

THESIS.

.Author - K. Ghosh.

Supervisor - Dr. D.L. Leete.

Department of Production Engineering.

The Design and Development of
High Speed Externally Pressurized Air Bearings
for an Internal Grinder.

1968/1969.

20th September, 1969.

ProQuest Number: 10832163

All rights reserved

INFORMATION TO ALL USERS

The quality of this reproduction is dependent upon the quality of the copy submitted.

In the unlikely event that the author did not send a complete manuscript and there are missing pages, these will be noted. Also, if material had to be removed, a note will indicate the deletion.



ProQuest 10832163

Published by ProQuest LLC (2019). Copyright of the Dissertation is held by Cranfield University.

All rights reserved.

This work is protected against unauthorized copying under Title 17, United States Code
Microform Edition © ProQuest LLC.

ProQuest LLC.
789 East Eisenhower Parkway
P.O. Box 1346
Ann Arbor, MI 48106 – 1346

SUMMARY.

Briefly, the basic theory of externally pressurized air bearings which provides shaft support, with very low friction in high speed devices, has been discussed in this paper. Some considerations have been given to the various factors such as axial flow, circumferential flow, dispersion effect, compressibility effect and hydrodynamic effect.

The main objective was to produce an air bearing system capable of running between 45,000 to 80,000 rev/min., driven by a belt, with the radial and axial stiffnesses of 250,000 lb/in. and 150,000 lb/in. respectively. Moreover, the air consumption rate had to be as low as possible, at a maximum supply pressure of 80 lb/in². (gauge), while at the same time the manufacturing method had to be as simple as possible.

With the aim of satisfying these requirements, an experimental internal grinding head was designed and manufactured. Throughout the manufacturing stages, the most conventional production techniques were used; even the radial clearance of 0.0005 ins. between the shaft and the journal bearing was achieved by grinding and lapping, and also the smallest journal orifice hole diameter of 0.004 ins. was successfully produced by using a small drill.

The tests consisted mainly of the verification of the predicted values of stiffness, volumetric air flow and free running and the effect of the driving method. The complete tests were carried out in two test rigs one for direct drive and another for belt drive.

The test results revealed a satisfactory correlation between predicted and measured values. At the supply pressure of 80 lb/in². (gauge), the stiffnesses of 235,000 lb/in. and 244,000 lb/in. at the centre position of the rear and the front journal bearings, and 190,500 lb/in. and 200,000 lb/in. for the front and the rear thrust bearings respectively, were obtained. Without any rotation, the total air consumption was 0.95 ft³/min., which was 1 $\frac{1}{2}$ % higher than the predicted value but, at 60,000 rev/min. due to hydrodynamic effects, the reduction of air consumption of 7 $\frac{1}{2}$ % of the no rotational condition was observed. An actual speed of 75,000 rev/min. was successfully achieved by the experimental head. Further increase of speed was limited by the pull and the vibration induced by the belt.

ACKNOWLEDGEMENTS.

The author wishes to thank Dr. D. L. Leete, Senior Lecturer in the Production Engineering and Industrial Administration Department, who in the capacity of personal supervisor to the author has given extremely useful guidance and direction in this research work,

The author is indebted to Professor H. W. Shaw, Head of Department, for his valuable suggestions and advice throughout the work.

The author wishes gratefully to acknowledge the facilities made available by Mr. J. B. Elwell, Production Manager and Mr. J. D. Harrison, Chief Engineer of Heald Machines Ltd, Biggleswade. They are sincerely thanked for their constant support and encouragement, without which this work would not have been possible. Also the author wishes to express his appreciation to Mr. T. L. Woodsworth, Development Engineer of the Heald Machines Ltd. Biggleswade, for his help throughout the work.

Thanks are also due to the staff of the Machine Tool Laboratory, Engineering Workshop and Metrology Laboratory of the College of Aeronautics, for their help and in particular, Messrs: L. G. Webb, M. A. Wooding and H. A. Loffe of the Machine Tool Laboratory, and Mr. F. L. Clarke of the Engineering Workshop, for their keen interest and assistance with the manufacture and assembly of experimental apparatus.

Finally, Mrs. K. Ghosh is thanked for her extreme patience and untiring efforts in the typing of this thesis.

List of contents.

Summary	1
Acknowledgements	2
List of contents	3
Nomenclature	5
1.1. Introduction	9
2.1. Theory	12
2.2. Principle	12
2.3. Pressure drop in a bearing and its effect	13
2.4. Axial flow model	15
2.5. Influence of circumferential flow	16
2.6. Dispersion effect	17
2.7. Compressibility effect	18
2.8. Influence of Reynolds equation on speed	18
2.9. Effect of orifice configuration	19
2.10. Effect of number of orifices	20
2.11. Effect of orifice position	21
2.12. Effect of length to diameter ratio	21
2.13. Principle of thrust bearing	21
2.14. Dynamic properties	23
2.14.1. Pneumatic hammer	23
2.14.2. Whirl instabilities	24
2.14.3. Mechanical critical speeds	26
2.14.4. Unbalance	27
3.1. Design	28
3.2. Journal bearing	28
3.3. Thrust bearing	32
3.4. Summary	35
3.4.1. Journal bearing	35
3.4.2. Thrust bearing	35
3.5. Stiffness and flow	36
3.6. Stability	40
3.6.1. Pneumatic hammer	40
3.6.2. Whirl instabilities	41
3.6.3. Mechanical critical speeds	43

3.6.4.	Unbalance	44
3.6.5.	Measurement of radial stiffness	44
3.7.	Friction	44
3.8.	Reynolds number and its effect	46
3.9.	Supply hole	48
3.10.	Influence of material properties	50
3.10.1.	Machinability	50
3.10.2.	Physical and metallurgical stability	50
3.10.3.	Thermal conductivity and expansion	51
3.10.4.	Strength and Elasticity	51
3.10.5.	Chemical stability	51
3.11.	Materials	51
3.11.1.	Alloy steel	52
3.11.2.	Bronzes	52
3.12.	Air supply	52
3.13.	Manufacturing capability	53
3.14.	Overall dimensions	54
3.15.	Safety factor	54
4.1.	Manufacturing technique	55
4.2.	Bearings	55
4.3.	Shaft	56
4.4.	Orifices	57
4.5.	Assembly	58
5.1.	Description of tests	60
5.2.	Apparatus	60
5.3.	Experimental results	61
5.3.1.	Load capacity and stiffness	61
5.3.2.	Volumetric air flow	62
5.3.3.	Free running	62
6.1.	Discussion	63
7.1.	Conclusions	66
8.1.	Figures	67
9.1.	References	139
10.1.	Bibliography	144
11.1.	Future work	149

NOMENCLATURE.

γ	C_p/C_v ratio of specific heats of air, 1.41 - (10)*.
μ	Viscosity of air, 2.62×10^{-9} lb. sec / in ² - (20).
δ	Non-dimensional orifice compensation factor.
Λ	Non-dimensional instability parameter.
Λ	Non-dimensional compressibility number.
Λ_m	Non-dimensional mean compressibility number.
Λ_s	Non-dimensional feeding parameter or restrictor coefficient.
ν	Kinematic viscosity, 2.28×10^{-5} in ² / sec ² - (20).
ρ	Density of air, 1.15×10^{-7} lb. sec ² / in ⁴ - (10).
ϵ	Eccentricity ratio, e/h .
ω	Angular velocity of shaft, rad / sec.
ω_{c1}	Cylindrical whirl, rad / sec.
ω_{c2}	Conical whirl, rad / sec.
a	Radius of orifice, inch.
d	Diameter of orifice, inch.
d_r	Diameter of orifice pocket, recess or feeder hole, inch.
d_s	Diameter of supply hole in main body, inch.
e	Eccentricity, bearing-journal centre distance, inch.
f_n	Natural frequency of shaft, cycle / sec.
g	Gravitational acceleration, ft / sec ² .
h	Radial clearance between shaft and bearing, inch.
h_r	Depth of recess or feeder hole, inch.
l	Distance of plane of orifice holes from end of bearing, inch.
m_r	Mass of rotor, lb. sec ² / in.
n	Number of orifices.

* Indicates reference number.

$\bar{n}$	Half speed whirl, 2 - (41).
r	Shaft radius, inch.
A	Effective orifice hole area, in^2 .
C	Diametral clearance, inch.
C_d	Coefficient of discharge, 0.8 - (8).
C_L	Non-dimensional load coefficient based on length and diameter of bearing, both journal and thrust.
C'_L	Non-dimensional load coefficient based on diameter of bearing.
C_{LO}	Non-dimensional theoretical short journal bearing load coefficient.
C_O	Choking diametral clearance, inch.
D	Diameter of shaft, inch.
D_w	Diameter of grinding wheel, inch.
E	Modulus of elasticity, $30 \times 10^6 \text{ lb} / \text{in}^2$.
F_d	Bearing dimension factor.
F_f	Frictional horse power, H.P.
F_g	Gas properties, ft.
F_p	Pressure ratio factor.
F_r	Radial force, lbf.
F_t	Tangential force, lbf.
G	Slot factor.
I	Moment of inertia, in^4 .
J	Half distance between centres of journal bearings, inch.
K	Journal bearing air film stiffness - spring constant, lb/in .
K'	Non-dimensional stiffness parameter.
K_A	Thrust bearing air film stiffness, lb/in .
K_g	Gauge pressure ratio - bearing pressure factor.
L	Length of bearing, inch.

L_1	Distance between journal bearing centres, inch.
M	Mach number, 1 - (9).
$\bar{M}$	Total mass flow, lb/sec.
M'	Non-dimensional flow factor.
$\bar{M}_i$	Total mass flow in ideal condition, lb/sec.
$\bar{M}_o$	Mass flow for single orifice, lb/sec.
N	Required speed, rev/min.
N_c	Critical speed before turbulence, rev/min.
N_{c1}	Cylindrical whirl speed, rev/min.
N_{c2}	Conical whirl speed, rev/min.
N_{c3}	Mechanical critical speed, rev/min.
N_m	Speed of motor, rev/min.
N_w	Speed of grinding wheel, rev/min.
P	Perimeter of recess, inch.
P_a	Ambient pressure, lb/inch ² .
P_c	P_d / P_s - critical pressure ratio.
P_d	Orifice downstream pressure (Absolute), lb/inch ² .
P_m	Effective mean pressure, lb/inch ² .
P_o	Supply pressure (Gauge), lb/inch ² .
P_s	Supply pressure (Absolute), lb/inch ² .
Q	Quantity of flow, ft ³ /min.
Q_t	Total quantity of flow, ft ³ /min.
R	Gas constant, 2.47×10^5 in ² /sec ² . (10).
R_A	Reaction load on front bearing, lb.
R_B	Reaction load on rear bearing, lb.
R_c	Radius of ring of orifices for thrust bearing, inch.

R_e	Reynold's number.
R_i	Inner radius of thrust bearing, inch.
R_o	Outer radius of thrust bearing, inch.
R_w	Radius of grinding wheel, inch.
T	Absolute temperature, $^{\circ}$ Rankine.
T_f	Frictional torque, lb. in.
T_{ft}	Total frictional torque, lb. in.
T_m	Motor torque, lb. in.
T_w	Grinding wheel torque, lb. in.
U	Linear velocity of shaft surface, in/sec.
V_c	Recess pocket volume, inch ³ .
V_s	Surface velocity of grinding wheel, ft/min.
W	Load, lb.
W_i	Load in ideal condition, lb.

1.1. INTRODUCTION.

The basic principle of air bearings was propounded over a century ago by Hirn (1). This was not discussed again until Kingsbury (2) built an air lubricated journal bearing and published some excellent experimental data in 1897. It is only in the last decade that the full potential of air bearings has begun to be appreciated. This sudden revival was due to the persistent demands of industry, in some specialized fields, where high speeds and lower frictional torque of a bearing are the main requirements.

Many applications of air bearings in Machine tools and measuring instrument design in United Kingdom are given by Wunsch (3); also a comprehensive account of their various applications in United States are stated by Sternlicht (4). But for the purposes of this research work, it is convenient to consider only the application of high speed rotary air-bearings in machine tool design.

Amongst all possible applications of the high speed air bearing in the machine tool field, there is very little doubt that the grinding machine is likely to be most appropriate. With the exception of largest grinders, the loads on the bearings are fairly small, moreover, they are not periodic. Both these factors favour air bearings, while at the same time the precision nature of the grinding process, and the high speeds require a bearing which is as vibration free and cool running as possible.

While considering the use of high speed air bearings, a preliminary survey revealed that as early as 1952, such externally pressurized air bearings for high speed internal grinders were being described by Mueller (5) from the U.S.A., Sheinberg (6) from Russia and Lehmann (7) from East Germany. It can, therefore, be seen that the advantages of using air bearings under conditions of correspondingly light loads and high speeds have been recognised for some time. However, it should be pointed out that none of them has used a belt to drive the grinding spindle, which obviously has the disadvantage of vibration induced by the belt.

There are certain obvious advantages which could be achieved the use of externally pressurized air spindles. The problem of heating and friction is virtually eliminated, since the arrangement of the pressurized bearing allows the air to be forced into the clearance, thus dissipating the heat generated during running; also the low viscosity of air in comparison to oil results in extremely low viscous friction properties even at high speed and temperature. Again this

type of bearing has the ability to support the safe working load without any metal to metal contact even at zero speed, which eliminates surface wear and the problem of 'stickslip', thus giving unlimited troublefree service.

Also the presence of a complete air film around the circumference of the spindle at all times results in a higher rotational accuracy than the spindle geometry would normally allow, due to the averaging action of the air film. There is also a unique advantage associated with the use of air as lubricant; no other lubricant is so widely available or could be so easily discharged without contaminating the surrounding atmosphere. This also leads to simplified design at a low cost basis. Finally, an added feature of the externally pressurized grinding head is that it would not require any 'warm up' period prior to grinding operation.

Against all these advantages, of course, it is necessary to consider some disadvantages in using the air bearing. One of the major problems is the need for a supply of clean and reasonably dry air.

Small amount of oil in the line could be tolerated, but unfortunately a small solid particle has an adverse effect on the bearings. Also argument has often been put forward about its low load carrying capacity; although this argument applies for self-acting bearings, for the case of externally pressurized air bearing, the load capacity is directly dependent on the supply pressure, and the industrial compressed air pressure is 100-120 lb/in², this restricts the design of air bearings.

Another points sometimes raised as an objection to the use of air bearings in an industrial environment is the need for, and cost of, a supply of clean, dry, compressed air which includes the cost of filtration unit and a somewhat higher running cost. In the long term, however, it will undoubtedly be cheaper to improve the quality of the factory air supply by the installation of better quality compressors and air dryers and, most important, by proper maintenance. This latter factor will, in many installations, also reduce costly air flow losses from leaks in factory lines to an extent far outweighing the commonly small air flow requirement of the air bearings in use.

However, in the light of many advantages likely to be achieved by using air bearings in high speed internal grinding applications, a prototype grinding head was designed and manufactured in collaboration with a machine tool firm. This research work was carried out from the basic design to the final development. While the designing work of air bearings was carried out, every precaution was taken to avoid any

special manufacturing technique to reduce the cost. This aspect of manufacturing method has been further discussed in the latter section.

2.1. THEORY.

The large scale use of externally pressurized air bearings has been generally hindered by the cumbersome methods of calculation required. This is why some difficulty was experienced in designing the high speed air bearing for the internal grinder. As the main object of this project was to design and develop an air bearing, some basic knowledge of the way in which these bearings work and of concepts like flow model, orifices, compensation and dynamic characteristic seemed desirable.

The theoretical equations for the idealized hydrostatic air bearing, assuming purely axial flow in the clearance space, has been verified by Shires (8), being based on the well known equation for flow in a slot and flow through an orifice. In this theory some assumptions have been made which are summarized as follows.

1. There are no pressure losses upstream of the throat of the orifice.
2. The pressure immediately downstream of the orifice is equal to the static pressure in the orifice throat.
3. The flow through the orifice is adiabatic.
4. The flow in the clearance between shaft and the journal is axial.
5. Inertia forces are negligible compared with viscous forces.
6. Laminar flow conditions exist at all points in the clearance space.
7. The temperature is constant throughout the clearance space.
8. The pressure drop in the clearance space is linear.

