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A REPORT ON THE STATUS OF GAS LUBRICATION FOR TURBOMACHINERY

MASTER

by

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OUTLINE

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NOMENCLATURE

C	Radial clearance, in.
D	Bearing diameter, in.
F	Frictional force, lb.
f	Dimensionless force - 1/S
g	Acceleration, in/sec.^{2}
L	Bearing length, in.
m	Mass, lb.sec.^{2}/in.
N	RPS
P	Pressure over projected area, psi
P_a	Ambient pressure, psia
P_L	Loading pressure, psid
R	Bearing radius, in.
W	Bearing load, lb.
S	Sommerfeld Number - $\frac{\mu N}{P} \left(\frac{R}{C}\right)^2$
β	Angle between load line and center of bearing arc
ϵ	Eccentricity ratio
Λ	Compressibility number - $\frac{6\mu\omega}{P_a} \left(\frac{R}{C}\right)^2$
μ	Viscosity, lb.sec/in.^{2}
ϕ	Attitude angle, degrees
ω	Angular velocity, rad/sec.

Superscript

—	Dimensionless

I. INTRODUCTION

In the last decade gas bearing technology has come of age and numerous applications have emerged. Most of these are in the fields of specialized turbomachinery and instruments. This paper is devoted entirely to the former.

The purpose of the authors is to discuss some of the reasons why gas bearings are being applied and to outline several of the factors that must be considered in selection and design. At the same time, attention is focused to the limitations of current knowledge and on a number of remaining problem areas. Illustrations are given for rotors supported in self-acting and in externally pressurized gas bearings. These rotors range from a fraction of a pound to several hundred pounds in weight.

The authors feel very strongly that gas bearings are no longer an art or a demonstration model.. A well defined technology has been established and accounts for the many successful applications. This is not to say that further technological advances are no longer needed. In fact this paper stresses several problem areas and limitations in order to alert the designer and to caution him against misapplications.

II. DISCUSSION

The growth of gas bearing applications can best be answered by raising the following question:

Why Gas Bearings?

Immediately a number of advantages emerge, such as:

1.) No contamination of the system environment by the lubricant.
2.) Operation over a wide temperature range.
3.) Resistance to thermal breakdown of the lubricant.
4.) Resistance to damage by radioactivity.
5.) High reliability and long life.
6.) Low and constant friction.
7.) Close position control and low vibration.
8.) Minimized leakage from gas system.
Etc.

Both commercial and military groups are taking these advantages and applying them to many products. Some of these are listed in Table I.

As evidenced from Table I, the application of process fluid lubrication to compressors and turbines offers many advantages. It is for this reason that many successful applications in this area have been developed over the last few years.

The Office of Naval Research early recognized the potentials of gas bearings and coordinated a government sponsored, interagency program to support fundamental work in a number of companies, institutes and universities. This continuing program has been the single most important factor in the establishment of gas bearing technology. Fundamental and systematic advances in knowledge have also come from a number of sources overseas including, in particular, the United Kingdom Atomic Energy Authority, joint efforts by the Japanese Universities and industry as well as research institutes and universities in the Soviet Union and Rumania.

The technology is quite complex because it involves nonlinear mechanics and a wide range of interrelated parameters. Without a clear understanding of this technology one must resort to trial and error procedure which can be extremely costly. In some cases, in fact, the development effort suffered because of this approach and was stopped after a considerable expenditure of funds.

A clear understanding of gas bearing technology still does not guarantee successful applications. This is because a machine is composed of many other elements beside bearings. The behavior of many of these elements is sometimes unknown and the interactions in the system are even less well understood. Ill defined forces due to magnetic and electrical dissymmetries of high frequency motors are a good example of such design pitfalls. Several disciplines, e.g., heat transfer, aerodynamics, stress analysis, material engineering and electrical engineering enter into a successful design.

To illustrate the effect of a few of the parameters and some of the interactions an example is given below:

Example

It is required to design a motor driven 100 HP compressor, which would operate on self-acting gas bearings. Aerodynamic considerations require the compressor to operate somewhere between 12,000 and 24,000 RPM. The example is simplified by assuming that within this speed range there are no rotor instabilities or dynamic problems and no problems of heat transfer or stress. There still remain a number of considerations, a few of which are chosen here to illustrate some of the interactions.

Experience has shown that in the range of 100 HP the weight of a hollow rotor assembly is in the order of 1 lb/HP (Fig. 15). On the other hand solid rotors would be about 1.5 lb/HP. From the standpoint of material compatability and without external pressurization for starts and stops, the load on projected bearing area should be about 2 psi. This requires for the 100 HP hollow rotor a bearing area of 50 square inches. With bearings of L/D = 1 the two bearings would, therefore, be 5" x 5". For a solid rotor the loading would go to 3 psi or the bearing area would have to be increased to 75 sq. in. With a larger diameter rotor the ratio of 1.5 lb/HP may be exceeded; this is an undesirable condition. The required starting motor torque for the hollow rotor using compatible materials with a coefficient of friction of 0.1 would be 25 in.lb. This presents a high starting torque, which requires oversized motor windings and power supply. With the solid rotor the situation would be 50 percent worse.

At 24,000 RPM a hollow steel rotor 5" O.D. and 4.5" I.D. has a centrifugal radial growth of 1.9×10^{-3} in., while a solid rotor grows only 0.4×10^{-3} in. The radial growth of the hollow rotor is excessive unless a design was provided to compensate for it. Yet radial growth or decrease in radial clearance with increase in speed can be taken advantage of since there exist regions in the stability plot where the instability threshold rises with reduction in clearance. This observation was illustrated in Ref. 2 and will be discussed further in this paper. One could cite other interactions but this will suffice to illustrate the point that successful turbomachinery application demands an understanding of the interactions and a proper balance between the various factors.

Rotor-Bearing Considerations

Numerous reports and papers have covered various aspects of gas bearings and rotor dynamics. Since reference cannot be made to all of them, only a recent bibliography (Ref. 1) is referred to. In addition reference is made to publications by the authors which are used in this paper.

In Refs. 2, 3 and 4 the effect of bearing grooving, position of venting orifice clearance, L/D, eccentricity ratio, mass to weight ratio and compressibility number were studied as a function of load carrying capacity and threshold of Half Frequency Whirl. This paper extends the past work. It selects a commonly used bearing with L/D = 1 and shows the effect of ambient pressure, external force, bearing arc, and direction of force on load carrying capacity and threshold of instability.

Plain Cylindrical Journal Bearings

Load carrying capacity, attitude angle and friction force as a function of compressibility number and eccentricity ratio are shown in Figs. 1, 2 and 3 respectively. Note that theory and practice are in good agreement. The effect of clearance on the threshold of instability is shown in Fig. 4. Two theoretical approaches, the Linearized PH-Quasi Static and the Non-Linear Galerkin's Method are compared to available experimental data. It has been noted that the Linearized PH-Quasi Static theory (Ref. 5) always gives safe criteria and that the trends are similar between theory and experiment. The effect of L/D ratio on the threshold of instability is shown in Fig. 5. Note that the lower the L/D ratio, the higher the threshold speed. This means of raising the instability threshold is achieved at the cost of increased unit loads and thinner films. The effect of eccentricity ratio on threshold of instability, using Galerkin's Method, is shown in Fig. 6. This figure gives a complete stability map for a bearing with L/D = 1. Similar maps are given in Ref. 6 for other L/D ratios. The effect of orifice venting on threshold of instability is also shown in Fig. 7. Note that an appreciable increase in threshold speed can be achieved by proper location of the venting orifice.

The above discussion has pointed out some factors which influence the rotor-bearing design. It is evident that even with a simple geometry, such as plain cylindrical journal bearings, the number of design factors is large. This bearing configuration is of interest because of its simplicity, high load carrying capacity and ease of manufacture to the tolerances required. Furthermore, it expands uniformly and is more resistant to thermal distortion than other less symmetrical bearing types. Its main disadvantage is its low threshold of instability. Nevertheless, the plain cylindrical journal bearing or one with very few minor modifications is adequate in many applications.

Partial Arc Journal Bearings

Applying most of the advantages and recognizing the limitations of plain cylindrical journal bearings, MTI has developed a bearing configuration which has appreciably higher threshold of instability and one whose fabrication is only slightly more complex. This configuration is discussed next.

In Fig. 7 it was shown that the threshold of instability could be raised by the elimination of subambient pressures; this was accomplished by an orifice which introduced ambient pressure to the diverging region of the bearing. In Fig. 6 it was shown that the threshold of instability could be raised by increasing the applied force. It was further recognized that it would be desirable if the applied force increased with speed. All of these features have been incorporated in a design patented by MTI as shown in Fig. 8.

The load carrying arc of the bearing consists of plain surface. The diverging film arc of the bearing is grooved. This region generates hydrodynamic pressures which are above ambient. The magnitude of the pressure is a function of bearing geometry and compressibility number. In this way a force is generated which increases with speed. The load carrying capacity is also a function of compressibility number and bearing arc. Such a configuration, therefore, generates above ambient pressures in both converging and diverging regions of the bearing.

Fig. 9 shows load carrying capacity and attitude angle as a function of bearing arc and eccentricity ratio for a given compressibility number and L/D ratio. Similar curves have been obtained for other compressibility numbers and L/D ratios. Note from this figure that the decrease in load carrying capacity between 360^{o} and 120^{o} bearing arc is only about 50 percent. In most cases the higher threshold speed raises the load carrying capacity which more than compensates for the reduction caused by the bearing arc change. The higher threshold has many other advantages in turbomachinery applications. Fig. 10 shows two such bearing configurations. Using this principle, tests were conducted to show the effect of several parameters on the threshold of instability. Fig. 11 shows the test rig and instrumentation. The test rig is instrumented to study load carrying capacity, attitude angle and rotor stability as a function of bearing geometry, compressibility number, weight, mass, unbalance forces, magnetic forces and hydrostatic forces. Some of the results are illustrated in Figs. 12, 13 and 14.

Fig. 12 shows whirl threshold speed as a function of ambient pressure, angle β, and hydrodynamic load P_L. Note the large gain in whirl threshold speed between the $\beta = 0^{o}$, $P_L = 0$ psid and $\beta = +12^{o}$, $P_L = 5$ psid. In fact a 12,000 RPM rotor is unstable at ambient pressures above 35 psia with the first bearing and it is stable all the way up to 500 psia with the second bearing.

Fig. 13 shows whirl threshold as a function of hydrodynamic pressure, ambient pressure and L/D ratio. Note the change of whirl threshold with change in L/D ratio. For the $p_a = 500$ psia and 12,000 RPM condition, discussed before, the L/D = 3/4 bearing will require less than half of the hydrodynamic load for stabilization. Further improvements can be obtained by varying the angle β.

The manufacture of these bearings is quite simple. Several techniques can be used, e.g., etching, plating, machining, etc. The particular application will generally dictate the best method.

While these bearings possess numerous advantages they have the disadvantage of load direction sensitivity. This can be overcome by making the bearing longer with sets of grooves all around the circumference but displaced axially. In effect, this is equivalent to several adjacent bearings with the plain and grooved portions at different angles with respect to the load vector. Such a design is cumbersome and makes the machine longer and more expensive. For good design one should know the magnitude and direction of the load. Thus, the disadvantage only presents itself where the load direction is variable. This brings us to the next point in the discussion.

Dynamic Forces and Instabilities

There are a number of sources of dynamic forces on the rotor. Some of these are internal to the machine (Ref. to Fig. 14) and others are external. The most common is the mechanical unbalance force which produces Synchronous Whirl. The amplitude and phase angle of this whirling motion has been analyzed for rotors in plain cylindrical journal bearings and is well understood (Refs. 7 and 8).

Motor driven rotors can impose a number of dynamic forces (see Fig. 14). During starting the rotor may have to pass through a number of electrical resonances. The force magnitude at these resonances may be several g's as seen in Fig. 14. In general electrical motors also produce magnetic forces with a frequency of two per revolution. The presence of these forces, together with the once per revolution mechanical unbalance produces a three leaf pattern.With electrical power cut off the three leaf pattern reduces to once per revolution synchronous whirl as shown in Fig. 14. With motor slip the three leaf pattern resembles a propeller. In addition, magnetostrictive forces in a motor produce excitation whose magnitude is time dependent and whose resonant frequency depends on electromechanical design. This force produces a beat (see Fig. 14). The magnitudes and frequencies of all the electrical forces are much less predictable than the mechanical forces and can cause considerable trouble.

Aerodynamic pulsating forces are extremely severe in the compressor surge range. Their magnitude and condition of occurrence depends on both the aerodynamic (impeller) and the system designs. While methods are available for reasonably accurate predictions of the surge line, the calculation of force magnitude in the neighborhood of surge is not readily attainable.

A number of other resonances can be troublesome. For example, a flexibly mounted bearing can be excited by external vibrations which are transmitted through the casing. In the case of externally pressurized bearings, pneumatic hammer is another source of trouble. In addition the self-excited fluid film instabilities must, under all conditions, be avoided. When such instabilities set in the self-acting fluid film forces will rapidly go down to zero and the fluid film is unable to support any load. (See Fig. 14.) This subject has been studied quite extensively and stability maps at least for plain cylindrical bearings and symmetrical rotors are available (Refs. 5, 6 and 9).

