A THEORY ON THE CONTAMINANT SENSITIVITY OF ELASTOMERIC, RECIPROCATING, HYDRAULIC PRESSURE SEALS

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1964

Submitted to the Faculty of the Graduate College
of the Oklahoma State University
in partial fulfillment of the requirements
for the Degree of
MASTER OF SCIENCE
May, 1973

A THEORY ON THE CONTAMINANT SENSITIVITY OF
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PRESSURE SEALS

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PREFACE

This thesis introduces a theory on the contaminant sensitivity of an elastomeric, reciprocating hydraulic seal, presents experimental verification of the theory, and recommends topics for future investigation. An introduction into the problems involved in sealing and a summary of previous investigations on the subject of sealing theory, wear, and lubrication are included.

At this time, I should like to express my gratitude to Dr. E. C. Fitch, Jr., for allowing me the opportunity to continue my education. His guidance, encouragement, and personal esprit de corps have been an extremely valuable asset to me during my graduate work.

In addition, I would like to express my appreciation to my constituents at the Fluid Power Research Center at Oklahoma State University for their assistance over the past two years.

Most of all, I wish to extend my gratitude to my wife, Kathy, for her understanding, patience, and mature viewpoint during my graduate endeavors at Oklahoma State University.

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CHAPTER I

INTRODUCTION

Since the day man first conceived the idea of utilizing a fluid to transmit and transform power, he has been faced with a major problem, that of retaining the working medium within specified boundaries. Depending upon economics, physical limitations, and the nature of the surfaces comprising the boundaries of a fluid power system, various means have been devised to accomplish the task of sealing.

In the due course of transforming and transmitting fluid power, there are two types of sealing functions to be fulfilled. These are associated with surfaces having relative motion and with surfaces having no relative motion. In the case where two surfaces have a relative motion, the motion may be reciprocating or rotary. It should be realized that in the imperfections of the surfaces where sealing occurs, the heterogeneous characteristics of the sealing material and the physical properties of the fluid both influence the integrity of the sealing mechanism. The need for seal designs to overcome these adverse influences was immediately recognized. Thus, with the introduction of modern fluid power machinery, the need came for drastic improvement of some of the very basic components of a fluid power system, the hydraulic sealing elements.

Sealing elements are made in a seemingly infinite variety of shapes and sizes. Seal composition may be natural or synthetic and can be composed of various materials from vegetable fibers such as hemp or exotic metals. Yet, even with all of the existing technology at his disposal, the designer still must rely on experience when considering each application of a sealing element.

The area of sealing element design is an art because of the complete lack of seal performance and design models in the industry. Without such models, the synthesis of seals to meet the requirements of specific applications is a game of chance. The demands of sophisticated fluid power systems require that the performance, reliability, and life of seals be predictable and mathematically represented. Even though extensive experimental work and some emperical analysis is evidence in the literature, most of the effort to apply modern control theory to the sealing phenomenon has been concentrated in the applications where rotary motion is in effect between the sealed surfaces. This one-sided effort has caused the area of sealing as applied to reciprocating motion to be virtually untouched, especially in the field of hydraulic fluid power.

One of the most important and demanding of the applications involving reciprocating motion is in the fluid power actuator, also known as the hydraulic clyinder. Although there are "static" sealing element applications in hydraulic cylinders, the most critical sealing areas are the sealing of the piston to barrel surfaces and the rod to the end-cap surfaces. The requirements which must be satisfied involve fluid retention under "static" as well as reciprocating motion.

Fluid bypassing the seal or what is more commonly called leakage is the parameter by which a hydraulic seal's performance is measured. Of major concern to the user of a reciprocating hydraulic seal is its performance or leakage characteristics over a specified total distance traveled. Also of concern to the user is the leakage occurring under "static" conditions when no relative motion of mechanical parts is present.

The content of this report contains the results of the development and verification of a mathematical model which describes the sealing characteristics of a reciprocating hydraulic seal throughout its useful life. In order to establish a realistic mathematical description, such factors as seal geometry, material properties, abrasive wear action and sliding distances were reflected. Since the elastomeric type of reciprocating seals is the most prevalent type utilized currently and the scope of this investigation had to be restricted to a manageable level, only the elastomeric seals are considered in this study.

An elastomeric seal accomplishes the sealing function by conforming as nearly as possible to the macroscopic discontinuities in the adjacent surface with which it is in contact. Since this type of seal, however, cannot exactly conform to such an irregular surface, capillary size passages are created through which the system fluid can flow. Also, as the seal moves relative to the sealed surface, a hydrodynamic film may be established depending on the surface velocity, pressure differential, seal material properties, fluid properties, etc. An understanding of the sealing mechnism relative to the elastomeric seal is essential to the development of a mathematical model. To gain this understanding required the personal involvement of the investigator in conducting a logical test program. The details regarding the test program are presented as supporting evidence for the validation of the proposed model.

CHAPTER II

PREVIOUS INVESTIGATION

Introduction

A comprehensive literature survey revealed that the majority of the mathematical analysis done on reciprocating seals has been performed in Europe (1) (2) (3) (4). Based upon the author's eight years of industrial experience in the area of reciprocating seals and cylinders, there has not been a mathematical model developed that will predict the leakage characteristics of reciprocating hydraulic seals. This is not to say that excellent experimental data has not been generated in the past because both seal and component manufactureres have conducted extensive test work.

Hydrodynamic and Elastohydrodynamic Lubrication

The literature reveals that technical information pertaining to flow of fluid past seal surfaces stems from the area of thin film lubrication, specifically from theories in hydrodynamic and elastohydrodynamic lubrication. Application of the theories has been quite extensive within the past few years, and it is from this work that a breakthrough in reciprocating seal technology is most likely to occur.

Previous investigators have separated hydrodynamic film flow into two distinct components (1) (2) (3). One component is termed

Poseuille flow and stems from the pressure gradient across the film. The second flow component, labeled Couette flow, is due to the motion of the seal relative to the sealed surface. In the hydrodynamic approach to a thin film lubrication problem, an assumed film profile is utilized to solve for the pressure distribution within the film. If one of the boundaries which form the film surface is flexible, such as a seal, the film profile is very difficult to describe because of local deformation. Theyse (1) suggests a solution to the problem by stating that this elastic deformation is so much larger than the hydrodynamic film thickness that the pressure profile within the lubricating film is not altered. Therefore, the film profile can be calculated from an assumed pressure distribution. Thus, the Poiseuille component of hydrodynamic film flow which occurs past a moving elastomeric seal can be calculated by utilizing this value of the known film thickness. However. Theyse (1) further states that the surface profile must be very smooth in order for Poiseuille flow to exist. Thus, the thin film lubrication theory is not applicable in the case of rough surface profiles.

The component of leakage flow known as Coutte flow was discussed by Dowson et al. (2) in reference to work accomplished by Dr. Denny (3). This flow component was relegated to a very minor role by Dowson et al. (2) due to its insignificant net effect in seal leakage.

It should be pointed out that in the work reported by Theyse (1), Dowson et al. (2) and Denny (3) concerning the Poiseuille flow component that the film surface was considered to be smooth. To the author's knowledge, the fact that Poiseuille flow exists in the interstices formed by a smooth seal and a rough surface has not been previously recognized.

The elastohydrodynamic lubrication analysis as opposed to the hydrodynamic analysis considers that the pliable surface of a seal will deform under loading, which results in Hertzian-type contact stresses. Since elastomeric seals are manufactured from such a pliable material when compared to the magnitude of the existing loading pressures, the theory of elastohydrodynamics should apply. Reference (2) indicates, however, that the possibility of seal to surface contact still exists, even under lightly loaded conditions. Theyse (1) points out that because the analysis of a reciprocating seal by the elastohydrodynamic method is very complex, a full treatment has not been made to date.

Surface Topography

The subject of sealing surface topography must be considered an important area influencing the leakage characteristics of reciprocating seals. According to Greenwood (5), the surface finish generated by most machining methods follows a Gaussian distribution. The height of asperities and, therefore, the depth of valleys resulting from a honing operation can best be represented according to plots presented by Schlesinger (6), by a normal distribution. However, Roth (7) presented a statement made by L. G. Gitzendamer and F. O. Rathbun, Jr., of General Electric, that the cost per solution using a Gaussian model for accurately describing such surfaces and a large digital computer would be prohibitive. Therefore, a simplified model offers the only means of describing the surface topography of a machined surface Roth (7) offered such a model by closely indicating that for machined surfaces the profile can be closely approximated by triangular asperities with amplitude A/2 having wave length ℓ and that the enclosed angle on such a triangle lies between 170° and 180°.

