ME 65-587-3

The University of Tennessee
Department of Mechanical and Aerospace Engineering

THEORETICAL AND EXPERIMENTAL STUDIES OF VISCO-TYPE SHAFT SEALS

Semi-annual Progress Report
October 15, 1964 - April 15, 1965

by

William K. Stair

Investigation conducted for the
National Aeronautics and Space Administration
under
Research Grant NsG-587

Reproduction in whole or in part is permitted
for any purpose of the U.S. Government

May 21, 1965

Knoxville, Tennessee

ABSTRACT

/2/99

This report, the third of a series, outlines the progress made on an analytical and experimental investigation of the visco-type shaft seal. This research is being conducted under Research Grant NsG-587 sponsored by The National Aeronautics and Space Administration.

A critical review of all available data led to the identification of specific areas which require analytical and/or experimental study. Based on this study an experimental facility was designed to permit the experimental investigation of those problems relevant to the United States space activity. The test facility construction, which is essentially complete, is described.

Author

TABLE OF CONTENTS

I. INTRODUCTION

An investigation embracing the theory and performance of the visco-type shaft seal under laminar and turbulent conditions was started on April 15, 1964 at The University of Tennessee in the Department of Mechanical and Aerospace Engineering. The investigation is being conducted for The National Aeronautics and Space Administration under Research Grant NsG-587, and this report presents the progress of the investigation for the period - October 15, 1964 through April 15, 1965.

Background information indicating the reasons for the initiation of this study and a resume of previous investigations of the visco seal has been previously reported (1).

II. OBJECTIVES

The general objective of this program is the theoretical and experimental study of the visco seal under laminar and turbulent conditions. Considering previous work and the current needs of the space effort the specific tasks incorporated as part of this investigation include:

1. Experimental verification of the optimum seal geometry suggested by existing laminar visco seal theories.
2. Development of a turbulent theory of visco seal performance.
3. Determination of the optimum seal geometry for turbulent conditions.
4. Experimental evaluation of the sealing coefficients and dissipation function under turbulent conditions.
5. Theoretical and experimental evaluation of the effect of eccentricity on the sealing coefficient and dissipation function under laminar and turbulent conditions.
6. Study of the cause of "seal break" and its effect on visco seal

performance.

7. Investigation of the ability of the visco seal to sustain radial hydrodynamic loads and to generate self-aligning forces.
8. Investigation of the effect of cyclic pressure changes on visco seal performance.
9. Evaluation of visco seal performance when non-Newtonian fluids are employed as the sealant.
10. Study of the visco seal performance, under both laminar and turbulent conditions, when the sealant fluid contains both the liquid and vapor phases.
11. Investigation of visco seal performance when operated at ambient pressures below the vapor pressure of the sealant.
12. Determination of the effect of thread shape on visco seal performance.

III. ACTIVITIES

Facility Construction

During the period covered by this report the major effort has been devoted to the construction and assembly of the experimental test facility which has been previously described (1). Figure 1 is a photograph of the test equipment which shows the general arrangement of the three major units: (a) the test section and drive stand, (b) the hydraulic supply system, and (c) the instrument and control stand. Figure 2 presents a close-up view of the test section and drive train. The test sleeve shown in Figures 1 and 2 is a dummy unit which was utilized during the operational test of the test sleeve support bearing system. The test sleeve and one of the test spindles is shown in Figure 3.

The arrangement of the visco seal test section is shown schematically in Figure 4. The test sleeve is restrained in the sleeve support block by eight hydrostatic radial bearings and a hydrostatic thrust plate. Flow control values are used for compensation of the radial bearings and the thrust bearing is effectively orifice compensated. It was anticipated that the orifice compensated thrust plate might be subject to instability. However, the preliminary operational check of the sleeve support system, which employed a dummy test sleeve, was conducted with no indication of instability. The detail design of the test sleeve support system is presented in Appendix A.

The rather complicated test sleeve support system was dictated by the need to determine the effect of eccentricity on visco seal performance and the need for precise values of the dissipation function as indicated by objectives 4 and 5 presented above. The test sleeve can be positioned with eccentricities from zero to 0.003" which will permit the visco seal to operate at eccentricity ratios from zero to approximately 0.9 depending upon the particular test spindle being used.

The design of the test spindle is shown in Figure 5 which also includes the initial thread dimensions of each of the first ten test spindles. Spindles 2 through 10 are scheduled for two regrinding operations which will provide the possibility of 28 test geometries. Table 1 shows the test spindle thread parameters currently being considered.

1. The hydraulic supply system has been completed and an operational check of the test sleeve support system has been satisfactorily concluded.

2. The main visco seal test stand has been completed. Operational checks have been made on the spindle drive train and the sealant supply and pressure control system.
3. The instrument and control panel has been completed and operational checks and calibration are in progress for the visco seal torque, speed, pressure, and clearance indicating systems.
4. Ten test spindle blanks (complete except for threads) have been machined and the thread geometry shown in Table 1 has been finished for spindles 1 through 4.

Analysis

While the major effort has been devoted to the construction and operational testing of the experimental apparatus, parallel study and analysis has continued during this report period.

A detail analysis of the visco seal under all operation conditions has been undertaken. The first part of this study which considers the laminar concentric operation has been completed (2). The second part of the study dealing with turbulent conditions is approximately 10 percent complete. In addition, a thesis being prepared under auspices of the grant deals with the turbulent operating conditions with particular emphasis being placed on the optimum thread parameters. Depending upon the progress of this latter study, the fractional factorial laminar experiment may be delayed by finishing spindles 5 through 10 with thread dimensions which are more desirable for turbulent operation.

