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TABLE OF CONTENTS

LIST OF FIGURES

TABLE

NOMENCLATURE

A	Area
b	Shaft circumference
h	Water film thickness
H	Head
K	Constant, conversion from heat power to mechanical power
l	Seal length
P	Pressure
Q	Volumetric seal leakage
t	Temperature
v	Surface velocity of shaft
μ	Viscosity, absolute
ρ	Weight density

ABSTRACT

A sealing device utilizing water ice to form a very close-fitting liquid water capillary seal about a rotating shaft has been designed, built, and tested. The experimental shaft diameter was 6 in. (15.24 cm) and it rotated at 600 rpm. The observed leakage rates were below 0.05 in^3/sec (0.8 cm^3/s) for differential pressures up to 600 psi (4150 kPa). The maximum power required to drive the shaft was less than 500 W. A set of basic equations was developed to predict the seal performance. Experimental results indicate the basic equations are sufficiently accurate for future seal designs.

ADMINISTRATIVE INFORMATION

The work reported herein was carried out by the Propulsion and Auxiliary Systems Department under Element 61152N, Task Area ZR 0220601, Work Unit 2723-152.

INTERNATIONAL SYSTEM OF UNITS

The International System of Units is used in this report. Where English units are used in addition to the SI units for clarification, they are in closed brackets.

INTRODUCTION

Ice has particular qualities which are considered exploitable in naval applications. The availability of the raw material (seawater) is one of these qualities. The process of forming it at the point of its engineering use merely by removing heat is a second quality. Its application as a very low friction sliding bearing is well known to anyone who has skied or slid down ice-covered steps in winter time. Other characteristics that indicate exploitation are its resistance to heat transfer, its electrical resistance, its hardness, its strength, and its being fireproof.

To introduce the Navy to the possibility of exploiting ice as an engineering material a mechanism was devised to demonstrate some of its particular capabilities and, in addition, to show how it might have shipboard applicability. A capillary shaft seal was proposed as the mechanism for development since not only does it provide a seal against entrance of water but it incorporates many features of journal bearings.

This report describes the design and operation of one type of ice seal, along with a discussion of the ancillary refrigeration equipment required for this device. Metric units are used throughout this report followed in most cases with English units. When English units are presented they are enclosed in parenthesis.

SEAL TEST DESIGN AND CONSTRUCTION

PRIMARY PARTS

The seal that was constructed, as illustrated in Figure 1, consists of six primary parts.

Part 1 - Pressure Housing

This part is made of mild steel in two pieces and contains water which would be seawater in an actual application. One opening on the top of the housing is for connection to a water supply and the other is fitted with a pressure gage. One hole at the bottom of this housing is plugged, the other fitted with a drain line and valve, which is closed during operation.

Part 2 - Insulator and Bearing

The insulator is made of Plexiglas and restricts heat flow to the seal housing, part 3, from the pressure housing, part 1. It further acts as a bearing during operation without ice. It is 2.5 cm (1 in.) in length, 15.24 cm (6 in.) in bore diameter, and the shaft supports about 295 N (66 lb) rotating at about 600 rpm. The radial clearance between the insulator startup bearing and the shaft is 0.127 mm (0.005 in.) The minimum film thickness during operation is calculated to be 0.01397 mm (0.00055 in.). Variation in coefficient of thermal expansion between the shaft material and the Plexiglas of this insulator was accounted for in the design.

Part 3 - Seal Housing

The seal housing is composed of a thick-walled bronze sleeve with a helical groove cut in its outer surface. A second part of the seal housing is composed of a thin-walled bronze sleeve which fits snugly over the grooves in the thick sleeve. The outer sleeve is fitted with two pipe connections which line up when assembled with the beginning and end of the helical groove in the inner sleeve. The inner surface of the thick-walled sleeve is threaded. When assembled, a radial clearance of 3.175 mm (0.125 in.) exists between the seal housing and shaft. It is in this gap, 3.175 mm (0.125 in.) thick, 15.24 cm (6 in.) in diameter, and 22.86 cm (9 in.) in length, that the water-sealing ice forms. The fine threads on the seal housing inner surface are for gripping the ice that forms there.

Part 4 - Insulator

This insulator was provided to restrict heat transfer between the seal housing and the leakage collection system, part 5. It is made of Plexiglas with an axial length of 2.54 cm (1 in.) and a radial clearance of 3.175 mm (0.125 in.) with the shaft.

Part 5 - Leakage Collector

The collector is bolted with the insulator, a gasket, and O-ring to the seal housing. It is made of mild steel, has a 3.175 mm (0.125 in.) radial clearance with the shaft, and is fitted with a drain hose which carries leakage to a measuring bucket.

Part 6 - Shaft

The shaft, made of stainless steel, is hollow with a 15.2 cm (6 in.) O.D. and a 12.42 cm (4.89 in.) I.D., and has a polished surface. Each end of the shaft was plugged and one end was fitted with a 2.54 cm (1 in.) diameter stub extension for driving.

ASSEMBLY

To assemble the stationary parts, all the pieces making up the right-hand half of the assembly were bolted to the right-hand seal housing. All the pieces making up the left-hand half of the assembly were bolted to the left-hand seal housing. The two halves of the pressure housing were then bolted together completing the assembly of the stationary parts. All of the bolts going through the insulators were lightly tightened to ensure that excessive stress would not occur by constraining relative deflection due to variation in coefficients of thermal expansion of the various materials. Insertion of the shaft completed the assembly.

SUPPORT EQUIPMENT

Figure 2 is a schematic of the test facility showing the support equipment. A more detailed schematic of the glycol mixture cooler is shown in Figure 3. A description of the parts shown in Figure 3 is tabulated in Table 1.

TABLE 1 - PARTS LIST FOR GLYCOL COOLER

Item	Description
A	Model WH15LE Water-Cooled Condensing Unit, 208/3/60, for use with Refrigerant-502. Complete with compressor, water-cooled condenser, water regulating valve, dual pressure switch, receiver, and liquid shutoff valve. Manufactured by Dunham-Bush
B	Liquid drier, Sporlan Model C-163
C	Contactor, 3-pole, 30 amp, 208 V collector
D	Sight glass, Sporlan Model SH-13FA1
E	Expansion valve, Sporlan Model FGE-1ZP35
F	Solenoid valve, Sporlan Model B6F1, 208/230 V
G	Heat exchanger, Dunham-Bush Model B200XS
H	CPR valve, Sporlan Model CROT-6, 7/8 in., 0/60 psig
I	Hot gas regulator, Sporlan Model ADRSE-2, 1/2 in., 0/30 psig
J	Hot gas solenoid valve, Sporlan Model B9S2, 208/230 V
K	Thermostat, White Rodgers Model 1609-101
L	Evaporator, Dunham-Bush Model HX-40
M	Pump, Lancaster Model M-903, 230/1/60, 1/2 hp

Coolant System

As shown in Figure 2 there are two water-glycol mixture coolers. Valving is provided such that the coolers can operate in parallel, split (i.e., each cooler providing for one seal), or one cooler providing both seals. Figure 2 shows the split mode valve settings. A thermostatic switch is available to regulate the temperature of the water-glycol mixture to within about 3 C (37.4 F) by cycling on and off a Freon compression unit.

Water Supply to Pressure Housing

Seawater is provided by a motor-driven piston displacement pump with a capability of providing about 12.5×10^{-5} m^3/s (2 gal/min) up to 4136 kPa (600 psi). The pump takes its suction from a 0.0038 m^3 (10 gal) capacity sump. The pump, being a piston type, acts as a check valve when not being driven. It is operated by an on-off switch or by a variable set point pressure switch in the pump discharge line. To reduce the frequency of starts and stops on the pump, the pump discharge line is fitted with a 0.038 m^3 (1 gal) accumulator. Connected into the pump discharge line is the pressure housing supply line with a needle-type valve. The pump discharge line and the pressure housing are fitted with pressure gages. The flow path of the water is from the sump to the accumulator, from the accumulator to the pressure housing, through the seals to the collection system, and back through transparent hoses to the sump. During leakage measurement, the leakage is diverted to graduated containers at the sump.

Mechanical Drive

The seal shaft is driven by a 2 hp, 1750 rpm motor. For full speed, i.e., about 600 rpm seal shaft speed, the motor is fitted with a 0.102 mm (4 in.) V-belt pulley sheave. This sheave drives a 30.5 cm (12 in.) V-belt pulley sheave mounted on a shaft supported on either side in ball bearing pillow blocks. Rotation of the seal shaft is attained via the 30.5 cm (12 in.) sheave shaft, a paraflex coupling, a BLH torquemeter, a second paraflex coupling, and finally, the seal shaft stub extension.