On the other hand, the surface profile for a seal will be smooth relative to that of the machined surface. The very nature of the seal material and the manufacturing process insures such a smooth surface.

Wear Consideration

The phenomenon of wear has been the target of a great many investigations. One such investigation resulted in a very comprehensive literature survey on wear by the Fluid Power Research Center at Oklahoma State University (8). It was from this work that much insight was gained in the area of seal wear.

One of the leading tribologists in this country, Dr. E. F. Finken (9), indicates that most of the wear which occurs in hydraulic components may be classified under two types, adhesive wear and abrasive wear. Adhesive wear occurs when materials are brought into sliding contact and a transfer of metal particles occurs. Abrasive wear involves the removal of material from a surface by a cutting or ploughing action. Abrasive wear is further divided into two-body and three-body wear as illustrated in Figure 1. The wear which takes place between an elastomer surface and a metal surface is a combination of both two-body and three-body abrasive wear.

Lomakin (10), while conducting tests on mud pumps which incorporated steel barrels and elastomeric (rubber) sealing rings, observed that particles entrained in the slurry became imbedded in the surface of the elastoner (Figure 2). This phenomenon would create two-body abrasion, when considering the particles and the ring as one-body. The report by Lomakin (10) also brought out several conclusions on the wear

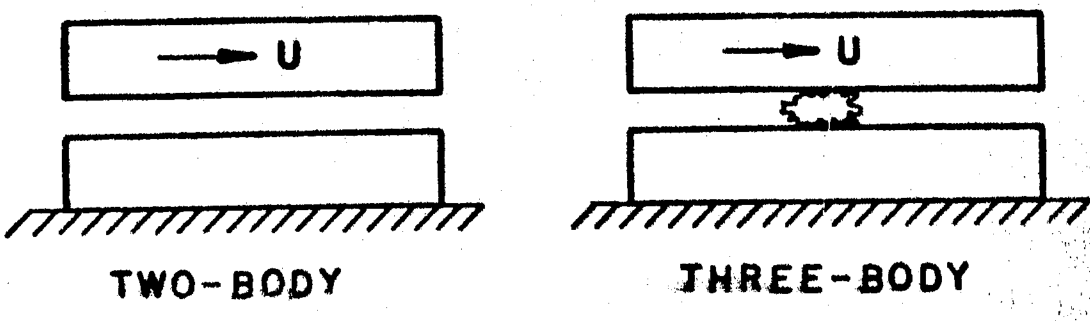

Figure 1. Illustration of Two- and Three-Body Wear

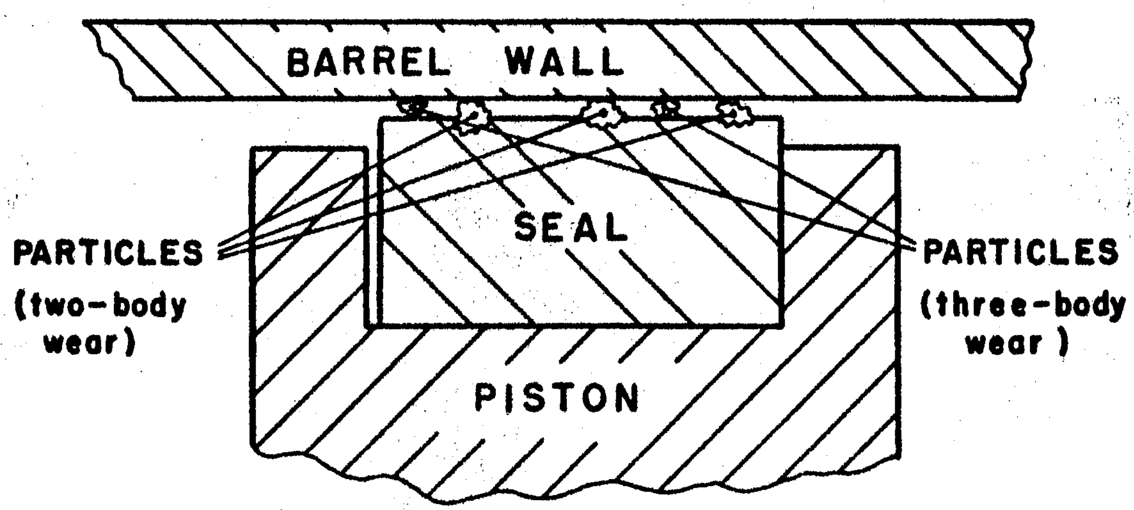

Figure 2. Illustration of Abrasive Wear in Seal Application

characteristics of two surfaces between which a lubricant film exists.

1. The smaller the grain, the more intensive the wear of sleeve material (60μ to 200μ).
2. The viscosity of the working fluid greatly affects the wear of the sleeve and the sealing material.
3. An increase in the contaminant concentration increases sleeve wear (.15% to 20% by weight).
4. Increases in pressure between rubbing surfaces decreased wear when a clay solution was used and increased wear when water was used (14.7 - 570#/in.^{2}) (a clay solution is known to have good viscous property).
5. Rate of motion does not have an effect on wear.
6. Wear is directly proportional to distance traveled.
7. Increased roughness in the surface finish increases wear (3.6μ in. - 100μ in.).

Lomakin (10) also presented a graphical illustration of his wear test results, and a plot of the wear versus operating time graph is shown in Figure 3. The region depicted here by "T" in Figure 3 is a "break-in" region and should be of considerable interest to users of hydraulic seals.

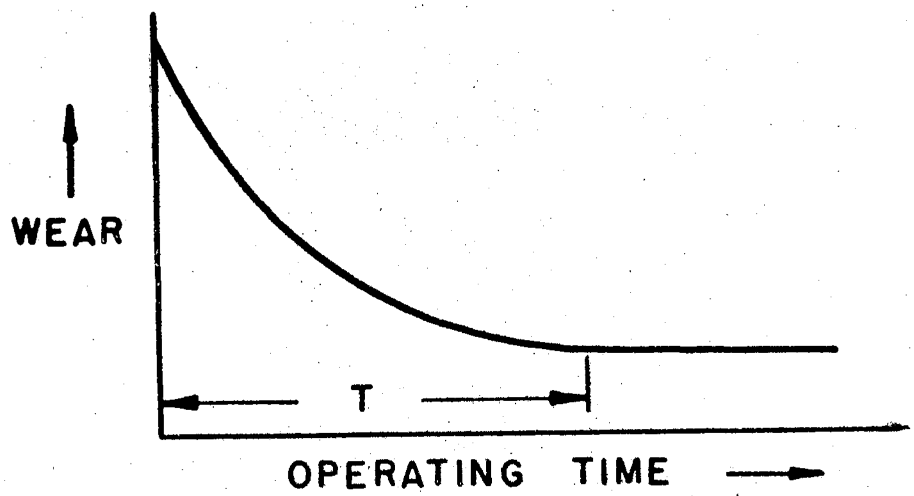

Figure 3. Wear Versus Operating Time

According to Thompson and Bocchi (11), it has been reasonably well established that the wear of dry solids resulting from sliding contact is governed by Archard's Law:

$$\frac{dw}{dL} = k P_1$$

where:

w = volume of wear

k = material const

P_1 = load

L = displacement through which the solids are moved relative to each other.

Thompson and Bocchi (11) approached the problem of wear lubricated surfaces by assuming that the force separating two surfaces in relative motion had two components. The elastohydrodynamic film was assumed to support part of the load while the remainder was supported by the reactive force created by the contact of surface asperities protruding through the film. This work implies, but never utilizes, the capillary flow concept.

Conclusions

Based upon the extensive literature survey conducted for this study, the following conclusions can be stated regarding the work of previous investigations:

1. Two components of hydrodynamic thin film flow associated with reciprocating seals have been recognized -- Couette and Poiseuille.
2. Every attempt to describe the flow between a reciprocating seal and a stationary surface which was presented assumed smooth surfaces and neglected the wearing action.
3. The flow theories presented ignored the possibility of flow occurring in the capillary passages created by the seal contacting the top of the stationary surface asperities.
4. Surface imperfections such as those resulting from surface machining will invalidate the flow relationships currently utilized.
5. The abrasive wear which will result in the presence of contaminant has been given very little attention.

CHAPTER III

THEORETICAL DEVELOPMENT OF THE MODEL

The leakage characteristics of a reciprocating hydraulic seal are dependent upon a great many factors. Because of these factors, one must pursue a course of divide and conquer if an adequately incorporated mathematical model is to be achieved. The first step in the analysis of any engineering problem is to identify the problem area. To satisfy this step, it is assumed that the fluid by-passing a reciprocating pressure seal may take two paths as illustrated in Figure 4.