Dynamic forces and instabilities have thus far been the greatest barriers to successful applications. Numerous examples can be cited of unsuccessful applications in which dynamic conditions were ignored during design. This should not be surprising since similar situations exist with oil bearing supported turbomachinery. With gas bearings this condition is aggravated by the fact that rotors generally operate at high speed and a failure can be catastrophic to the rotor and bearings. The degree of damage depends almost entirely on material compatability since boundary lubrication is absent.

These points are discussed here to alert the designer to some of the factors that he must consider in his application. It cannot be emphasized too strongly that, new applications, the designer should always resort to careful analysis and well instrumentated test runs, rather than trial and error procedure.

III. EXAMPLES OF ROTOR-BEARING SYSTEMS

In the discussion of this paper several parameters and interactions were discussed. Awareness of these factors and fundamental understanding of the interactions were used in the development and design of several successful systems. Some of these are given in Table II.

Analysis of a number of self-acting motor driven and hybrid turbine driven machines indicate that for high HP (ratings > 200) considerable reduction in weight is possible by use of the latter drive (see Fig. 15). The motor driven rotors range between 12,000 and 24,000 RPM. Considerably higher speeds are possible with turbine drives which permit decrease in weight and more nearly optimum system design. Another advantage is the reduction in bearing size. With hybrid bearings the psi loading in numerous applications is comparable to oil lubrication which makes it two orders of magnitude greater than the self-acting bearings. Some additional comparisons between self-acting and externally pressurized journal bearings are given in Table III.

Several of the self-acting, motor driven rotors and hybrid, turbine driven rotors are shown in Figs. 16 and 17 respectively. Their weight-horsepower characteristics are included in Fig. 15. Included also are examples of self-acting, motor driven fans and circulators ranging from 0.4 - 75 HP (Fig. 18), and hybrid, turbine driven compressor, a feed water pump and generator (Fig. 19). Both larger and smaller rotors are being investigated for several new applications.

IV. PROBLEM AREAS

Even though gas bearings have been successfully applied to several machines, there still remain a number of problem areas which demand further developments. Some of these are listed in Table IV.

CLOSING REMARKS

1.) Gas bearing technology is sufficiently developed that numerous turbomachinery applications are now possible that take advantage of process fluid lubrication.

2.) The next few years will see very rapid increase in the number of commercial as well as military uses of gas bearings. Development costs are warranted because of the many advantages of process fluid lubrication.

3.) Expensive, often futile, trial and error as well as misapplications can be prevented by careful consideration of the many performance parameters and system interactions.

4.) There remain a number of problem areas; however, these are defined and will yield to additional research and development.

5.) Most applications to date have been motor driven machinery with self-acting bearings. It is expected that turbine driven machinery with hybrid bearings will comprise an increasingly larger share of future turbomachinery applications because of higher speed and greater economy of design. This is particularly true in the higher power ranges (greater than 200 HP).

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TABLE I

ADVANTAGES AND APPLICATIONS

	Industry	Applications
No Contamination	Nuclear	Pumps, compressors, blowers, circulators, turbines, motors
	Chemical	Compressors, turbines, expanders
	Food	Compressors, blowers, circulators, motors
	Pharmaceutical	Compressors, blowers, circulators, motors
	Refrigeration	Compressors, expanders
	Medical	Dental drills
Low-High Temperature	Refrigeration	Cryogenic compressors, expanders
	Space	Turbines, compressors, generators
	Utility	Turbines, compressors, generators
	Nuclear	Pumps, compressors, turbines, motors
Radiation	Nuclear	Compressors, turbines
	Space	Compressors, turbines, generators
High Reliability and Long Life	Nuclear	Compressors, blowers, circulators, turbines, motors, generators
	Space	Compressors, turbines, motors, generators, gyros, disks, drums
	Textile	Spindles
	Consumer	Motor, generators, compressors, blowers, circulators, fans, recorders, computers
Low and Constant Friction	Electrical	Dynamometers
	Military	Gyros, accelerometers, turn tables, optical scanners
Close Position Control and Low Vibration	Machine Tool	Grinders, turn tables
	Consumer	Recorders, computers, dental drills

TABLE II

SPECIFICATIONS FOR TYPICAL MACHINES

Machine type	Design is by:	Status	Bearing design: Type	Bearing design: gas used	Performance (actual or predicted): speed, rpm	drive hp	at inlet: flow, cfm	at inlet: T, °F	at inlet: P, psi	head, ft.	coeff (1-stage) Q/ND^3	coeff (1-stage) ψ
Compressors, Blowers and Circulators												
1 Regenerative	MTI	delivered	self-acting	He	12,000	8	6.1	100	492	6,600	.00603	2.820
2 Centrifugal	MTI	delivered	self-acting	He	12,000	75	972	600	500	7,560	.09230	0.710
3 Modified centrifugal	MTI	delivered	self-acting	He	18,000	10	58.5	1,000	500	11,250	.00625	0.657
4 Centrifugal	MTI	in constr.	self-acting	He	23,000	140 (aero)	1,332	245	236	21,800	.09890	0.695
5 Two-stage centrifugal	MTI	in constr.	self-acting	--	10,800	35	400	-	-	6,120	.06940	0.540
6 Regenerative	MTI	delivered	self-acting	Air	3,600	0.125	9	70	14.7	638	.01545	2.210
7 Two-stage regenerative	MTI	designed	self-acting	Air	12,000	10	37.4	200	15	12,450	.01759	1.610
8 Two-stage centrifugal	MTI	preliminary design	self-acting	Mix.	52,500	59 (aero)	336	70	14.5	63,500	.05340	0.632
9 Axial	MTI	preliminary design	self-acting	Air	30,700	250 (aero)	8,430	90	14.5	10,365	.83600	0.270
10 Axial	MTI	preliminary design	self-acting	Air	18,280	250 (aero)	12,880	90	14.5	6,770	.84800	0.270
11 Viscous (Double-sided)	MTI	preliminary design	self-acting	Air	24,000	-	0.55	150	20	15,270	.00018	1.25
12 Centrifugal	GE	delivered	ext.-press.	He	24,000	10	57.4	180	215	9,220	.01477	0.688
13 Centrifugal	GE	delivered	ext.-press.	N_2	24,000	10	47.4	180	215	3,350	.04113	0.518
Turbines												
14 Axial	MTI	in constr.	self-acting	He	23,000	140 (aero)	-	1,600	250	-	-	-
15 Re-entry impulse	MTI	in constr.	self-acting	Air	11,500	2 (aero)	-	100	26.7	-	-	-
16 Axial	MTI	preliminary design	self-acting	Mix.	52,500	59 (aero)	-	1,450	48	-	-	-

(cont'd. on next page)

(cont'd)

TABLE II

Machine type	Design is by:	Status	Bearing design: Type	Bearing design: gas used	Performance (actual or predicted): speed, rpm	drive hp	at inlet: flow, cfm	at inlet: T, °F	at inlet: P, psi	head, ft.	coeff. (1-stage): Q/ND^3	coeff. (1-stage): ψ
Electrical												
17 Motor-Generator	MTI	proposed	self-acting	He	3,600	10KW	-	-	-	-	-	-
Test Rotors												
18 Vertical, multistage	MTI	delivered	self-acting	-	10,800	40	-	-	-	-	-	-
19 Turbo-compressor	MTI	on test	ext.-press.	N_2	35,000	35	-	-	-	-	-	-
20 Single stage	MTI	tested	self-acting	Air	24,000	10	-	-	-	-	-	-
21 Single stage	MTI	tested	self-acting	Air, He	24,000	10	-	-	-	-	-	-
22 Single stage	MTI	constructed	self-acting	Air	18,000	10	-	-	-	-	-	-
23 Single stage	MTI	constructed	self-acting	Air, He	18,000	10	-	-	-	-	-	-
Turbo Pump												
24 Feed Water Pump	GE & MTI	designed	ext.-press.	steam	-	-	-	-	-	-	-	-

Q = actual inlet cfm; N = rpm; D = impeller tip dia., ft; ψ = head coeff. = g x head/U^2; U = tip speed, ft/sec

TABLE III

SELF-ACTING VS. EXTERNALLY PRESSURIZED BEARINGS

Bearing system	Operating requirements				
	Load-carrying capacity	Max. stable speed	Starting torque	Materials compatibility	Auxiliary equipment needed
Externally pressurized (hydrostatic)	Depends on supply pressure and bearing geometry	Very High	Very Low	Minimum problem	Separate compressor or other continuous high press. supply
Self-acting (hydrodynamic) with dry starting	Limited by speed, bearing geometry and fluid properties	Depends mainly on bearing and rotor geometry, fluid properties	Depends on choice of materials as limited by fluid properties	Carefully select materials for minimum wear during starting & stopping	None
Self-acting with hydrostatic starting and stopping	Limited by speed, bearing geometry, and fluid properties	Depends mainly on bearing and rotor geometry, fluid properties	Very low	Minimum problem	Reservoir of fluid for starting, stopping
Hybrid (combined externally pressurized and self-acting)	Depends on supply pressure, bearing geometry, speed, and fluid properties	Depends mainly on bearing and rotor geometry, fluid properties	Very low	Minimum problem	Separate compressor or other continuous high-pressure supply

TABLE IV

PROBLEM AREAS

Rotor-Bearings	Problem Areas
Self-Acting (Hydrodynamic)	Instability (Fluid Whirl, Resonant Whip) Dynamic Performance (Critical Speeds, Synchronous Whirl, Shock, Acceleration, Response to External Vibration) Performance under Misalignment Materials (Compatability, Fretting, Coefficient of Friction, Stability)
Hybrid (Externally Pressurized Bearing with Rotation)	Instability (Fluid Whirl, Resonant Whip, Pneumatic Hammer) Dynamic Performance (Critical Speeds, Synchronous Whirl, Shock, Acceleration, Response to External Vibration) Performance under Misalignment Lockup Pressurized Flow Requirements Materials (Erosion, Fretting, Stability)
Electrical Motors	Magnetic Dissymmetry Electrical Resonances Compatability of Materials with Gases and Vapors Unavailability of Motors with High Power, High Speed and Light Weight Unavailability of Motors with High Temperature Insulation
Reliability	Effect of Time on Performance