A second thesis dealing with the detail design and fabrication of the visco seal experimental facility is approximately 75% complete. It is noted that all theses prepared under the auspices of this grant are

TABLE 1

VISCO SEAL TEST SPINDLE PARAMETERS

No	c	h	α	β	γ
1	0.0035	0.010	14.5	3.86	0.625
2	0.002	0.013	9.67	7.5	0.3
3	0.002	0.013		7.5	0.5
4	0.002	0.013		7.5	0.7
2A	0.003	0.012		5.0	0.3
3A	0.003	0.012		5.0	0.5
4A	0.003	0.012		5.0	0.7
2B	0.005	0.010		3.0	0.3
3B	0.005	0.010		3.0	0.5
4B	0.005	0.010		3.0	0.7
5	0.003	0.0195	5.81	7.5	0.3
6	0.003	0.0195		7.5	0.5
7	0.003	0.0195		7.5	0.7
5A	0.0045	0.018		5.0	0.3
6A	0.0045	0.018		5.0	0.5
7A	0.0045	0.018		5.0	0.7
5B	0.0075	0.015		3.0	0.3
6B	0.0075	0.015		3.0	0.5
7B	0.0075	0.015		3.0	0.7
8	0.003	0.0195	20.15	7.5	0.3
9	0.003	0.0195		7.5	0.5
10	0.003	0.0195		7.5	0.7
8A	0.0045	0.018		5.0	0.3
9A	0.0045	0.018		5.0	0.50
10A	0.0045	0.018		5.0	0.7
8B	0.0075	0.015		3.0	0.3
9B	0.0075	0.015		3.0	0.5
10B	0.0075	0.015		3.0	0.7

A = First regrind
B = Second regrind

to be submitted as reports to NASA.

A comparative study of the concentric laminar visco seal theories was started. In this effort the theories presented by Asanuma (3), Zotov (4), and Boon and Tal (5) are used to compute the visco seal performance for the same seal geometries. The Zotov theory utilized is the modified form presented by McGrew and McHugh (6), and the Boon and Tal theory employed is in the modified form presented by Stair (2). While this work is not complete, the sealing coefficient based on these three analyses in summarized in Table 2. While the differences in the sealing coefficients as determined by these three theories are small in many cases, distinctly different patterns are noted. An attempt is being made to relate these differences to the assumptions employed in the various analyses. Also, the performance profiles being developed by this study will be used in the comparative evaluation of the data produced in the experimental investigation.

The review and evaluation of technical publications dealing with the visco seal and related devices has continued during this period. Also, a limited effort has been invested in the study of currently proposed space power apparatus in an effort to see how the visco seal may be applied and the problems which might be anticipated.

<u>Activity Level</u>

During the period October 15, 1964 - April 15, 1965 the following personnel were working on visco seal research:

W. K. Stair, Director, - 1/4 time

C. F. Bowman, Graduate Assistant - 1/3 time

R. H. Hale, Graduate Assistant - 1/3 time

TABLE 2

A COMPARISON OF SEALING COEFFICIENTS

α	γ	β	Λ_S	Λ_M	Λ_A
8	0.3	3	19.8	26.8	28.3
	0.3	5	13.0	13.6	15.9
	0.3	7	13.8	12.6	15.7
	0.5	3	16.8	22.6	24.2
	0.5	5	11.6	11.9	14.2
	0.5	7	12.9	11.5	14.7
	0.7	3	19.7	26.8	28.7
	0.7	5	13.0	13.6	16.2
	0.7	7	13.8	12.6	16.0
14	0.3	3	13.5	16.7	19.0
	0.3	5	12.9	11.3	15.1
	0.3	7	17.9	14.4	19.7
	0.5	3	11.9	14.3	16.6
	0.5	5	12.1	10.3	14.1
	0.5	7	17.4	13.8	19.0
	0.7	3	13.5	16.7	19.1
	0.7	5	12.9	11.3	15.2
	0.7	7	17.9	14.4	19.8
20	0.3	3	12.4	13.3	16.6
	0.3	5	15.0	11.9	17.4
	0.3	7	23.2	18.2	25.9
	0.5	3	11.2	11.5	14.8
	0.5	5	14.4	11.2	16.6
	0.5	7	22.8	17.7	25.4
	0.7	3	12.4	13.3	16.7
	0.7	5	15.0	11.9	17.5
	0.7	7	23.2	18.2	25.9

Λ_S based on Boon and Tal theory

Λ_M based on Zotov theory

Λ_A based on Asanuma theory

The above schedule will be followed through June 5, 1965. For the period June 14 through September 15, 1965 the above personnel will devote full time to the visco seal project. In addition, a third graduate assistant has been engaged who will devote full time to the grant for the summer period.

IV. PROPOSED SCHEDULE

During the period April 16, 1965 - October 15, 1965 the following tasks are scheduled to be completed:

1. Complete performance tests and calibration of all visco seal instrumentation.
2. Conduct reproducibility test series using test spindle No. 1 and prepare a report covering this test series.
3. Conduct fractional factorial experiment using test spindles No. 2 through No. 10 for both laminar and turbulent flow. This experiment series is designed to provide data necessary to accomplish the objectives 1,3,4,5, and 6, under II above.
4. Complete the analysis of the turbulent visco seal.
5. Prepare a series of test spindles having the optimum geometry for turbulent operation.

   NOTE: Depending upon the progress of item 4, this work may be introduced ahead of the tests scheduled in 3 above.
6. Prepare design drawings of the tester modifications required to investigate the hydrodynamic load capability of the visco seal.

NOMENCLATURE

a	axial length across land, inches
b	axial length across groove, inches
c	radial clearance, inches
h	screw thread depth, inches
α	screw thread helix angle, degrees
β	$(h+c)/c$, dimensionless
γ	$b/(a+b)$, dimensionless
Λ	sealing coefficient, dimensionless

REFERENCES CITED

1. Stair, W. K., "Theoretical and Experimental Studies of Visco-Type Shaft Seals," Mechanical and Aerospace Engineering Research Report ME 64-587-1, October 23, 1964. Semi-annual progreas report on NASA Grant NsG-587.