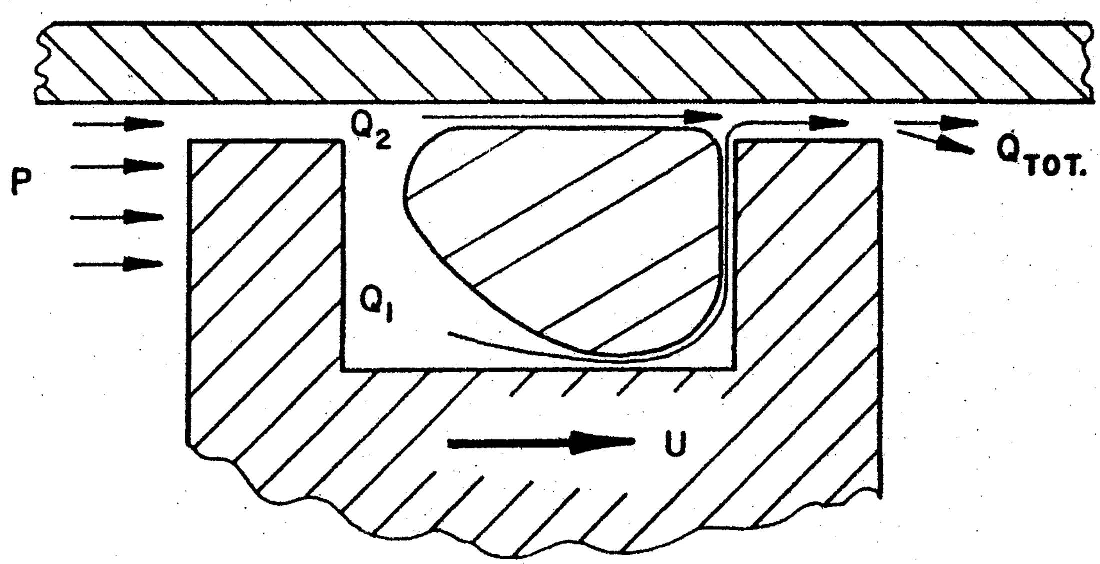

Figure 4. Seal Application

From eight years of personal experience in designing and testing such seals, the author has found that the leakage Q_1 is infinitesimally small relative to Q_2. Leakage Q_2 may occur in any application where two sealing surfaces have a relative motion to each other. According to the literature (1) (2) (3), the leakage Q_2 is described by both Couette and Poiseuille flows and conforms to elastohydrodynamic or hydrodynamic theories. However, there are topographical effects not considered in the application of these theories. In fact, the literature cited in the previous investigation indicates the necessity for a "considerable break-in time" before the application of either theory is valid (1).

Geometric Considerations

In order to conduct any sort of analysis of the leakage-wear mechanism associated with a hydraulic seal, the geometry of the flow paths must be considered. Therefore, the leakage path that flow Q_2 follows, as illustrated in Figure 4, must be closely defined.

The path for the leakage flow Q_2 can be divided into two specific zones: (1) the zone between the sealing surface and the seal itself, and (2) the porous zone created by surface finishing or normal wear processes. The first zone is called the hydrodynamic thin-film region and is geometrically described using conventional film thickness calculations and the seal configuration. The second zone represents a totally new leakage path concept and is thoroughly described in the remaining part of this section.

The general practice in industry is to prepare the internal surface of a hydraulic cylinder barrel by honing. Due to the reciprocating and rotary motion of the hone head and the cutting action of the grit in the hone stones, the generated surface is somewhat helically shaped. Prolifometer recordings of such honed surfaces made by the author show that these surfaces are described by small triangular-shaped peaks and valleys (asperities) having a width to depth ratio of approximately 10.

For modeling purposes, the cross-section of the leakage path formed by the walls of the asperities and the seal are accepted as triangular in shape and can be illustrated as in Figure 5. It can be noted that the seal forms the base for the triangle by conforming to the tops of the asperities of the sealed surface. Due to the fact that a seal has a finite width, some equivalent flow passage length, L_e, must be considered. This flow path is designated as an equivalent length because the actual trajectory of the asperities cannot be defined due to their random nature. One such triangular-shaped (capillary size) flow path formed at the sealed surface as shown in Figure 5 can be illustrated by the wedged-shaped volume as in Figure 6. The legs of the triangle in this cross-section represent the walls of the asperities or scratches generated by the surface finishing process. The base of the triangle is the seal material itself or the thin film boundary and moves relative to the triangle due to the reciprocation action under consideration. System fluid is forced to move within the triangular-shaped passages located around the full periphery of the seal.

The quality of a surface finish is presented in terms of the arithmetic average height of the asperities. For example, a 20AA surface finish exhibits a cross-sectional profile having triangular-shaped asperities with an arithmetic average height of 20 microinches. Thus, the triangular-shaped flow passage as depicted in Figures 5 and 6 can be

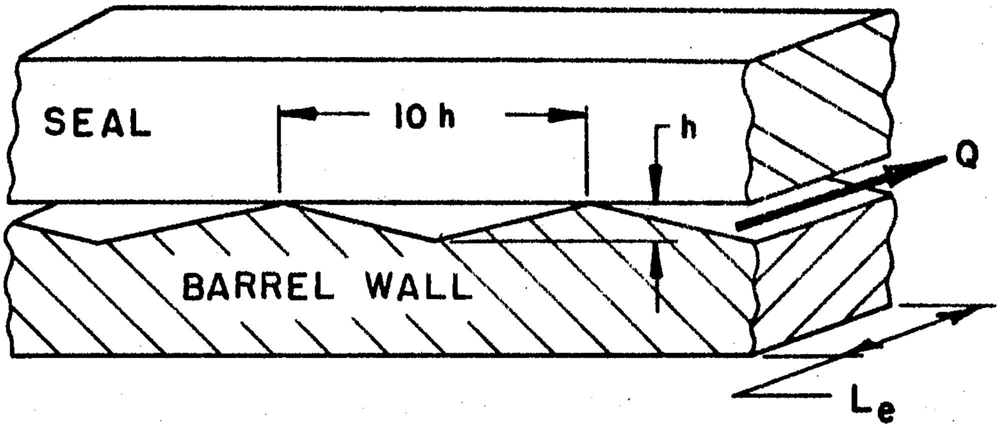

Figure 5. Leakage Path Model

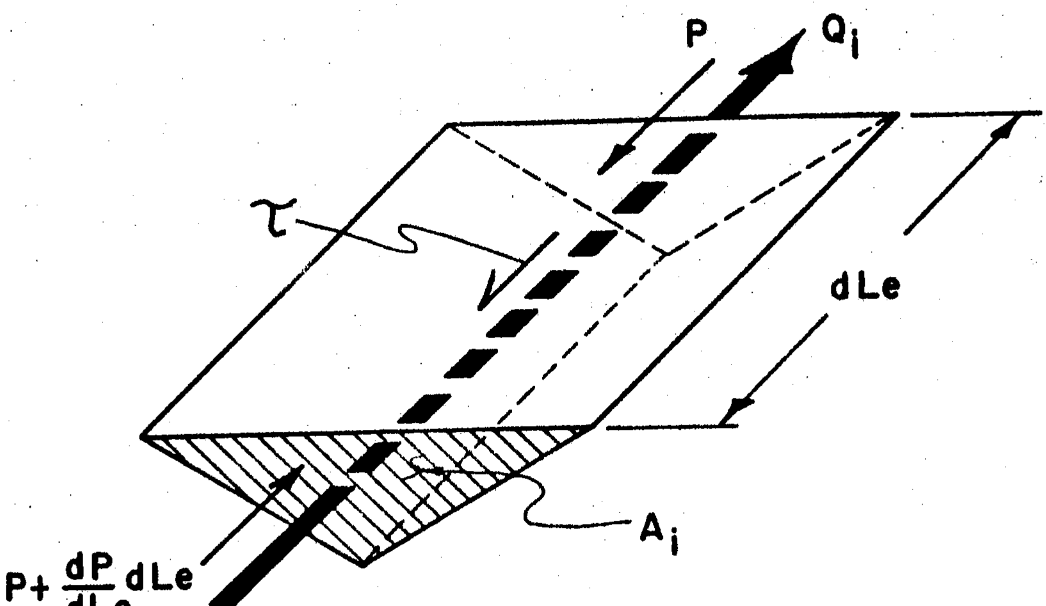

Figure 6. Capillary Model

essentially zero when associated with the large asperity heights produced by a normal honing process. Thus, since the finish of the sealed surface prior to any break-in or run-in is characterized as a honed surface, it is reasonable to assign the initial leakage to a flow component other than the conventional hydrodynamic thin film flow components. Therefore, the initial leakage must be attributed to interstitial flow.