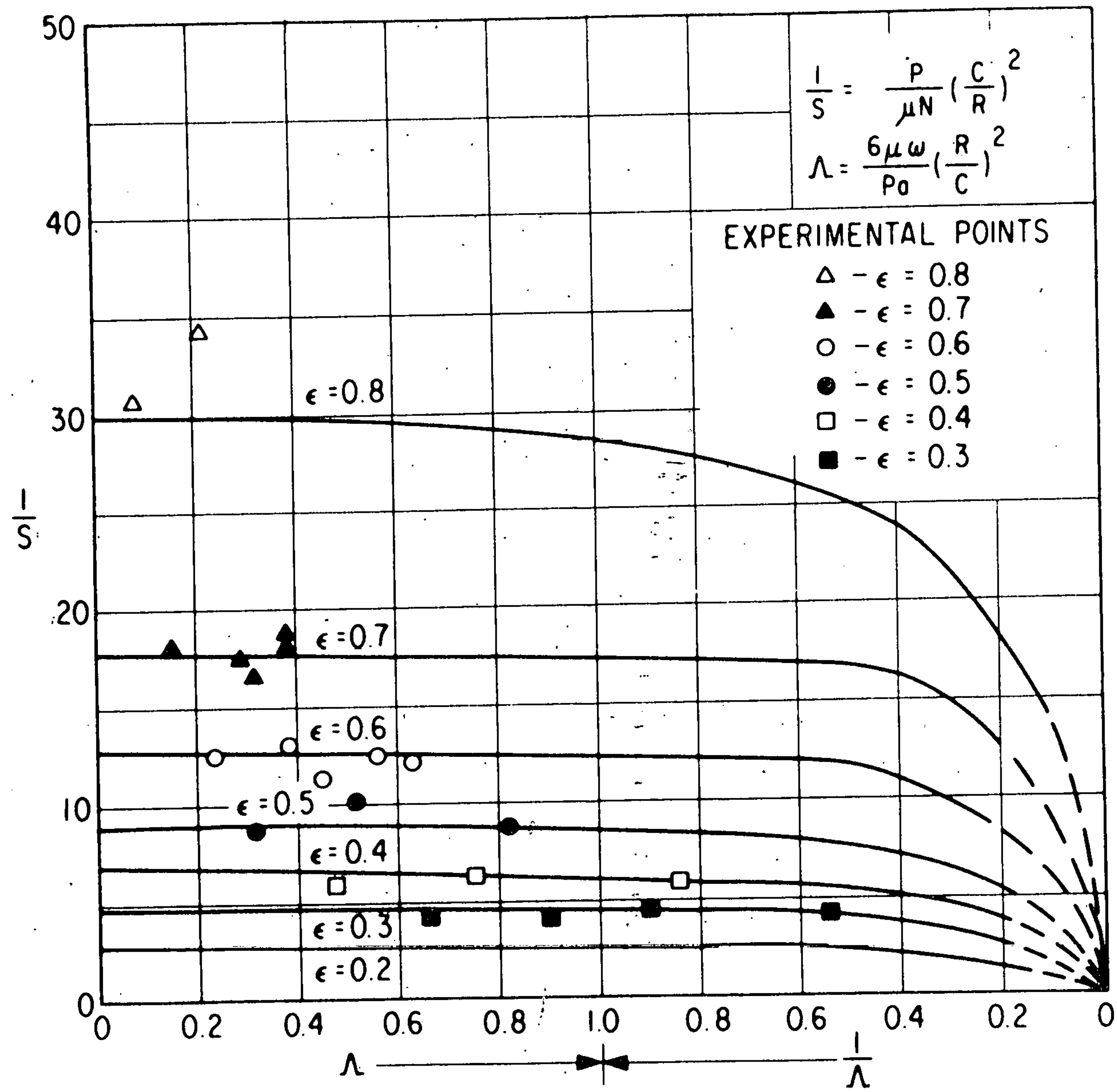
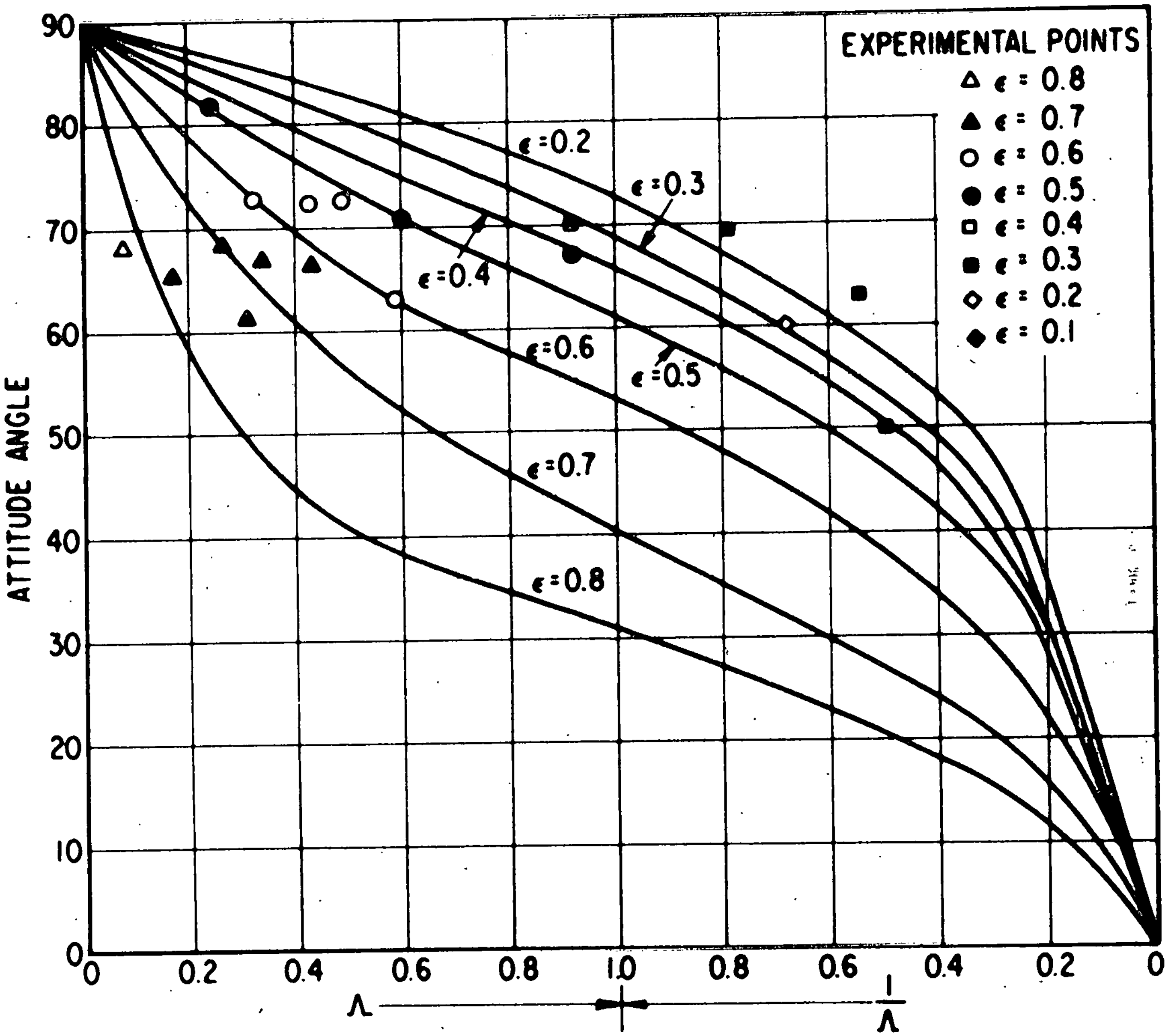

1/S = P/(μN) (C/R)^2
Λ = 6μω/Pa (R/C)^2
EXPERIMENTAL POINTS
△ − ε = 0.8
▲ − ε = 0.7
○ − ε = 0.6
● − ε = 0.5
□ − ε = 0.4
■ − ε = 0.3
ε = 0.8
ε = 0.7
ε = 0.6
ε = 0.5
ε = 0.4
ε = 0.3
ε = 0.2
1/S
Λ
1/Λ
0
10
20
30
40
50
0.2
0.4
0.6
0.8
1.0

Fig. 1 - 8.5



EXPERIMENTAL POINTS
△ ε = 0.8
▲ ε = 0.7
○ ε = 0.6
● ε = 0.5
□ ε = 0.4
■ ε = 0.3
◇ ε = 0.2
◆ ε = 0.1
ε = 0.2
ε = 0.3
ε = 0.4
ε = 0.5
ε = 0.6
ε = 0.7
ε = 0.8
ATTITUDE ANGLE
90
80
70
60
50
40
30
20
10
0
0
0.2
0.4
0.6
0.8
1.0
0.8
0.6
0.4
0.2
0
Λ
1/Λ

Fig. 2-B5.

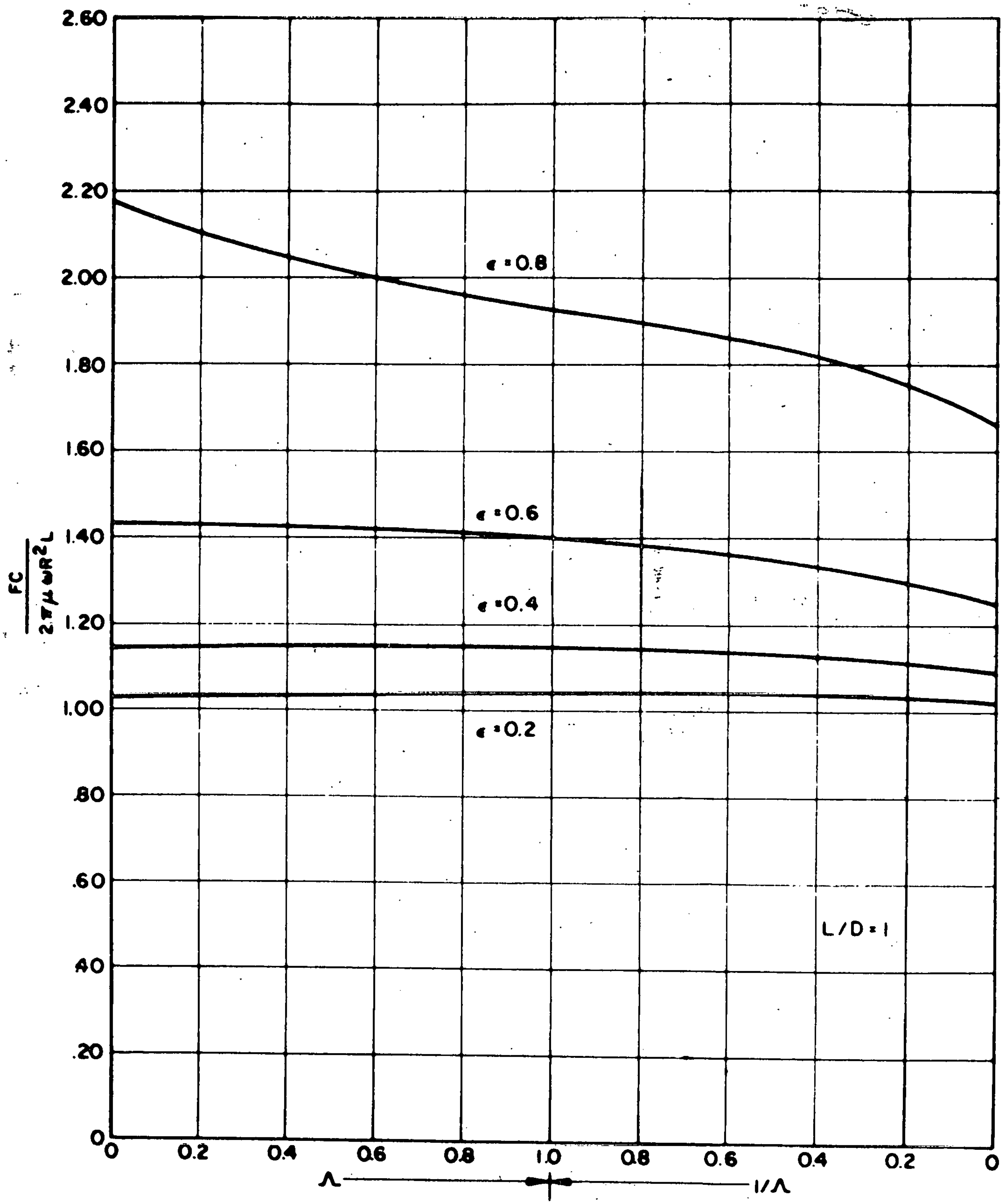

2.60
2.40
2.20
2.00
1.80
1.60
1.40
1.20
1.00
.80
.60
.40
.20
0
FC / 2π/μ ωR²L
ε = 0.8
ε = 0.6
ε = 0.4
ε = 0.2
L/D = 1
0
0.2
0.4
0.6
0.8
1.0
0.8
0.6
0.4
0.2
0
Λ
1/Λ

Fig. 3 - B.S.