2. Stair, W. K., "Analysis of the Visco Seal, Part I - The Concentric Laminar Case," Report No. ME 65-587-2, The University of Tennessee, January 18, 1965.

3. Asanuma, T., "Studies on the Sealing Action of Viscous Fluids," International Conference on Fluid Sealing, Paper A3, April 17-19, 1961, BHRA, Harlow, Essex, England, 26 pages.

4. Zotov, V. A., "Research on Helical Groove Seals," Machine Design and Calculation (Russia), Issue No. 10.

5. Boon, E. F., S. E. Tal, "Hydrodynamische Dichtung fur rotierende Wellen," Chemie-Ing.-Technik., Vol. 31, No. 3, Jan. 31, 1959, pp. 202-212.

6. McGrew, J. M., J. D. McHugh, "Analysis and Test of the Screw Seal in Laminar and Turbulent Operation," The General Electric Advanced Technology Laboratories, Report No. 63GL66, May 3, 1963.

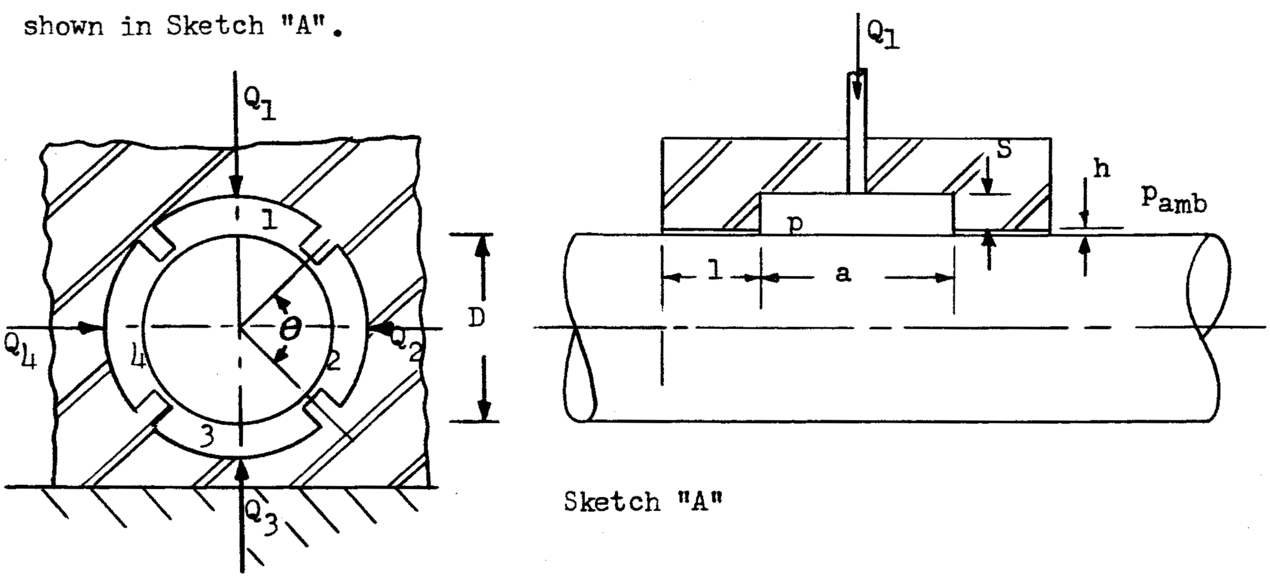
APPENDIX - VISCO SEAL SLEEVE SUPPORT BEARING DESIGN

The visco seal test sleeve and spindle are arranged as shown in Figure 4. The support structure consists of a machined steel bearing block incorporating eight hydrostatic radial pads and an axial thrust pad. The radial load is due to the weight of the test sleeve and the hydrodynamic film force resulting from eccentric operation of the visco seal. Under planned operating conditions the estimated total radial load is of the order of 100 lb_f. The steady axial load will be limited to about 75 lb_f with short duration loads of approximately 150 lb_f.

The load capacity of the support system is of minor importance in this design since loads of these magnitudes could be carried with relatively small bearings. The primary purpose of the sleeve support bearings in the visco seal test apparatus is to permit: (a) the precise measurement of the visco seal torque with a minimum restraint due to static friction, and (b) the positioning of the test sleeve in order to control the eccentricity between the test sleeve and spindle.

Radial Bearings

The arrangement of one set of radial pads and the nomenclature is shown in Sketch "A".



Sketch "A"

Assumptions:

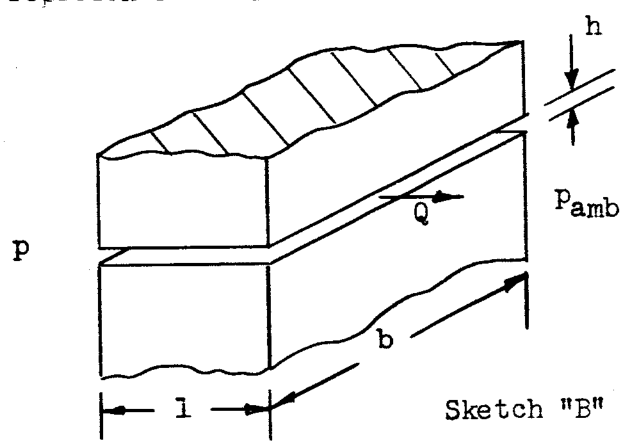
1. The bearing leakage is axial only,

2. $\frac{dp}{dl}$ = constant,

3. $S \gg h$,

4. Steady incompressible flow.

The annular slot of thickness h is the leakage flow path and, since $h \ll D$, can be represented as in Sketch "B".