The moving boundary flow, generally referred to as Couette flow, results from the adherence of fluid to the moving boundary. In a reciprocating seal application, the magnitude of Couette flow occurring in one direction is essentially equal to that which occurs in the opposite direction. Thus, a unique situation exists in a hydraulic cylinder application in that the net effect of Couette flow is zero.

Interstitial flow is leakage that occurs due to the differential pressure across the seal contact area and within the interstices which are bounded by the asperities and the fluid film. These fluid passages are the ones illustrated in Figures 5 and 6. Interstitial flow is a Poisseille type flow and, to the author's knowledge, has not been recognized in the literature as an important leakage phenomenon for seals. It is this flow consideration that has been made the operational or monitored variable to assess the performance, reliability, and life of a seal.

In order to derive a relationship for the interstitial flow, a force balance technique is applied to a representative interstice. This interstice is illustrated in Figure 6. The force on the fluid within the interstice is given by

$$f_P = A_i \Delta P \qquad (3\text{-}1)$$

explicitly defined as having a height of 20 microinches and a base of 200 microinches (considering the 10:1 width to depth ratio).

The surface profile of unused honed cylinder barrels exhibits a continuum of sharp-edged asperities, thus forming interstices conducive to flow. Therefore, the number of triangular-shaped flow passages to be considered is equal to the periphery of the seal divided by the base of the triangular asperities defined by the value of the surface finish.

Flow Considerations

According to previous investigations, the hydrodynamic leakage flow Q_2 is the summation of two flow components. However, these investigations have completely ignored the capillary flow which exists in the interstices created by normal machining processes. Therefore, to be complete, the total leakage flow must be described by the following three components:

1. Poiseuille Flow
2. Couette Flow
3. Interstitial Flow.

The hydrodynamic flow is the flow within the thin fluid film thickness or between the tops of the asperities and the seal. The thickness of the film is dependent upon the contact pressure which the seal makes with the barrel wall, fluid bulk properties, and the relative velocity of the parts. The contact pressure is a function of both the mechanical loading on the seal and the hydrostatic loading due to differential pressure. In accordance with Theyse (1), a considerable run-in time is required before thin film flow is established. In relation to this study, Theyse's (1) results indicate that thin film flow is where:

A_i = Cross-sectional area, and

ΔP = Pressure differential across the seal.

The cross-sectional area of the triangular-shaped interstice is equal to one-half the product of the base and height. Considering the proportionality of the base and height, the force on the fluid column can be written as

$$f_p = 5h^2 \Delta P \tag{3-2}$$

where:

h = Arithmetic average of the asperity height.

To determine the total external force associated with the interstitial flow area, the force on the single representative flow path must be multiplied by the number of such interstices along the sealed perimeter as presented earlier. Therefore, the total driving force causing the interstitial leakage is given by the expression

$$F_p = \pi r \Delta P h \tag{3-3}$$

where:

r = Radius of the sealed perimeter.

To evaluate the interstitial drag forces on the leakage fluid within a single interstice, an equivalent hydraulic diameter is used and is given by reference (12) as follows:

$$d = 4\ A_i/\text{wetted perimeter.}$$

Substituting appropriate terms for the triangular-shaped passage yields

$$d = 20\ h^2/(10h + 2\sqrt{25\ h^2 + h^2}\ .$$

Simplifying gives

$$d \cong h. \tag{3-4}$$

Thus, the interstitial drag force on the leakage fluid within a single interstice is given by Fitch (13) as follows:

$$f_d = \frac{\mu\ A_s\ V}{h/8} \tag{3-5}$$

where:

A_s = Surface area of the interstice

μ = Absolute viscosity of the fluid

v = Average velocity of the leakage fluid.

Since the surface area of the interstice is equal to the perimeter defined by the equivalent hydraulic diameter times its equivalent length (L_e), Equation (3-4) can be utilized to produce the following relationship:

$$A_s = \pi\ h\ L_e. \tag{3-6}$$

The velocity of the flow through a single interstice when the leakage flow rate is q_i as reported by Fitch (13) can be expressed as

$$V = \frac{qi}{\pi\ h^2/4}. \tag{3-7}$$

Thus, Equations (3-6) and (3-7) can be substituted into Equation (3-5) to give the total drag force acting on the leakage fluid in N interstices along the seal perimeter and is given by

$$F_d = \frac{32\ \mu\ L_e\ Q_i}{h^2} \tag{3-8}$$

where:

$$Q_i = N\ q_i.$$

Since a force balance must exist in all stable, steady-state hydrodynamic systems, the driving force F_p on the fluid contained within the interstices must be equal to the resistive forces (F_d); hence,

$$\frac{32\ \mu\ L_e\ Q_i}{h^2} = \pi\, r\, \Delta Ph.$$

Rearranging and solving for the total interstitial flow gives

$$Q_i = \frac{\pi\, r\, \Delta Ph^3}{32\, \mu\, L_e}. \tag{3-9}$$

For Equation (3-9) to uniquely apply to a reciprocating seal situation, the interstitial flow must be expressed in terms of leakage volume per cycle. Using the symbol T for cycle time (seconds per cycle), the total interstitial flow per cycle for a reciprocating cylinder application can be expressed by the following model:

$$Q_c = \frac{\pi\, r\, T\, \Delta Ph^3}{32\, \mu\, L_e}. \tag{3-10}$$

Equation (3-10) is capable of predicting the interstitial leakage for a given seal and cylinder barrel combination and under specific operating conditions.

Wear Considerations

One of the objectives of the research conducted for this thesis was to develop a mathematical model for describing seal leakage flow as a function of contaminant wear. The model derived for representing steady-state interstitial leakage (Equation 3-10) will be utilized as the basis for a wear model.

The wear associated with lubricated surfaces having relative motion can be described by the Archard-Finken Wear Theorem (9) as given by the expression

$$W = K_1 B F D \tag{3-11}$$

where:

W = Volume of wear material

K_1 = Archard wear constant

B = Finken lub. constant

F = Unit force normal to contact area

D = Total sliding distance.

The wear volume (W) may be described as the product of the wear depth (h_w) and the area of contact (A_c). Hence, using Equation (3-11), the wear depth may be expressed by

$$h_w = K_2 D \tag{3-12}$$

where:

$$K_2 = \frac{K_1 B F}{A_c}.$$

As the piston and seal of a hydraulic cylinder reciprocate the wear depth (h_w) increases and the asperity height (h) decreases, thus the wear height is equal to the original asperity height (h_o) minus the asperity height (h) at the end of any intervening time. Therefore, the wear relation (Equation 3-12) can be written to express the value of the asperity height after a given wear period as follows:

$$h = h_o - K_2 D. \tag{3-13}$$

The general wear model for a reciprocating, elastomeric, pressure seal is obtained by combining Equations (3-10) and (3-13). The form of the model is

$$Q_c = \frac{\pi r T \Delta P}{32 \mu L_e} [h_o - K_2 D]^3. \tag{3-14}$$

For a specific seal, test conditions, and cycle time, Equation (3-14) reduces to

$$Q_c = K_3[h_o - K_2 D]^3. \tag{3-15}$$

The constant K_3 can be evaluated by knowing the value of Q_c at one set of conditions.

An accelerated wear test provides a unique way of establishing the value of the interstitial flow (Q_c) separate from the total leakage flow which involves all three components. This can be accomplished by exposing the cylinder to fine contaminant and monitoring the leakage until no change in flow occurs. The difference between the total flow and the thin film hydrodynamic flow is the interstitial flow at any specified time.

The constant K_2 is a function of material and geometric properties of the barrel and the level of contamination in the fluid. A value of

K_2 may be determined for a specific barrel finish and a given contamination level by measuring the change in leakage flow during a designated number of cycles.

Another aspect of wear directly related to the performance of reciprocating pressure seals is the deterioration of the seal material itself. In such cases, the cylinder barrel is still being affected by the wear mechanism. However, as the seal material is scored by particulate contaminant, interstitial leakage takes place in the seal as well as the macroscopic surface of the barrel.

The influence of interstitial flow in the seal can be reflected by adding a term for the initial finish (h_o') of the seal and a seal material deterioration term within the brackets of Equation (3-15) as follows:

$$Q_c = K_3[h_o - K_2 D + h_o' + K_2' D]^3 \qquad (3\text{-}16)$$

where:

K_2' = Archard's wear constant associated with the wear volume of the seal material

h_o' = Initial arithmetic average surface finish of the seal.