Although the theory and principle discussed in this chapter is based on above assumptions, some considerations have been given to the circumferential flow, compressibility effect and dispersion flow.

2.2. PRINCIPLE.

A simplified externally pressurized journal bearing is shown in Fig. 1. In this, the pressurized air P_s is supplied in to the plenum chamber of the bearing shell, then it passes through the orifices into the clearance space between the shaft and the bearing at pressure P_d , and from there flows towards

the open end where it is exhausted to atmosphere at pressure P_a . As the air flows from the supply manifold to the open end of the bearing, is subjected to two aerodynamic resistances in series as shown schematically in Fig.2., that of orifice itself and that of the gap between shaft and shell.

When the shaft moves towards one of the orifices due to an applied load, the reduction of aerodynamic resistance of the gap increases the downstream pressure of the orifice, considering the case of two orifices on opposite diameters of a journal bearing, movement of the shaft toward one wall of the sleeve gives rise to a pressure differential across the shaft, the high pressure at the minimum clearance tending to centralize the shaft.

In the case of a full journal bearing, the total effect of the orifices is to centralize the shaft with the bearing, in which case all the restoring forces are balanced and there is no net force on the shaft when the shaft is central.

2.3. PRESSURE DROP IN A BEARING AND ITS EFFECT.

If each hole is considered to be nozzle in which the pressure is fed, then there lies a relationship between supply pressure P_s , which is given by Shires (9) as -

$$P_s/P_d = \left\{ 1 + \frac{\gamma-1}{2} \cdot M^2 \right\}^{\gamma/\gamma-1}$$

Since in ideal condition Mach No., $M = 1^*$ then,

$$P_s/P_d = \left\{ \frac{\gamma+1}{2} \right\}^{\gamma/\gamma-1} \quad 1$$

Again if the mass flow through each orifice is assumed to be independent of the clearance h , then within the laminar flow region the pressure loss along the gap is expressed by Shires (8) as -

$$P_d^2 - P_a^2 = \frac{2 \times 10^{12} \mu R T \bar{M}_0 L n}{\pi D h^3 g} \quad 2$$

and the total mass $\bar{M}$ flowing through these orifices is given by Gross (10) and Constantinescu(11) for laminar flow as -

$$\bar{M} = C_d A P_s n \left\{ \left(\frac{2\gamma}{\gamma-1} \right) R T \left[\left(\frac{P_d}{P_s} \right)^{\frac{2}{\gamma}} - \left(\frac{P_d}{P_s} \right)^{\frac{\gamma+1}{\gamma}} \right] \right\}^{\frac{1}{2}} \text{ lb/sec.} \quad 3$$

* Values and their References given in Nomenclature.

provided $P_d/P_s \leq P_c$, where the critical pressure ratio

$$P_c = \left[\frac{2}{\gamma+1} \right]^{\gamma/(\gamma-1)}$$

$$\text{If } P_d/P_s \leq P_c = 0.528,$$

the choking condition exists and the form of the pressure distribution is optimum.

The discharge coefficient C_d in equation 3 is dependent on both the critical pressure ratio and the design of the orifice. The coefficient of discharge is given by Dudgeon and Lowe (12) as -

$$C_d = 0.85 - 0.15 (P_d/P_s) - 0.1 (P_d/P_s)^2 \quad 4$$

It should be noted that equation 4 is applicable over the entire range of critical pressure ratio from 0 to 1.0. However, it has been found, that for choked orifices i.e. $P_d/P_s = 0.528$, C_d has a value very close to 0.8; without doubt this value is not constant over whole range of critical pressure ratio but, to avoid the complexity of dealing with a variable C_d , a constant value has been recommended by Shires (8).

Analysis of experimental data by Shires (9), reveals that the maximum working load which a bearing will support is dependent on the choking action, the supply pressure, its dimension, and the ratio of the actual clearance to the choking clearance, C_o . The value of the latter is derived from viscous flow theory, and for the laminar flow it is given by -

$$C_o = \frac{30.5 (\gamma+1/2)^{1/2} (3\gamma-1/\gamma-1)^{1/2} \mu g (\gamma g R T)^{1/2} \text{ L.A.n.}}{P_s [1 - (\gamma+1/2)^{2\gamma/(\gamma-1)} (P_d/P_s)^2] D} \quad \text{in}^3$$

Hence, by substituting the values of γ , g , R , T and μ .

$$C_o = \frac{2.19 \times 10^3 \times \text{L.A.n.}}{P_s [1 - 3.59 (P_d/P_s)^2] D} \quad \text{in}^3 \quad 5$$

Assuming the bearing to be designed with the optimum ratio, $C/C_o = 0.86$; the inlet hole area can be calculated from

$$A = \frac{718 \times C^3 \times D \times P_s}{l \times n} \left[1 - 3.59 (P_a/P_s)^2 \right] \text{ in}^2 \quad 6$$

2.4. AXIAL FLOW MODEL.

The pattern of flow in the gap of a pressurized bearing is complicated by the interaction of the orifices, the dispersion effect and the circumferential flow. However, if these factors are ignored, then the axial flow model holds good when length/diameter ratio is small i.e. when $L/D < 1$.

Shires (9) has applied a simple theoretical treatment which is based on the combination of a orifice and a slot. The mathematical model represents the continuously varying clearance space by discrete rectangular slots, each corresponding to one of the orifice in a row. In this model the extension of the equivalent slots represents the space between the row of orifices. The pressure between two orifices is assumed to be constant and equal to P_d , adjacent to the orifices in the corresponding equivalent slots.

Robinson and Sterry (13) have calculated a general expression for load coefficient C_L by summing the vertical components of the restoring forces for the equivalent slots. Also the total load capacity of the bearing has been calculated by extending this principle between rows of orifices, in terms of bearing dimensions, supply pressure, gas properties and eccentricity. The equation for an even number of orifice is

$$C_L = 2 \sin \pi/n \left\{ (1 - 2l/3l) [K_{g1} \cos \pi/n + K_{g2} \cos 3\pi/n + \dots + K_{gn/2} \cos (\pi - \pi/n)] \right\} \quad 7$$

where the load coefficient,

$$C_L = \frac{W}{LD(P_s - P_a)} \quad 8$$

In many practical cases the diameter of the shaft is fixed, and it is necessary to choose the values of L/D and l/L which will give the maximum load for a given pressure under these circumstances. The actual parameter is then the load coefficient based on diameters,

$$C_L' = \frac{W}{D^2 (P_s - P_a)} \quad 9$$

and bearing pressure factor,

$$K_g = \frac{P_d - P_a}{P_s - P_a} \quad 10$$

The influence of the axial pressure distribution on the journal bearing, and an equivalent slot based in Shires theory shown in Fig. 3.

Although the equations 8 and 9 apply to any value of eccentricity ratio ϵ actually it is found very difficult to determine the value of P_d (13). However, to overcome this problem if an idealized flow configuration as illustrated in Fig. 4 is considered, according to Dudgeon & Lowe, (12) the total flow may be expressed as -

$$\bar{M}_i = \frac{\pi D h^3 P_a^2}{6 \mu n R T (L - l)} \left(P_d / P_s \right)^2 \quad 11$$

The pressure immediately below the orifice P_d can then be determined by equating the equations 3 and 11. Also the load carrying capacity of the clearance space is found by determining the pressure at any point and integrating over the length of the slot. The net load determined in this manner is represented by -

$$W_i = \frac{P_a \pi D}{3 n} \left\{ \frac{L - l \left[2 \left(\frac{P_d}{P_a} \right)^2 - \left(\frac{P_d}{P_a} \right) - 1 \right] 3 l \left[\left(\frac{P_d}{P_a} \right)^2 - 1 \right]}{\left[\left(\frac{P_d}{P_a} \right) + 1 \right]} \right\} \quad 12$$

2.5. INFLUENCE OF CIRCUMFERENTIAL FLOW.

The axial flow model so far described gives a value of load coefficient which cannot be achieved in practice. In actual bearings, there is a peripheral leakage of air from the high pressure side to the low pressure side, which reduces the load capacity and stiffness below the one dimensional bearing values.

A theoretical treatment applicable to small eccentricities has been made by Pinkus and Sternlicht (14), but unfortunately the solution is only for a bearing with a single central row of orifices.

An empirical correction for the difference between the

load capacity of an actual bearing and the predicted values from the one dimensional model was derived by Shires (15) from the experimental results of Robinson & Sterry (13) and Shires & Pantall (16). The factor is -

$$\frac{C_L}{C_{L0}} = \frac{0.315 \left[\frac{\cosh(6.36 L/D) - 1}{\sinh(6.36 L/D)} + \tanh\left(6.36 \frac{L-2l}{D}\right) \right]}{\left(\frac{L-l}{D}\right)} \quad 13$$

This is based on bearings with eight orifices per ring and a further correction has to be made for different numbers of orifices as shown in the above reference. It is recommended that the above equation should not be used for values of $L/D < 0.5$, or for values $l/L < 0.25$, because of the uncertainty regarding dispersion losses when the orifices are positioned close to the end of the bearing.

2.6. DISPERSION EFFECT.

In the long bearing with inlet orifices the air flow spreads outwards from the inlet holes before flowing along and around the bearing to the exit. In the short bearing with inlet orifices the circumferential component of flow is much less but the spreading out from the orifices now occurs over a larger part of the bearing surface and consequently has a more significant effect on performance. It can be seen that, in the extreme case where the distance from the inlet orifices to the end of the bearing is small compared to the circumferential distance between them, the flow will be locally concentrated in line with the orifices and the circumferential distribution will be peaky and ineffective.

The quantitative effect of dispersion has been investigated by Dudgeon and Lowe (12), using an electrolytic tank analogue of the pressure field in the clearance space and the results have been used with considerable success in the prediction of journal bearing performance. Briefly, the effects of dispersion are to reduce the flow corresponding to a given differential pressure and to reduce the load capacity and stiffness of the bearing.

The flow dispersion co-efficients are functions of the length/width ratio and the source ratios and are independent of pressure ratio. The effects on load capacity and stiffness depend upon the ratio of length to width $\frac{L}{2\pi D}$ and source ratio $\frac{nd}{2\pi D}$.

2.7. COMPRESSIBILITY EFFECT.

As long as the ratio of supply pressure to exhaust pressure, P_s/P_a , is less than about 2, the compressibility of the air may be neglected in the analysis of air flow in such a bearing. However, when the ratio of P_s/P_a is high, the density of air does not remain constant. Thus as the density increases, as the pressure rises, the ratio $\bar{M}/\rho$ will decrease, and the rate of pressure rise will increase.

The compressibility effect becomes particularly noticeable when the pressure variation through the bearing in the direction of motion is large compared with the ambient pressure. Moreover, the effect becomes more complex when, in addition to compressibility, the practical bearing geometric shape, side leakage effects of a finite bearing, and the hydrodynamic effect of a hydrostatic bearing (i.e. Hybrid effect) are all taken into account. Some solution of this problem has been recommended by Heinrich (17) and Gröss (18). However, from a series of experiments Powell (19, 41) observed, that in the case of externally pressurized high speed spindles with small radial clearance, the hybrid effect exists. Therefore, to calculate the load capacity of this type of bearing, at any finite speed, it has been suggested to use a mean compressibility dimensionless number, given by-

$$\Lambda_m = \frac{6 \mu \omega}{P_m} \left(\frac{r}{h} \right)^2 \quad 14$$

where the effective mean pressure,

$$P_m = \frac{P_s + P_a}{2} \quad \text{lb/in}^2$$

2.8. INFLUENCE OF REYNOLDS EQUATION ON SPEED.

The critical velocity at which the turbulence is initiated is identified by a dimensionless parameter, Reynolds number, which is given by -

$$Re = \frac{\rho U h}{\mu} \quad 15$$

considering the total mass flow through orifices, the Reynolds number is given by Robinson and Sterry (13) as -

$$Re = \frac{12 \bar{M}}{\pi D \mu} \quad 16$$

Furthermore, Taylor (44) analysed the problem for the liquid between two concentric cylinders and developed a criterion for the initiation of turbulence under these circumstances. Fortunately, this can also be applied to the journal bearing with sufficient accuracy (20).

Taylor's criterion as modified by Wilcock (45) is -

$$\frac{\pi N_c D h}{60 \nu} = 41.1 \sqrt{\frac{D}{2h}} \quad 17$$

which gives speed N_c and by further substitution in equation 14, the critical Reynolds number is obtained.

2.9. EFFECT OF ORIFICE CONFIGURATION.

Theoretically the effective orifice area A varies according to the type of orifice is used. There are several types used in air bearings among them the most important types are shown in Fig.5.

Fig.5 (a) is a plain orifice without any recess, the air passes through the fixed orifice area $(\pi d^2/4)$, and then radially outwards through the effective area $\pi d h$, stated by Geary (21). The effective area varies with the clearance h . Therefore, in the case of a plain orifice, or an annular orifice as it is generally known, it would regulate the air flow entering the bearing clearance as long as,

$$\pi d h < \frac{\pi d^2}{4} \text{ i.e. } h < \frac{d}{4}$$

In the case of Fig.5 (b), this is a plain orifice with a recess, so that the area of the variable orifice $P.h$, where P is the perimeter of recess, is greater than the effective area of a fixed orifice $(\pi d^2/4)$. In this case the restriction of the gas flow would be carried out by the fixed orifice area $\pi d^2/4$ in the same way as externally pressurized liquid bearing. This is commonly known as simple orifice.

However, the above theory is not strictly true, as with the load applied, will be a variation of h , i.e. the area $P.h$ will fluctuate on either side of the orifice, so the air flow will be restricted partly by the annular orifice and partly by the simple orifice principles.

Of the two basic types of orifice, the first is easy to manufacture, and yields a more stable bearing as this type of

orifice supplies the least volume of air to the bearing. On the other hand, second type off orifice gives a higher stiffness, but it tends to be unstable. Licht, Fuller, & Sternlicht, (22) have attempted to analyse this phenomenon using a number of simplifying assumptions in order to indicate the primary causes of this undesirable action and to establish criteria for determining stability.

Their results do show that the air storage capacity of the bearing is of paramount importance and should be held to a minimum. This means the recess depth should be small in order to achieve stability and should have a depth roughly comparable to the gap thickness over the annulus.

As the depth of recess shown in Fig.6, is critical in relation to the orifice diameter, Holster (23) has suggested some limitation factor which is expressed as -

$$3.3 \pi d^2/4 \leq \pi d h_r (1 + h/h_r) \Rightarrow h_r \geq d/1.2 \quad 18$$

Also to avoid the instability and to achieve stiffness Lund (24) has included a non-dimensional compensation factor which is given as

$$\delta = \frac{a^2}{d_r h} \quad 19$$

2.10. EFFECT OF NUMBER OF ORIFICES.

The main experimental results obtained are those of Robinson and Sterry (13) on bearings with orifices terminating in circumferential slots, and those of Allen and Stokes (25) on bearings with orifice holes with flared exit. They found that the load coefficient C_L increases by increasing the number of orifices, and also that the local pressure depression between orifices became less significant. However, Robinson and Sterry checked that by increasing the number of orifices, 8 to 16, the load coefficient only increased by 20 percent whereas by decreasing this from 8 to 4 it reduced the load coefficient by 40 percent as shown in Fig.7.

However, the greater the number of orifices the smaller their diameter has to be and the greater the manufacturing difficulties encountered. According to Shires (9) this criterion could be optimized by using equation 6.

2.11. EFFECT OF ORIFICE POSITION.

The load carried by a bearing is influenced by the position of the rows of feed holes. In the work of Allen and Stokes (25), it was suggested that at constant pressure the load carrying capacity is maximum when the ratio of ℓ/L is 0.125 and the angular stiffness increases as this ratio decreases, as shown in Fig. 8 & 9 respectively. At these conditions, however, the gas flow is high, and the corresponding pump power is also high. The optimum position of the orifices is found when -

$$0.2 \leq \ell/L \leq 0.25$$

Also Whor and Lund (27) experimented with the effect of L/D ratio on the load carrying capacity, stiffness and flow, where non-dimensional factors K' , Λ_s and M' are used which confirms the above. In most cases, the bearing with two rows of orifices is optimized at a cost of greater mass flow.

2.12. EFFECT OF LENGTH TO DIAMETER RATIO.

As the length to diameter ratio of a bearing is increased its load coefficient C_L falls, due to the greater influence of the circumferential flow and the dispersion effect, the ratio ℓ/L also has a marked influence.