Sketch "B"

The leakage flow through the narrow slot is

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

From Sketch "A", $b = 2\pi D$ (for two sills). Thus,

$$Q = 0.524 \frac{\Delta p D h^3}{\mu l} \tag{2}$$

Assuming pad compensation by the use of flow control valves,

$$Q_1 = Q_2 = Q_3 = Q_4 = Q_p, \tag{3}$$

$$Q = Q_1 + Q_2 + Q_3 + Q_4, \text{ and} \tag{4}$$

$$Q_p = 0.131 \frac{\Delta p D h^3}{\mu 1} \tag{5}$$

Consider a radial load W along center line of pads number 1 and 3 in Sketch "A". Upon application of the load W, Q_1 remains equal to Q_3 if control valve compensation is employed. However, $p_1 \neq p_3$ and changes will occur in h_1 and h_3. Solving for pressure from equation (5)

$$p_1 = 7.64 \frac{Q_1 \mu 1}{D h_1{}^3}, \text{ and} \tag{6}$$

$$p_3 = 7.64 \frac{Q_3 \mu 1}{D h_3{}^3} \tag{7}$$

For equilibrium, the sum of vertical forces gives

$$F_V = p_1 A_1 + W - p_3 A_3 = 0 \tag{8}$$

The recess and sill areas are:

$$A_R = Da \sin \frac{\theta}{2}, \text{ and} \tag{9}$$

$$A_S = 2D1 \sin \frac{\theta}{2} \tag{10}$$

Based on assumption 2, the effective sill area is:

$$A_{SE} = D1 \sin \frac{\theta}{2}. \tag{11}$$

The effective pad area is the sum of A_R and A_{SE},

$$A_p = A_1 = A_3 = D(a+l)\sin\frac{\theta}{2}. \tag{12}$$

From (8)

$$W = p_3A_3 - p_1A_1 = A_p(p_3-p_1) \tag{13}$$

Let $h_3 = h - \delta$, and $h_1 = h + \delta$,

where: h = nominal clearance,

δ = displacement of sleeve axis.

Substituting (6) and (7) into (13)

$$W = 7.64\,\frac{Q_p\mu l A_p K}{D} \tag{14}$$

where: $$K = \frac{1}{(h-\delta)^3} - \frac{1}{(h+\delta)^3} \tag{15}$$

In normal bearing practice the control values would be adjusted for equal values of Q_p for each pad with the pressure changes in the pads compensating for changes in load. In the present application, however, the supply system will be adjusted to provide a fixed constant pressure and the motion of the sleeve center will be accomplished by adjusting the Q_p values. Thus, equating (6) and (7)

$$\frac{Q_1}{h_1^3} = \frac{Q_3}{h_3^3} \tag{16}$$

Design Selected

$D = 4.25"$, $a = 0.5"$, $l = 1.0"$, $h = 3"$, $\Delta p = 1000$ psa.

$A_p = 4.25 \times 1.5 \times \sin 45 = 4.51 \text{ in}^2$

The pads are arranged with centers on the vertical and horizontal center-lines as shown in Sketch "A". The selected hydraulic fluid viscosity is

Temperature-°F	μ- Reyns
100	6.5×10^{-6}
130	4.19×10^{-6}
210	1.28×10^{-6}

At an assumed average film temperature of 110°F, $\mu = 5.6 \times 10^{-6}$ Reyns. From equation (5) the flow per pad under concentric conditions is:

$$Q_p = \frac{0.131 \times 1000 \times 4.25 \times 0.003^3}{5.6 \times 10^{-6} \times 1} = 2.69 \text{ in}^3/\text{sec.}$$

The total concentric flow (for eight pads) is

$$Q_T = 8 \times 2.69 = 21.5 \text{ in}^3/\text{sec (5.58 gpm)}$$

To determine the stiffness of the bearing, assume Q_p is constant at the concentric value in each pad. The load W required to produce a deflection of $\delta = 0.001$ may be found from equation (14). For $h = 0.003$ and $\delta = 0.001$ equation (15) gives a $K = 1.094 \times 10^8$.

$$W = \frac{7.64 \times 2.69 \times 5.6 \times 10^{-6} \times 1 \times 4.51 \times 1.094 \times 10^8}{4.25}$$

$$W = 13{,}380 \text{ lb}_f.$$

Thus, the film stiffness under concentric conditions is

$$k = 13{,}380/0.001 = 13.38 \times 10^6 \text{ lb}_f/\text{in.}$$

For operation under eccentric conditions the flows to pads 2 and 4 are assumed to remain essentially at the concentric position values. Evaluating equation (5) with Δp constant,

$$Q_p = \frac{0.131 \times 1000 \times 4.25\ h^3}{5.6 \times 10^{-6} \times 1} = 9.95 \times 10^7\ h^3\ \text{in}^3/\text{sec}$$

For a $\delta = 0.0015''$, $h_1 = 0.0045''$, $h_3 = 0.0015''$, and $h_2 = h_4 \cong 0.003''$.

Therefore,

$$Q_T = 2 \left[Q_1 + Q_2 + Q_3 + Q_4 \right] \qquad \text{(for 8 pads)}$$

$$Q_T = 2 \left[9.2 + 2.69 + 0.34 + 2.69 \right] = 29.84\ \text{in}^3/\text{sec} \quad (7.75\ \text{gpm})$$