When a seal is prone to contaminant wear, the net result of the bracketed terms in Equation (3-16) can easily cause the interstitial flow of the operating seal to increase without bounds. Such results are typical of worn-out seals.

CHAPTER IV

EXPERIMENTAL VERIFICATION

The test program designed to provide the experimental verification for the proposed leakage-flow models utilized seals of various configurations and materials. The selection of values for the test parameters such as oil temperature, reciprocating velocity, cycle length, and pressure was based on those normally encountered in actual service. The contamination level in the fluid was closely controlled and verified by Oklahoma State University particle counting techniques. Particular attention was paid to the problem of obtaining accurate leakage readings, since the leakage represented the major assessment parameter. The basic experimental program which was conducted can be outlined as follows:

1. Designed and fabricated test equipment.
2. Established test procedure.
3. Compared experimental results with theoretically predicted values.

Test Facility

The test facility consisted of two separate hydraulic systems as shown on the following page by the schematic drawing (Figure 7). One system was utilized to provide a driving force for the moving piston and rod in the seal test fixture, while the second system provided test

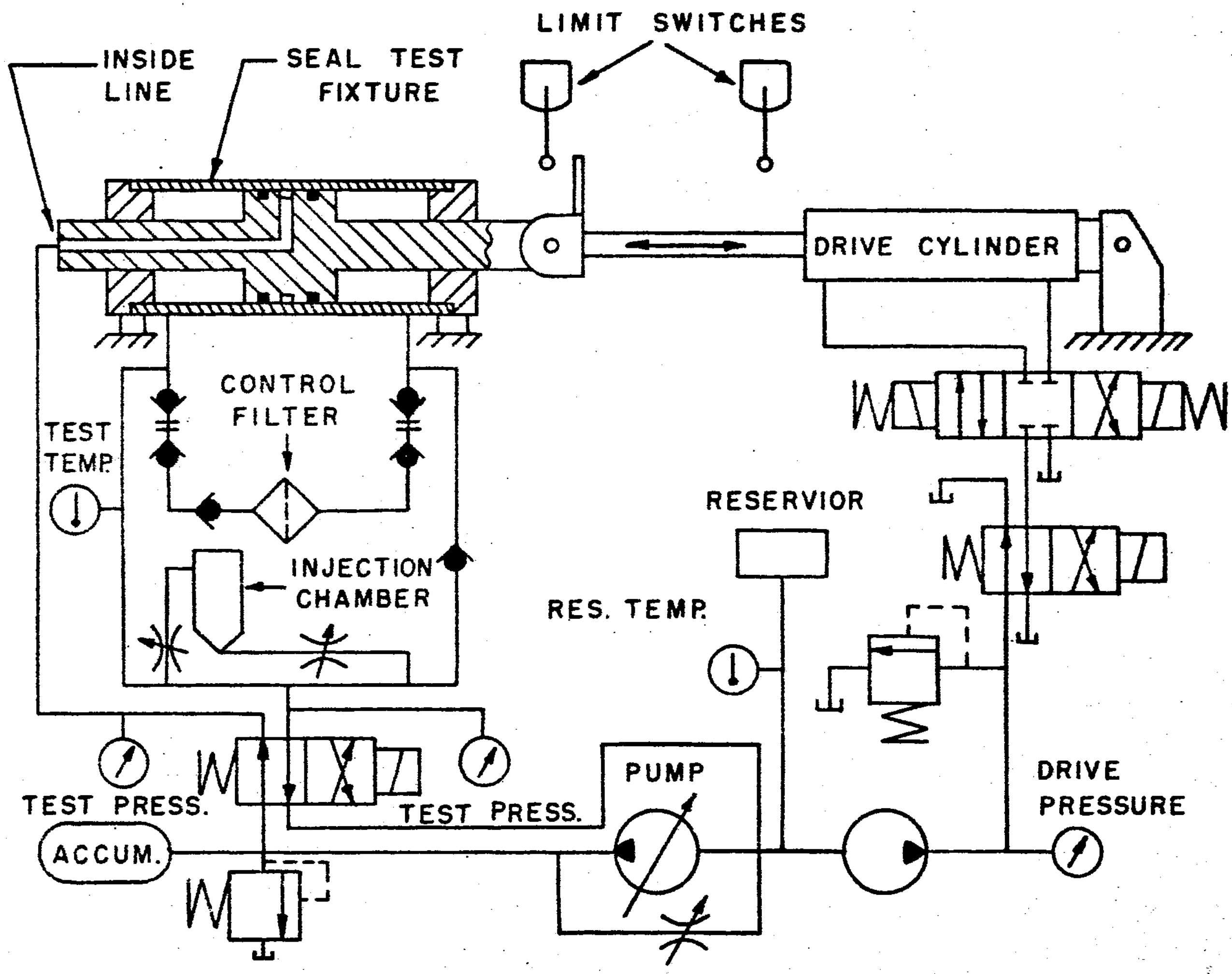

Figure 7. Test Stand Circuit

pressure for the seal test fixture and allowed a controllable contaminant environment.

A simple illustration of the balanced force test fixture is presented in Figure 8 and a complete description of the test system is contained in the procedures document of the Fluid Power Research Center (14). A photograph of the seal test facility is shown in Figure 9.

The piston adapter rings retaining the seals were machined according to the seal manufacturers suggested dimensions. The material used for the barrels was obtained from warehouse stock and was received prehoned to a specified size and finish. Oil temperature was monitored throughout the test by the use of a copper-constantan thermocouple installed directly in the fluid stream.

Control of the fluid contaminant level was accomplished by an injection loop and a filtering loop coordinated with the pumping action of the test fixture. Both loops could be segregated from the contaminant circuit during the cycle test. A stopwatch was used together with the driving cylinder pressure to monitor the cycling speed of the moving element. Leakage was measured in 10 ml. graduated cylinders.

Surface analyzing equipment was used to quantitatively appraise the inside bore surface finish. Profiles of barrels both before and after testing were obtained. The Clevite brush surface analyzer gave the profiles which are presented in Figure 10.

Method of Testing

To initiate the tests, new seal sets were installed into new barrels which constituted the test fixture. The filter system was then activated and the cycling motion initiated. During this phase, the