SEAL OPERATION

START-UP PROCEDURE

The procedure used for starting is as follows:

1. Set the coolant valving for split, parallel, or one unit serving both seals.

2. Set the thermostats to operate at a maximum coolant temperature of about -3 C (37.4 F) below the freezing point of the supply water when both cooling units are used and at maximum cooling capacity when only one unit is used.

3. Turn on the cooling unit or units, depending on operating mode.

4. Turn the pump unit on and off to maintain pressure between 689 and 4136 kPa (100 and 600 psi) in the accumulator.

5. Throttle down the needle valve in the supply line until a trickle of a few cubic centimeters per minute comes out the leakage lines, enough to provide a little lubrication to the Plexiglas start-up bearings.

6. The drive motor may be started at this time, but preferably should be started as the coolant supply temperature approaches the freezing temperature of the supply water.

7. After the drive motor has been started and the coolant temperature drops below freezing, ice begins to form in the gap between the seal housing and shaft. Rather suddenly the leakage decreases to a negligible amount, and the pressure reading at the pressure housing rapidly rises to equal that measured at the supply water accumulator. The seals are now fully operational. In several minutes they operate in a relatively steady state with the Freon compressors cycling to maintain the coolant temperature if both cooling units are being used and the water supply pump is cycling to maintain the supply pressure. If only one cooling unit is supplying both seals and the thermostatic switch is set at maximum cooling, then no cycling of the Freon compression unit will occur and a steady state condition occurs between -3 and -13 C (26.6 and 55.4 F).

Data were taken for the set conditions as follows:

1. Arrange the cooling system so that only one unit supplies coolant to both seals.

2. Set the thermostatic switch at its minimum setting to enable the unit to run continuously.

3. Maintain the water pressure in the pressure housing at the desired value by turning the supply pump on and off manually. Automatic pressure control by using the pressure switch in the supply line was too coarse for use in setting data points.

4. Record the data when a steady state condition exists, i.e., no temperature difference in any location over a 10-min period.

5. Set the next point by throttling down the water-glycol coolant flow to the seals. When the steady state is reached, record the data for the new setting.

6. Set the successive points by successive throttling down of the coolant until leakage becomes excessive.

The data taken were torque, leakage, coolant temperature to and from each seal, water supply temperature, pressure housing temperature, and temperature of each seal housing in two places.

The torque per seal was obtained by reading the level of torque on the torquemeter each minute for a period of 10 min, determining the average, and diving by two because the meter read the torque for both seals. The torque readings could not vary by more than a percent from the average torque during the 10-min period for the data to be acceptable.

The leakage was measured over a 10-min period also and the amount divided in two.

The thermocouples were read on a digital recorder which gave readings correct to 1 C. Steady state was considered to exist when these readings remained constant for the 10-min period. This means that the actual temperature at any location during a data recording period could have varied plus or minus 1/2 C.

SHUT-DOWN PROCEDURE

If the unit is to be shut down for a long period of time (in excess of 1 hr) it makes little difference in what sequence the equipment is secured. If the downtime is to be kept to a minimum, the following procedure is followed in order to ensure that the ice does not freeze to the shaft.

1. Turn off the cooling unit.

2. When leakage becomes significant, turn off the shaft drive motor.

3. To return to service, restart the shaft drive motor and cooling units.

PERFORMANCE

Calculation of the anticipated performance is developed from the idealized seal shown in Figure 4. The volumetric flow through a slot has been derived[1*] as:

$$Q = \Delta P\ b\ \frac{h^3}{12\mu\ell} \tag{1}$$

This leakage results from three sources of heat which must be removed by the refrigeration system. These are (1) the flow power, (2) the viscous power, and (3) the leakage cooling power. The flow power is:

$$\text{Flow Power} = Q\rho\Delta H \tag{2}$$

where ρ is the weight-to-volume ratio of water at the freezing point. The viscous power is by definition[1] $= \mu A\ v^2/h$. Substituting $h = \sqrt[3]{12\mu\ \ell\ Q/\Delta P\ b}$ from Equation (1) into this yields:

$$\text{Viscous Power} = \mu A\ v^2\ \sqrt[3]{\Delta P\ \frac{b}{12\mu\ell}\ Q} \tag{3}$$

The steady power absorbed to cool the leakage down to the freezing point is:

$$\text{Leakage cooling power} = Q\rho tK \tag{4}$$

where K is a conversion factor from heat to mechanical power.

Applying Equations (1) through (4) results in the values presented in Figures 5a through 5d when the values listed below are used.

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*The reference appears on page 26.

$$\rho = 10{,}050 \text{ N/m}^3 \qquad (0.037 \text{ lb/in}^3)$$

$$\ell = 0.2286 \text{ m} \qquad (9 \text{ in.})$$

$$b = 0.4788 \text{ m} \qquad (18.85 \text{ in.})$$

$$A = 0.1095 \text{ m}^2 \qquad (169.65 \text{ in}^2)$$

$$\mu = 17{,}925 \text{ N s/m}^2 \qquad (2.6 \times 10^{-7} \text{ lb sec/in}^2)$$

$$v = 4.65 \text{ m/s} \qquad (183.3 \text{ in/sec})$$

$$\Delta T = 22.2 \text{ C} \qquad (40 \text{ F})$$

Figure 6 results in summing up the powers in Equations (2), (3), and (4). Figures 7a through 7d and Figure 8 provide these results in English units.

RESULTS

The presentation of the experimental data for operation of the ice seal in seawater, as shown in Figures 9 and 10, compares quite well qualitatively with that predicted, even though there is quantitative variation which appears to increase with the pressure level at which the seal is operating.

No in-depth explanation for this variation can be advanced at this time due to the limited experience with the test apparatus, however, several points might be noted. The theory on which the predicted curves are based considered uniform film thickness over the entire area of the seal surface. A very preliminary finite element analysis of the problems indicates that this is not the exact case. The finite element analysis indicated that the water film is thicker on the upstream end of the seal. This is due to the thermal shape of the seal housing and because the upstream end of the seal is called upon to provide heat removal for the cooling down to the freezing temperature of the leakage. The finite element analysis predicted slightly higher viscous loss than did the approximate mathematical model used in this report. Full development of this finite element model may tend to reduce the quantitative variation between experimental results and theoretical predictions.

16

Comparison of the plotted data of Figures 9 and 10 with the curves of Figures 5a and 7a shows that the film thickness was approximately 0.01 mm (0.0004 in.) when 450 W or 0.536 shp are input, and represented the minimum film thickness obtained with the facility operating with only one cooling unit. With two cooling units in service it was possible to stall the drive motor at all pressure levels. To do so, since the motor can provide only 746 W (1 hp) per seal, means that the film thickness must have been on the order of 0.00635 mm (0.00025 in.). At these very small film thicknesses, just before stalling the motor, rubbing noises were heard.

To restart the shaft after a very short stoppage time, a procedure was developed whereby the cooler was completely shut off and the shaft continued to be rotated until leakage increased to about 0.5 to 0.6 cm^3/s (0.03 to 0.04 in^3/sec). It was then possible to shut down the shaft and restart it while maintaining pressure. Two to three minutes elapsed between the time the cooling unit was turned off and the time at which the leakage reached the rate above. Stopping the shaft while the cooling unit was left on resulted in a condition where it was impossible to restart until the cooler was shut off. After shutoff, pressure remained at the operating level prior to stopping the shaft for about 15 minutes; within another 15 minutes the shaft could be restarted, the cooler restarted, and the ice that melted, reformed.

The predicted curves of Figures 6 and 8 show a minimum power point defined at a particular leakage rate. Stable operation was only possible to the left of this point.

The seals operated for about 100 hr at various leakage rates and pressures from 0 to 4500 kPa (0 to 650 psi), the majority of the time being at the data conditions of 690, 1725, and 3450 kPa (100, 250, and 500 psi). The only failure to any item during the entire operation was the saltwater postive displacement pump which maintained simulated sea pressure. The failure resulted from starting the pump with the discharge valve closed causing blowout of the the O-ring seals.

DISCUSSION

Although this exercise showed that ice could be used to seal the gap between a rotating shaft and stationary penetration, it did not and was not planned to demonstrate a fully developed seal suitable for Fleet installation. It does show that the engineering of such a seal can be done with a great deal of confidence. There are several areas of concern for using such a seal. The first is the question of having to require cooling power at all times to the seal unit, i.e., it is not a passive system, and is, therefore, subject to power failure. A second is the question of freezing up in the the shaft stopped condition. A third is the control of film thickness at low speed. A fourth question is the reliability of the various components making up the glycol cooling system.