Robinson and Sterry (13) showed how the theoretical load coefficient, C_L , for a given pressure factor K_g , is affected by variation of the bearing length to diameter ratio. The maximum load coefficient occurs when the values of L/D is between 1 and 3, and the axial position of orifice ℓ/L ratio is between $\frac{1}{4}$ and $\frac{1}{3}$, as shown in Fig.10. For smaller values of L/D there is a tendency for the theoretical maximum to occur at lower values of ℓ/L .

2.13. PRINCIPLE OF THRUST BEARING.

In the Fig.11, two types of thrust bearing together with their pressure distributions have been shown. The Fig.11a, is central orifice thrust bearing with a concentric pocket. This type has the advantage of high load capacity, simplicity of manufacture and low air consumption. In the Fig.11b, a circular thrust bearing with a ring of orifices is shown. Although this has lower load capacity than the central orifice type, it offers some resistance to tilting forces. Moreover a thrust bearing with a ring of orifices is generally used, where a continuous central shaft has to be accommodated for the

driving arrangement.

The externally pressurized thrust bearing operates on the same principle as the journal bearing. The clearance between the pad and the bearing is reduced by the application of a load perpendicular to the surface of the bearing, thus reducing the aerodynamic resistance of the gap relative to that of the orifice, under equilibrium conditions the clearance adjusts itself to produce a total pressure force equal to the applied load.

Experimental results of Powell (26) (et.al) shown that the total load which the bearing will support is proportional to K_g , and is obtained by integrating the pressure over the total area. The solution for the thrust bearing shown in Fig.11a, is -

$$W = K_g (P_s - P_a) \frac{\pi (R_o^2 - R_i^2)}{2 \log_e (R_o/R_i)} \quad \text{lb.} \quad 20$$

for incompressible flow, the maximum occurs when $K_g = 0.69$, under which conditions,

$$W = 0.69 (P_s - P_a) \frac{\pi (R_o^2 - R_i^2)}{2 \log_e (R_o/R_i)} \quad \text{lb.} \quad 21$$

and the maximum stiffness is given by -

$$\frac{dW}{dh} = -0.69 (P_s - P_a) \frac{\pi (R_o^2 - R_i^2)}{2 h \log_e (R_o/R_i)} \quad \text{lb/in} \quad 22$$

the mass flow is given by Fuller (20),

$$Q = \frac{P_o \pi h^3 \times 60}{12 \mu \log_e (R_o/R_i) \times 1728} \quad \text{ft}^3/\text{min.} \quad 23$$

In the case of a thrust bearing having a ring of orifices at radius R_c , it is assumed that the outward mass flow is equal to the inward mass flow i.e. $R_c^2 = R_o R_i$, the total load capacity is given by,

$$W = 0.69 (P_s - P_a) \frac{\pi (R_o - R_i)^2}{\log_e (R_o/R_i)} \quad \text{lb} \quad 24$$

and maximum stiffness is given by -

$$\frac{dW}{dh} = -0.69 (P_s - P_a) \frac{\pi (R_o - R_i)^2}{h \log_e (R_o/R_i)} \quad 16/in. \quad 25$$

and the mass flow is given by Fuller (20),

$$Q = \frac{P_o \pi h^3 \times 60}{6 \mu \log_e (R_o/R_i) \times 1728} \quad ft^3/min. \quad 26$$

In both type of thrust bearing the load coefficient C_L is given by -

$$C_L = \frac{W}{\pi R_o^2 (P_s - P_a)} \quad 27$$

Considering the radius ratio R_o/R_i , the load capacity decreases with increase of R_o/R_i in the case of the central orifice; but for the ring orifice type, the load capacity increases. Both of them reach the optimum range when the radius ratio is 3.55, shown in Fig.12; and at the ratio of 3, both types of bearing require the same supply pressure to support a given load over the same total area. Again increase in R_o/R_i ratio decreases optimum orifice diameter.

2.14. DYNAMIC PROPERTIES.

Due to very small clearance, the high speed externally pressurized bearing in this application has the hydrodynamic effect; the analysis of dynamic properties and design must include the effects of rotation.

The hydrostatic bearing exhibits two types of instability; one of a pneumatic, self excited nature called pneumatic hammer, and the other is induced by the rotation of the journal which is known as hybrid bearing instability or whirl instability. Moreover, mechanical critical speed and the effect of unbalance on a high speed shaft must be considered at design stage.

2.14.1. PNEUMATIC HAMMER.

This instability occurs in bearings in which the air is fed through an orifice into a feeder hole or pocket of relatively large volume. Licht (22) et al. suggested that

in order that the pocket pressure can adjust quickly due to the variation in clearance, the pocket must be of small volume. The changes in pocket pressure could lag to such an extent that, following a random disturbance from equilibrium, an oscillation of increasing amplitude is generated.

Vohr (27) illustrated the relationship between the restrictor coefficient and the depth of feeder hole. Also he has recommended that to avoid pneumatic hammer, the feeder hole volume for the journal bearing should be,

$$V_c \leq \frac{0.25 \pi L D h}{n} \quad \text{in}^3 \quad 28$$

and for the thrust bearing,

$$V_c \leq \frac{0.25 \pi h (R_o^2 - R_i^2)}{n} \quad \text{in}^3 \quad 29$$

Again the work carried out by Fischer (28) et al. reveals that pneumatic instability could also occur due to high hydrostatic pressures, large diametral clearances and resonance in the pressure supply line system.

2.14.2. WHIRL INSTABILITY.

Rotating shafts are frequently supported in externally pressurized journal bearings. The contribution of the hydrodynamic pressures thus generated must be analysed where bearing stability is important. Lund (24) (29) has considered both static and dynamic performance of air lubricated hybrid bearings. This analysis was based on two major assumptions.

- a. The journal eccentricity is small compared to the radial clearance.
- b. The pressurized air is supplied to the bearing through a sufficiently large number of orifices that they may be represented by a line source in the centre plane of the bearing.

With these simplifications, the basic equation for the air film, Reynold's equation, becomes linear. The resultant forces on the bearing have been obtained for the steady state operating position, and then under the influence of small disturbance with a prescribed frequency. Knowing the forces, the equation of motion for the rotor are established and the

stability of the motion for the given frequency may be investigated, scanning the frequency range, the onset of instability can be determined.

Numerical results are given by Lund (29), Reiger and Wilcock (30), and Sternlicht and Rieger (31) in the form of several graphs, covering a wide range of design parameters for both the speed of threshold of instability and for the corresponding frequency. The following non-dimensional parameters have been used to predict the behaviour of instability. The feeding parameter or restrictor coefficient Λ_s is given as -

$$\Lambda_s = \frac{6\mu n a^2 (RT)^{1/2}}{\rho_s h^3 (1 + \delta^2)^{1/2}} \quad 30$$

the compressibility number or bearing number Λ as -

$$\Lambda = \frac{6\mu \omega}{\rho_a} \left(\frac{\gamma}{h} \right)^2 \quad 31$$

the load parameter or stiffness number K' as -

$$K' = \frac{W}{DL (\rho_s - \rho_a) \epsilon} \left[\frac{1 + \delta^2}{1 + \frac{2}{3}\delta^2} \right] \quad 32$$

The relation between these parameters is shown in their report. Also Lewis (32) et al. used these parameters to predict the stiffness and the behaviour of the system. Since the bearing stiffness can be given by

$$K = \frac{W}{\epsilon h} \quad \text{lb/in.} \quad 33$$

then equation 32 can be written as

$$K' = \frac{Kh}{DL(\rho_s - \rho_a)} \left[\frac{1 + \delta^2}{1 + \frac{2}{3}\delta^2} \right] \quad 34$$

when solved for stiffness gives

$$K = \frac{DL(P_s - P_a)}{h} \left[\frac{1 + 2/3 \delta^2}{1 + \delta^2} \right] K' \quad \text{lb/in.} \quad 35$$

Experiments have shown that in most two bearing systems, the unbalance is predominantly static, and the whirl instability usually occurs in the cylindrical mode or conical mode. To determine the cylindrical whirl speed for rigid shaft and low damping, Lund (36), and Sternlicht and Rieger (31) introduced another dimensionless instability parameter λ , and given as

$$\lambda = \frac{m_r h \omega_{c1}^2}{(P_s - P_a) L D} \quad 36$$

when solved for critical speed

$$\omega_{c1} = \frac{\lambda (P_s - P_a) L D}{m_r h} \quad \text{rad/sec.} \quad 37$$

Again Powell (37) and Whitley (38), experimented the conical whirl in a two bearing system. Analysis of their experimental results suggests that conical whirl occurs when

$$\omega_{c2} = \bar{n} \sqrt{\frac{2 K J^2}{I}} \quad \text{rad/sec.} \quad 38$$

2.14.3. MECHANICAL CRITICAL SPEEDS.

The mechanical critical speed must be considered in high speed operation. This critical speed is largely established by the elastic and mass properties of the shaft. This critical speed is also influenced by the flexibility of the bearings and their supports. Fischer (28) et al. showed that in order to run a shaft at the high speeds, it was necessary to calculate the shaft mechanical critical speeds quite accurately. One of the first observations made concerned the low restraining effect of the air bearings on the rotating shaft system. The assumption that the bearings acted as simple supports proved to be completely inadequate

and instead it was found that the bearings applied little, if any, restraint to the lightly loaded shaft.

2.14.4. UNBALANCE.

The amount of unbalance which can be tolerated in a high speed shaft will depend on the inversion speed of the system or the damping which is present in the bearings.

The basic idea in balancing is to make the gravity axis coincide with the geometric axis. The inversion speed which is usually lower than the operating speed and the unbalance in the rotor system must be small enough to permit running through this speed without causing damage to the bearing. Since at the inversion speed the shaft which was rotating about its geometric axis starts to rotate about its gravity axis, the distance between the two axes must be limited to somewhat less than the radial bearing clearance. If the shaft is operating at 50 percent eccentricity then the allowable distance between axes is reduced by 50 percent. Since housing distortion, manufacturing tolerances, and accidental cocking may lead to eccentricities greater than 50 percent a safe value for the allowable shift may be about 10 percent of the eccentricity. This value when multiplied by the shaft weight gives the maximum advisable in a high speed shaft supported by air bearings.

3.1. DESIGN.

Many of the advantages and disadvantages of air bearings have already been discussed. However, to achieve these advantages, a design study has been carried out in collaboration with a machine tool manufacturer. The main object of this research work is to design an externally pressurized air spindle with particular reference to machine tools ie. internal grinders, which is capable of running from a very low speed to a high speed of approximately 80,000 rev/min. This particular project was chosen because it enables a number of features of the air bearing design to be highlighted. It is also representative of a wide range of possible industrial applications where the use of a common air supply for lubrication can result in a simple and economical design with a number of useful advantages.

In a design problem of this type the prime concern of the designer is to consider the performance, such as the ability to hold close tolerances or the limitation of the run-out of unbalanced wheels, which may determine the minimum requirements of bearing stiffness and load carrying capacity. However, as far as air bearings are concerned these two design factors are more usually determined by the necessity for the bearings to be stable at the designed speed. Moreover, in a design exercise like this there are a number of boundaries which limit the range of solution. In air bearing design, apart from achieving an optimum solution, there are three other major boundaries, which are considered more fully in a later section.

1. Air Supply.
2. Manufacturing Capability.
3. Overall Dimensions.

In this chapter all the final calculations were done by two methods based on the work carried out in U.K. and U.S.A. respectively. Although, one set of calculations are presented, in the actual case the calculations were compared with others work to achieve the best design. Also the drawings based on these design calculations are shown in Fig. 13 - 23.

3.2. JOURNAL BEARING.

No design data is available for calculation, except the required stiffness of 180,000 lb/in. in the radial direction.* So, the main criterion was to achieve a stiffness greater

* Refers to existing grinding head.

than 200,000 lb/in. to avoid instability at high speed. According to the manufacturer, this stiffness is much higher than one expects from a high speed grinding machine. Nevertheless, a load on the bearing is estimated as described below. In grinding, the radial component of the cutting force exceeds the tangential component and usually lies between 1.6 times to 5 times the tangential component. Due to the presence of variable factors the resultant force on the spindle is taken as an approximation as equal to the radial cutting force, i.e. twice the tangential force obtained from consideration of the motor horse-power, as suggested by Stansfield, (33),

A 4H.P. motor at 2900 rev/min, is used in the present machine, therefore the motor torque,

$$T_m = \frac{H.P \times 33,000}{2 \pi N_m} \quad \text{lb.ft.}$$

$$T_m = \frac{4 \times 33,000}{2 \times 3.14 \times 2900} \quad \text{lb.ft.}$$

$$T_m = 7.24 \quad \text{lb.ft.}$$

Then the wheel torque at 45000 rev/min.

$$T_w = \frac{7.24 \times 2900 \times 12}{45000} \quad \text{lb.in.}$$

$$T_w = 5.6 \quad \text{lb.in.}$$

Taking 6500ft/min. as the surface velocity V_s of a grinding wheel recommended by the manufacturer, the minimum wheel size which could be used at 45000 rev/min. is given by the following equation,

$$V_s = \frac{\pi N_w D_w}{12}$$

$$\therefore D_w = \frac{6500 \times 12}{3.14 \times 45000}$$

$$D_w = 0.552 \quad \text{in.}$$

Taking the diameter of wheel approximately $9/16"$, then the tangential force is given by

$$F_t = T_w / R_w$$

$$F_t = \frac{5.6}{9/32}$$

$$\therefore F_t = 19.91 \quad \text{lb.f.}$$

Hence the radial force or resultant force,

$$F_r = 2 F_t$$

$$F_r = 19.91 \times 2 \quad \text{lb.f.}$$

$$F_r \approx 40 \quad \text{lb.f.}$$

Assuming that this force acts on a weightless shaft at a distance from the bearing centre as shown in Fig.24. Taking moment at R_B .

$$\sum R_A = 11 \times 40$$

$$\therefore R_A \approx 62.5 \quad \text{lb.} \quad \uparrow$$

$$\text{and} \quad R_B \approx 22.5 \quad \text{lb.} \quad \downarrow$$

Although nominal 100 lb/in^2 (gauge) compressed air supply is normally available from a workshop line, but for safety reasons 80 lb/in^2 is assumed to be used.

It has been shown by Shires, (9) (15), that the theoretical pressure fed journal bearing with recessed orifices, has a maximum load carrying capacity, when $K_g = 0.4$, $\epsilon = 0.5$, $P_s = 94.7 \text{ lb/in}^2$ (Abs), $P_a = 14.7 \text{ lb/in}^2$, $\gamma = C_p / C_v = 1.41$, then pressure drop P_d is given by -

$$K_g = \frac{P_d - P_a}{P_s - P_a}$$

$$\text{when } P_d / P_s \leq 0.528$$

$$\therefore 0.4 = \frac{P_d - 14.7}{94.7 - 14.7}$$

$$\therefore P_d = 46.7 \text{ lb/in}^2$$

Which satisfies the choking criterion given in Ref. (15), thus preventing instability due to over-expansion in the orifices.

Considering a bearing of orifice position $L/D = 0.25$

Optimum length to diameter ratio $L/D = 2$

Supply pressure (Absolute) $P_s = 94.7 \text{ lb/in}^2$

Number of orifice in each row $n = 8$

Load on bearing $W = 62.5 \text{ lb}$

Then the load coefficient based on diameters is obtained from equation 9,

$$C_L' = \frac{W}{(P_s - P_a) D^2} = C_L (L/D)$$

So from Fig. 25. $C_L' = 0.5$

$$\therefore D^2 = \frac{W}{(P_s - P_a) C_L'} \quad \text{in}$$

$$D^2 = \frac{62.5}{(94.7 - 14.7) 0.5} \quad \text{in}$$

$$D \approx 1.25 \quad \text{in.}$$

The values of orifice position, length to diameter ratio and number of orifices are taken as suggested in Ref. (13).

Hence,

$$L = 2.D \text{ inch.}$$

$$L = 2.5 \text{ inch.}$$

and $l = 0.625$ inch.