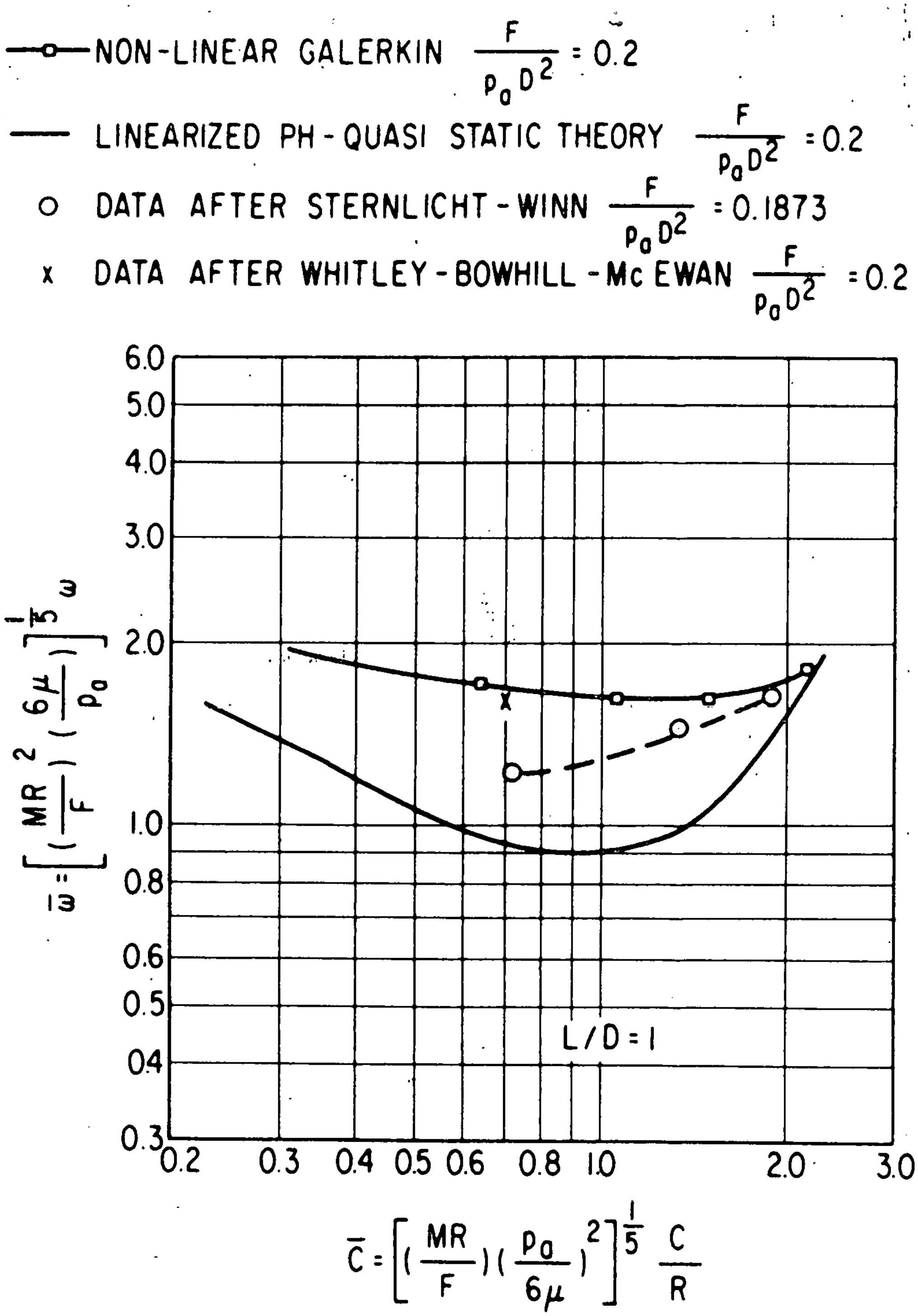
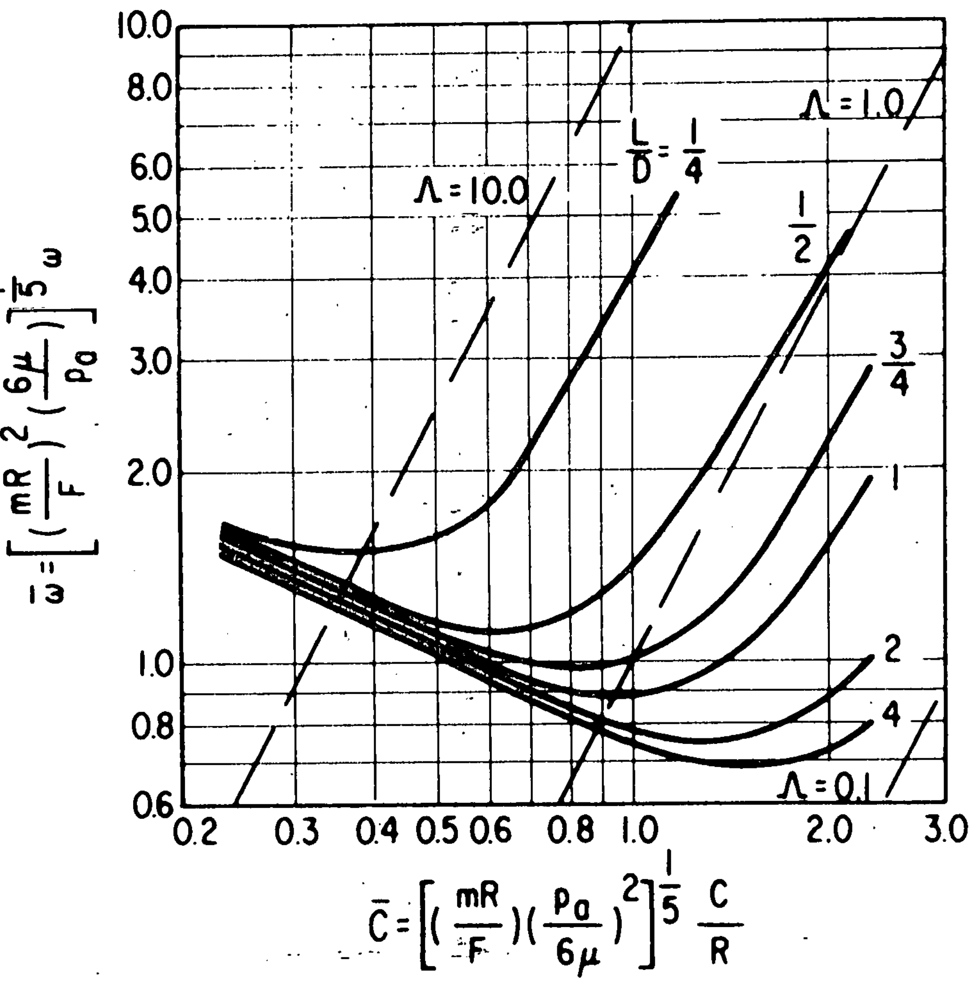

NON-LINEAR GALERKIN $\frac{F}{p_a D^2} = 0.2$
LINEARIZED PH - QUASI STATIC THEORY $\frac{F}{p_a D^2} = 0.2$
DATA AFTER STERNLICHT - WINN $\frac{F}{p_a D^2} = 0.1873$
DATA AFTER WHITLEY - BOWHILL - Mc EWAN $\frac{F}{p_a D^2} = 0.2$
6.0
5.0
4.0
3.0
2.0
1.0
0.8
0.6
0.5
0.4
0.3
0.2
0.3
0.4
0.5
0.6
0.8
1.0
2.0
3.0
L / D = 1
$\bar{\omega} = \left[\left(\frac{MR}{F} \right)^2 \left(\frac{6\mu}{p_a} \right) \right]^{\frac{1}{5}} \omega$
$\bar{C} = \left[\left(\frac{MR}{F} \right) \left(\frac{p_a}{6\mu} \right)^2 \right]^{\frac{1}{5}} \frac{C}{R}$

Fig. 4 - B.S.



10.0
8.0
6.0
5.0
4.0
3.0
2.0
1.0
0.8
0.6
0.2 0.3 0.4 0.5 0.6 0.8 1.0 2.0 3.0
$\Lambda = 10.0$
$\frac{L}{D} = \frac{1}{4}$
$\Lambda = 1.0$
$\frac{1}{2}$
$\frac{3}{4}$
1
2
4
$\Lambda = 0.1$
$\bar{\omega} = \left[\left(\frac{mR}{F} \right)^2 \left(\frac{6\mu}{p_a} \right) \right]^{\frac{1}{5}} \omega$
$\bar{C} = \left[\left(\frac{mR}{F} \right) \left(\frac{p_a}{6\mu} \right)^2 \right]^{\frac{1}{5}} \frac{C}{R}$

Fig. 5-8.5.

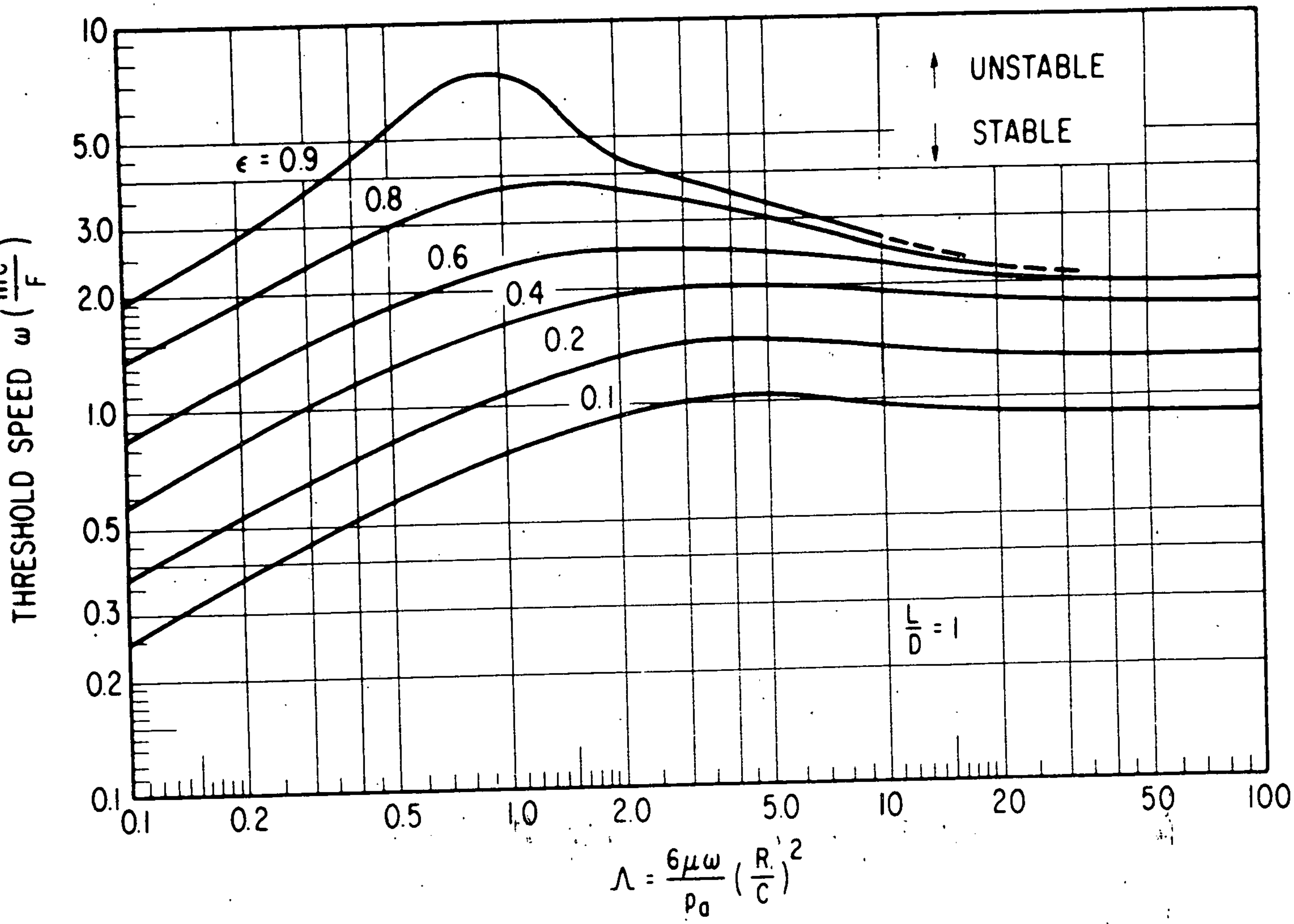

UNSTABLE
STABLE
ε = 0.9
0.8
0.6
0.4
0.2
0.1
L/D = 1
THRESHOLD SPEED ω (mC/F)^1/2
Λ = (6μω/pa)(R/C)^2
10
5.0
3.0
2.0
1.0
0.5
0.3
0.2
0.1
0.1
0.2
0.5
1.0
2.0
5.0
10
20
50
100

Fig. 6-BS.

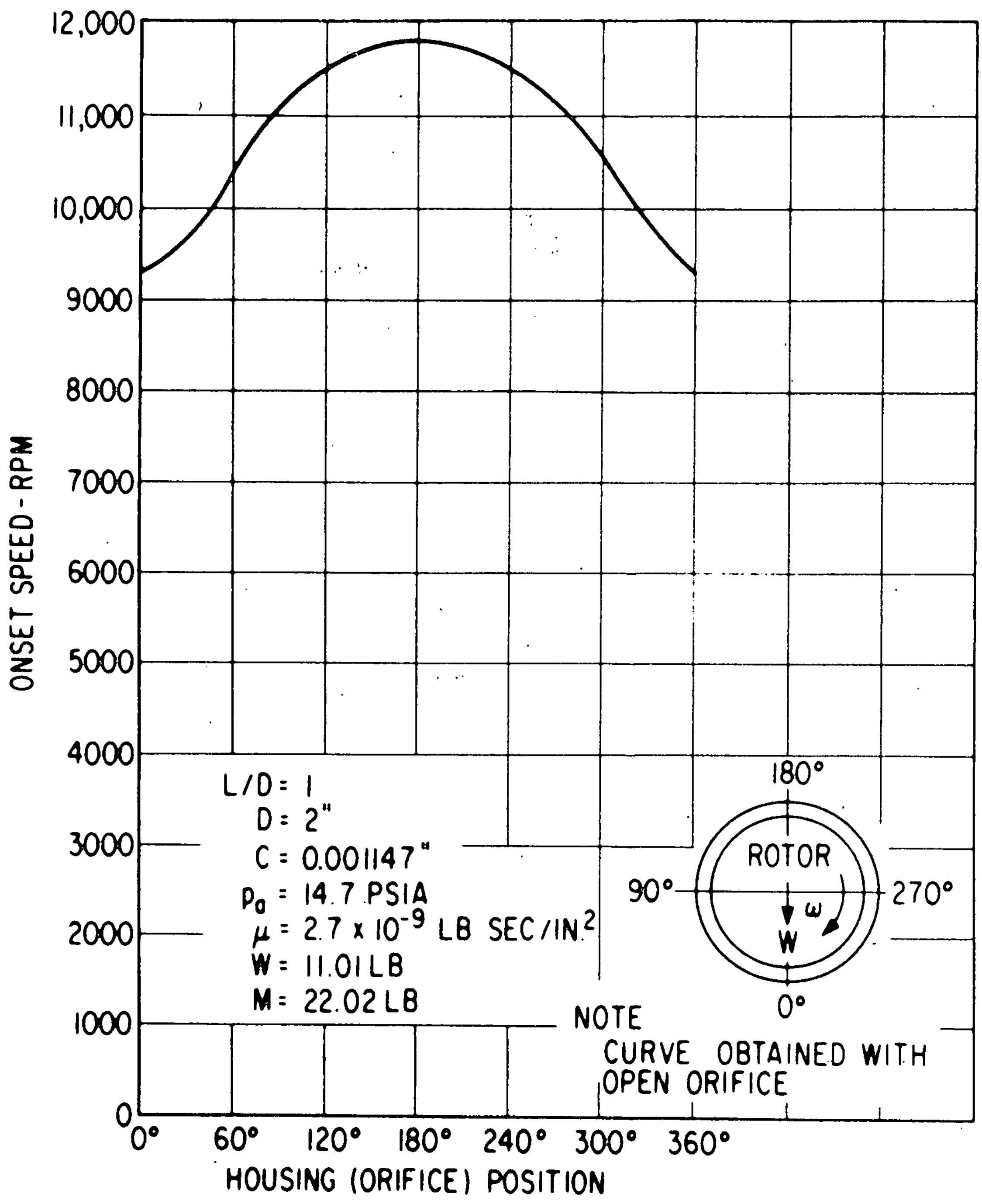
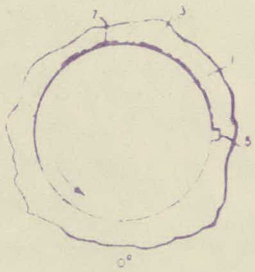
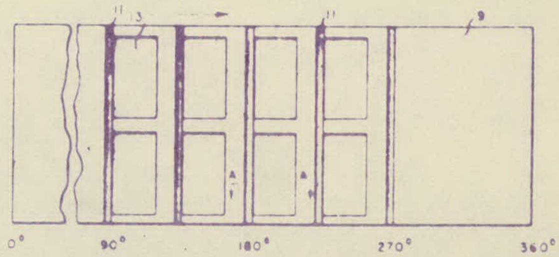
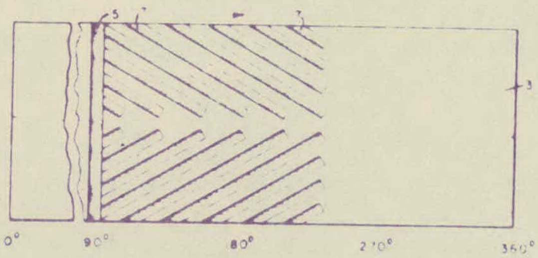
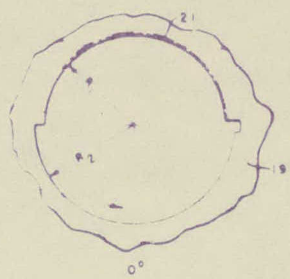