In discussing the first question, a review of the power requirements must be made. The rate of thermal energy which must be removed from the seal is:

$$\text{Power (thermal) removed} = Q\,P + \mu A v^2\, \sqrt[3]{\Delta P \frac{b}{12\mu} \ell Q} + Q\rho t K + \text{conduction loss} \qquad (5)$$

The first three terms are from Equations (2), (3), and (4). The conduction loss term represents the requirement to overcome heat passing in through the insulation surrounding the seal and the heat passing to the leakage water film from the rotating surface. To have a safe stable system, the energy removal rate must be above the minimum power point shown on the predicted curves of Figures 6 and 8. The experimental results indicated no problem with powers two times the minimum. The flow at the minimum power point is

$$Q = \left[\frac{1}{3}\,\mu A v^2\, \sqrt[3]{\Delta P \frac{b}{12\mu\ell}} \Big/ \Delta P + \rho \Delta t K\right]^{3/4} \qquad (6)$$

For the model, this results in a value of 1 cm^3/s (0.061 in^3/sec) at the full pressure condition. Using this value in Equation (5) yields 393 W (3485 in-lb/sec); 1344 Btu/hr + conductive loss is the minimum power requirement. Using a multiplier of two and adding 10 percent as an estimate for the conduction loss yields a refrigeration requirement of 825 W (7319 in-lb/sec), 2822 Bru/hr, or 1.1 hp. This was obtained in the model test by using half the capacity of a 1119 W (1-1/2 hp) motor-driven refrigeration unit. A unit three times the diameter and one-third the rotational speed would require a 3357 W (2.25 hp) motor-driven refrigeration unit. In neither case is this a significant amount of power. The power could be mechanically extracted directly from the propeller shaft. As an aside, it might be noted that the refrigeration requirements of the model, 825 W (2822 Btu/hr), could be met for 3-1/2 hours of operation at full speed and pressure with a 0.208 m^3 (55 gal) drum of water-glycol mix initially at -18 C (0 F) and a hand-powered circulation system.

The second question is a specific point in the general performance of the seal at various speeds. Figures 6 and 8 show some curves which approximate the relationship of power versus flow at 500 rpm at a variety of pressures for the model tested. It can be seen that there is a minimum power point on these curves occurring at a flow rate approximated by Equation (6). Figure 11 is a plot of these minimum powers (excluding the conduction loss) versus speed, and Figure 12 is a plot of shaft torque required at these minimum power points. Stable steady state operation is always to the left of the minimum point requiring more power than shown on Figure 11 except at zero speed. At zero speed, and even very close to it, control is difficult. At zero speed, the minimum power point exists at zero flow. Operation must be to the right of the minimum power required or in the unstable region, or the shaft will freeze up because of rapid cooling when the excessive heat is removed. Near zero speed, it would be extremely difficult to control the exact requirements of heat removal. For instance, at 1 rpm, applying Equations (4) and (5) yields a minimum power of 0.000306 W (0.002707 in-lb/sec) at a flow rate of 29.33 x 10^{-7} cm^3/s (1.79 x 10^{-7} in^3/sec) excluding conduction loss. This results in a torque of 0.00022 Nm (0.00196 in-lb). If 100 W more than this were removed, it would result in an increase in torque to 71.9 Nm (640 in-lb) which shows that very large torque can be absorbed with little increase in cooling. It is possible but more difficult to operate in the unstable region by using the time delay built into the system. This is done by intermittently stopping and starting the flow of coolant to the seals. The leakage rate may not be allowed to exceed a point on the curve of Figures 6 and 8 where the power removal required by the coolant exceeds the capacity of the coolant system. This was the method used to permit start up after short duration shutdowns. In addition to controlling the cooling, heating the shaft (or controlling the conduction loss terms of Equation (5)) would improve low-speed operation and startup.

The question of reliability of a sealing system can only be answered from the experience of approximately 100 hr of failure-free operation. This is not a long period of time and cannot certainly eliminate the concern for the very long period of time that would be required for a real seal system. The heart of the system is a Freon compression unit and a water circulating pump, both of which could be provided with redundancy. In addition, a little storage capacity for the coolant provides a great deal of system inertia as stated previously on page 12.

The ice shear stress at the surface between the ice and seal housing was 48.6 kPa (7 psi). This is very conservative for ice loading which is quoted as having a shear strength on the order of 70 to 150 psi. The cost of purchasing the cooling units and other material required for the model seal was about $7,000. To assemble them, in the form the concept was evaluated, cost an additional $20,000.

CONCLUSIONS

The concept of using ice to seal the space between the rotating shaft and a stationary hull penetration has been demonstrated. Equations required to design such a seal have been developed. Seals of the type evaluated limit leakage more than the seals presently employed. The ice seal requires a constant supply of coolant at about 5 to 10 C below the freezing point of the fluid being sealed. Further work is needed to prove the feasibility of this concept for applications of naval interest.