Again limiting minimum diametral clearance as 1×10^{-3} inch. for laminar flow, the inlet hole area is calculated from equation 6.

$$A = \frac{718 \times C^3 \times D \times P_s}{l \times n} \left[1 - 3.59 \left(P_a / P_s \right)^2 \right] \text{ in}^2$$

Substituting the value of, $D = 1.25 \text{ in.}$, $P_s = 94.7 \text{ lb/in}^2 (\text{Abs.})$,

$P_a = 14.7 \text{ lb/in}^2$, $l = 0.625 \text{ in.}$, $n = 16$ (8 per row.),

$C = 1 \times 10^{-3} \text{ in.}$, and $A = \pi d^2 / 4 \text{ in}^2$

$$\therefore d^2 = \frac{718 \times 1 \times 10^{-3} \times 1.25 \times 94.7 \times 4}{3.14 \times 0.625 \times 16} \left[1 - 3.59 \left(\frac{14.7}{94.7} \right)^2 \right] \text{ in}^2$$

$$d = \sqrt{\frac{77,700 \times 10^{-9}}{7.85}} \text{ in.}$$

from which -

$$d \approx 0.0035 \text{ inch.}$$

$d = 0.004$ inch, taken due to manufacturing difficulties, orifice shown in Fig. 20.

3.3. THRUST BEARING.

The design of the thrust bearing is based on the principle of the orifice and slot combination which has been used by Shires (15). In the calculation, bearing pressure factor $K_g = 0.69$ is taken, which gives the maximum load carrying capacity as stated by Powell (26) et al.

As a driving arrangement has to be provided for the grinding head, the thrust bearing with a ring of orifices is best suited for the purpose, so that the shaft can be extended through the centre hole of the bearing. Also the internal radius R_i of the bearing is taken as the radius of the shaft, and the external radius R_o is assumed as $2 R_i$. These two

parameters were governed by the diameter of the shaft and the condition $R_c^2 = R_o R_i$ respectively. So assuming a thrust bearing of,

$$\text{Internal radius } R_i = 0.625 \text{ inch.}$$

$$\text{External radius } R_o = 1.25 \text{ inch.}$$

When pressure factor $K_g = 0.69$, the total load carrying capacity is obtained from equation 24.

$$W = 0.69(P_s - P_a) \frac{\pi (R_o - R_i)^2}{\log_e R_o/R_i}$$

$$W = 0.69(94.7 - 14.7) \frac{3.14(1.25 - 0.625)^2}{\log_e 1.25/0.625}$$

$$W = 0.69 \times 80 \times 3.14(0.625^2/0.6931)$$

$$\therefore W = 97.5 \text{ lb.}$$

Again from the slot and orifice theory given by Shires (15), the clearance h is obtained as,

$$G = \frac{P_a/P_s}{(1 - P_a/P_s)^{1/2} (1 + P_a/P_s)} \times \frac{24\mu(2gRT)^{1/2}}{P_a g} \times \frac{C_d n d^2 \log_e(R_o/R_i)}{32 h^3}$$

$$G = F_p \times F_g \times F_d$$

Designing for maximum stiffness, the value of the slot factor $G = 1.25$ and $C_d = 0.8$ for choked hole has been recommended by Shires (15), as the value of C_d does not remain constant for unchoked feed hole. Also $P_a = 14.7 \text{ lb/in}^2$ at 15°C , the value of $F_g = 5.66 \times 10^{-6} \text{ ft.}$ as shown in Fig.26.

Assuming, Diameter of orifice $d = 0.015$ (See Fig. 21.)

Number of holes $n = 8$

$$1.25 = \frac{14.7/94.7}{\sqrt{1 - 14.7/94.7 \times (1 + 14.7/94.7)}} \times 5.66 \times 10^{-6} \times 12 \times$$

$$\times \frac{0.8 \times 8 \times 0.015^2 \log_e 2}{32 h^3}$$

$$h^3 = \frac{0.1553}{\sqrt{0.8447 \times (1.1553)}} \times 67.92 \times 10^{-6} \times \frac{31.16 \times 10^{-6}}{1.25} \text{ in.}^3$$

$$h^3 = \frac{0.147 \times 67.92 \times 10^{-12}}{1.25} \text{ in.}^3$$

$$h \approx 6.4 \times 10^{-4} \text{ in.}$$

$$h = 7.0 \times 10^{-4} \text{ in., taken due to measuring difficulties.}$$

To satisfy the assumption of the inward mass flow to be equal to the outward mass flow i.e. $R_c^2 = R_o R_i$, the radius of the ring for the orifices is given by,

$$R_c = \sqrt{R_o R_i}$$

$$R_c = \sqrt{1.25 \times 0.625}$$

$$R_c = 0.885 \text{ in.}$$

3.4. SUMMARY.

Before proceeding to the second set of calculations, the results which have been obtained so far summarized below.

3.4.1. JOURNAL BEARING. (See Fig. 15, 16.)

a.	Length of bearing	L	$= 2.5$	in.
b.	Length of orifice position from end of bearing	l	$= 0.625$	in.
c.	Diameter of bearing	D	$= 1.25$	in.
d.	Diametral clearance between shaft and bearing	C	$= 1 \times 10^{-3}$	in.
e.	Diameter of orifice	d	$= 4.0 \times 10^{-3}$	in.
f.	Total number of orifices	n	$= 16$	
g.	Load carrying capacity	W	$= 62.5$	lb.
h.	Supply pressure (absolute)	P_s	$= 94.7$	lb/in. ²

3.4.2. THRUST BEARING. (See Fig. 16, 19.)

a.	External diameter of thrust bearing	D_o	$= 2R_o = 2.5$	in.
b.	Internal diameter of thrust bearing	D_i	$= 2R_i = 1.25$	in.
c.	Clearance between pad and bearing for each side	h	$= 7.0 \times 10^{-4}$	in.
d.	Diameter of orifices	d	$= 15 \times 10^{-3}$	in.
e.	Number of orifices on each bearing	n	$= 8$	
f.	Diameter of orifice position	D_c	$= 2R_c = 1.77$	in.
g.	Load carrying capacity	W	$= 97.5$	lb.
h.	Supply pressure (absolute)	P_s	$= 94.7$	lb/in. ²

3.5. STIFFNESS AND FLOW.

The following calculations are based on the results obtained previously and also the effect of non-dimensional parameters on the mass flow, stiffness, instabilities etc. has been shown.

Taking the same values as previously calculated and to achieve the highest stiffness and stability for a journal bearing, the non-dimensional orifice compensation factor δ is obtained from equation 19,

$$\delta = \frac{a^2}{d_r h}$$

assuming diameter of recess pocket $d_r = 0.125$ in.,

$$\delta = \frac{0.002^2}{0.125 \times 0.0005}$$

$$\delta = 0.064$$

$$\text{also } \delta^2 = 0.0041$$

Again δ has been used in the calculations of Λ_s and K' to predict the static properties of a 360° journal bearing. The feeding parameter or restrictor co-efficient Λ_s is obtained as given in equation 30,

$$\Lambda_s = \frac{6\mu n a^2 (RT)^{1/2}}{P_s h^3 (1 + \delta^2)^{1/2}}$$

$$\Lambda_s = \frac{6 \times 2.62 \times 10^{-9} \times 16 \times 0.002^2 (2.47 \times 10^5 \times 530)^{1/2}}{94.7 \times 0.0005^3 (1 + 0.0041)^{1/2}}$$

$$\Lambda_s = 0.975$$

Also the flow parameter is given by Rieger and Wilcock (30) as

$$M' = \frac{6\mu RT \bar{M}}{\pi P_s^2 h^3}$$

Hence the total mass flow of a bearing is

$$\bar{M} = \frac{M' \pi P_s^2 h^3}{6 \mu R T}$$

$$\bar{M} = \frac{M' \times 3.14 \times 94.7^2 \times 0.0005^3 \times 32.2 \times 12}{6 \times 2.62 \times 10^{-9} \times 2.47 \times 10^5 \times 530}$$

$$\bar{M} = M' \times 6.6 \times 10^{-4} \text{ lb/sec.}$$

Taking into consideration the compressibility effect of an externally pressurised bearing, the bearing number or compressibility number Λ as suggested by Lund (29), Rieger and Wilcock (30), Sternlicht and Rieger (31), Ausman and Wildman (34), and Pantall (35), then from equation 31,

$$\Lambda = \frac{6 \mu U r}{P_a h^2} \quad \text{when } U = \frac{\pi N D}{60} \text{ in/sec.}$$

$$\therefore \Lambda = \frac{6 \mu \pi N D r}{P_a h^2 60}$$

$$\Lambda = \frac{6 \times 2.62 \times 10^{-9} \times 3.14 \times 80,000 \times 1.25 \times 0.625}{14.7 \times 0.0005^2 \times 60}$$

$$\Lambda = 14.$$

The relation between the flow parameter M' and the feeding parameter Λ_s , along with the pressure ratio P_s/P_a is shown in Fig.27.

Due to extreme difficulty in calculating the stiffness of a journal bearing, Lewis (32) et al. suggested the use of a dimensionless load parameter K' , which is obtained from equation 35,

$$K = \frac{LD(P_s - P_a)}{h} \left[\frac{1 + \delta^2}{1 + \frac{2}{3}\delta^2} \right] K' \quad \text{lb/in.}$$

Substituting the values of all the terms except K' , which is obtained from Fig.28. in the equation,

$$K = \frac{2.5 \times 1.25 (94.7 - 14.7)}{0.0005} \left[\frac{1 + 0.0041}{1 + 0.0027} \right] 0.5 \text{ lb/in.}$$

when solved,

$$K = 251,000 \text{ lb/in.}$$

Also the value of M' for $P_s/P_a \approx 7$ is obtained from Fig.27. and substituted in

$$\bar{M} = M' \times 6.6 \times 10^{-4} \quad \text{lb/sec.}$$

Hence, the total mass flow for a journal bearing is,

$$\bar{M} = 0.394 \times 6.6 \times 10^{-4} \quad \text{lb/sec.}$$

$$\bar{M} = 2.60 \times 10^{-4} \quad \text{lb/sec.}$$

Therefore, when density of air is 0.08 lb/ft^3 , the total quantity for two journal bearings is

$$Q = \frac{2.60 \times 10^{-4} \times 60 \times 2}{0.08} \quad \text{ft}^3/\text{min.}$$

$$Q = 0.39 \quad \text{ft}^3/\text{min.}$$

The calculation for the thrust bearing is based on Shires theory discussed earlier. Using equation 25, the stiffness is -

$$\frac{dW}{dh} = K_A = \frac{0.69 (94.7 - 14.7) \times 3.14 (1.25 - 0.625)^2}{0.0007 \times \log_e \frac{1.25}{0.625}}$$

$$K_A = - \frac{0.69 \times 80 \times 3.14 \times 0.3905}{0.0007 \times 0.6931} \text{ lb/in.}$$

Minus sign only indicates -ve slope, then

$$K_A = 150,000 \text{ lb/in.}$$

The total quantity of flow for a thrust bearing is obtained from equation 26,

$$Q = \frac{P_o \pi h^3 \times 60}{6\mu \log_e \frac{R_o}{R_i} \times 1728} \text{ ft}^3/\text{min.}$$

$$\text{Where } P_o = (P_s - P_a) \text{ lb/in.}^2$$

Substituting the values,

$$Q = \frac{80 \times 3.14 \times 0.0007^3 \times 60}{6 \times 2.62 \times 10^{-9} \times \log_e \frac{1.25}{0.625} \times 1728} \text{ ft}^3/\text{min.}$$

$$Q = 0.2744 \text{ ft}^3/\text{min.}$$

Then for two thrust bearings,

$$Q = 0.2744 \times 2 \text{ ft}^3/\text{min.}$$

$$Q = 0.5490 \text{ ft}^3/\text{min.}$$

Therefore, the total quantity of flow for journal and thrust bearings is,

$$Q_t = (0.390 + 0.5490) \text{ ft}^3/\text{min.}$$

$$Q_L = 0.939$$

$$\text{ft}^3/\text{min.}$$

3.6. STABILITY.

Rotor - bearing instability is one of the severest problems in the application of the air lubricated bearing. To achieve a successful air bearing, the four main factors which has been discussed in the previous chapter are calculated here.

3.6.1. PNEUMATIC HAMMER.

As pneumatic hammer is caused by the large volume of the recess pocket, it is important to restrict this volume to a minimum as suggested by Vohr. (27), which is given in equation 28, for a journal bearing.

$$V_c \leq \frac{0.25 \pi L D h}{n} \quad \text{in}^3$$

$$V_c \leq \frac{0.25 \times 3.14 \times 2.5 \times 1.25 \times 0.0005}{16} \quad \text{in}^3$$

$$V_c \leq 0.768 \times 10^{-4} \quad \text{in}^3$$

Knowing the size of the recess hole $d_r = 0.125$ inch, then the recess depth h_r is given as,

$$h_r = \frac{4 V_c}{\pi d_r^2} \quad \text{in}$$

$$h_r = \frac{4 \times 0.768 \times 10^{-4}}{3.14 \times 0.125^2} \quad \text{in}$$

$$h_r = 6.27 \times 10^{-3} \quad \text{in.}$$

Hence to avoid pneumatic hammer and instability, the recess depth should not be greater than 0.006 in.

In the case of the thrust bearing, the calculated hole size is based on the principle of the annular orifice, so

the problem of pneumatic instabilities does not arise.

3.6.2. WHIRL INSTABILITIES.

An examination of the existing grinding head design revealed that the length of the rotor for the air bearing must not be greater than the existing one due to the limitations of space; also the weight of the rotor should be as low as possible to avoid the whirling speed coinciding with the running speed. Because of dynamic stability considerations, the translatory critical speed of the rotor in its bearings has to be either below 0.7 N, or above 1.3 N, as suggested by Rieger (30).

An approximate weight of the rotating components were calculated for the given length and diameter i.e. $W_r \approx 4.5 \text{ lb}$; see Fig. 17, 23, so the mass of the rotor per bearing m_r in a two bearing system is,

$$m_r = \frac{4.5}{2 \times 32.2 \times 12} \frac{\text{lb. sec}^2}{\text{in}}$$

$$m_r = 0.00582 \frac{\text{lb. sec}^2}{\text{in}}$$

Then knowing the governing factors,

$$\text{Bearing number} = 14$$

$$\text{Feeding parameter} = 0.975$$

$$\text{Pressure ratio } P_s/P_a \approx 7$$

The dimensionless instability parameter Λ of a hybrid journal bearing is obtained from Fig. 29, as recommended by Lund (36), and Sternlicht and Rieger (31). Hence, by substituting the value of Λ , the critical speed or cylindrical whirl speed is obtained from equation 37,

$$\omega_{c1}^2 = \frac{3(P_s - P_a) L D}{m_r h}$$

$$\omega_{c1} = \sqrt{\frac{3 \times 80 \times 3.122}{5.82 \times 10^{-3} \times 5 \times 10^{-4}}} \quad \text{rad/sec.}$$

$$\omega_{c1} = 16,200 \quad \text{rad/sec.}$$

$$\therefore N_{c1} = \frac{60 \times 16,200}{2 \times 3.14} \quad \text{rev/min.}$$

$$N_{c1} = 153,600 \quad \text{rev/min.}$$

Also the onset speed of shaft conical whirl is given by Powell (37) and Whitley (38), and from equation 38,

$$\omega_{c2} = \bar{n} \sqrt{\frac{2 K J^2}{I}} \quad \text{rad/sec.}$$

When J = Half distance between centres of journal bearings.
See General Assembly Fig. 13, 14.

I = Moment of inertia of rotor.

$\bar{n} = 2$ for half speed whirl.