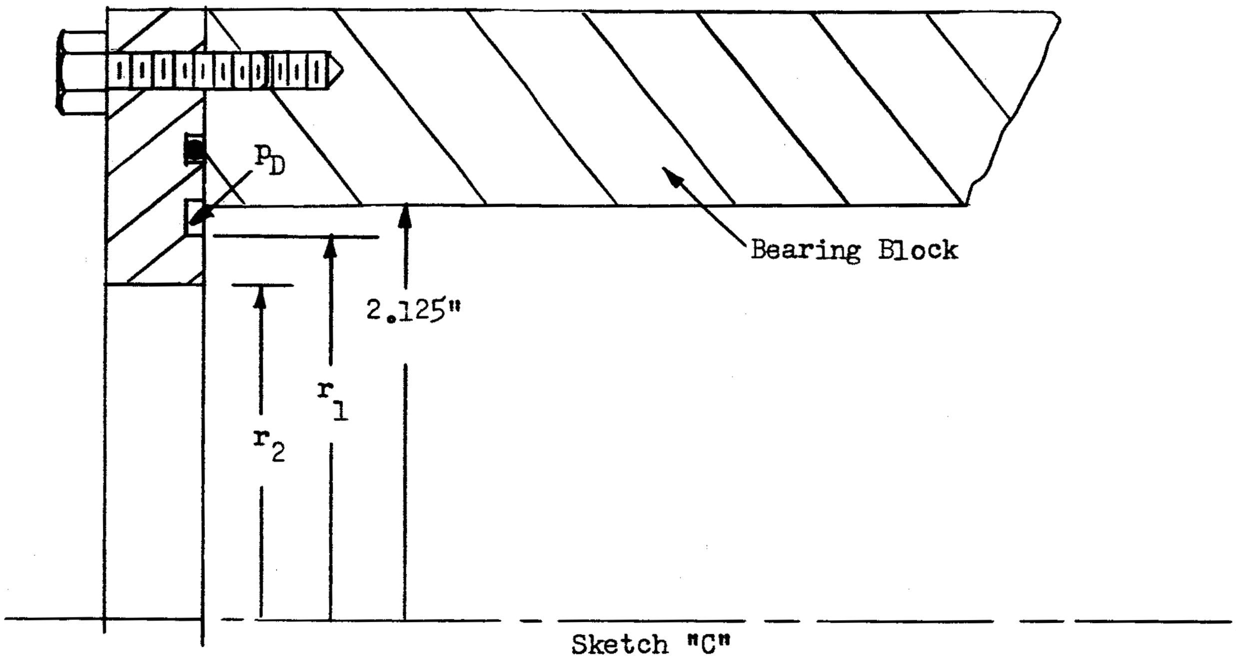
At other deflections the total flow becomes:

	Q_T	
δ	in^3/sec	gpm
0.0005	22.48	5.84
0.001	24.34	6.32
0.0015	29.94	7.77
0.0020	36.56	9.48
0.0025	44.78	11.63

Each of the eight flow control valves selected for this application can be manually adjusted to deliver from 0 to 19 in^3/sec.

Thrust Bearing

Since the thrust capacity is very low the function of the thrust bearing is to provide axial positioning with a minimum of rotational restraint. The leakage flow over the outboard sill of one set of radial bearings is used as the supply for the thrust pad which is arranged as shown in Sketch "C".



Sketch "C"

Assuming steady incompressible flow the radial flow across the thrust pad is

$$Q = \frac{\pi h^3 \Delta p}{6\mu \ln r_1/r_2} \qquad r_1 > r_2 \tag{17}$$

The flow for concentric operation of the radial bearings with $\mu = 5.6 \times 10^{-6}$ Reyns and p = 1000 psi is 5.38 in^3/sec which is the flow over one sill of one set of radial pads. For r = 2.125", r_1 is selected as 2". Now, r_2, h_1 and the thrust supply pressure p_D is unknown. However, the thrust force F_T varies from 0 to 150 lb_f.

$$F_T = \int p dA = p_D A_E$$

$$A_E = \pi(2.125^2 - 2^2) + \frac{\pi}{2}(2^2 - r_2^{\,2}) = \frac{\pi}{2}(5.04 - r_2^{\,2})$$

If r_2 is selected as 1.75" then $A_E = \pi$ in^2, and

$$P_D = \frac{150}{\pi} = 47.7 \text{ psi for maximum axial load and}$$

$$P_D = \frac{75}{\pi} = 23.85 \text{ psi for normal axial load.}$$

$$\ln(r_1/r_2) = \ln(2/1.75) = 0.133$$

(From (17) the normal thrust film thickness is found as

$$h = \left[\frac{4 \times 6 \times 5.6 \times 10^{-6} \times 0.133}{\pi \times 23.85}\right]^{1/3} = 6.3 \times 10^{-3} \text{ inches.}$$

and under maximum load h becomes 5×10^{-3} inches.