21

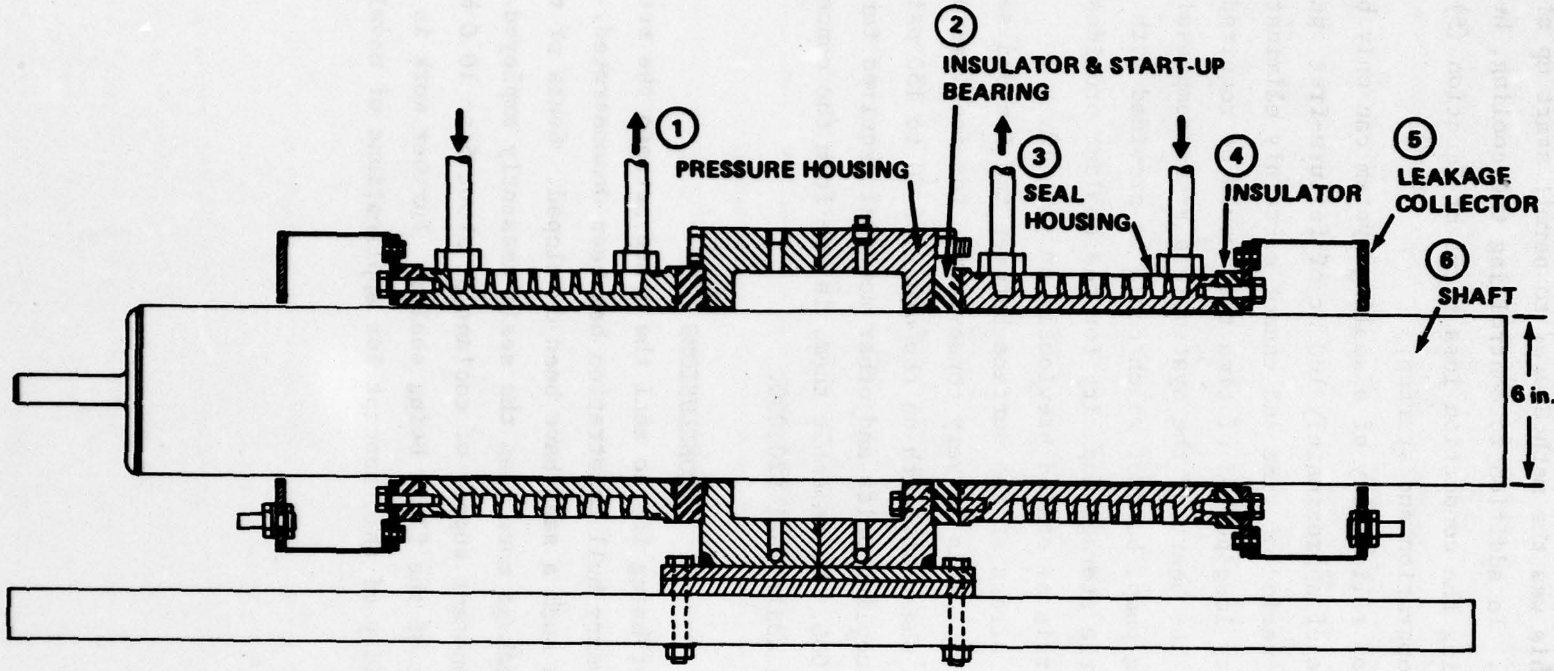


Figure 1 - Six-Inch Ice Seal Test Assembly

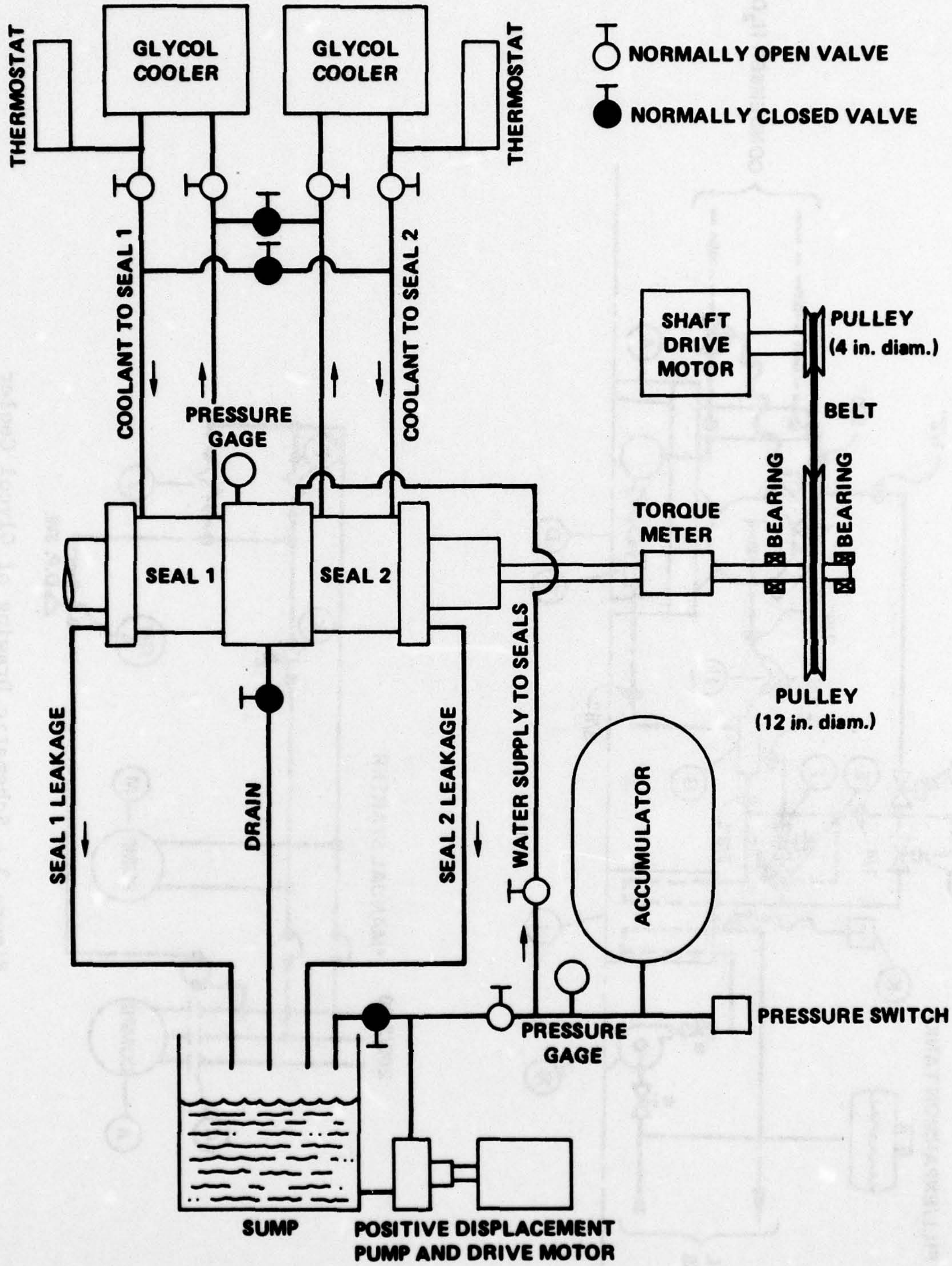


Figure 2 - Ice Seal Test Facility: Schematic Diagram

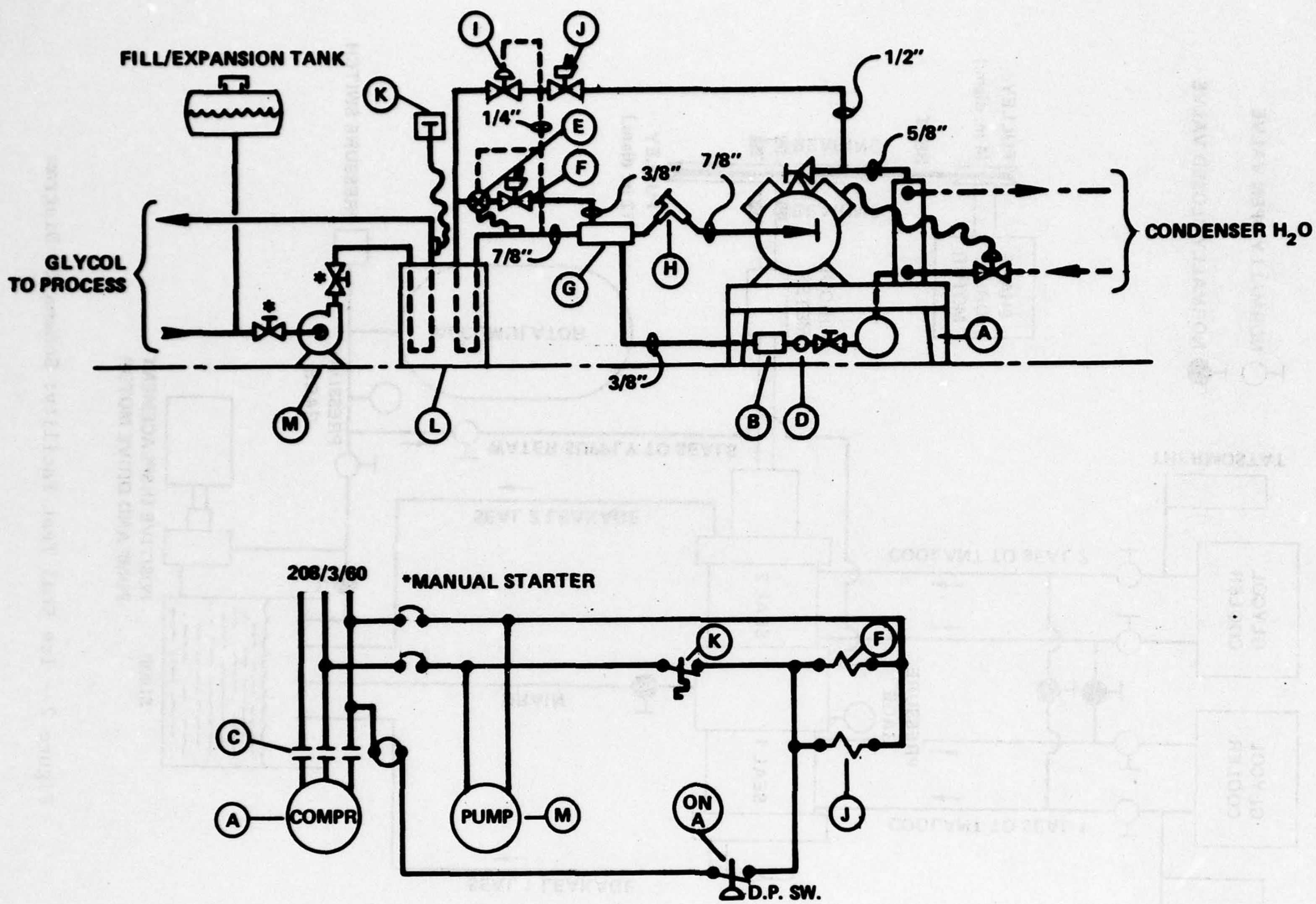
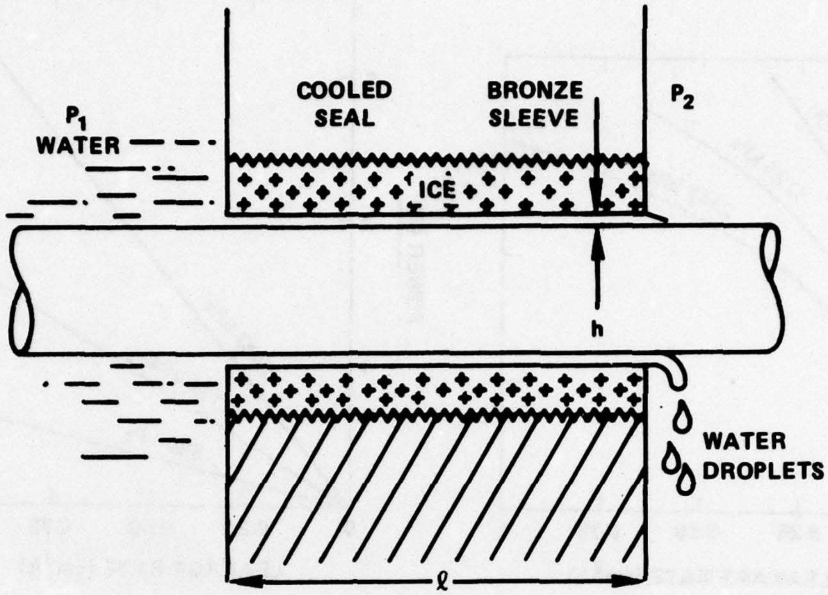