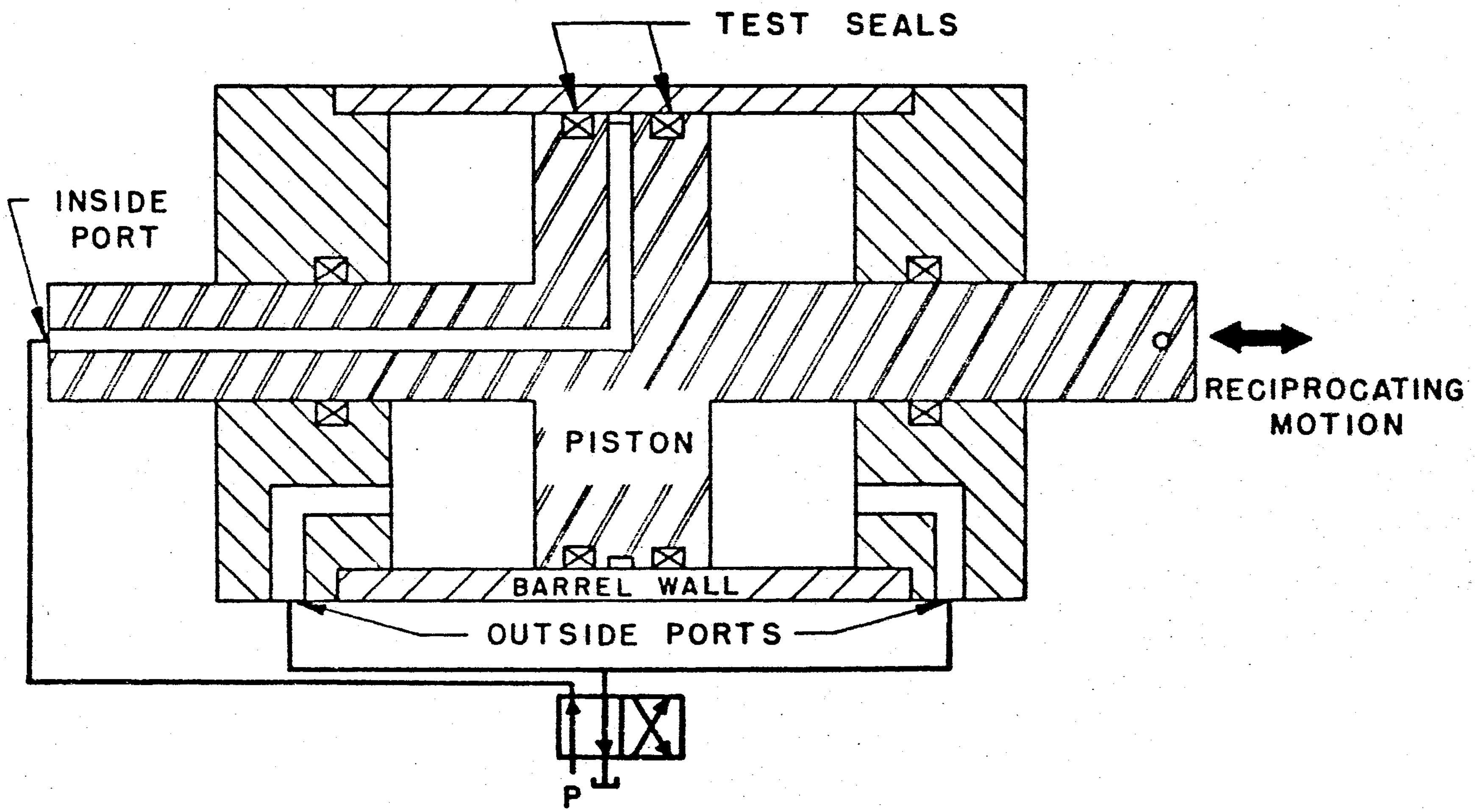

Figure 8. Test Fixture

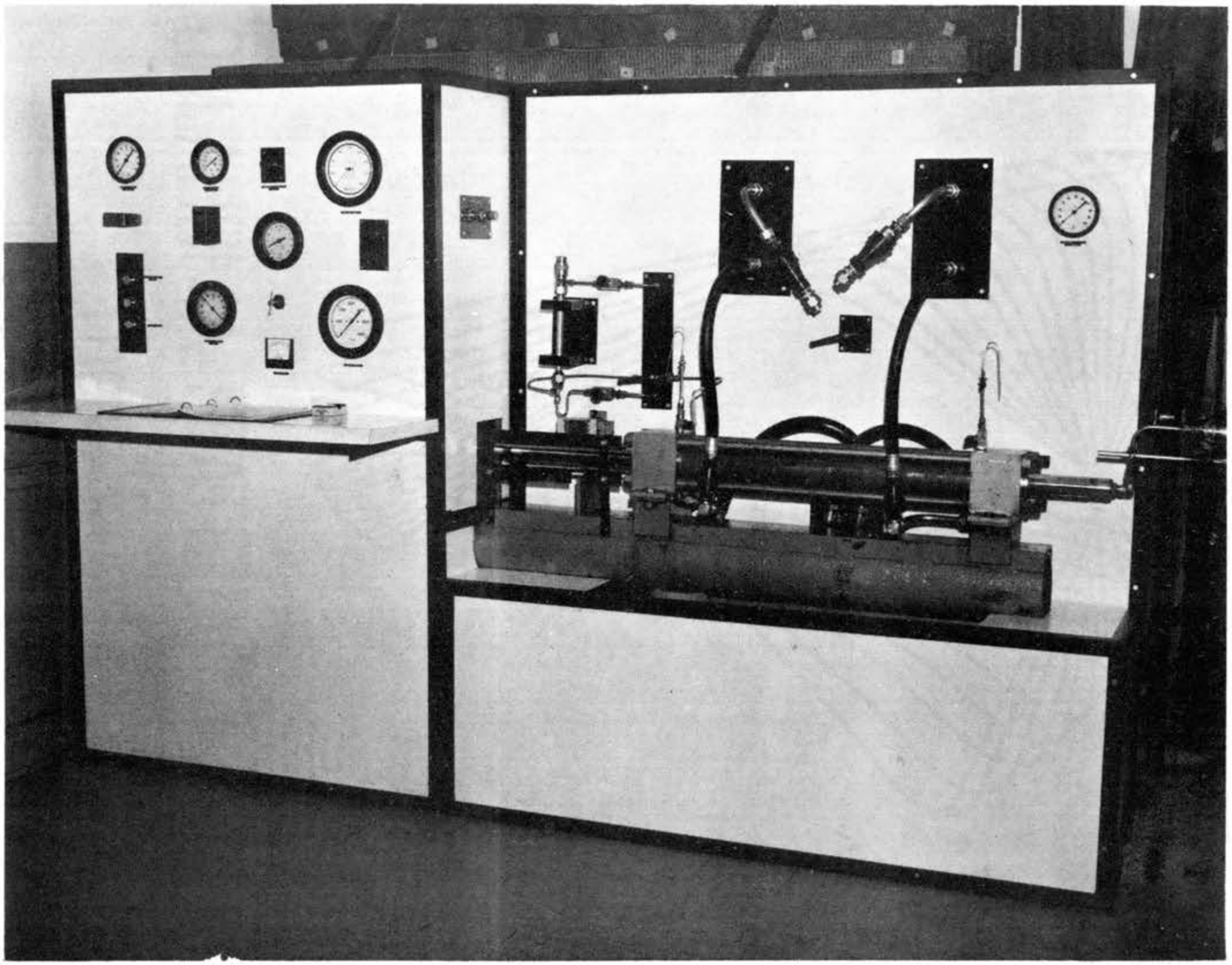

Figure 9. Picture of Test Stand

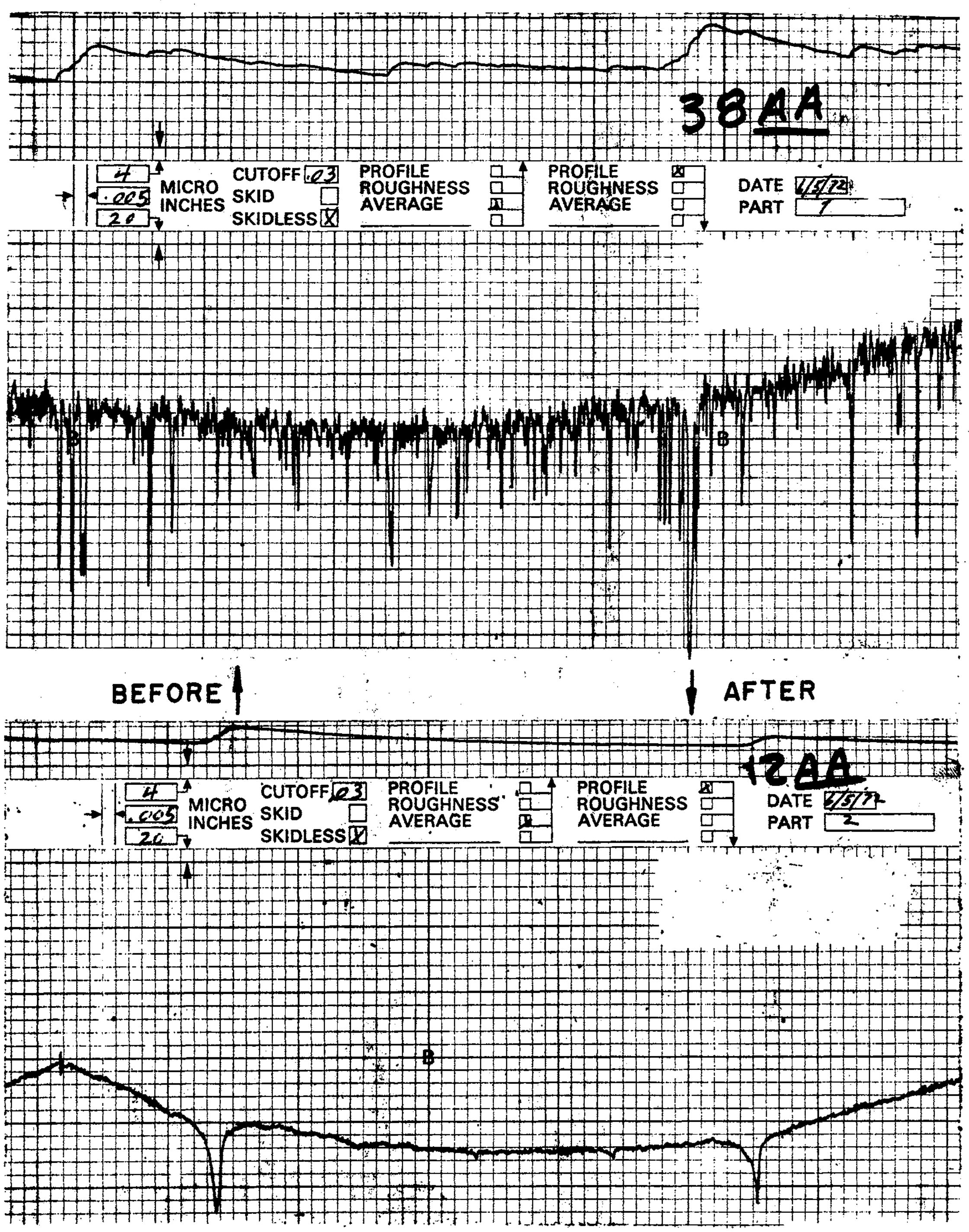

Figure 10. Example of Readings on Barrel Surface Finish

fixture is acting as a pump, circulating the fluid through the filter. The maximum pressure in the test fixture during filtering was 80 psi. When the required fluid cleanliness level was achieved (approximately 100 cycles), the filter was disconnected, and the internal pressure of the fixture was increased to 2000 psi.