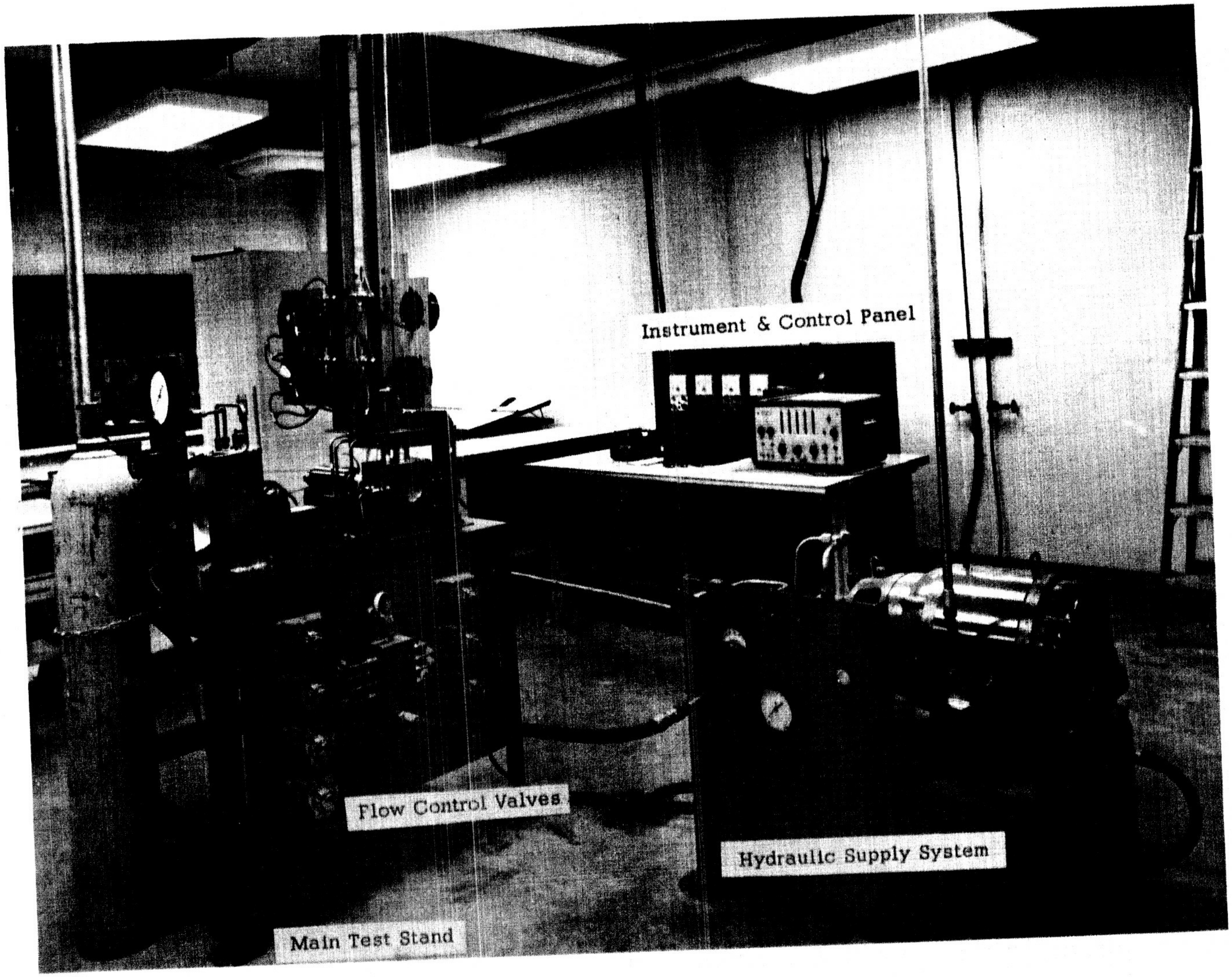


Figure 1 Visco Seal Test Facility

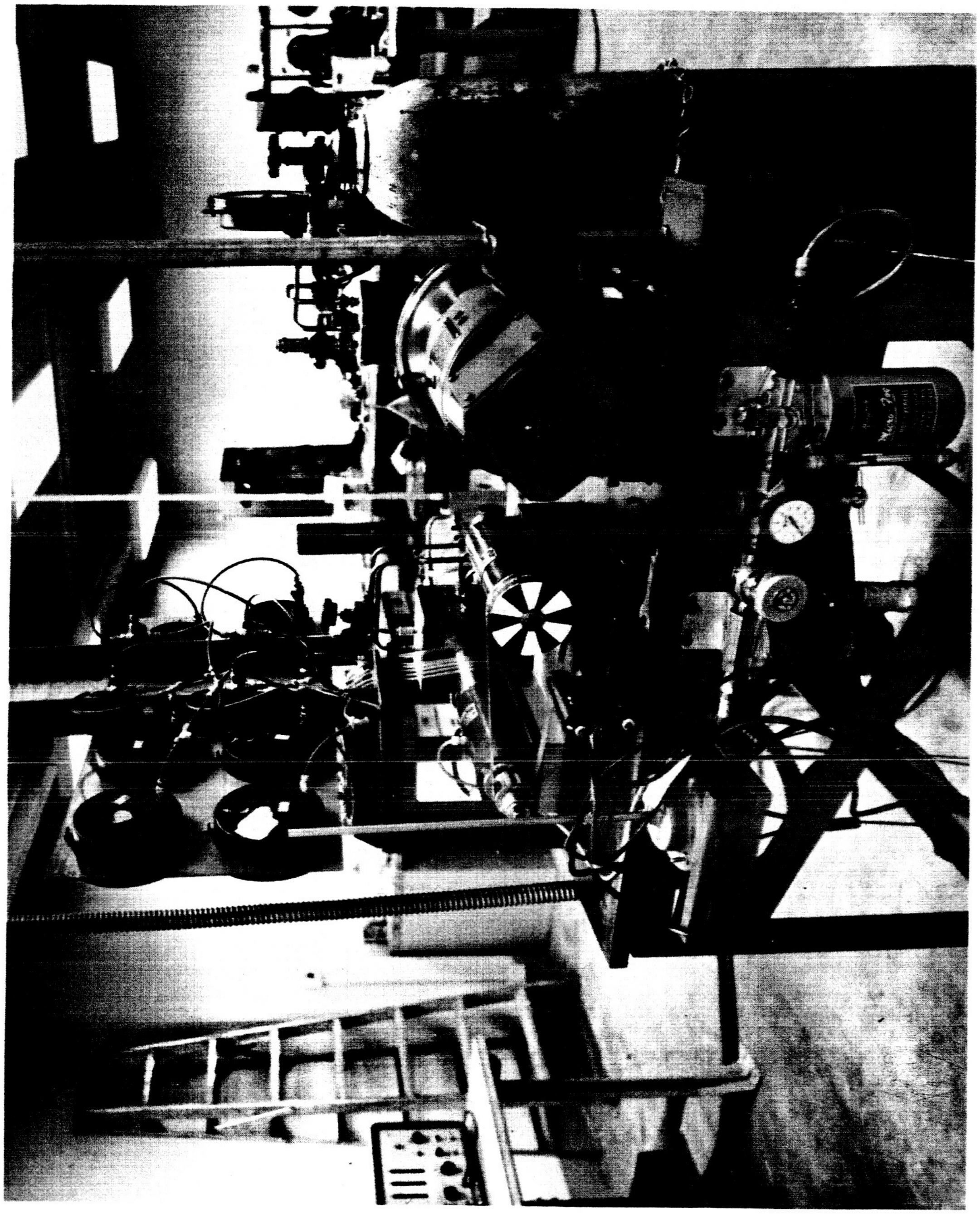