Figure 3 - Schematic Drawing of Glycol Cooler



P_1	PRESSURE OF WATER
P_2	AMBIENT ROOM PRESSURE
ΔP	P_1-P_2
b	CIRCUMFERENCE OF SHAFT
Q	VOLUMETRIC LEAKAGE
h	LEAKAGE FILM HEIGHT FORMED BY FRICTIONAL HEATING
μ	AVERAGE VISCOSITY OF WATER FORMED IN THE GAP
ΔH	PRESSURE DIFFERENCE CONVERTED TO m (OR in.) OF HEAD OF WATER
A	SURFACE AREA OF THE SEAL
v	SHAFT SURFACE VELOCITY
ΔT	DIFFERENCE IN TEMPERATURE BETWEEN THE WATER SUPPLIED AT P_1 AND ITS FREEZING POINT TEMPERATURE
ℓ	SEAL LENGTH

Figure 4 - An Idealized Ice Seal

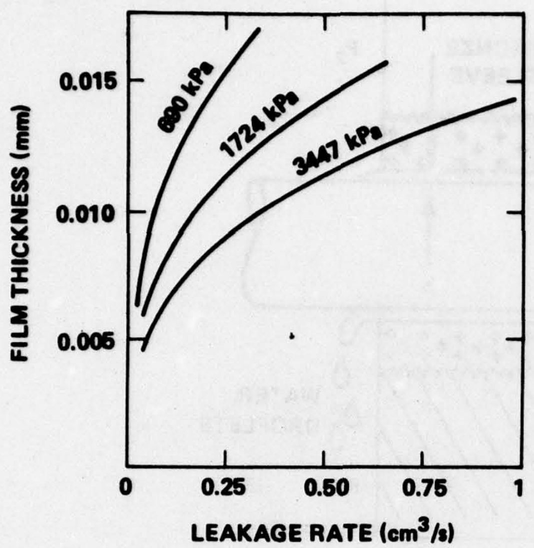


Figure 5a - Leakage versus Film Thickness per Equation (1)

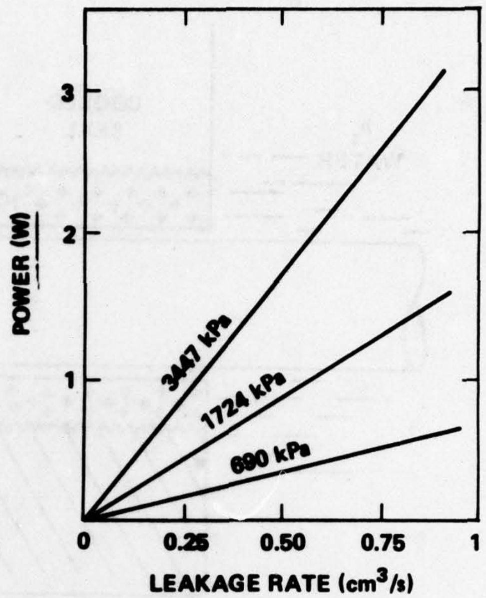


Figure 5b - Flow Power versus Leakage Rate per Equation (2)

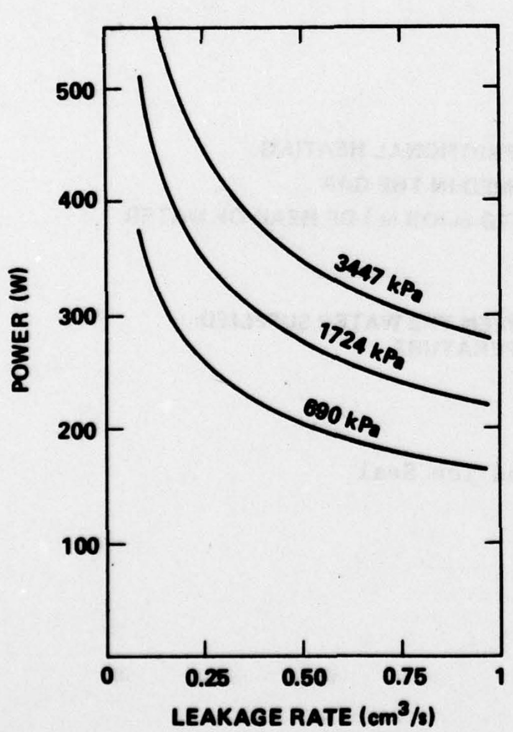


Figure 5c - Viscous Power versus Leakage Rate per Equation (3)

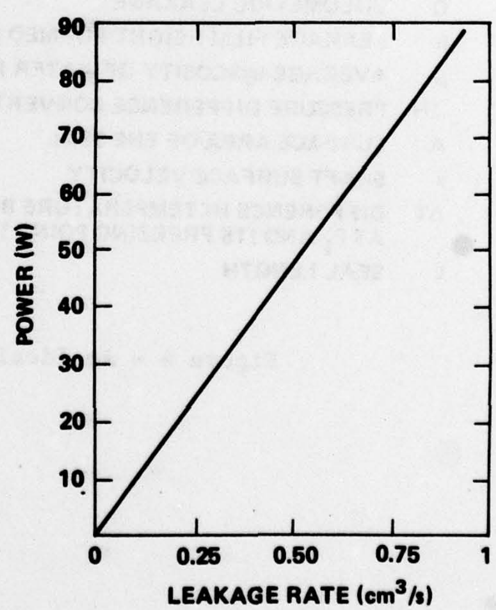


Figure 5d - Leakage Cooling Power versus Leakage Rate per Equation (4)