The control logic in the hydraulic circuitry sequenced the internal fixture pressure alternately to either the inside or outside port as the piston moved through one complete cycle. Two movements, right to left and then left to right, constituted a cycle. This action was continued for 800 cycles of the moving piston. A leakage test was then conducted by removing the inside port supply line and collecting leakage flow from the inside port during ten cycles of the moving piston. During this ten-cycle phase, test pressure is supplied to the outside port while the piston is moving left to right and, therefore, the seals are pressurized for one-half the cycle.

Test pressure on the fixture was then lowered and a fluid sample extracted. Cycling motion was resumed, and three grams of classified AC Fine Test Dust with a particle size range of 0-20 micrometers was then injected into the test fixture at a controlled rate to bring the gravimetric level of the fluid to approximately 300 milligrams per liter. Following the 50-cycle injection and circulating phase, the cycling pressure was increased to 2000 psi. Testing was continued for 980 cycles under these conditions. Following this cycle period, a 100-cycle filter phase was initiated.

Before initiating the second contaminant injection, a ten-cycle leakage reading was taken. The second size range was 0-40 micrometers. The same method of testing was continued through particle size ranges of

0-60 and 0-80 micrometers. Piston velocity was maintained at 40 feet per minute; fluid temperature was 145°- 155°F. and stroke length was two feet during the described test.

Measurements of the barrel surface finish were made before and after the wear test to establish initial and final values of the surface finish.

Presentation of Data

The seals tested in the conduction of this study are presented in Table I. Leakage readings associated with each seal at equally spaced contaminant size range intervals constitute the data. The values indicated in the table are the leakage as measured in milliliters per ten cycles.

TABLE I

SEAL LEAKAGE

	OSU Seal Number			
Injection	112	118	98	11
Initial	10.0	10.7	20.5	7.7
0-20	8.7	8.5	8.5	7.0
0-40	7.7	8.2	4.5	14.5
0-60	7.7	7.7	3.5	13.5
0-80	6.5	8.0	3.5	66.0
0-20	6.7	8.0		
0-40	4.9	7.5		
0-60	5.6	7.2		
0-80	4.9	6.2		
0-20	4.3			
0-40	5.6	7.5		
0-60	5.6	7.7		
0-80	5.3			

The materials and design configurations of the seals tested represent those most commonly used throughout the fluid power industry.

Discussion of Results

The purpose of the verification phase of this study was to establish the validity of the proposed flow model. Since the thin film hydrodynamic flow portion of seal leakage flow has been covered quite extensively by other investigators, verification of the newly introduced interstitial flow will be presented here. This model is represented by Equation (3-16) and is rewritten here for convenience:

$$Q_c = K_3[h_o - K_2D + h'_o + K'_2D]^3$$

where:

Q_c is the interstitial flow per cycle.

Equation (3-16) indicates that if the seal itself were completely unaffected by abrasive wear and the surface finish of the barrel would continually improve, a decrease in leakage flow would result. To demonstrate the validity of this implication, seals #112 and #118 were selected for test. These seals were formed from urethane, one of the most abrasion-resistant, elastomeric materials known. The data for these seals as given in Table I shows that the abrasive wear which results in the presence of contaminant reduced the total leakage flow past the seal. Since it can be assumed that the seal material did not deteriorate, this reduction in leakage can only be produced by a reduction in interstitial flow caused by a reduction in average asperities height at the barrel walls. Furthermore, to verify this conclusion, surface finish measurements were made on the cylinder barrel both before and after testing. The barrel surface had a 20×10^{-6} inch average asperity height prior to testing which was reduced to 10×10^{-6} inch at the conclusion of the test.

Another seal which is considered by the industry to be very insensitive to contaminant was seal #98. The test data, in a most classical way, proves the existence of the same phenomenon expressed by Equation (3-16) and illustrated by seals #112 and #118. Seal #98 was extra special in that a 40AA finish (40×10^{-6} inches) barrel was used to conduct the test. At the conclusion of the test, the surface finish had been reduced to approximately 15AA. In view of this "rough" barrel consideration, the validity of the interstitial flow theory is difficult to refute.

When a seal material is susceptible to rapid deterioration under abrasive wear conditions, the interstitial leakage (according to Equation (3-16)) becomes predominant in the seal itself and not in the interstices of the barrel surface. To simulate the "soft seal" condition, seal #11 was selected. The formulation of the seal material was plastic in nature and could be scratched with the finger nail. The exact compound used to fabricate the seal was not disclosed by the manufacturer.

The test results on seal #11, which are presented in Table I, dramatically illustrate seal material deterioration by the ten to one increase in dynamic flow leakage. After the test, an examination of the sealing surfaces showed a barrel finish of 10AA (10×10^{-6} inches) compared to a 20AA finish prior to the test, while the seal exhibited the normal characteristics of an abrasive assault.

Based upon the results of the "soft seal" test, the following statements can be concluded:

1. The average roughness of the barrel surface will decrease by wear processes independent of the action occurring on the seal.
2. Interstitial flow through the interstices of the barrel surface can decrease even though the interstitial flow through the abraded surface of the seal increases.
3. The cubic effect on the seal wear term ($K_2'D$) in Equation (3-16) can far overshadow all other factors influencing the leakage flow past a seal.

The interstitial flow as influenced by the finishes on the barrel and seal surfaces together with the hydrodynamic thin film flow regimes interact in a most complex manner. Based upon the insight gained from the experimental verification phase of the study, a qualitative visualization of the flow interaction can be presented as shown in Figure 11. Each flow effect is illustrated separately as well as represented in overall (total) effect. The curve for the total effect is a general reflection of the apparent life cycle of all reciprocating seals.

The experimental verification phase of the study has revealed a possible technique for evaluating the value of the hydrodynamic thin film flow associated with a reciprocating seal. This technique became apparent as a result of calculations to establish the capability of Equation (3-16) to predict the final interstitial flow.

Since seals are normally formed by a molding process, the initial seal finish (h_o') is essentially equal to zero. Thus, with a "rough