12,000
11,000
10,000
9000
8000
7000
6000
5000
4000
3000
2000
1000
0
ONSET SPEED - RPM
0°
60°
120°
180°
240°
300°
360°
HOUSING (ORIFICE) POSITION
L/D = 1
D = 2"
C = 0.001147"
p_a = 14.7 PSIA
μ = 2.7 x 10^{-9} LB SEC/IN^2
W = 11.01 LB
M = 22.02 LB
180°
ROTOR
90°
270°
ω
W
0°
NOTE
CURVE OBTAINED WITH OPEN ORIFICE

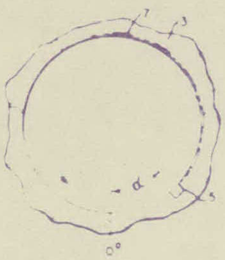
Fig. 7-8-5

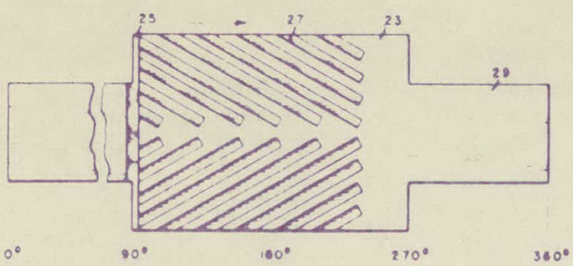


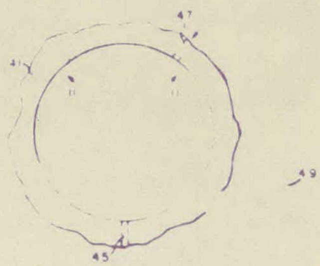


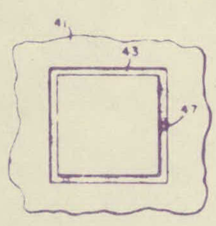












INVENTORS
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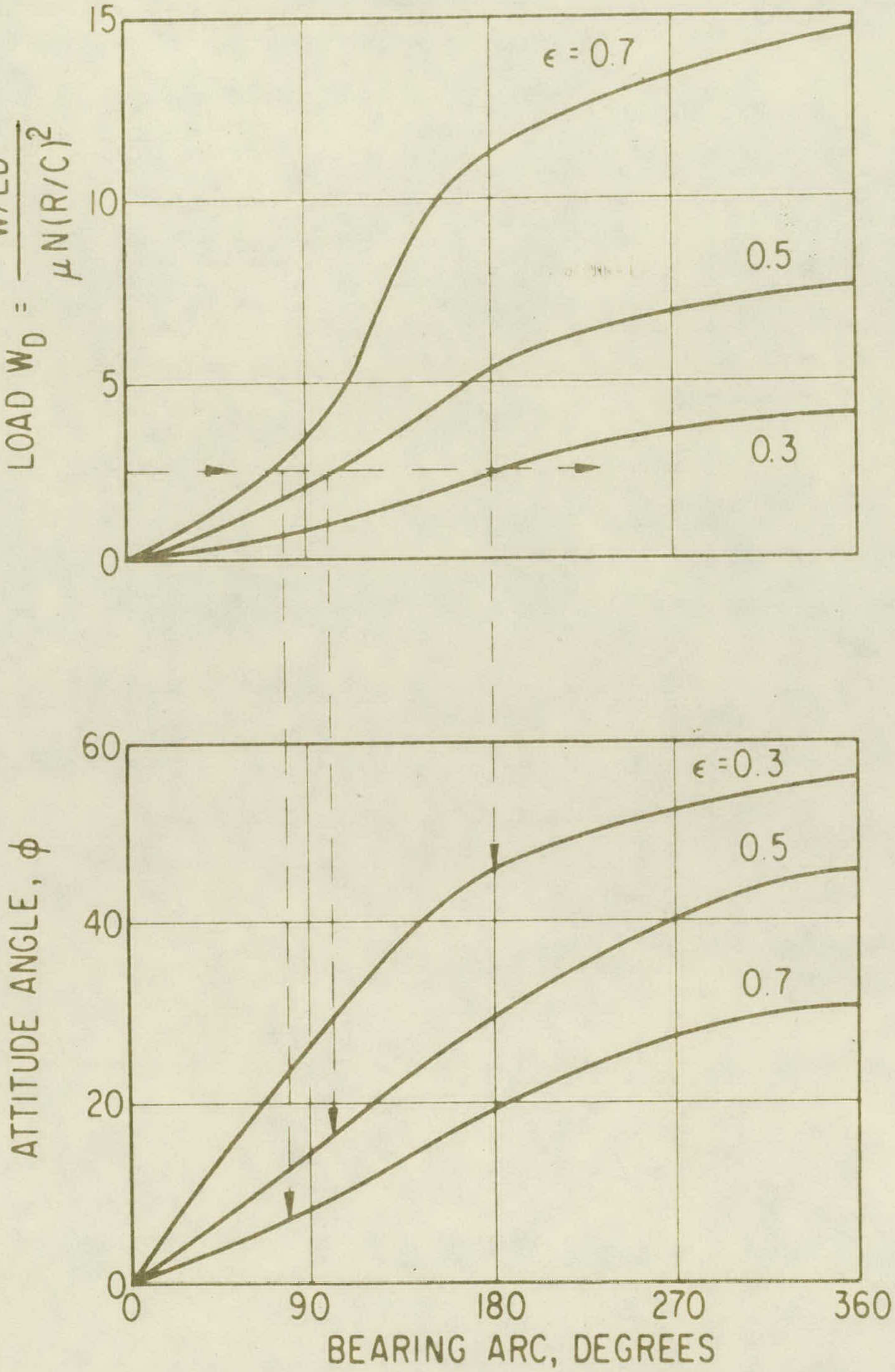
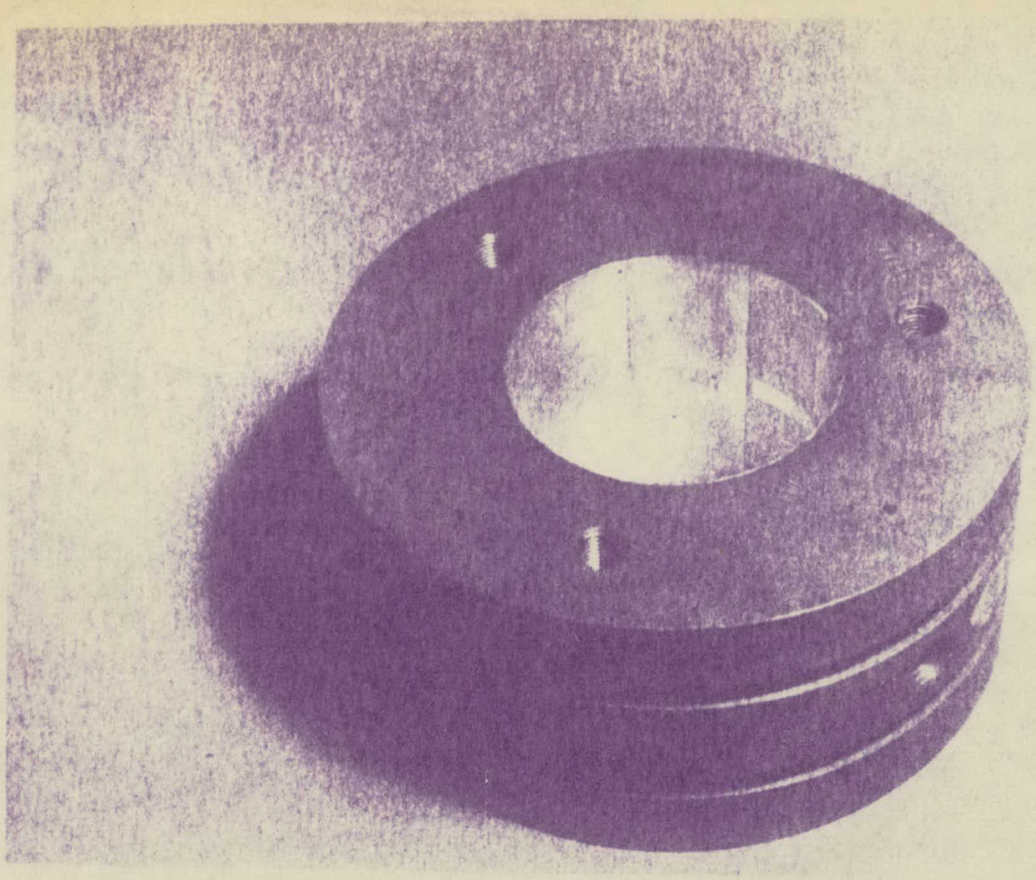
BENO STERNLICHT
BY
[illegible]
Attorney



LOAD $W_D = \frac{W/LD}{\mu N(R/C)^2}$
$\epsilon = 0.7$
0.5
0.3
15
10
5
0
ATTITUDE ANGLE, ϕ
$\epsilon = 0.3$
0.5
0.7
60
40
20
0
0
90
180
270
360
BEARING ARC, DEGREES