Substituting the values of J = 3.406 ins., K = 251,000 lb/in.,
and $I = \frac{\pi D^4}{64} \text{ ins}^4$.

$$N_{c2} = \frac{2 \times 60}{2 \times 3.14} \sqrt{\frac{2 \times 251,000 \times 3.406^2 \times 64}{3.14 \times 1.25^4}} \quad \text{rev/min.}$$

$$N_{c2} = 19.15 \frac{502,000 \times 7041}{3.14 \times 2.435} \quad \text{rev/min.}$$

$$N_{C2} = 130,000 \text{ rev/min.}$$

Hence, both the critical speeds i.e. cylindrical whirl and conical whirl of shaft are greater than,

$$\begin{aligned} 1.3 N &= 1.3 \times 80,000 \\ &= 104,000 \text{ rev/min.} \end{aligned}$$

3.6.3. MECHANICAL CRITICAL SPEEDS.

The importance of mechanical critical speed has already been discussed. So considering the shaft as a "free-free" beam, the natural frequency of the shaft is given by Timoshenko (39) as;

$$f_n = 3.56 \sqrt{\frac{EI}{m_r L^3}} \quad \text{C/sec.}$$

$$\therefore N_{C3} = 3.56 \times 60 \sqrt{\frac{30 \times 10^6 \times 3.14 \times 125^4}{2 \times 0.00582 \times 64 \times 6.812^3}}$$

$$= 213.5 \sqrt{\frac{230.5 \times 10^6}{234.5}}$$

$$= 213.5 \sqrt{0.983 \times 10^4}$$

$$N_{C3} = 211,000 \text{ rev/min.}$$

3.6.4. UNBALANCE.

Naturally, at high rotational speeds, unbalance becomes very important since unbalance of the shaft may generate forces that will exceed the safe load capacity of the bearing film. The degree of which unbalance must be eliminated may be ascertained by considering the action that occurs when the shaft system passes through the inversion speed. The calculation shown here is based on Fischer (28) et al.

Calculated weight of shaft	$=$	4.5	lb.
Radial clearance	$=$	5.0×10^{-4}	in.
Then maximum unbalance	$=$	$4.5 \times 5.0 \times 10^{-4}$	
	$=$	22.5×10^{-4}	lb.inches.
	$=$	0.036	oz.inches.

Assuming 50 percent eccentricity as taken in 3.2. i.e.

$$\epsilon = \frac{e}{h} = 0.5$$

Maximum unbalance	$=$	0.036×0.5	
	$=$	0.018	oz.inches.

However, as indicated in the above reference, it is safe to balance to about 1/10th of the maximum allowable unbalance.

Maximum unbalance	$=$	$0.018 \times 1/10$	
	$=$	0.0018	oz.inches.

3.6.5. MEASUREMENT OF RADIAL STIFFNESS.

To predict the whirl onset speeds of an air bearing system a precise knowledge of the radial stiffness of the journal bearing is of importance. Although an estimate of the radial stiffness of the bearings were obtained in the design stage, it is often of advantage to measure the radial stiffness of the bearings before the machine is run for the first time. This measurement technique has been further discussed in section 5.

3.7. FRICTION.

The friction of all gas bearings is very low compared to oil bearings and consequently it is not often a major design

consideration. In air bearing machines, mechanical losses due to friction are usually negligibly small and are often less than the windage loss. Nevertheless, bearings for high speed applications, which have to be designed with a small clearance in order to achieve adequate stability, can absorb appreciable power.

The friction torque of a two bearing system is given by the well known Petroff (46) equation, which is further substantiated by Rieger (40)

$$T_f = \frac{2\pi \mu r^3 L \omega}{h} \quad \text{in. lb.}$$

$$\text{Where } \omega = \frac{2\pi N}{60} \quad \text{rad/sec.}$$

Substituting the values in the above equation

$$T_f = \frac{2 \times 3.14 \times 2.62 \times 10^{-9} \times 0.625^3 \times 2.5}{5.0 \times 10^{-4}} \times \left(\frac{2 \times 3.14 \times 80,000}{60} \right) \quad \text{in. lb.}$$

$$T_f = 0.1684 \quad \text{in. lb.}$$

The friction torque in a loaded circular thrust bearing with plane surfaces is given by Powell (41) (42) as,

$$T_f = \frac{\pi \mu \omega}{2h} (R_o^4 - R_i^4) \quad \text{in. lb.}$$

Substituting the values,

$$T_f = \frac{3.14 \times 2.62 \times 10^{-9}}{2 \times 0.0007} \left(\frac{2 \times 3.14 \times 80,000}{60} \right) (1.25^4 - 0.625^4)$$

$$= 0.0452 \quad \text{in.lb.}$$

Therefore, the total friction torque T_{ft}

$$T_{ft} = 0.1684 + 0.0452$$

$$T_{ft} = 0.2136 \quad \text{in.lb.}$$

Since friction horsepower F_f , given as,

$$F_f = \frac{2 \pi N T_{ft}}{12 \times 33,000}$$

$$F_f = \frac{2 \times 3.14 \times 80,000 \times 0.2136}{12 \times 33,000}$$

$$F_f = 0.27$$

3.8. REYNOLDS NUMBER AND ITS EFFECT.

Turbulence in the lubricant film becomes a very real problem for a shaft running at high speed. To achieve a stable system, the speed has to be restricted to maintain the laminar flow condition. For the calculated design, the critical speed is obtained by using the equation 17,

$$\frac{\pi D N_c h}{60 \nu} = 41.1 \sqrt{\frac{D}{2h}}$$

Where ν is the kinematic viscosity.

$$N_c = \frac{41.1 \times 60 \times \nu}{\pi \times D \times h} \sqrt{\frac{D}{2h}} \quad \text{rev/min.}$$

$$N_c = \frac{41.1 \times 60 \times 2.28 \times 10^{-2}}{3.14 \times 1.25 \times 5.0 \times 10^{-4}} \sqrt{\frac{1.25}{0.001}}$$

$$N_c = 1,013,000 \quad \text{rev/min.}$$

Hence, the critical speed is much higher than the required running speed. Again the Reynolds number for the calculated design assuming 80,000 rev/min., is given by,

$$Re = \frac{U h}{\nu}$$

$$\text{Where } U = \frac{\pi N D}{60} \quad \text{in/sec.}$$

$$Re = \frac{\pi N D}{60} \times \frac{h}{\nu}$$

$$Re = \frac{3.14 \times 80,000 \times 1.25}{60} \times \frac{0.0005}{2.28 \times 10^{-2}}$$

$$Re = 116$$

Therefore, the Reynolds number is within the laminar flow region.

3.9. SUPPLY HOLE.

To allow a smooth flow of air and to avoid starvation at the orifice, it is important to make the supply hole area much greater than that of the orifices. This was carried out as a ratio of supply area to orifice area as shown below.

For the front bearing -

Area of orifice hole x number of holes < area of supply hole

$$\text{Total area of orifices} = \frac{\pi d^2}{4} \times 16 \quad \text{where } d = 0.004 \text{ in.}$$

$$= \frac{3.14 \times 0.004^2 \times 16}{4} \quad \text{in.}^2$$

$$= 201 \times 10^{-6} \quad \text{in.}^2$$

$$\text{Assuming a supply hole } d_s = 0.125 \quad \text{in.}$$

$$\text{Area of supply hole} = \frac{\pi d_s^2}{4} \quad \text{in.}^2$$

$$= \frac{3.14 \times 0.125^2}{4} \quad \text{in.}^2$$

$$= 12250 \times 10^{-6} \quad \text{in.}^2$$

$$\text{Then the ratio.} = \frac{\text{supply area}}{\text{orifice area}}$$

$$= \frac{12250 \times 10^{-6}}{201 \times 10^{-6}}$$

$$\frac{60}{61}$$

Considering the rear bearing, which has the same hole area as the front one and also the hole area of thrust bearings. For design simplification, one supply hole used in both cases, as shown in Fig. 14.

For the rear bearing -

$$\begin{aligned} \text{Total area of thrust orifices} &= \frac{\pi \times d^2}{4} \times 16 \text{ in}^2 \text{ where } d = 0.015 \text{ in.} \\ &= \frac{3.14 \times 0.015^2 \times 16}{4} \text{ in}^2 \\ &= 2830 \times 10^{-6} \text{ in}^2 \end{aligned}$$

Total area of rear bearing to be supplied = Area of thrust orifices
+ area of journal orifices.

$$\text{Total area} = (2830 + 201) \times 10^{-6} \text{ in}^2$$

$$= 3031 \times 10^{-6} \text{ in}^2$$

$$\text{Assuming a supply hole } d_s = 0.25 \text{ in.}$$

$$\text{Area of supply hole} = \frac{3.14 \times 0.25^2}{4} \text{ in}^2$$

$$= 49100 \times 10^{-6} \text{ in}^2$$

$$\text{Then the ratio} = \frac{49100 \times 10^{-6}}{3031 \times 10^{-6}}$$

16

Hence, both holes are large enough to supply the orifices.

3.10. INFLUENCE OF MATERIAL PROPERTIES.

The selection of air bearing materials is governed by many factors; most of them are essentially the same that apply to the selection of oil - lubricated bearing materials, but their relative importance may be different. One of the main differences arises from the fact that air bearings require parts made much more accurately than oil bearings, and also that the materials must be compatible with respect to friction without lubricant. The fulfilment of these two main conditions with one combination of material is the main reason of difficulty, and in most cases a compromise must be made. Although air bearings have been used for some years, very little information about the performance of various combinations of materials is available.

While considering the influence of the bearing and shaft materials on the performance of an air bearing, five main factors were taken into account which affect the performance.

3.10.1. MACHINABILITY.

The minimum film thickness between the two bearing surfaces before any contact of the surface asperities occurs is mainly a function of the accuracy of the bearing surfaces. To some extent this accuracy depends on the bearing materials. Some materials, in particular hard metals like steels, are best suited for the machining techniques used for air bearings than soft material, like babbitt-metal.

3.10.2. PHYSICAL AND METALLURGICAL STABILITY.

Once good geometry and surface finish have been achieved, it is important that this precision is not lost by distortion during or before test, which can be either permanent or temporary. Permanent distortions are mainly due to insufficient metallurgical stability and should be minimized by proper heat treatment before the final machining operation. This heat treatment or stress - relieving technique is well known in

industry and commonly used on materials like steels, cast iron, etc. The temporary distortion is caused by overload, particularly on thrust bearings, and by uneven temperature distribution.

3.10.3. THERMAL CONDUCTIVITY AND EXPANSION.

Self generated and external heating problems can lead to uneven temperature distribution and hence deformation of the system. The shear of the air film generates heat which has to be conducted away through one of the bearing parts.

In a 360° journal bearing the air film thickness varies around the circumference and the energy produced also varies. Theoretical and experimental investigations by Powell (43), show that this does not normally lead to an uneven temperature distribution because of the good distribution of the energy by moving gas. The heating up will be even and leads to growth of both parts, which does not affect the performance provided the coefficients of expansion of the shaft and the bearing are similar. However, the last condition is not practicable from the production and the cost aspects. Moreover, in case of overloading, if seizure takes place, then both shaft and bearings will be damaged if they are manufactured from same material.

3.10.4. STRENGTH AND ELASTICITY.

With most air bearing applications problems arising from insufficient strength of materials will not be encountered. However, deformations due to load can assume significant proportions, especially with pivoted pad or thrust bearings the preventive measures are necessary. The main criterion in selecting a material is the modulus of elasticity, as the natural frequency of the system is directly proportional to it, and at high speeds this is one of the governing factors.

3.10.5. CHEMICAL STABILITY.

The materials used in an air bearing must resist chemical attack by the surrounding atmosphere. Also particular attention must be given to selecting rust free material because of water particles in the supply line.

3.11. MATERIALS.

A large number of materials can be used for air bearings. Their suitability for particular applications has been considered as discussed earlier. The importance of the various desirable properties varies and depends on the type

of bearing.

Metals are an obvious choice for an air bearing, mainly because they are strong, often relatively cheap, have good thermal properties and in most cases there is extensive knowledge of technology for them, which means either that they are easy to machine, or that the difficulties have been solved. Disadvantages are their generally poor rubbing performance when used as a pair and their often poor chemical stability.

Among all the ferrous and non-ferrous metals Mild steel, Alloy steels and Stainless steels for the shaft, and Bronzes and Monel for the bearings were considered. On final evaluation two metals were found which satisfy the requirements.

3.11.1. ALLOY STEEL.

Among many varieties, En 19 steel was found to be the most suitable for the shaft of the air bearing. Due to small additions of Mn, Cr and Mo, it has greater strength and better chemical properties, and also to some extent improved rubbing properties. Its machinability, high tensile properties and impact properties are good. This particular grade is suitable for surface treatment, which improves the scoring resistance. Stress relieving at 670 - 780°C for 2 - 3 hours is required. The test shaft was chrome plated with 0.015 ins. radially to avoid rust. Moreover, it is cheaper than some of the more expensive steels, such as En 23.

3.11.2. BRONZES.

In general, they have reasonable anti-seizure properties and are used extensively in oil bearings. The best are phosphor bronzes and lead bronzes. They are most useful for lightly loaded bearings; however their performance at higher loads and in corrosive atmospheres is doubtful. In these alloys, a small amount of self alignment is possible because they yield in a plastic manner, and since the alloy is self annealing at the normal working temperature, the embrittling tendency of this kind of cold working is obviated. Finally the phosphor bronzes are cheaper than the lead bronzes. The journal and thrust bearings used in this research were made of phosphor bronze, and stress relieved at 200-300°C for approximately 2 hours.

3.12. AIR SUPPLY.

For an externally pressurized air bearing, the availability of a suitable air supply is always a matter of the greatest

importance. The two most important factors which have been considered in the design were clean air and constant supply.

Of these two factors, cleanliness is important in relation to the supply of compressed air. Almost all industrial compressed air is unsuitable for direct use in air bearings due to the presence of large quantities of impurities. The most common of these are solid particles, oil and water. The solid particles are potentially the most dangerous, but are most easily guarded against by the use of a 5 micron or finer sintered bronze, ceramic or carbon absorber filter. The performance of a bearing is not directly affected by the presence of oil, but water in the form of mist does cause corrosion, and in the droplets form would impinge on the surface opposite the feed hole causing pitting. Oil in minute quantities may be an advantage because it could prevent damage during accidental overload and it also helps to keep the shaft rustfree. But one disadvantage is that when the spindle is standing overnight, the oil settles to form a film between the shaft and the bearing, and when started it produces a drag due to the higher viscosity of the oil and the small clearance of the system. However, to overcome this problem effectively, flushing devices could be incorporated. In view of the poor quality of air and the short life of filtration equipment, the experimental grinding head was designed so that it could be easily dismantled for periodic cleaning. For test purpose two types of filtering unit are used; the first one contains a sintered bronze filter and the second one is a carbon absorber type with a flushing unit.

In order to make this grinding head economically viable, some safety features have to be included to avoid damage resulting from air supply failure during use. Firstly, an adjustable pressure switch operates electrical contacts at 10 lb/in² below the design pressure, thus preventing the motor from starting unless the bearings are pressurized. Secondly, if during the operation the air supply should fail below the set level, the same switch must break the power supply to the motor, and then immediately switch in a d.c. supply to apply the eddy-current brake to stop the spindle quickly. These safety features were not included in the experimental stage for economic reason and also due to lack of space in the test rig.

3.13. MANUFACTURING CAPABILITY.

In order to achieve high stiffness and low consumption of air, the air bearing has to be designed with small clearances. These small clearances demand high precision bearing surfaces with closely controlled roundness. However, since a smaller clearance demands smaller orifice holes, for

the load capacity to be maintained, the difficulty in drilling the small holes are often the limiting factor in the design.

If perfect geometry of the bearing surfaces is assumed and the ability to drill very small orifice holes is available, then the load capacity of the air bearing is independent of clearance as discussed by Rieger and Wilcock (30). In practice, the minimum clearance that could be used is dependent on the accumulated geometric error in the bearing surfaces or the limitation imposed by orifice size.

3.14. OVERALL DIMENSIONS.

In most designs there is usually some restriction on overall size. Since with air lubrication the bearings usually occupy a large part of the volume within the machine, considerations of overall size are often of great importance. As far as this experimental head is concerned, the length of the shaft was made similar to that of the existing ball bearing shaft due to the limitation of space. Again considerable care was taken in designing the bearing mounting to minimise distortion of the bearing surfaces in the event of over-clamping. This is one of the main reason that the bearings are clamped axially rather than radially, which are shown in Fig. 16, 22. Moreover, tolerance of the bearings and the housing, Fig. 14, 15. must be clearance fit to avoid distortion.

3.15. SAFETY FACTOR.

Having determined the size of the bearing and the shaft, an important question arises before accepting the final design. Are the values of the applied loads inserted as they stand, into the load capacity expressions or should some safety factor be introduced? The answer to this usually depends upon the certainty with which the applied load has been ascertained. For practical purposes, the load capacity values are usually accurate enough, within limitations. However, in the case of grinding, the applied load is difficult to ascertain accurately. It was considered that, in the case of an experimental load when running at high speed, very little force would be exerted on the bearing during the grinding process. According to the manufacturer, the grinding head itself is overpowered when a 4H.P., motor is used. Hence, it is considered that no safety factor is required when the load on the bearing is calculated on the assumption of 4H.P., motor and 100 percent efficiency of driving arrangement.