Figure 2   Visco Seal Test Section and Drive

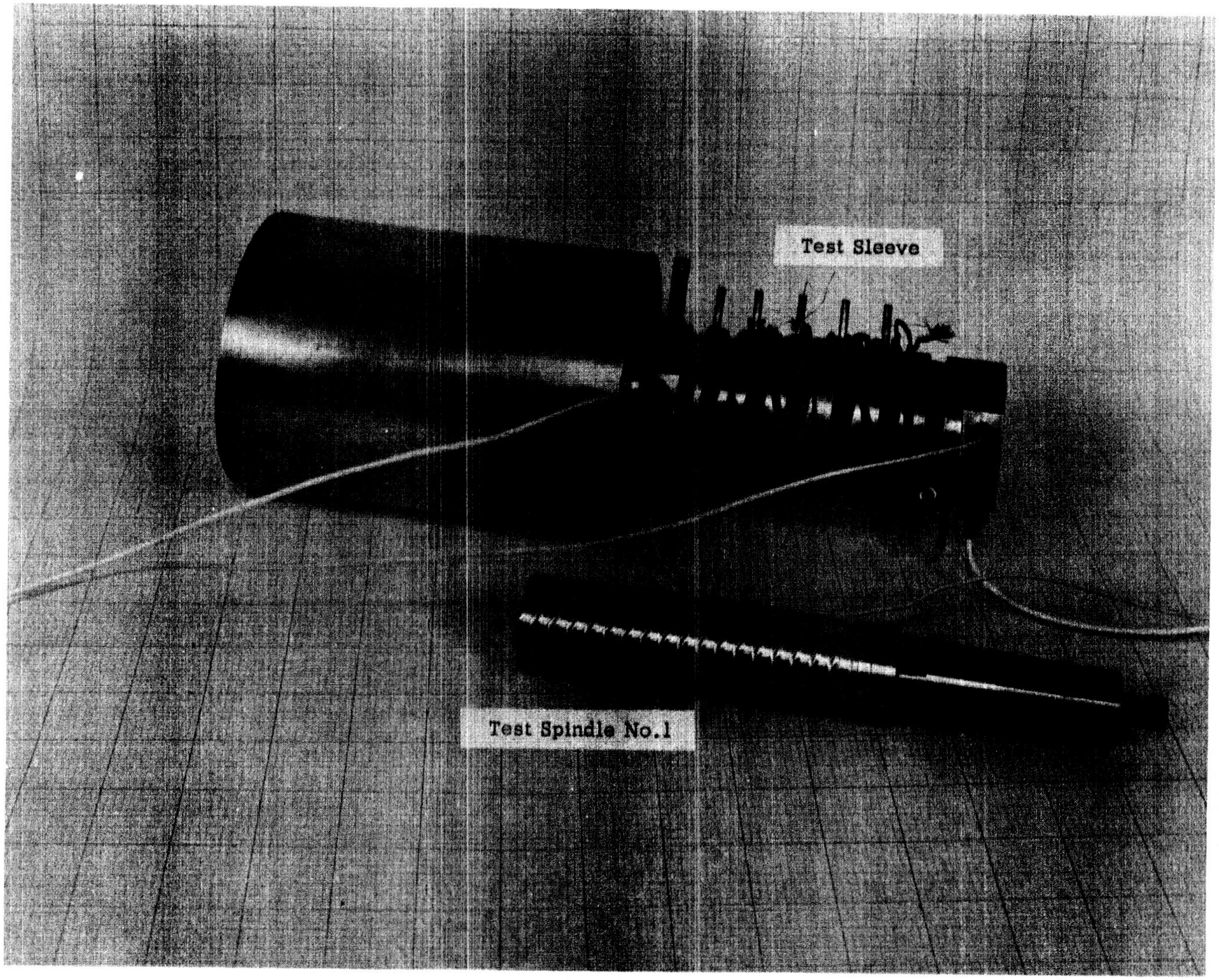
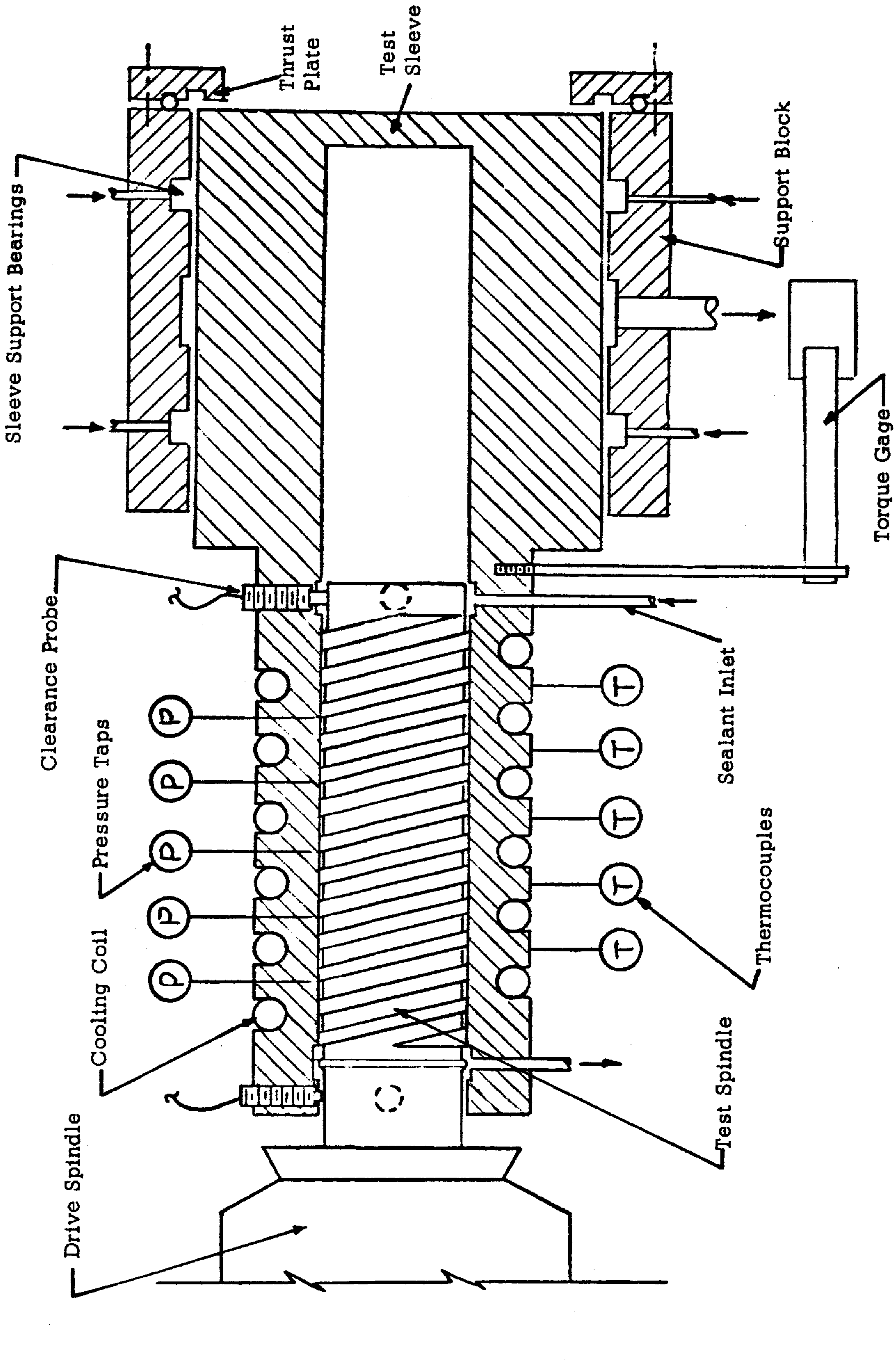