Fig. 9-85



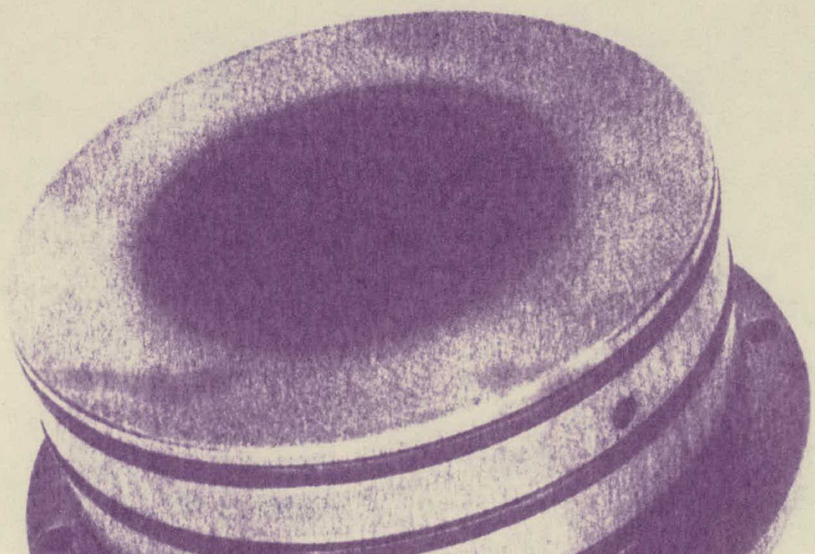
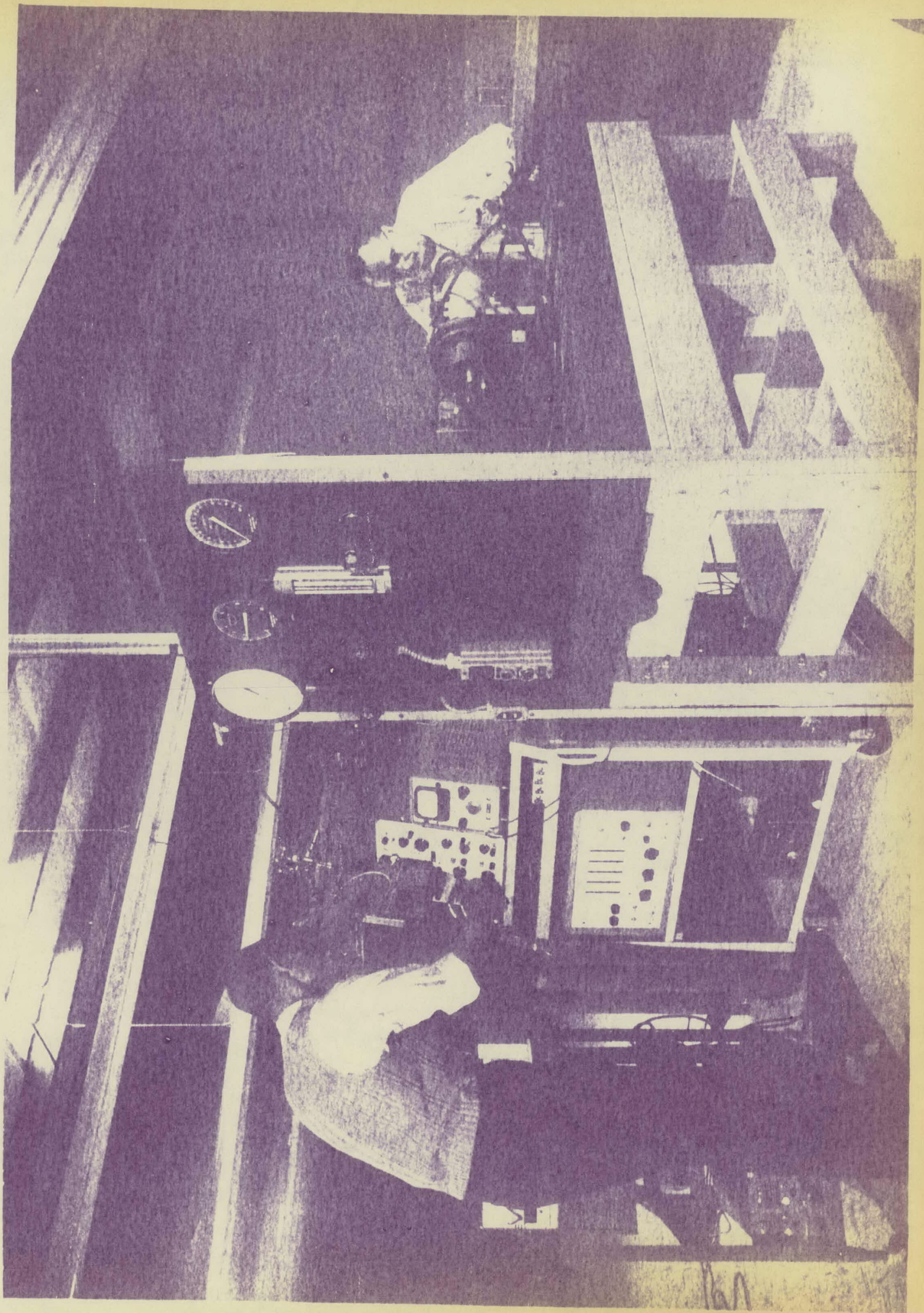
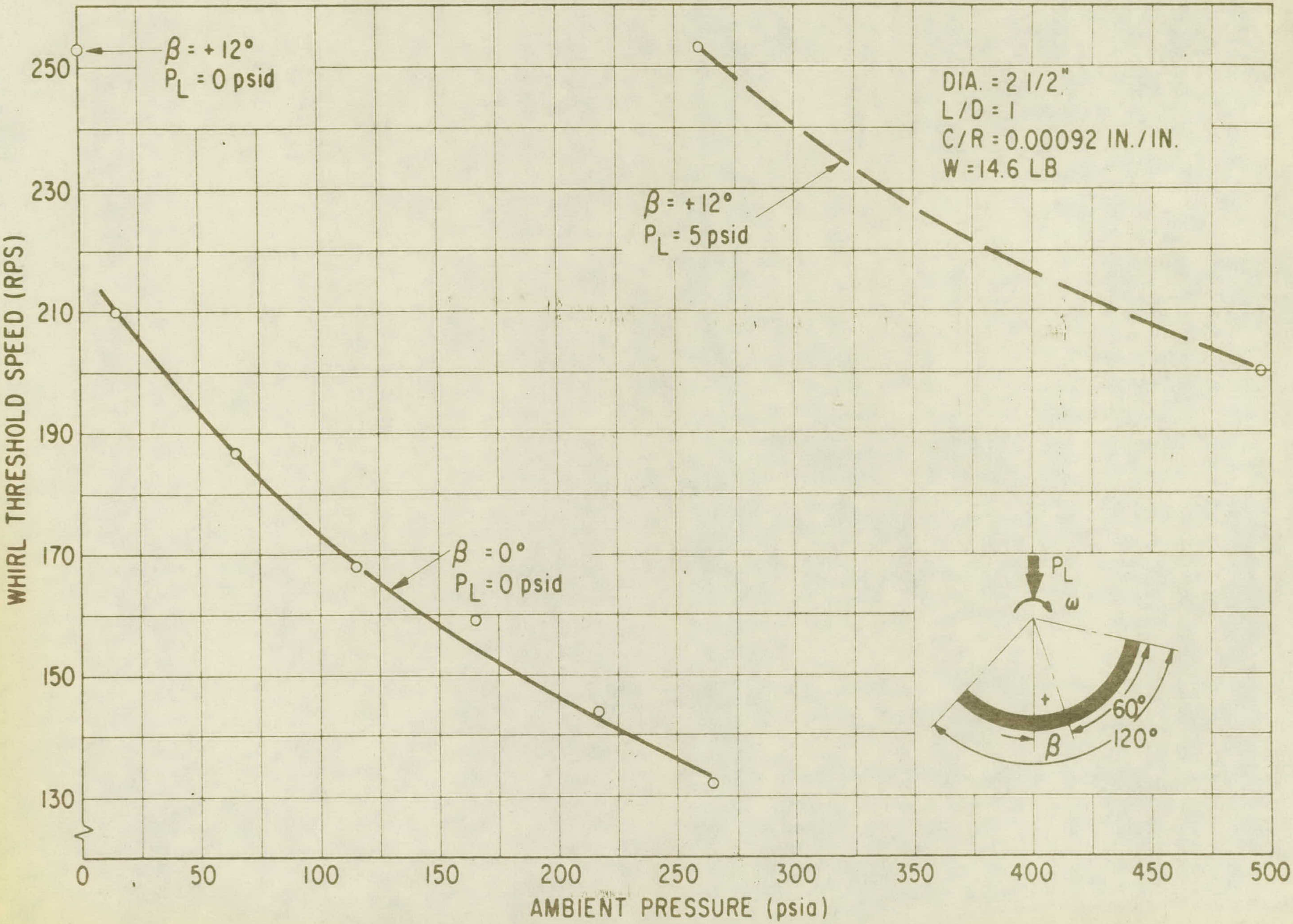


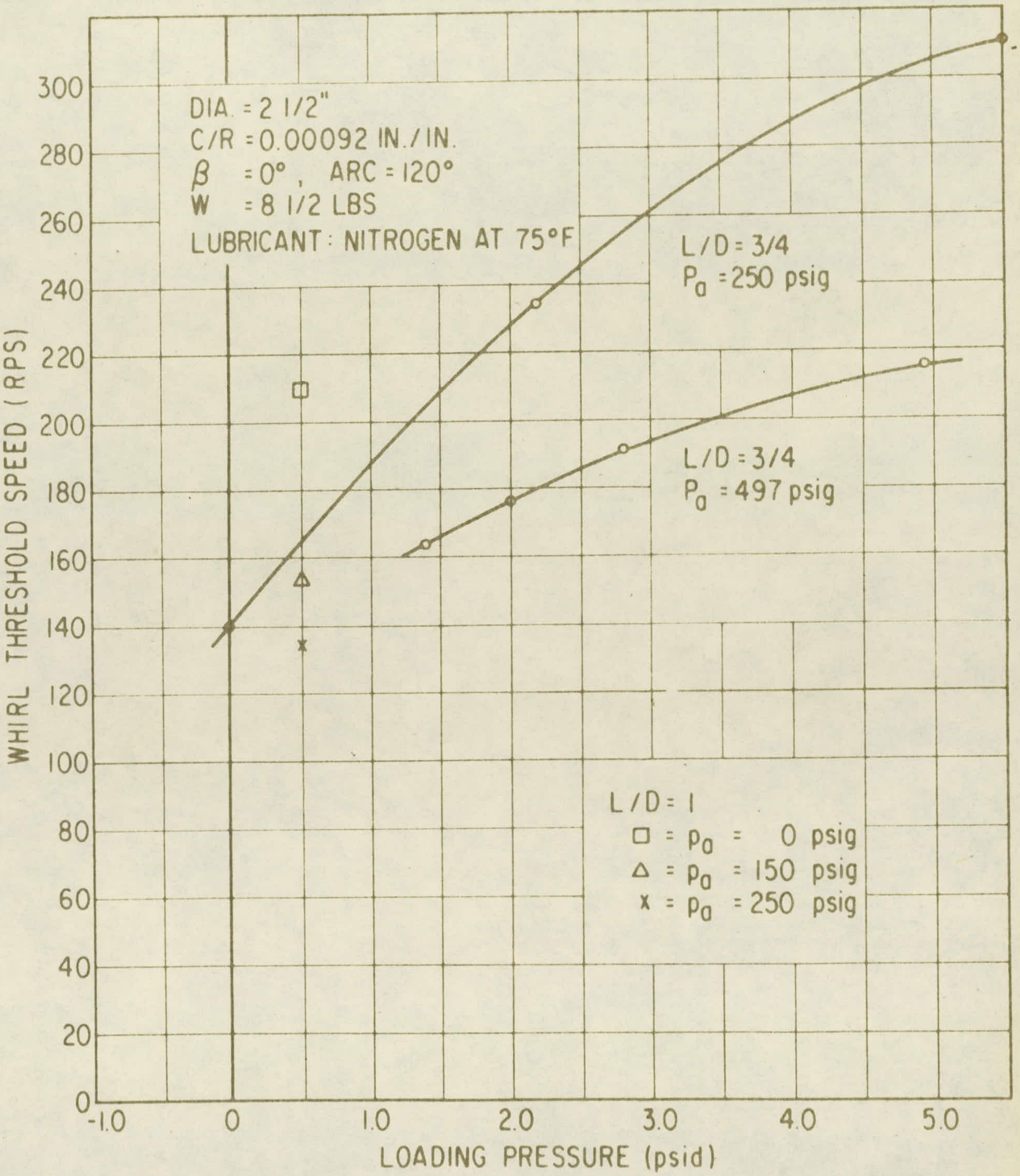
Fig 10





β = +12°
P_L = 0 psid
DIA. = 2 1/2"
L/D = 1
C/R = 0.00092 IN./IN.
W = 14.6 LB
β = +12°
P_L = 5 psid
β = 0°
P_L = 0 psid
P_L
ω
60°
120°
β
WHIRL THRESHOLD SPEED (RPS)
250
230
210
190
170
150
130
0
50
100
150
200
250
300
350
400
450
500
AMBIENT PRESSURE (psia)

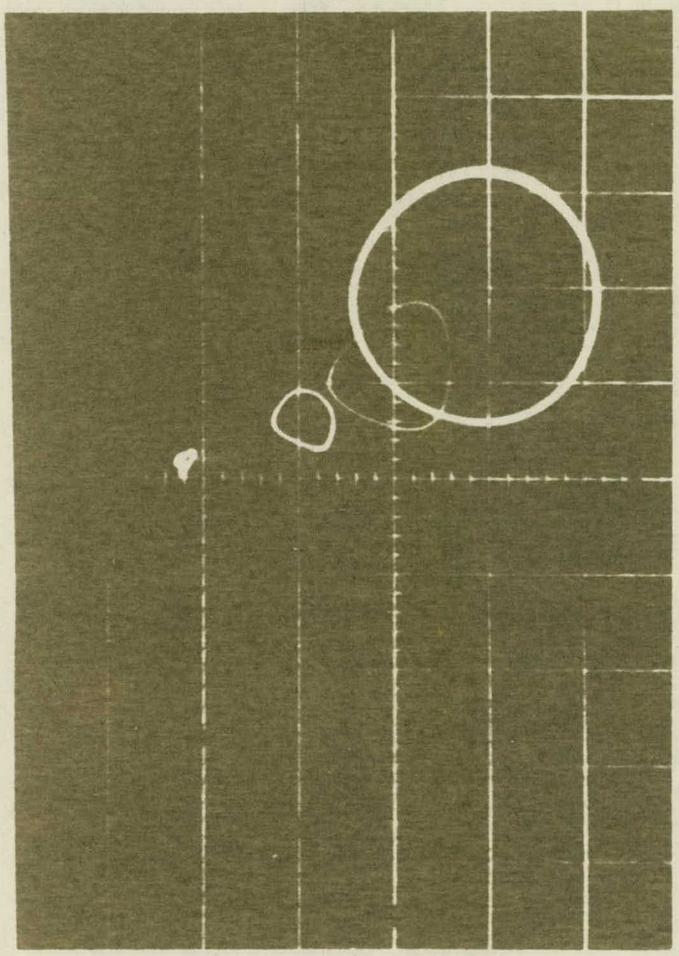
Fig. 12 B.S.