4.1. MANUFACTURING TECHNIQUE.

In the production of air bearings, there is a wide variation in manufacturing and inspection methods, that can be, or have been employed. However, in the case of this experimental grinding head, every step has been taken to avoid the use of special manufacturing and inspection techniques. This was done mainly for two reasons, to lower the cost per unit to produce it and to make it economical proposition, and to achieve a functional piece of equipment within the limited amount of time available. Although the conventional production processes were used, experience has shown that it is important for a design engineer to appreciate their special significance in air bearing technology.

Almost all the components of the experimental air bearings were manufactured by sub-contractor, care was taken not to involve them in greater details of special manufacturing technique except that prior to final machining the shaft and bearings must be stress relieved as per design specifications stated earlier. After all the components were received from the sub-contractor and passed by inspection then the following finishing processes were carried out in the machine tool workshop.

4.2. BEARINGS.

After mounting the 'O' ring onto the bearings, Fig. 31, 32, they were assembled with the head casting, Fig. 14. then both the bearings were lapped in line. Throughout the process two lapping shafts were used, one with thrust pad and another without it. These two are exact replicas of the actual shaft manufactured from Meehanite and contains spiral groove on the bearing position as shown in Fig. 32, 33, respectively.

The lap without the thrust pad was ground to a close slide fit to the bearing bore, charged with 1 micron diamond compound and carefully located in the bore. Lapping was then commenced imparting a rotary action while at the same time incorporating a slight reciprocating movement. It was important that at no time during this operation did the lap protrude from either end of the bore in order to avoid bell mouthing; this condition can take on serious proportions in a relatively short space of time. Also lapping was carried out in a vertical machine to avoid the effects of gravity load on the lap.

Intermediate inspection assessment of the parallelism of the bores was taken at regular intervals; this was achieved by using an air gauge connected to a Sigma air unit, Dialair, type no. DN 1, serial no. PNC587, which gave accuracy in the order of 10 micro-inches. This measuring device, after being

set to a particular range by Reference ring gauges, certified by the National Physical Laboratory, was inserted into the bore and a thorough assessment made of its entire length; from these results any necessary corrective lapping action was then taken until the required parallelism of 10 micro-inches had been successfully achieved. A similar check was again made at an axis 90° to the former position and by comparing these results an accurate picture of its roundness was obtained.

Then the head assembly was thoroughly cleaned and the other half of the thrust bearing Fig. 34, was fitted. Using the finish bore as datum, the second lap with thrust pad was used to achieve the squareness of the thrust faces Fig. 31, 34, to the bore axis. This lapping operation was carried out using a part rotary, clock and anticlock wise motion, and the spindle being indexed $20^\circ - 30^\circ$ at regular intervals. Before final lapping of the thrust faces all the thrust orifices, Fig. 21, were inserted and araldite was used to hold them in position. This was carried out to ensure that the orifices were lapped flat with the thrust faces. While lapping, the intermediate assesment of the squareness of both faces was obtained by auto-collimation.

With the completion of final ultrasonic cleaning processes the grinding head with bearings was then ready for final checking in metrology section. Before final inspection the sub-assembly was allowed to stand for 16 hours in the conditions of a dustfree atmosphere and controlled temperature, in this time any dimensional variation due to temperature change will have stabilised. The results of the final inspection along with Talyrond and Talysurf traces are shown in Fig. 35, 36, 37,.

4.3. SHAFT.

In the manufacture of the shaft, Fig. 17, the turning, grinding and stress relieving were carried out by the sub-contractor, the journal diameters were produced 0.020in. smaller than the required diameter. The shaft was then hard chrome plated with 0.030ins. on diameter for the reason stated earlier.

After initial inspection, the shaft was then accurately set between the polished centres of a Studer cylindrical grinding machine and the journal diameters were ground to give a size of 1.252 ins. i.e. 0.002ins. above the nominal size required. This 0.002ins. being found from experience to be adequate to gain confidence between the machining and inspection processes. The shaft was then checked by a Sigma air gauging comparator which was set against a specially prepared stack of Reference slip gauges corresponding to this

size. The overall measurement accuracy of the comparator is in the order of 10 micro-inches. At this stage the shaft was also checked for roundness and surface finish. This inspection stage was carried out to ensure compliance of the grinding machine accuracy with the requirements.

The shaft was then returned in between the centres and final grinding of the journals and the thrust plate was carried out to obtain the total clearance in the order of 0.001in. and 0.0014in. respectively. Throughout the finish grinding, the journal diameters were constantly checked, for absolute size and parallelism by Sigma comparator as described earlier. Also a Ferranti Mitronic measuring instrument, serial no. 1246, was used to check the squareness of the thrust plate in relation to journal diameter. It is of interest to note, that grinding was used as the finishing process rather than honing or lapping. This was because honing will not improve straightness and parallelism, due to the abrasive stones being much shorter than the length of the shaft, and lapping is a slow and expensive process.

After the final grinding, the shaft was balanced to 0.0007 oz.ins. at 6000 rev/min., ultrasonically cleaned and allowed to stand for at least 16 hours in the metrology room to obtain temperature stability. The journal diameters were then measured for absolute size, roundness, parallelism and surface finish, using a similar set-up as for journal bearings, and the results are given in Fig. 38, 39, 40.

4.4. ORIFICES.

In the externally pressurized air bearing applications the orifice diameter is large enough to cause no difficulty in manufacture. It is only on the smaller sizes, below approximately 0.015 ins. diameter, that difficulties may be experienced due to the small size coupled with difficulty of access by the drill to the orifice hole, or on account of the hardness of the material.

For the experimental bearings, the thrust orifices, Fig. 21, with 0.015 ins. diameter hole presents no production problem but the journal bearings orifices, Fig. 20, with a hole of 0.004 ins. diameter is certainly a manufacturing problem. However, there are two methods which could be used to obtain satisfactory results, one of them is by drilling and the other one which has proved successful for production is that of spark erosion technique. By the first method it is less costly but difficult to produce, and the second method is easier to produce but most expensive.

The orifices of the experimental head were produced by a small air drill attached to the quill of a conventional drilling machine, a suitable fixture was used to hold the orifice blank in position, the holes and their locations were checked under a travelling microscope, which gave a magnification in the order of $\times 100$.

4.5. ASSEMBLY.

During assembly it is essential that care and thought is given to the assembling of the various components. Although there is no fixed rule for assembling externally pressurized air bearings, some precautions are necessary to achieve the best results. For example assembling the thrust bearing, in such a way that subsequent distortions or other effects, which may affect satisfactory running of the shaft, are prevented.

After the final inspections all the components were dismantled, noting the fixing position of the bearings, before the final assembly all the orifices Fig. 20, were inserted into the journal bearings and held together by araldite. In extreme cases there was some interference between the orifice diameter and the hole in the bearing, special care was taken to avoid distortion of the bearing roundness during assembly. Then both the bearings were checked to ensure that no distortion had taken place. Although the above procedure was time consuming, experience has shown that it is safer to fit the orifices after lapping to achieve satisfactory results from the bearing and to avoid any 'trapping' of foreign particle between orifice pocket and orifice itself.

Finally, all the small components were once again ultrasonically cleaned. In the case of the body, Fig. 14, it was simply washed with trichlorethylene, but not the industrial grade which leaves a stain mark, followed by a wipe with a clean and soft tissue paper. Use of free compressed air is not suggested due to the possible presence of solid particles, which may impinge on the bearing surface. After cleaning, throughout the assembly procedure, the working surfaces of shaft and the bearings must never be touched by hand; this precaution is necessary to avoid any finger print marks on the working surfaces, which may contain some moisture and often corrosive fluids.

Prior to assembly all the components were sub-assembled as shown in Fig. 41. The complete assembly was then carried out in a dustfree atmosphere. While assembling, extra precaution was necessary to avoid any rotational movement of the shaft without air; it was essential to use some filtered air during

assembly, which floated the shaft rather than having metal to metal contact, also overtightening of thrust or journal bearing was avoided.

So far the physical manufacturing processes have been discussed, but some consideration must be given to the human element. The correct selection of personnel to carry out the assembly, and the final steps of manufacture is very important for the application of air bearings. Some of the qualities that are necessary are patience, a good sense of touch, a background of precision fitting, a high sense of responsibility, commonsense, and lastly, but by no means least, interest.

Finally, to achieve a successful air bearing, inspection is probably the most important item of all. In all cases a very much higher standard and thoroughness of inspection is required than is usually encountered. In principle, all the inspection items previously mentioned have to be checked when applicable, or at the very least the effects on performance of possible inaccuracies recognized. For example, efforts to make a design accurate within a few percent are completely wasted if the bearing is not within design tolerance in all respects and any inaccuracies not revealed.

5.1. DESCRIPTION OF TESTS.

The following gives details of the test rigs and the measuring instruments used. The Fig. 42. shows the first test rig along with the measuring equipment and Fig. 43. is the second rig which has been used with the same set of measuring equipment.

5.2. APPARATUS.

The first test rig contained the experimental head clamped on a plate, and another high frequency motor was mounted on a pair of vee blocks in line with the grinding head. This high frequency motor was used to drive the head at the required speed, the driving power was transmitted to the grinding head by means of a flexible plastic tube, which acted as a direct coupling. The use of plastic tube as a driving medium is justified for two main reasons; it overcomes any misalignment between the grinding head and the motor due to its flexible nature, and it avoids any vibration induced by the motor due to its damping characteristic. The driving motor was connected to a high frequency converter and cold water circulated around the stator to keep the unit cool at high speed.

The air was supplied to the grinding head through two filters as described earlier and a flow meter. The whole test rig was fitted with two 45° angle brackets. This arrangement permitted direct displacement readings from the Vibrometer probe and the Oscilloscope, during vertical loading on the shaft.

The instrumentation of test rig consisted of several measuring equipments which are described below. Among these the most important are the inductive transducers, types TW5-2/A, manufactured by Vibro-Meter Corporation. Two pairs of these transducers were used for the front bearing and another two pairs for the rear bearing. All of them were fitted so that by setting the test rig at on 45° angle the position of the pair of transducers becomes vertical and horizontal respectively, as shown in Fig. 44. These transducers are small in diameter, and were fitted to the head by means of a special holder, as illustrated in Fig. 45. (a,b). These probes provide a noncontacting method of monitoring radial displacements and amplitude of vibrations of the rotor. The transducers were connected to a six channel 8kc carrier frequency amplifier, type 8-MCA-1/A*, manufactured by Vibro-Meter Corporation, shown in Fig. 45c. Initially the transducers were set by two bench micrometers with known gaps and without the shaft. Then the amplifier was calibrated to obtain the zero scale reading. Finally, the calibration was such that the value of 0.01 Volt correspond to a distance of 1micro-inch, was readily discernible.

* Serial No. D 307, D 308, D 313, D 314, D 316, D 358.

Two X-Y curve tracing oscilloscopes, Tektronix, type 535 A, serial no. 101032 and type 535-56, serial no. 8129, were used to display the locus of the rotor centre and to check the radial displacement readings, the sensitivities of both the instruments were equalised and they were set against a known value of Vibrometer scale reading. In conjunction with the above mentioned measuring instruments, Sigma 31, serial no. 31 321, and Ferranti Mitronic equipments, serial no. 1246, were also used. These two were capable of giving direct displacement readings in the order of 10 micro-inches, which were used during load capacity and stiffness measurements. Supply pressures were measured with two pressure gauges which were attached with the filters, and air flow was measured with a calibrated flow meter, Rotameter type, serial no. R 307298.

The second test rig comprised the same air spindle and high frequency motor. But in this case both of them were mounted horizontally. The power was transmitted to the air spindle by means of two pulleys, shown in Fig. 46, 47, and a neoprene impregnated flat belt. The high frequency motor was clamped on a 12 'V' block, and this in turn was mounted on a sliding plate. This arrangement simplified the belt adjustment. The same set of measuring equipments were used, which has been described along with the first rig.

5.3. EXPERIMENTAL RESULTS.

Before carrying out any test the shaft was allowed to run freely at 5000 rev/min to observe the dynamic behaviour of the system. Next, the tests were carried out in three stages, namely,

1. Load capacity and stiffness.
2. Volumetric air flow.
3. Free running.

5.3.1. LOAD CAPACITY AND STIFFNESS.

The static loading test were carried out for both the journal and the thrust bearings. The loads were mounted on to a hanger attached to the shaft. The loading point of the front bearing was $1 \frac{5}{8}$ " from the centre line of the bearing, and for the rear bearing this distance was $2 \frac{1}{4}$ ". For the thrust bearing test the loads were directly mounted along the shaft. The test was started without any weight or rotational movement, eventually the maximum load of 50lbs. was used with increments of 5 lbs, and at three different pressures of 60, 80, and 100 lb/in².

Three sets of readings for each pressure were taken using Ferranti Mitronic and Sigma '31' equipments, and also these readings were compared with the readings of the Vibrometer instrument. The readings of load with the resulting deflection are shown in Fig. 48, 49, for two journal and thrust bearings. Moreover, the relationship of load against deflection with different pressures and stiffness against pressure are shown in Fig. 50 to 55.

5.3.2. VOLUMETRIC AIR FLOW.

Total air flow was measured by using a calibrated flow meter as described earlier. At first, the total air flow was recorded for a range of different pressures from 20 to 100 lb/in², with increments of 20 lb/in² and without any load or rotation. Then the same test was repeated with 50lbs weight at each end of the shaft, and the corresponding air flow was measured. Finally, the supply pressure was kept constant at 80lb/in², and the air flow was measured at different rotational speeds ranging from 10,000 to 60,000 rev/min., at no load condition. All the results of these tests and their relationship are shown in Fig. 56, 57, respectively.

5.3.3. FREE RUNNING.

This was carried out on the two test rigs which have been described earlier. The first trial run ranging from 0 to 50,000 rev/min. was achieved in the first rig, Fig. 42. Although this system did provide a higher speed without any ill effects, due to the centrifugal force the failure of the plastic tube coupling prevented it from driving the spindle at an increased speed.

To overcome this limitation imposed by the coupling, the second test rig was used. Moreover, within the speed range of 50,000 rev/min. the results of both the tests are compared to obtain the corresponding vibration induced by the belt. With the belt driving system, an actual speed of 75,000 rev/min, was achieved. In both tests the vibration amplitudes were measured directly from the Vibrometer, the results and their relationship are given in Fig. 58, 59. In the case of the second test the frequency was measured from the Oscilloscope to calculate the actual speed of the system, given in Fig. 60, thus eliminating the belt slip which was present at this high speed. At this stage, further increase in speed resulted in failure of the system; an investigation showed that the shaft had touched the rear end of the rear journal bearing.

6.1. DISCUSSION.

The immediate fact emerging from the experimental results is that the data which has been used in designing the air bearings gives reasonably accurate predictions of load capacity, stiffness and volumetric flow of air.

Although the results of load capacity and the resultant deflection show that only 50lbs. weight has been used for these tests, this was done mainly to avoid damaging the bearings, which are designed for eccentricity ratio $\epsilon = 0.5$, i.e. 50% of radial clearance. As can be seen, the loads are not applied on the centre line of the bearing, but 1.625ins. and $2\frac{1}{4}$ ins. distance from the centre of front and rear bearings. However, if the loads are shifted to the bearings centre line then the load bearing capacities are increased to 61.5lbs. and 65.5lbs. for front and rear bearings respectively (See Fig. 61).

The relationship of stiffness against supply pressure P_0 (gauge), Fig. 54. shows remarkable correlation between the predicted and experimental results. A point to note that in the case of stiffness, linearity exists, but the rear bearing shows slightly higher stiffness than the front one. This could be attributed to the thrust bearings or the smaller gap, influencing the stiffness of the rear bearing. Again Fig. 54, shows that at 80lb/in² (gauge) supply pressure the measured stiffness of the front and rear bearing is lower than the predicted value by 6% and $2\frac{1}{2}$ %, respectively. This discrepancy could be mainly for two reasons, the calculated stiffness is based on non-dimensional parameter Λ_s and P_s/P_a ratio, but does not take into consideration any circumferential flow, hence, the assumption of axial flow theory gives a slightly higher value than the experimental bearing. Also the accumulated manufacturing tolerances of orifice hole, shaft diameter and bearing bore may have a significant effect on both stiffness and load carrying capacity.