Figure 3 Test Sleeve and Test Spindle No. 1



Thrust Plate
Test Sleeve
Sleeve Support Bearings
Support Block
Torque Gage
Clearance Probe
Sealant Inlet
Pressure Taps
P
T
Thermocouples
Cooling Coil
Test Spindle
Drive Spindle

FIGURE 4

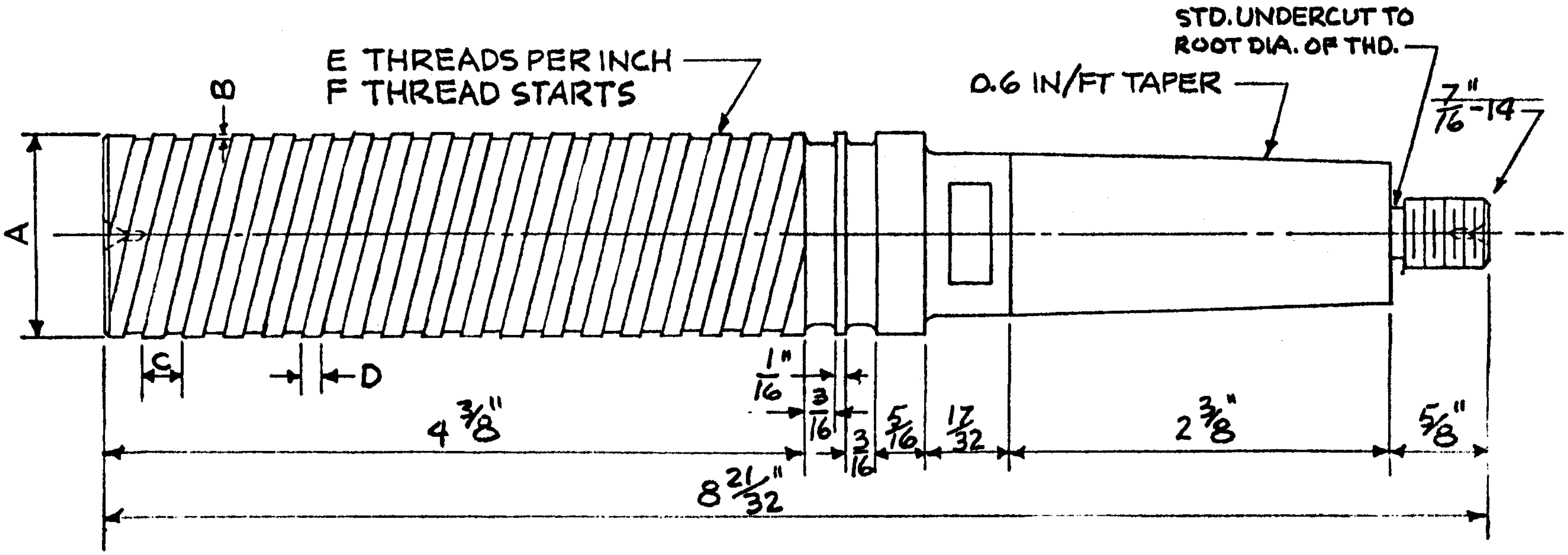


FIGURE 5

SPINDLE Nº	DIMENSIONS					
	A	B	C	D	E	F
1	1.2432	0.010	0.253	0.095	0.988	4
2	1.2462	0.013	0.167	0.1168	1.5	"
3	"	"	"	0.0834	"	"
4	"	"	"	0.050	"	"
5	1.2442	0.0195	0.200	0.140	2.5	2
6	"	"	"	0.100	"	"
7	"	"	"	0.060	"	"
8	"	"	0.360	0.252	0.695	4
9	"	"	"	0.180	"	"
10	"	"	"	0.108	"	"

UNIVERSITY of TENNESSEE
KNOXVILLE, TENNESSEE

VISCO SEAL
TEST SPINDLE

DRAWN BY: C.F. BOWMAN	DATE: 9/1/65	DWG
CHECKED BY:	DATE:	Nº VS-8-A