DIA. = 2 1/2"
C/R = 0.00092 IN./IN.
β = 0°, ARC = 120°
W = 8 1/2 LBS
LUBRICANT: NITROGEN AT 75°F
L/D = 3/4
Pa = 250 psig
L/D = 3/4
Pa = 497 psig
L/D = 1
□ = pa = 0 psig
△ = pa = 150 psig
x = pa = 250 psig
WHIRL THRESHOLD SPEED (RPS)
LOADING PRESSURE (psid)
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300
-1.0
0
1.0
2.0
3.0
4.0
5.0

Fig. 13. - B.S.

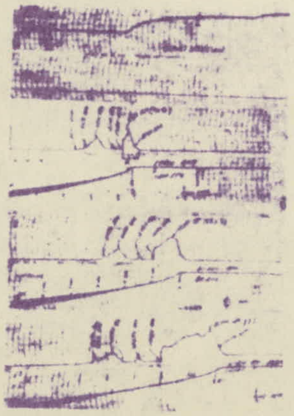
SYNCHRONOUS WHIRL



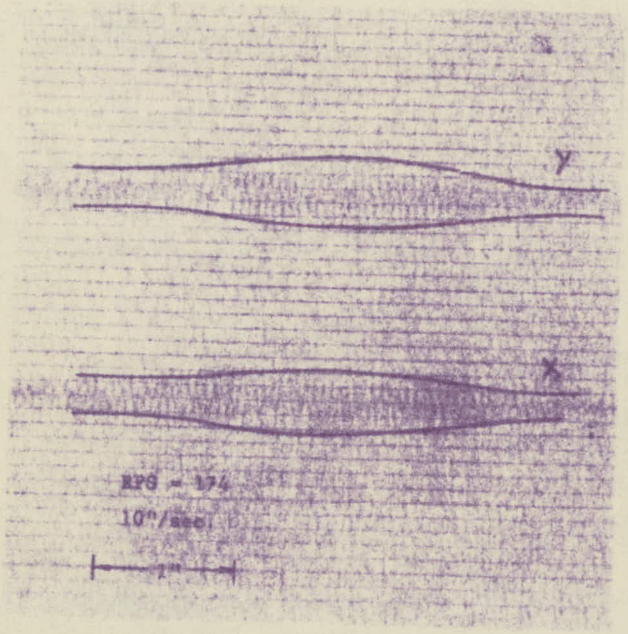
100, 200, 300 and 400 cps

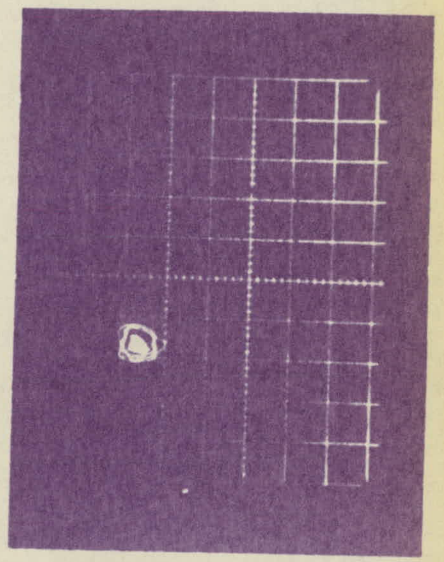
Fig. 14a

ELECTRICAL

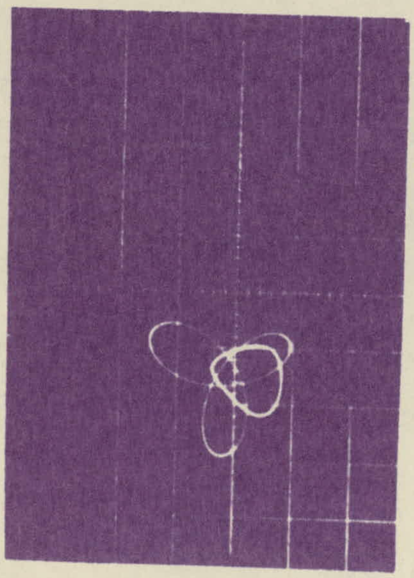


Electrical Resonance





Beat Frequency

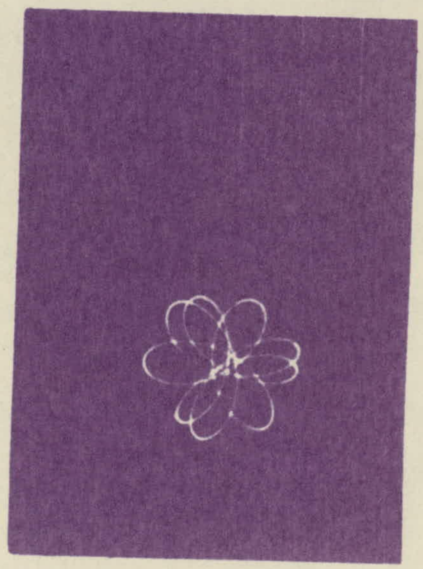


Mechanical $Ae^{i\omega t}$

Electrical $Be^{-i\omega t/2}$

$Ae^{i\omega t} + Be^{-i\omega t/2}$

20 mV/cm

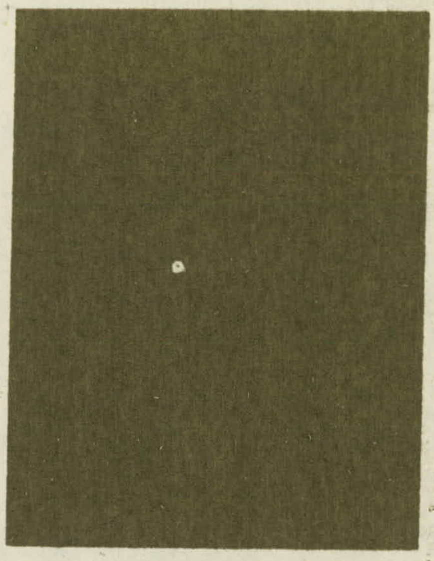


Electro-mechanical with Slip

FIG 14

SELF-EXCITED INSTABILITIES

Stable $Ae^{i\omega t}$



5000 rpm, 220 μ in/div

Unstable $Ae^{i\omega t} + Be^{i\omega t/2}$

Self-Acting

12,400 rpm, 220 μ in/div

Unstable - Hybrid

P_s = 275 psig P_a = 30 psig

317 cps

Fig. 14c

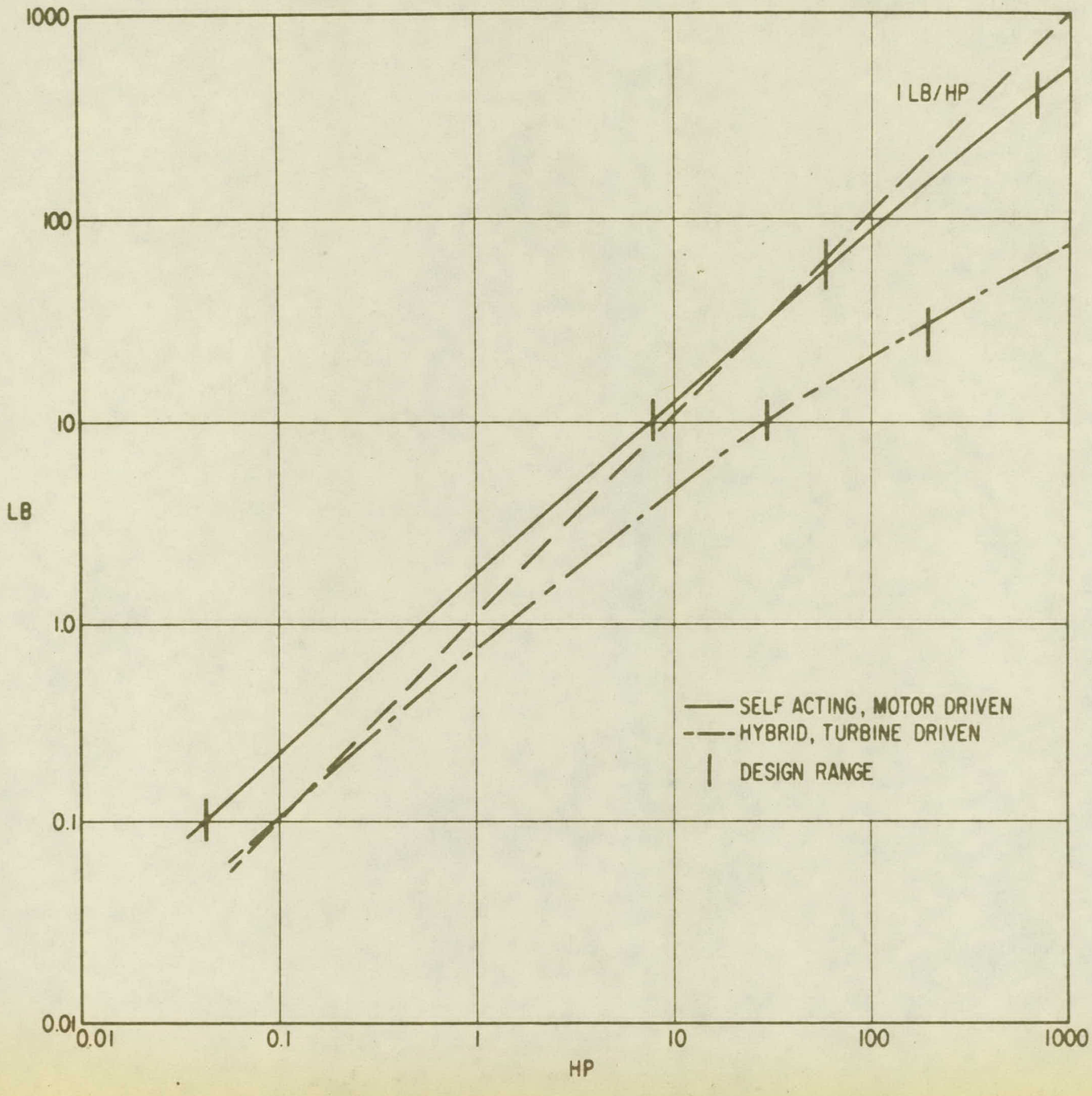

1000
100
10
1.0
0.1
0.01
LB
0.01
0.1
1
10
100
1000
HP
1 LB/HP
SELF ACTING, MOTOR DRIVEN
HYBRID, TURBINE DRIVEN
DESIGN RANGE

Fig. 15 - 83.