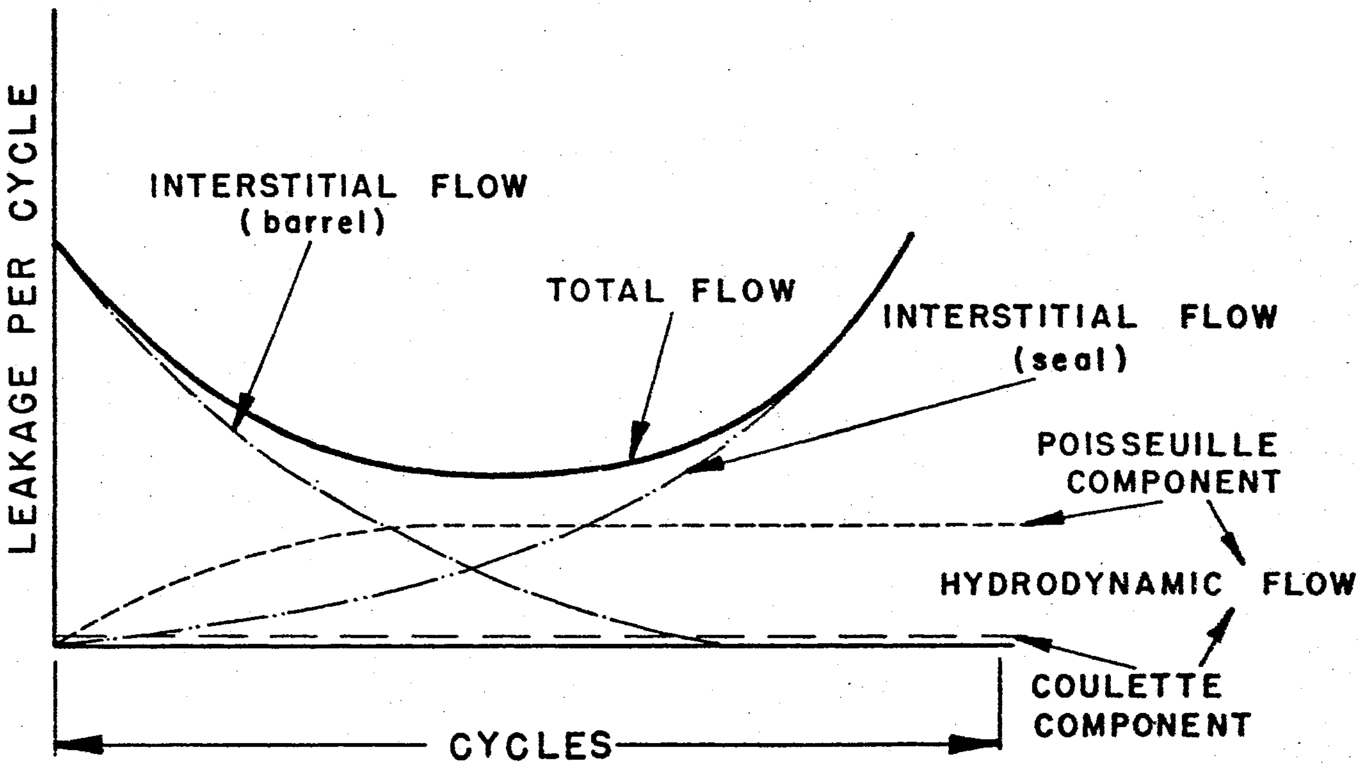

Figure 11. Illustration of Flow Interaction

barrel", the hydrodynamic leakage is equal to zero as pointed out by Theyse, and the total initial leakage (Q_o) is equal to the interstitial barrel surface leakage only. This can be expressed as follows:

$$Q_o = Q_c = K_3 h_o^3 .$$

The final total leakage (Q_f) is equal to the hydrodynamic thin film flow (Q_H) plus the final interstitial leakage. For an abrasive resistant seal, the final interstitial leakage is K_3 times the cube of the final average asperity height, h_f, of the barrel; hence,

$$Q_f = Q_H + K_3 h_f^3 .$$

The values for Q_o and Q_f may be taken directly from Table I, while values for h_o and h_f were obtained from the measurements on the test barrels. The calculated values of K_3 and Q_H for seals #112, #118, and #98 are presented in Table II. Values of flow in Table II are given in terms of cubic inches per cycle. Table II shows that, for the two urethane seals #112 and #118, the values for Q_o, Q_f, and Q_H are consistent. Comparing the values of Q_H with the respective values of Q_f in Table II shows that some interstitial leakage still occurs in the barrel surface finish. The values of hydrodynamic thin film flow given by the calculations appear compatible with those determined by other techniques.

TABLE II

PREDICTED FLOW

OSU Seal No.	Q_o	Q_f	K_3	Q_H
112	.061	.040	7.6×10^{12}	.032
118	.065	.043	8.1×10^{12}	.035
98	.125	.021	1.87×10^{12}	.018

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

The field of activity associated with the sealing of internal working fluid in today's fluid power systems is an area of advancing technology. Generally speaking, however, the application of seals to fluid power systems is still an art. An assessment of the knowledge of the modern seal industry as evidenced in the literature revealed a need for studies in the area of reciprocating hydraulic pressure seals.

With the emphasis currently being placed on higher reliability in fluid power systems, one must be able to predict the performance of a component during its useful life. To have the capability of predicting the performance and operational life of a hydraulic seal, the design engineer must have access to relationships which reveal the interaction of variables affecting seal performance. Seal performance is influenced by variables inherent to the fluid power system and to seal geometric and material design properties.

The purpose of this report was to develop a mathematical model by which the operational performance of an elastomeric, reciprocating hydraulic pressure seal working in a known contaminant environment might be predicted. To this author's knowledge, no such model is contained in the literature and is not available to the industry today. As documented by this report, areas such as abrasive wear, topology, and hydrodynamics were recognized as contributing factors required in the synthesis of such a seal model.

Based upon the broad considerations reflected by the seal model and its demonstrated validity, the author maintains that a number of significant contributions have been made by this study. Some of the more prominent contributions are listed as follows:

1. Recognition of the fact that interstitial flow due to surface effects as well as hydrodynamic thin film flow is contributory to total seal performance.
2. Consideration of abrasive wear phenomena as a prime factor in predicting seal life and performance.
3. Initiation of a test procedure by which seal parameters may be evaluated.
4. A means of quantitatively predicting the seal's performance over a specified period of time.
5. Implication of steps which might be taken to improve product reliability and performance.

Recommendations

Studies on contaminant sensitivity of hydraulic seals are in their infancy and represent an area in which major contributions to the fluid power industry could be made. Investigations into the parameters affecting the contaminant sensitivity of seals would also lead to a better understanding of the various phenomena involved in the sealing process.

The primary target area for future studies should be the evaluation of the properties of the seal's deterioration constant (K_2'). This constant is, of course, different for each type of seal, and analysis of the relationship of the properties governing the value of K_2' is a key consideration. The results presented in this report should provide the foundation for many new studies in the future.

BIBLIOGRAPHY

1. Theyse, F. H., N. V. Warkspoor, Amsterdam, Delft University of Technology, Netherlands. "The Inverse Hydrodynamic Theory and Its Application in the Design of Controlled Leakage Seals Between Moving Parts." Proceedings of the Third International Conference on Fluid Sealing, British Hydromechanics Research Association, Cambridge, England (1967), Sec. F2, pp. 17-32.

2. Dowson, D., and P. D. Swales, Department of Mechanical Engineering, University of Leeds. "An Elastohydrodynamic Approach to the Problem of the Reciprocating Seal." Proceedings of the Third International Conference on Fluid Sealing, British Hydromechanics Research Association, Cambridge, England (1967), Sec. F3, pp. 33-44.

3. Denny, D. F. "Leakage Characteristics of Rubber Seals Fitted to Reciprocating Shafts." Paper 22, Proceedings of the Conference on Oil Hydraulic Power Transmission and Control. I. Mech. E. (1961), pp. 259-268.

4. Blok, H., and J. Koens. "The 'Breathing' Film Between a Flexible Seal and a Reciprocating Rod." Research Report, Proccedings I. Mech. E., Vol. 180, pt. 3B (1965), pp. 221-223.

5. Greenwood, J. A. "The Area of Contact Between Rough Surfaces and Flats." Transactions of the American Society of Mechanical Engineers, 89, Ser. F, 1 (January, 1967), pp. 81-91.

6. Report of the Research Department of the Institution of Production Engineers (Dr. George Schlesinger, Director), January, 1942.

7. Roth, Dr. A. "Sealometry and Sealography." Proceedings of the Third International Conference on Fluid Seals, (Soreq Nuclear Research Center, Israel). British Hydromechanics Research Association, Cambridge, England, Sec. C2 (1967), pp. 9-23.

8. Basic Fluid Power Research Program, Component Wear, Annual Report No. 5, Sec. 71-5 (July, 1971).

9. Finkin, E. F. "Mechanical Wear and Fluid Power Systems." Lecture to National Fluid Power Association, 1969 Annual Convention.

10. Lomakin, V. S. "Investigation of Wear of Sleeves and Piston Rings of Mud Pumps." Friction and Wear in Machinery. Transactions of the American Society of Mechanical Engineers, Vol. 11 (1956), pp. 39-68.

11. Thompson, Robert A., and William Bocchi. "A Model for Asperity Load Sharing in Lubricated Contacts." Presented as an ASLE Paper at the ASME/ASLE Lubrication Confrence, Pittsburg, Pa. (October 5-7, 1971).

12. Parker, Jerald D., James H. Boggs, and Edward F. Blick. Introduction to Fluid Mechanics and Heat Transfer, Copyright 1969. Reading, Mass.: Addison-Wesley Publishing Co., Inc.

13. Fitch, E. C., Professor, Mechanical Engineering, Oklahoma State University. "A Basic Science Program in Filtration Mechanics," Oklahoma State University Engineering Research Bulletin, Part No. 126 (November, 1962).

14. Recommended Procedures for Evaluating Fluid Power Components and Systems. FPRC No. 72.1 prepared by OSU-MENDC Hydraulics Program Personnel (March, 1972), Fluid Power Research Center, Oklahoma State Univ., pp. 228-230.

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