Figure 5 - Leakage Rate

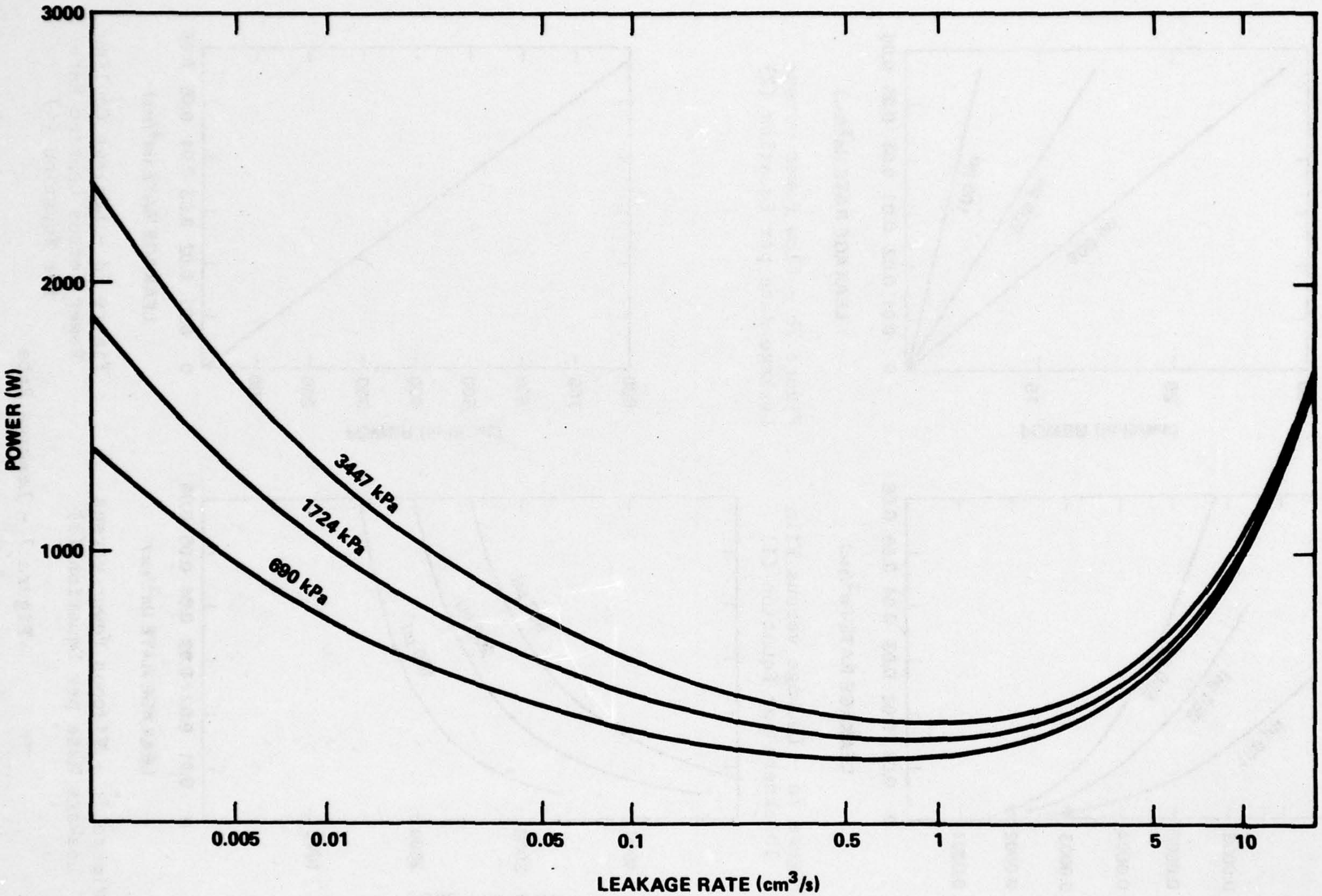


Figure 6 - Power Removed by Coolant versus Leakage Rate

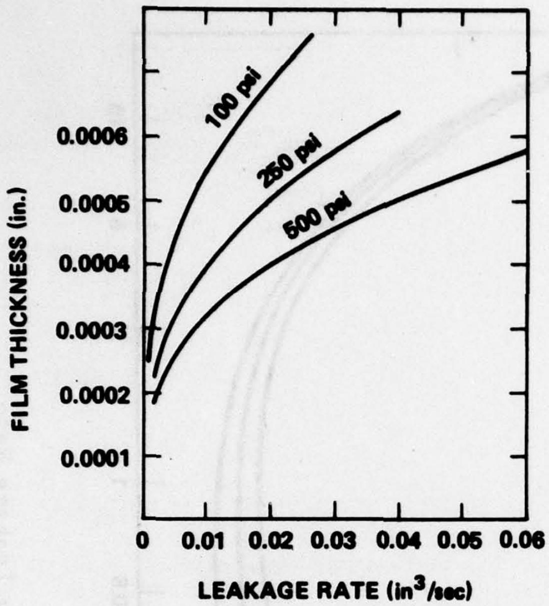


Figure 7a - Leakage versus Film Thickness per Equation (1)

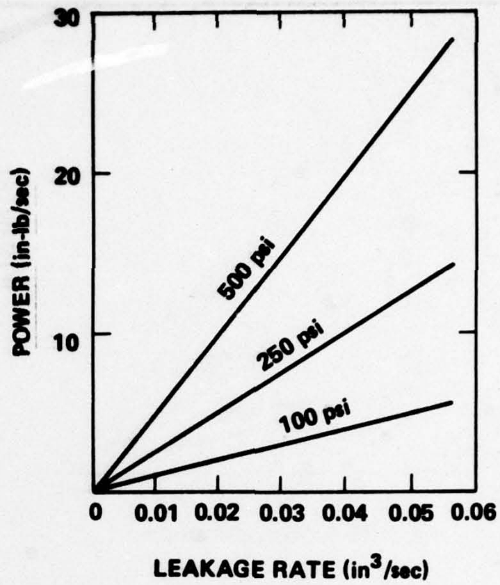


Figure 7b - Flow Power versus Leakage Rate per Equation (2)

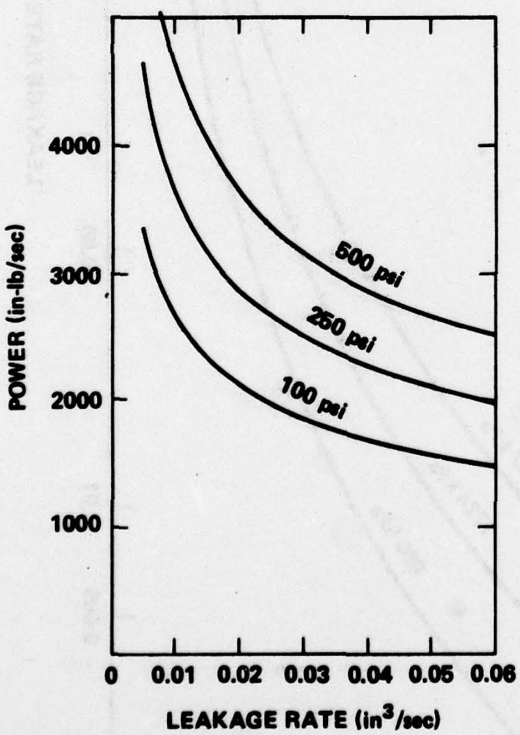


Figure 7c - Viscous Power versus Leakage Rate per Equation (3)

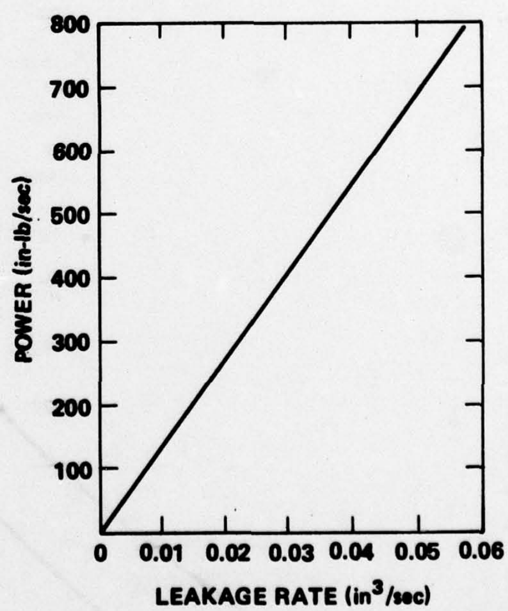


Figure 7d - Leakage Cooling Power versus Leakage Rate per Equation (4)

