increments.

5. Pressure Gage: 6" Crosby-Ashton, Style AC10, phenol case and ring, range 0-1000 psi, 5 psi increments, back connection, steel bourdon tube.

6. Strobotach: General Radio Co., Type No. 631-B, Serial No. 12038.

7. Dead Weight Gage Tester: Manufacturer--Ashcraft Gauge Division of Manning, Maxwell and Moore, Inc.; Type 1313A, Serial No. 2-50.

8. Electric Timer: Standard, Electric, 0.01 minute increments.

APPENDIX C

TABLE I

TACHOMETER CALIBRATION DATA

Tachometer rpm	Strobotac rpm
3000	3300
4000	4275
5000	5250
6000	6270
7000	7285
8000	8400
9000	9520
Vibration	----
13000	13930
14000	14800
15000	15720
16000	16320
17000	17380
18000	18485
19000	19455
20000	20570
21000	21570
22000	22690
23000	23700

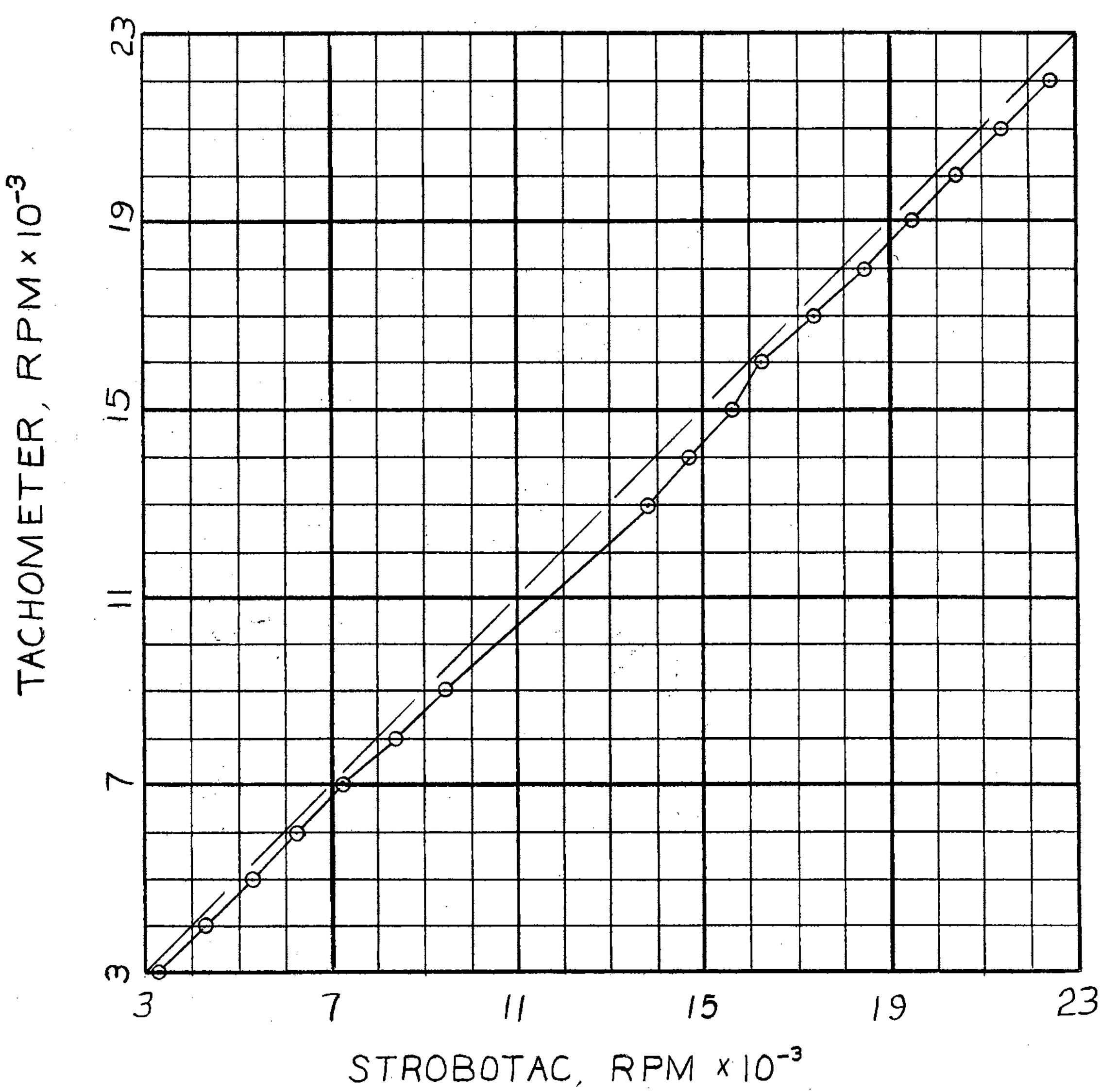

Figure 9. Tachometer Calibration Curve

VITA

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Candidate for the Degree of

Master of Science

THESIS: DESIGN AND OPERATION OF A HIGH-SPEED TEST FACILITY FOR ROTATING FACE SEALS

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