The slope of the curve in Fig. 51, 52, gives the lowest stiffness when the eccentricity ratio ϵ approaches 0.5. Higher values could be obtained by using a lightly loaded shaft or a low eccentricity ratio. Generally, at the higher supply pressure and lower eccentricity ratio the perturbation theory given by Lund (24), could be used with an accuracy of 10%. Moreover, the experimental results do under estimate the load carrying capacity, as the hybrid or hydrodynamic effect is not taken into account. Powell (19), showed experimentally that within a certain range the load capacity increases with shaft speed due to its hybrid effect, but this becomes insignificant at high speed due to the compressibility effect: the load carrying capacity tends to become independent of speed when large values of Λ are attained.

The thrust bearing gives an experimental stiffness value of approx. 12% higher than the predicated one. This may be due to the design data which is based on a clearance gap $h = 0.001 \text{ in.}$, and a pressure factor $K_q = 0.69.$, but at smaller clearances the value of K_q increases thus increasing the stiffness. Hence, some correction factor is required. The Fig. 52, 53, shows a linear relationship between load against clearance; however, at high speed, due to compressibility effects and inertial effects this linearity does not exist.

Again Fig. 55, shows that the stiffness of the front thrust bearing is 3% lower than the rear bearing. Once again, this is most likely due to the smaller size of exhaust hole between rear journal and front thrust bearings, which does not allow the exhaust air to escape satisfactorily. To overcome this problem, it is advisable to use the larger size of exhaust hole, approximately 0.187 in. diameter.

In general, a linear relationship between mass flow and supply pressure is indicated in Fig. 57; this can be explained by Shires (8) theory, considering each bearing as a complex nozzle, convergent linear in form. Discharge through the convergent section is closely linear with pressure, and through the clearance also where turbulent (axial) conditions exist.

As shown in Fig. 57, the measured airflow when the shaft is concentric with the bearing, is in excellent agreement with the predicted values. This undoubtedly is due to using the value for the discharge coefficient $C_d = 0.8$. Values of airflow measured for the eccentric case i.e. radial load of 50 lbs each end applied to the shaft, do not vary significantly from the concentric case. However, when airflow is measured with a constant supply pressure of 80 lb/in^2 (gauge) and with variable speed, it shows a $7\frac{1}{2}\%$ less consumption at 60,000 rev/min. than the non-rotational condition. This is attributable to the basic hydrodynamic effect which generates a hydrodynamic film due to the rotational movement of the shaft, thus reducing the airflow consumption.

Finally, the figure 59 shows the relationship of the vibration amplitude of the eccentricity with the shaft speed obtained from both types of driving method. A and B denotes * the amplitude of vibration, taken in vertical direction at front and rear end of the shaft, respectively.

In the first test when the shaft is driven directly, it exhibits a small unbalance whirl which is synchronous to the shaft speed. The amplitude relationships between the synchronous vibrations of the front and the rear bearing in general are not the same; therefore, they seem to depend on the distribution

of unbalance mass in the shaft and the small difference of stiffness between the two bearings. As the shaft speed is increased, two resonant speeds are observed in general when the supply pressure is kept constant at 80 lb/in² (gauge).

As the shaft speed, could not be further increased due to the centrifugal force acting on the plastic tube coupling, therefore, the shaft is driven by a belt in the second test rig. A violent vibration is observed around the speed of 12000 rev/min. and the amplitude of the vibration at both the bearings further increased with increase of speed. As can be seen from figure 59, at 75,000 rev/min. the amplitude of vibration at the rear bearing is twice that at the front one; moreover, there is some phase difference between both ends. Further increase of speed could not be obtained due to the 'whipping' effect of the belt. Also at this speed the belt vibration introduces the bending effect of shaft, which can be seen from the eccentricity value of the shaft at the rear bearing. Thus the eccentricity ratio $\epsilon = e/h = 0.6$, exceeds the designed eccentricity ratio. Moreover, another factor which may have prevented from obtaining higher speed, is the unbalance effect present in the shaft. The balancing of a shaft, within the required degree of accuracy is difficult to achieve by conventional methods. To overcome this problem it is advisable to balance the shaft in its own air bearings or on a pair of slave bearings.

The accuracy of the shaft speed was checked with the frequency readings of the oscilloscope, thus eliminating the belt slip. On analysing the results shown in figure 60, it can be stated that the accuracy is within 2 $\frac{1}{2}$ % of the driving speed of the high-frequency motor.

7.1. CONCLUSIONS.

The excellent correlation between the predicted values and the test results, obtained from the experimental grinding head, indicates that the approximate assumptions used in simplifying the design problem were justified. This simplified design data can be used to predict the stiffness, load capacity, air consumption and stability, within the degree of accuracy imposed by the manufacturing tolerances.

However, before embarking on the design of a high speed device extreme care must be taken regarding several factors. In all cases it is desirable, for stability, that the difference between the supply pressure P_s and the downstream pressure P_d be small. A condition wherein the orifice is choked will tend to cause the bearing to be unstable unless the recess depth is made very small. The orifice size is limited however, since it functions as a flow restrictor and in this way contributes to the stiffness of the bearing but, within these limitations the largest possible orifice hole diameter and smallest possible recess depth should be used.

Moreover, in any high speed air bearing to avoid the whirling speed every effort should be made to achieve the highest stiffness. Hence, the bearings should be designed so that maximum possible advantage can be gained from the hydrodynamic effect. Also when the supply pressure is greater than the ambient pressure, the compressibility effect should be taken into consideration; the noncompressible theory overestimates the bearing stiffness. As mentioned earlier, within manufacturing limitations, the radial clearance should be kept small in order to increase the stiffness and to reduce the air consumption.

The inability of obtaining the higher speed than 75,000 rev/min. indicates, that some other driving method should be used to avoid the belt vibration, or some form of damping device must be introduced to overcome this problem. Also out of balance effect of the shaft imposes a serious limitation in designing a successful high speed air bearing system.

It can be concluded, that a high speed air bearing could be designed successfully, if one can overcome the difficulties in manufacturing the components, and given due considerations to the above factors. However, much experimental work remains before the performance of this air bearing could be said to have been clearly defined.

8.1. FIGURES.

1.	Configuration of a typical externally pressurized air journal bearing.	71
2.	Schematic flow path through two orifices.	72
3.	(a) Pressure profile of a complete journal bearing, and (b) equivalent slot for an orifice based on axial flow theory.	73
4.	Idealized model for pressure distribution in clearance space of an air-bearing.	74
5.	Some types of orifice used in air bearings.	75
6.	Configuration of a typical recess orifice.	76
7.	Influence of number of orifices on journal bearing load coefficient.	77
8.	Influence of fractional distance of orifice from bearing ends on load capacity.	78
9.	Influence of fractional distance of orifice from bearing ends on angular stiffness.	79
10.	Relationship of load coefficient with fractional distance of orifice from bearing ends and length to diameter ratio.	80
11.	Two types of circular thrust bearing with pressure distribution.	81
12.	Optimum load coefficient of thrust bearings.	82
13.	General assembly of externally pressurized air-bearing for internal grinder.	83
14.	Head casting (Body).	84
14A	View 'A' of head casting.	85
14B	View 'B' of head casting.	86
14C	View 'C' of head casting.	87
14D	View 'D' of head casting.	88

15.	Front bearing.	89
16.	Rear journal bearing and front bearing.	90
17.	Spindle.	91
18.	Rear thrust bearing cover.	92
19.	Rear thrust bearing.	93
20.	Orifice (Journal Bearing).	94
21.	Orifice (Thrust Bearing).	95
22.	Front cover.	96
23.	Slinger.	97
24.	Load acting on journal bearings.	98
25.	Influence of length to diameter ratio on journal bearing load capacity.	99
26.	Influence of gas properties on orifice diameters, for ambient pressure. After Shires, (15).	100
27.	Hydrostatic journal bearing, restrictor coefficient, Λ_s , versus flow factor, M' .	101
28.	Hydrostatic journal bearing restrictor coefficient, Λ_s , versus load or stiffness parameter, K .	102
29.	Hydrostatic journal bearing, compressibility number, Λ , versus instability threshold parameter, Λ .	103
30.	Front journal bearing.	104
31.	Rear journal bearing & front thrust bearing.	105
32.	Lap with thrust pad.	106
33.	Lap without thrust pad.	107
34.	Rear thrust bearing before lapping.	108
35.	Showing measurements of journal bearings.	109

36.	Talyrond traces showing roundness accuracy of journal bearings.	110
37.	Talysurf traces showing surface finish of journal bearings.	111
38.	Measurements of shaft diameters.	112
39.	Talyrond traces showing roundness accuracy of shaft journal diameters.	113
40.	Talysurf traces showing surface finish of shaft journal diameters.	114
41.	Complete arrangement showing the components.	115
42.	First test rig with measuring equipments.	116
43.	Second test rig showing driving arrangement.	117
44.	Showing transducers position and driving arrangements.	118
45(a & b)	Sketches of transducer and holder,	119
45c.	Circuit drawing shown schematically.	120
46.	Driving pully.	121
47.	Pully for experimental head.	122
48.	Results of load capacity and stiffness tests of front journal bearing and rear journal bearing.	123
49.	Results of load capacity and stiffness tests of front and rear thrust bearings.	125
50.	Relationship between applied load and shaft deflections, (front position).	127
51.	Relationship between applied load and shaft deflections, (rear position).	128
52.	Relationship between applied load and clearance of front thrust bearing.	129
53.	Relationship between applied load and clearance of rear thrust bearing.	130

54.	Relationship between radial stiffness and supply pressure of journal bearings.	131
55.	Relationship between axial stiffness and supply pressure of thrust bearings.	132
56.	Results of volumetric flow measurements.	133
57.	Relationship between air flow and supply pressure and speed.	134
58.	Results of free running tests.	135
59.	Relationship of amplitude (eccentricity) against frequency for direct and belt drive.	136
60.	Results of belt slip measurements.	137
61.	Sketch of bearing centre, probe centre and loading positions.	138

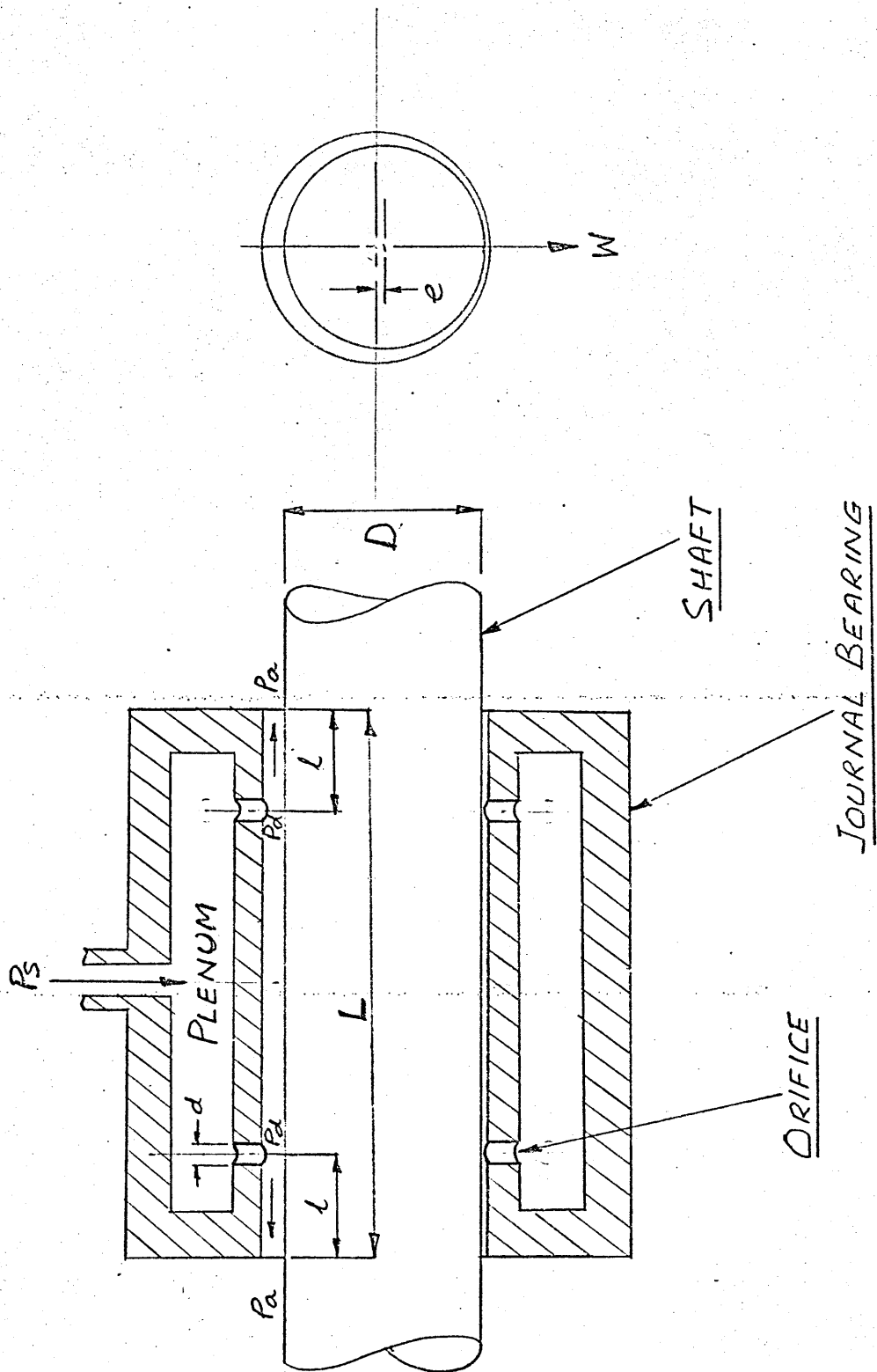


Fig. 1 — Configuration of a typical externally pressurized air journal bearing.