Figure 7 - Leakage Rate

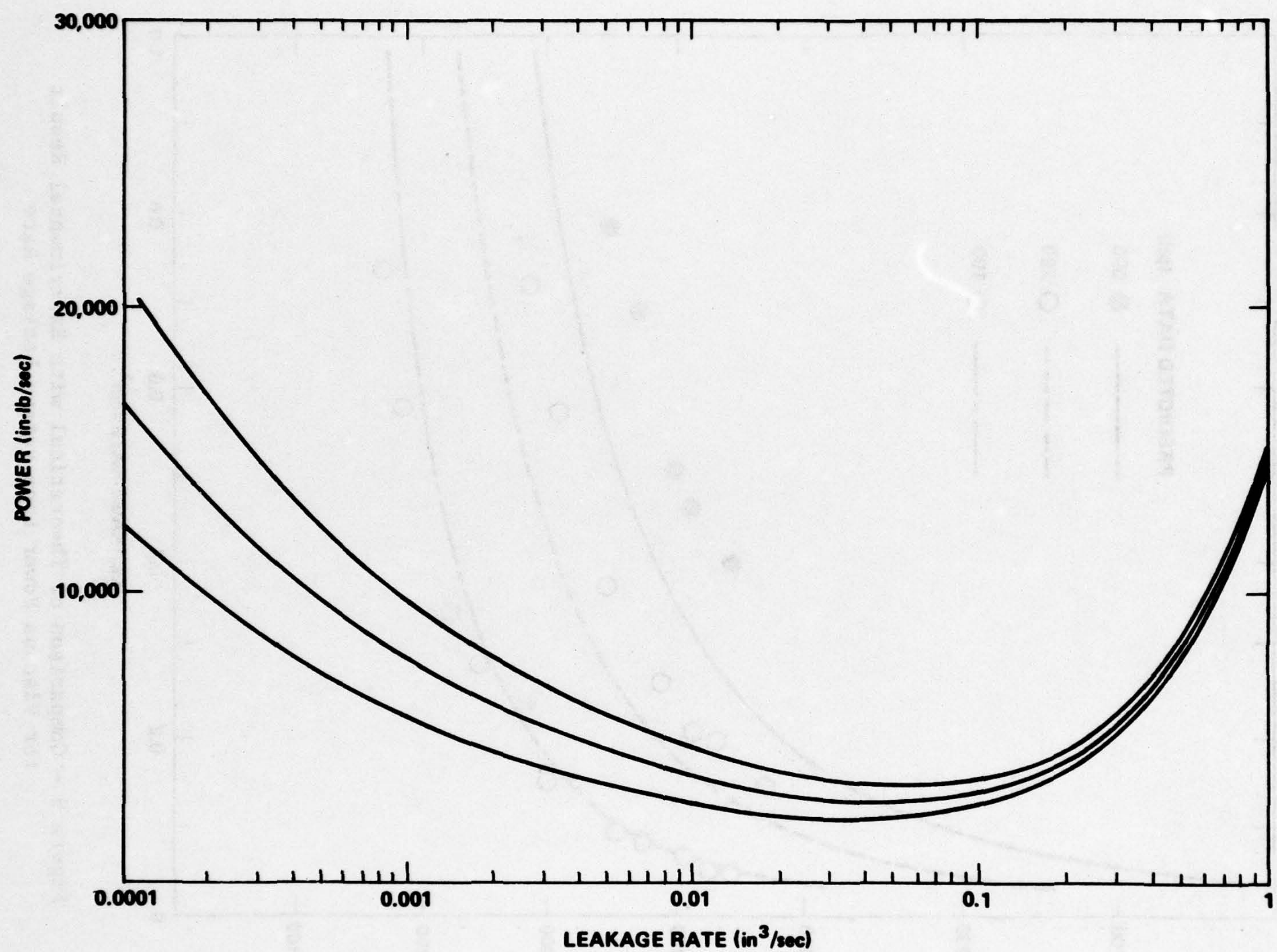


Figure 8 - Power Removed by Coolant versus Leakage Rate

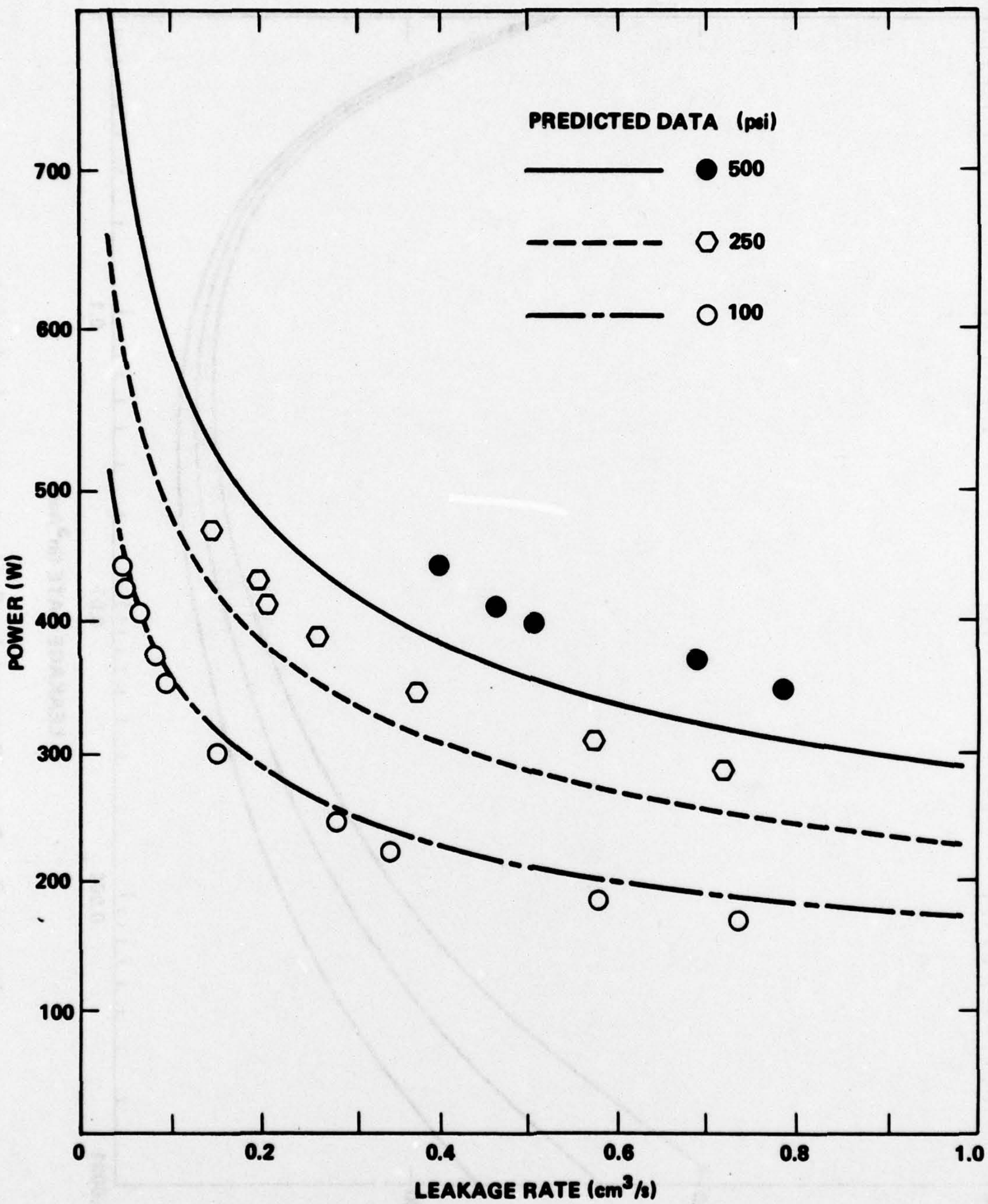


Figure 9 - Comparison of Theoretical with Experimental Result for Viscous Power Loss versus Leakage Rate

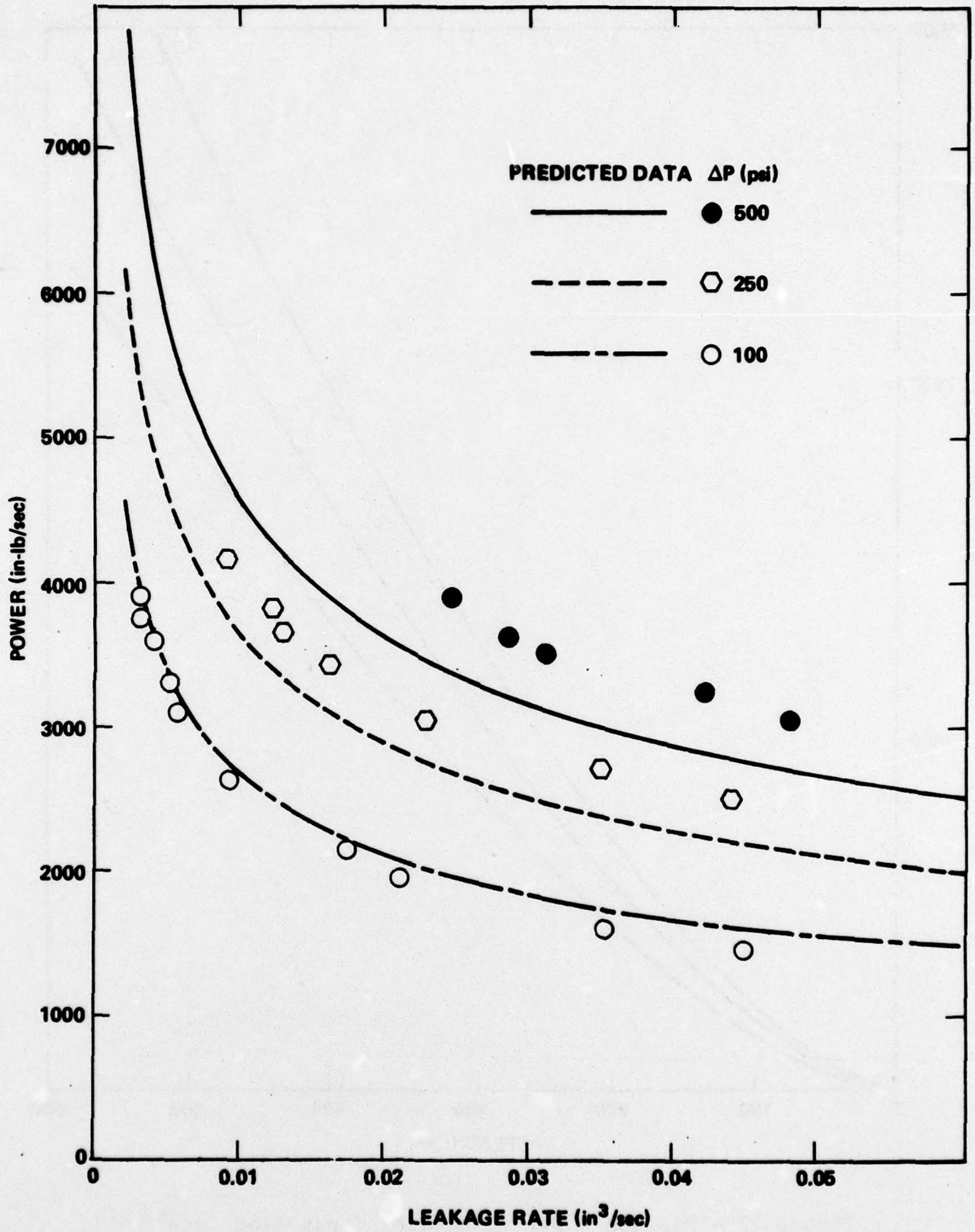


Figure 10 - Comparison of Theoretical Prediction with Experimental Results for Viscous Power Loss versus Leakage Rate