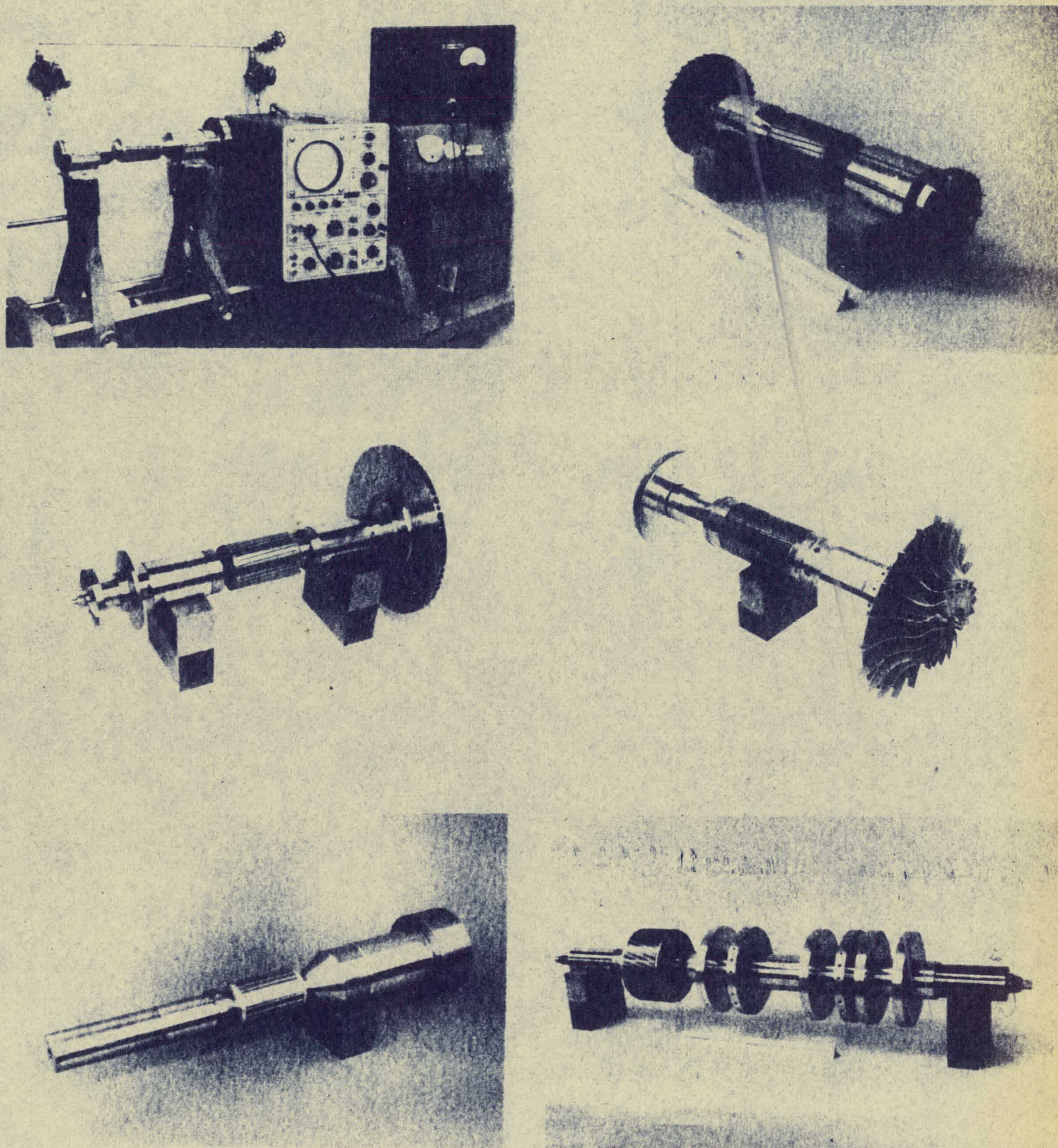
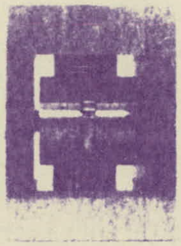
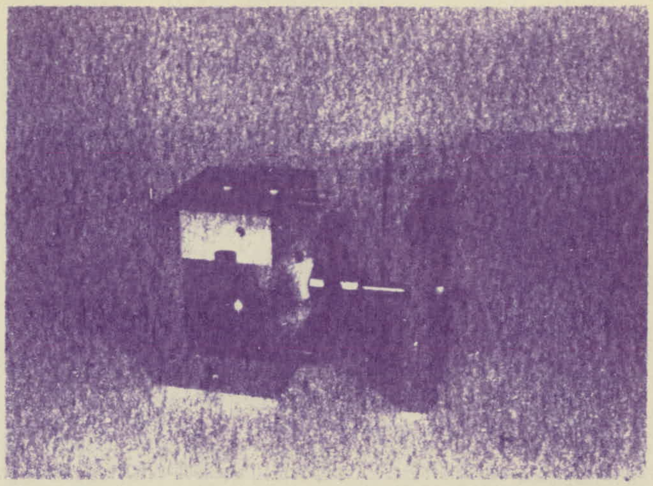


Fig 16 B.S.

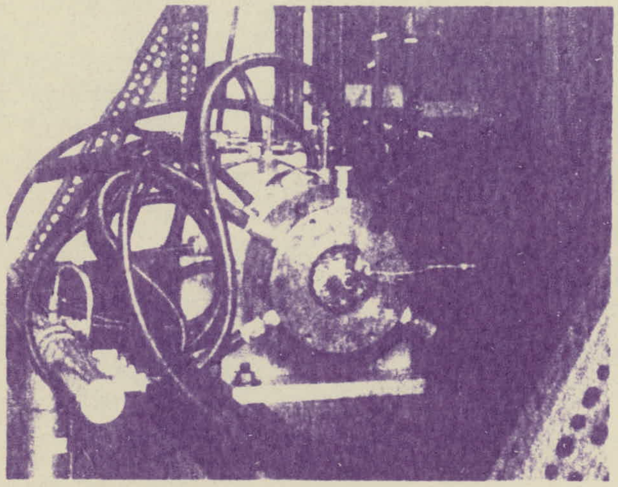
Test Rotor



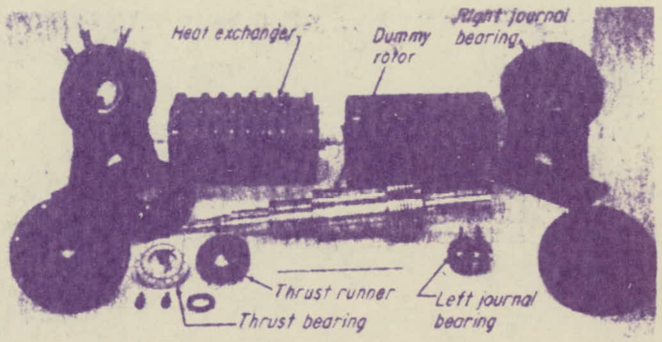
Test Rotor



100,000 RPM Rotor Test Rig

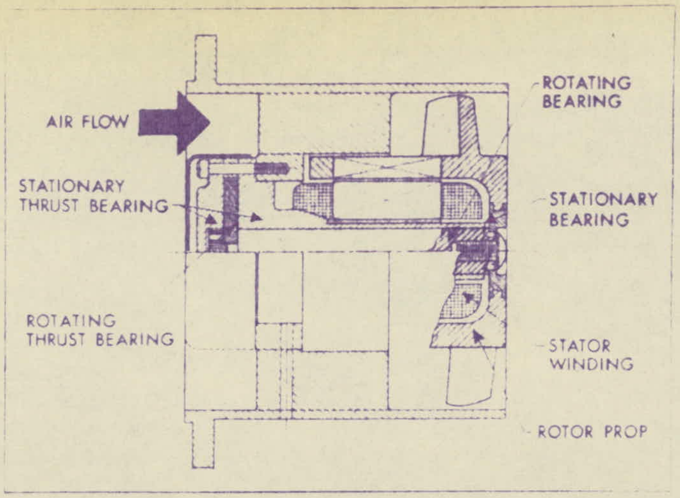


Test Rotor



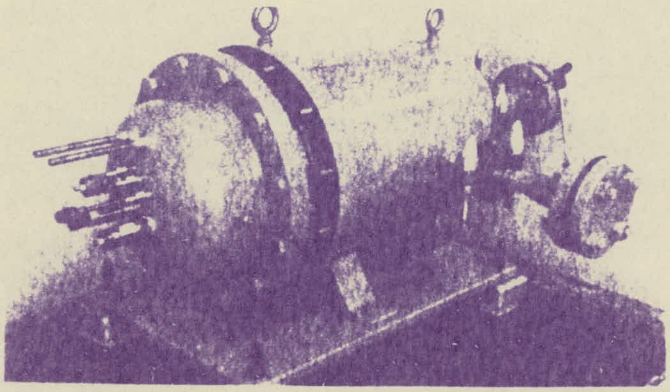
for externally-pressurized gas bearing system, closed-cycle gas turbine (item 19, in table)

Fan



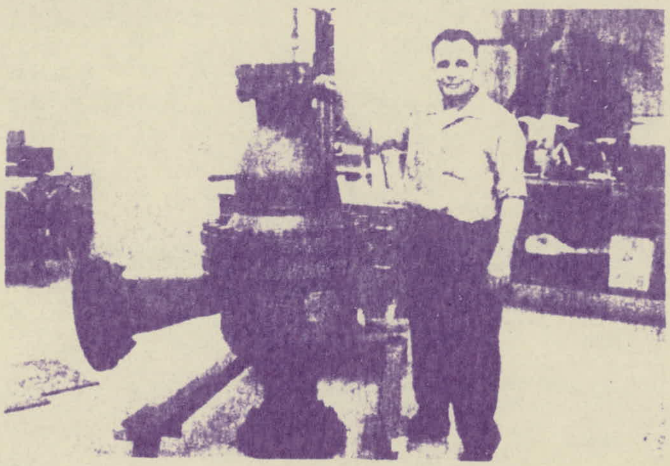
24,000 RPM, 0.4 HP

Helium Circulator

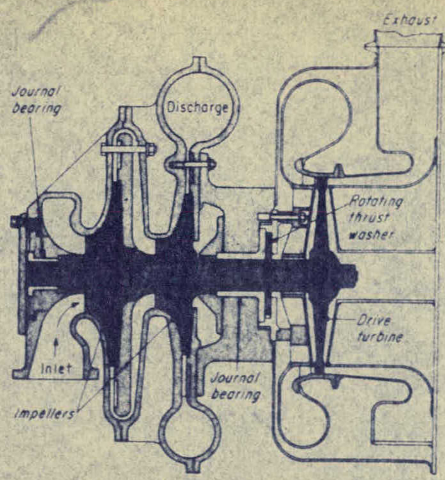


12,000 rpm, 8 hp, 100 F, 500 psi, self-action bearings (item 1, Table II)

Helium Circulator

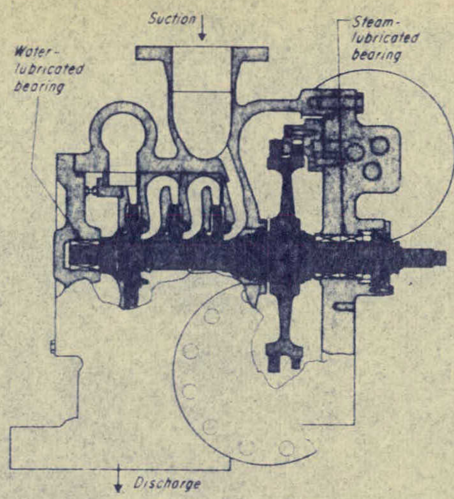


12,000 rpm, 75 hp, 600 F, 500 psi, self-acting bearings (item 2, Table II)



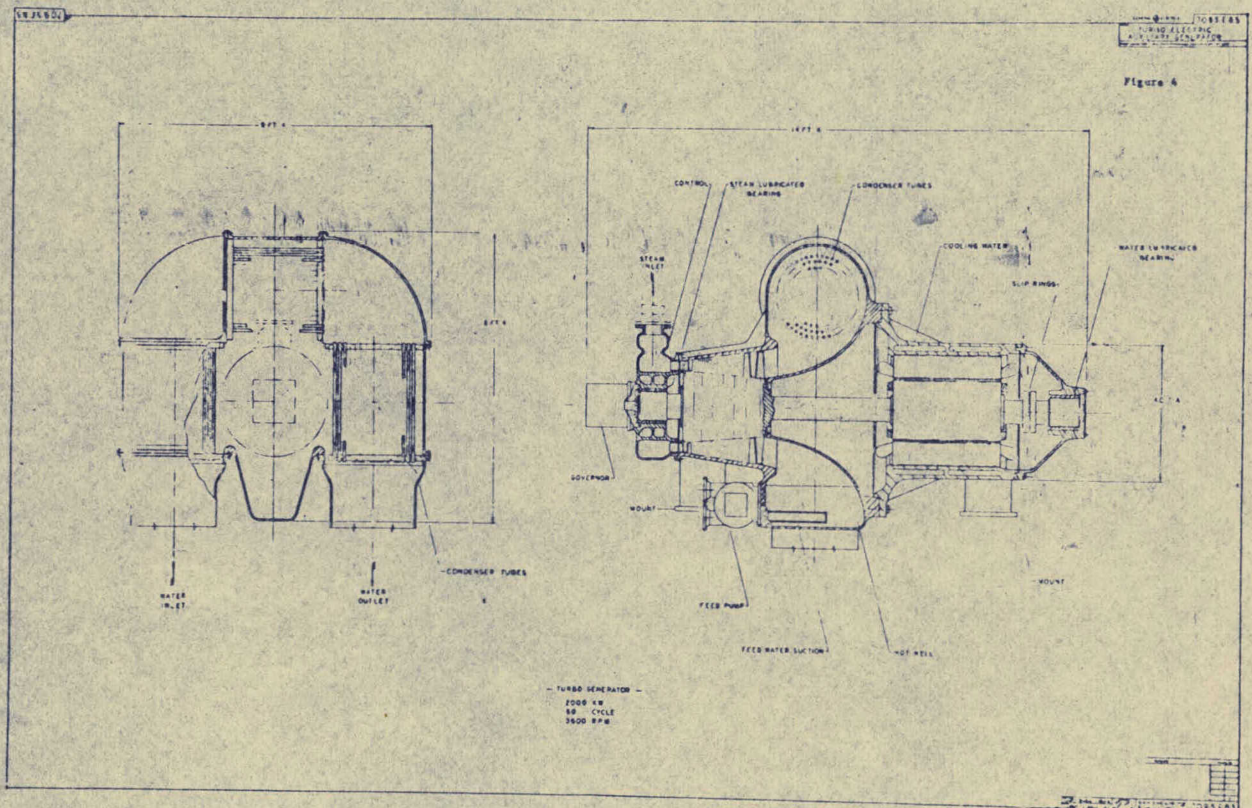
Turbine-Driven Compressor
supported on gas self-acting journal and thrust bearings

(items 8 and 16, Table II)



Turbine-Driven Pump
supported on one steam bearing and one water bearing

(item 20, Table II)



Turbo Generator

2000 KW, 60 cycle, 3600 rpm

Fig. 19 - 85