30

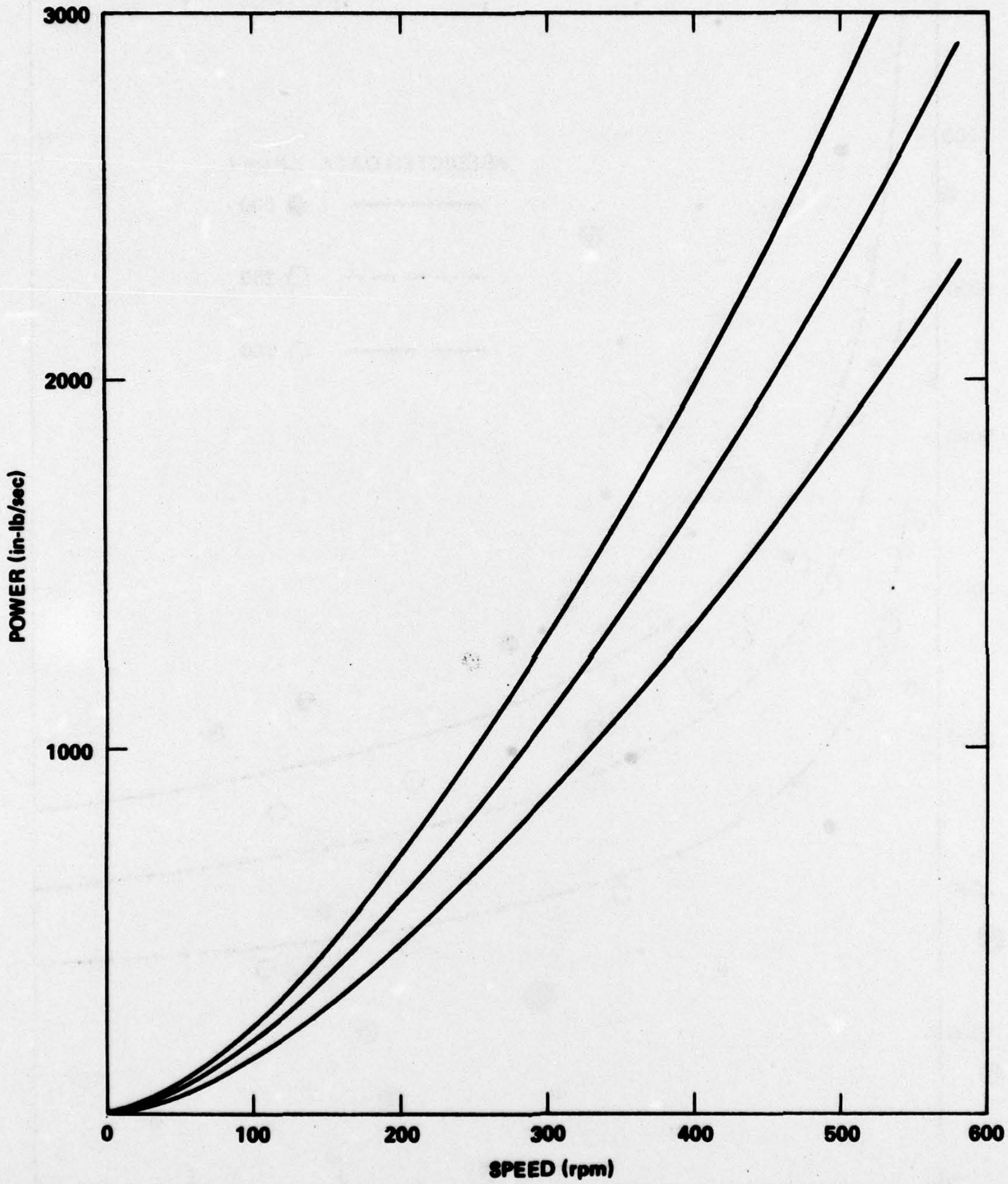


Figure 11 - Minimum Power (Excluding Conduction Loss) versus Speed for a Variety of Sealing Pressures

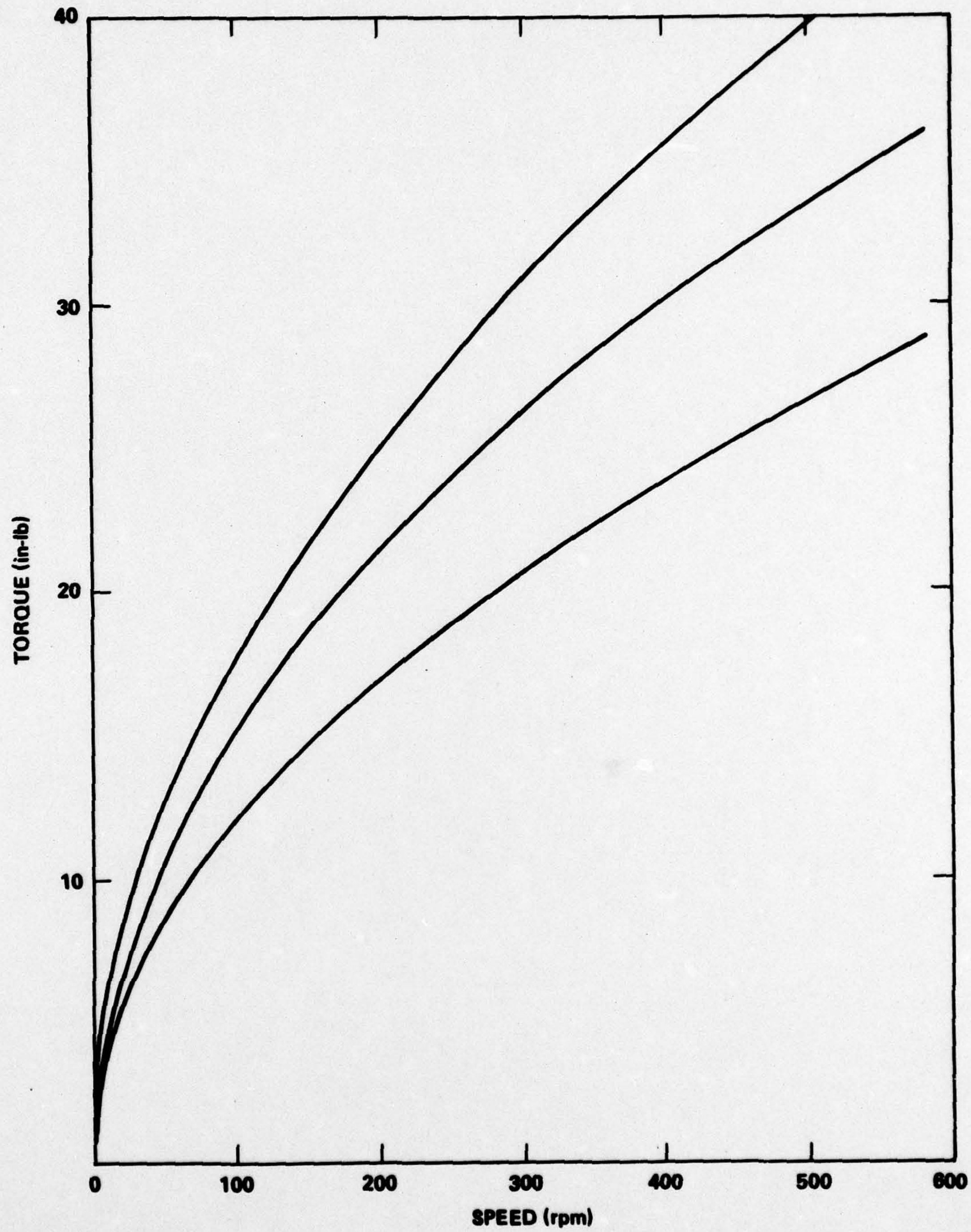


Figure 12 - Minimum Torque versus Speed for a Variety of Sealing Pressures

REFERENCE

1. Fuller, D. D., "Theory and Practice of Lubrication for Engineers," John Wiley and Sons, Inc., New York (1956).

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