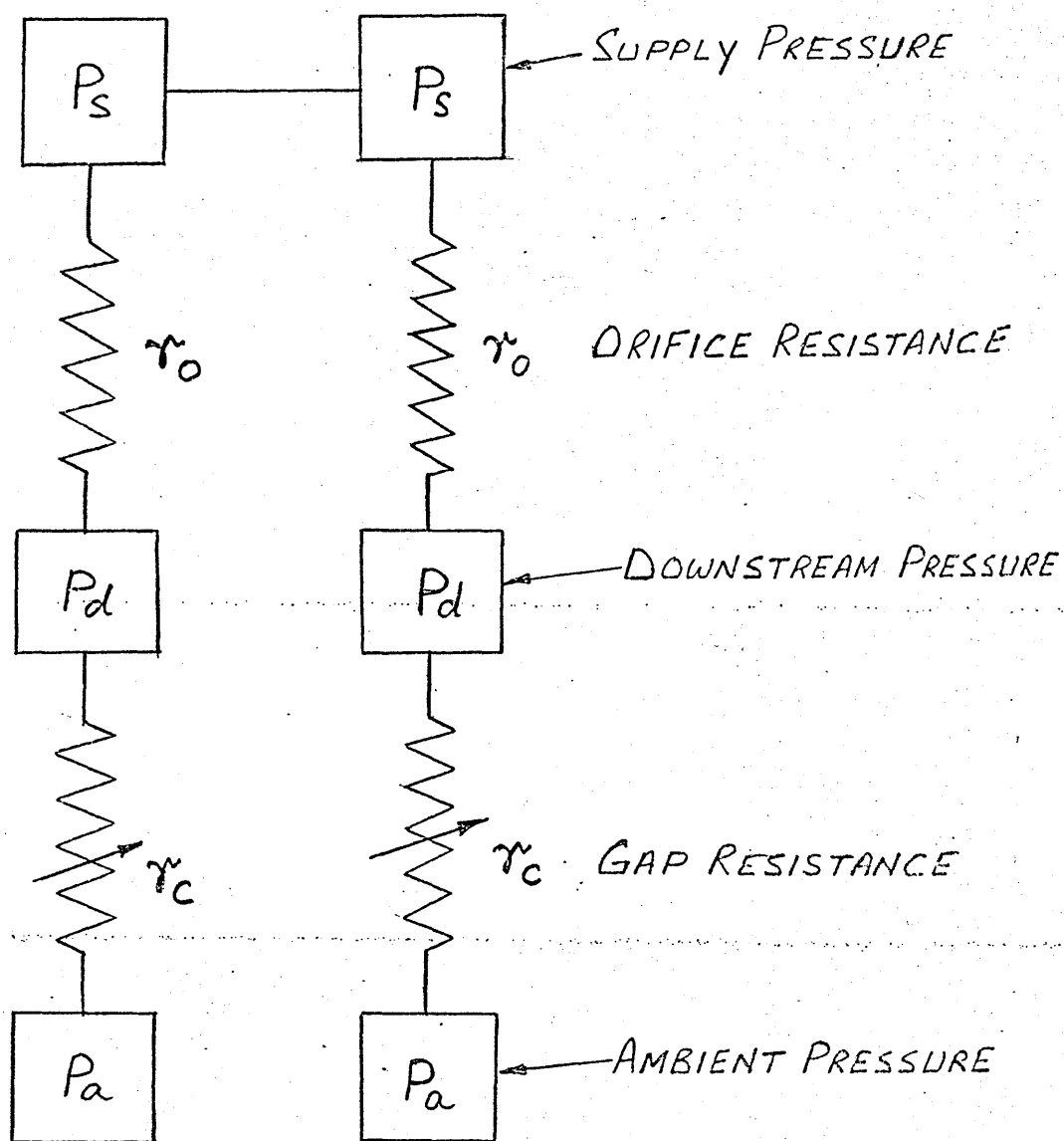
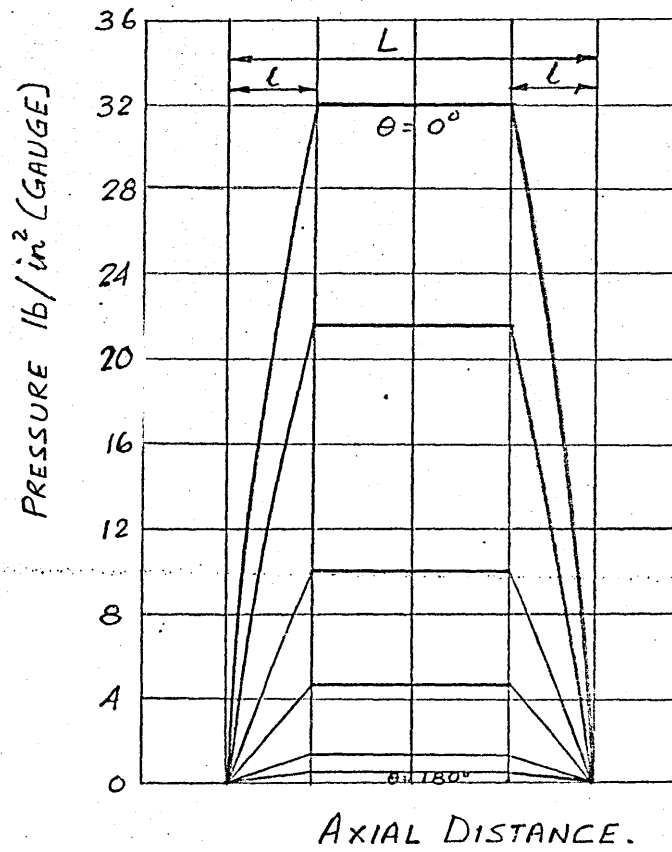


Fig. 2 — Schematic flow path through two orifices.

(a)



(b)

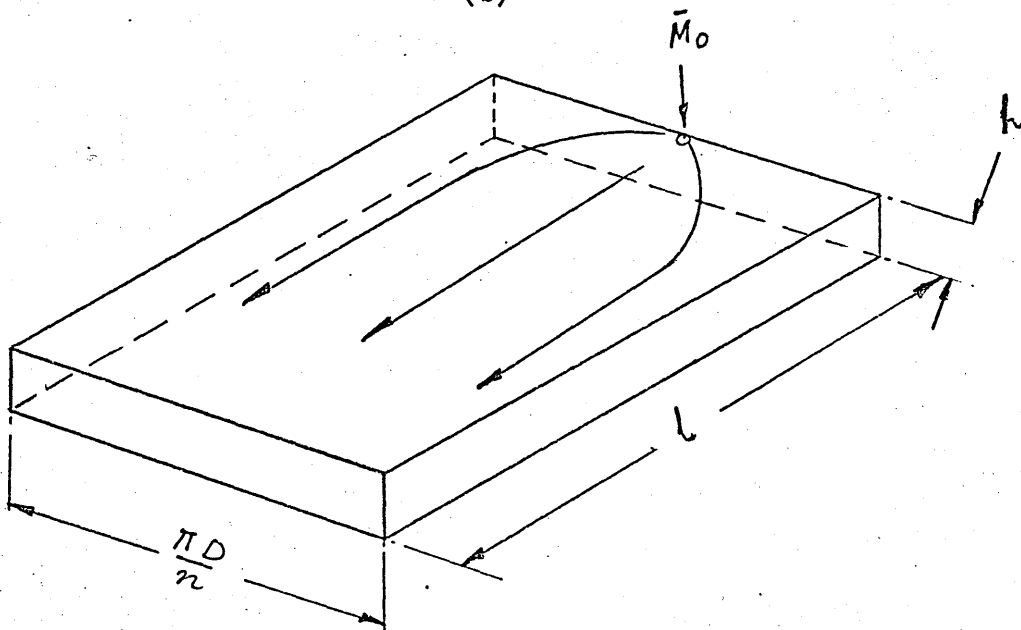
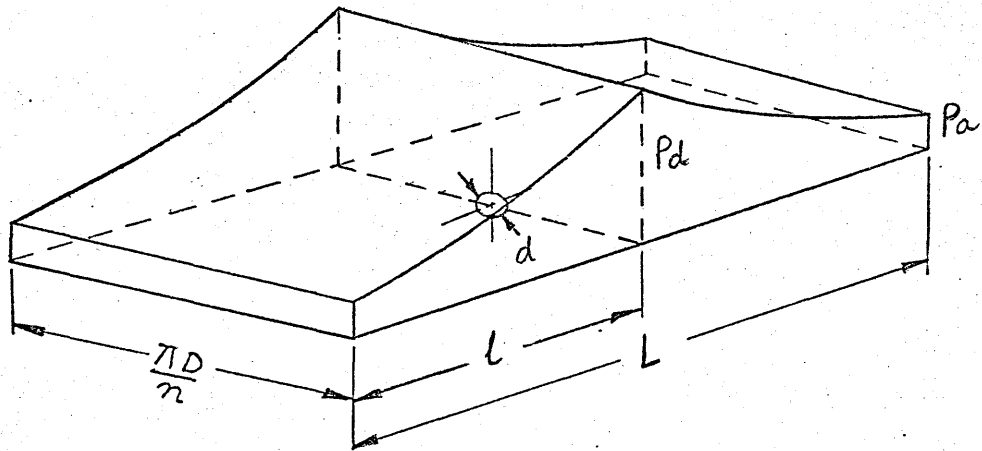
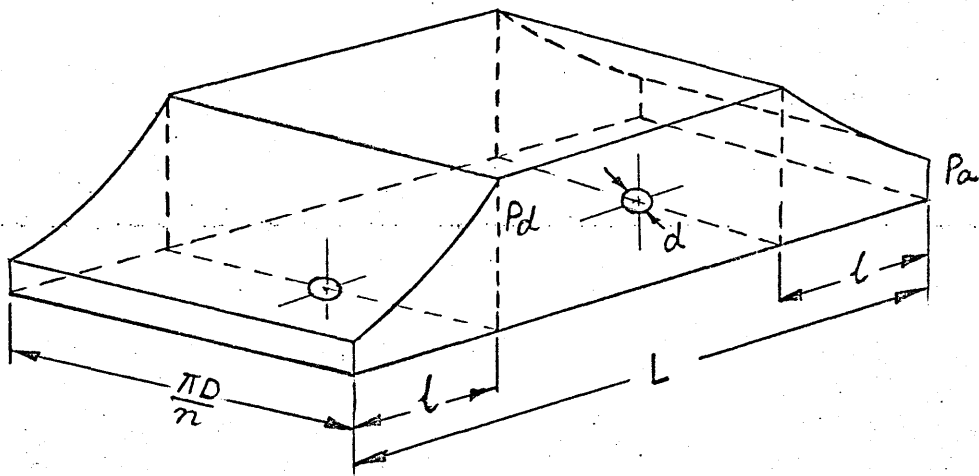


Fig. 3 — (a) Pressure profile of a complete journal bearing, and (b) equivalent slot for an orifice based on axial flow theory.

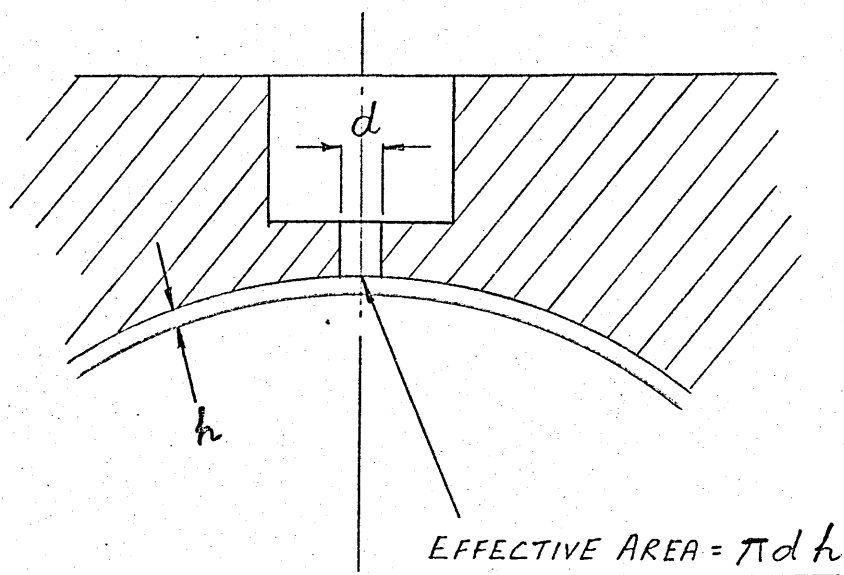


(a) Single central orifice.

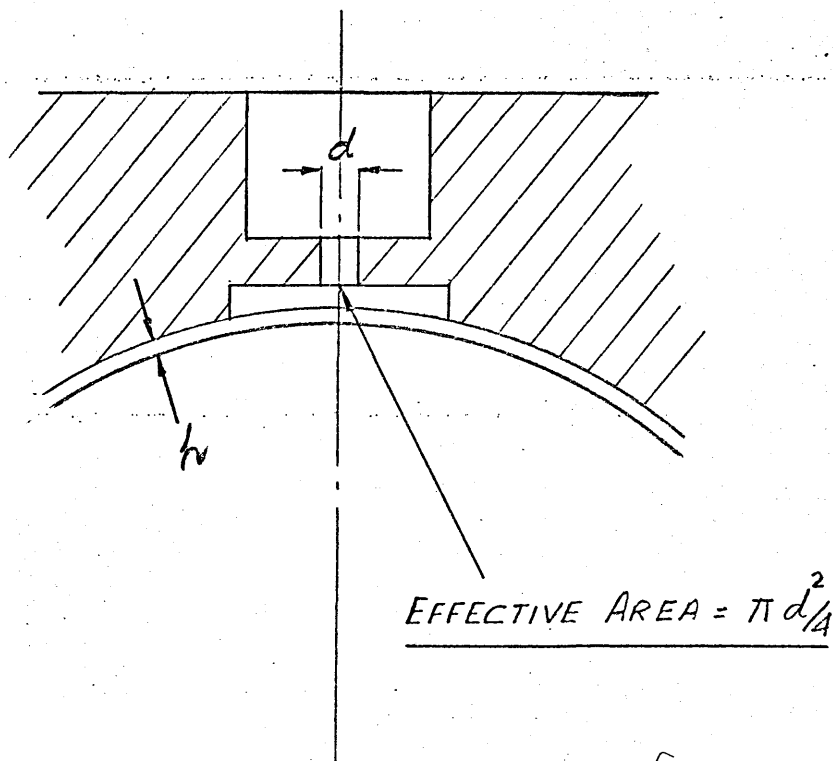


(b) Two orifices in one channel.

Fig. 4 — Idealized model for pressure distribution in clearance space of an air-bearing.



(a) Plain or annular orifice.



(b) Simple or recess orifice.

Fig. 5 -- Some types of orifice used in air bearings.

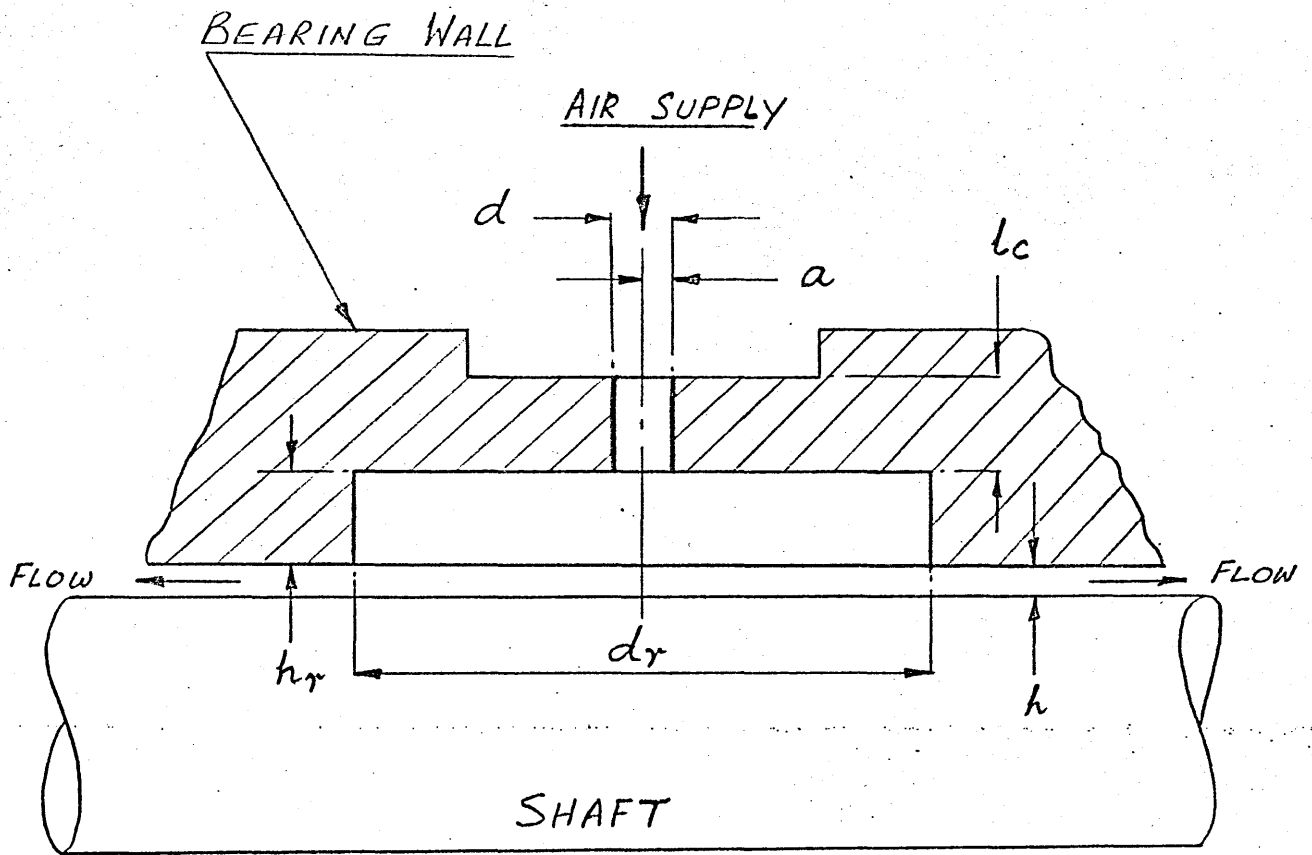


Fig. 6 -- Configuration of a typical recess orifice.

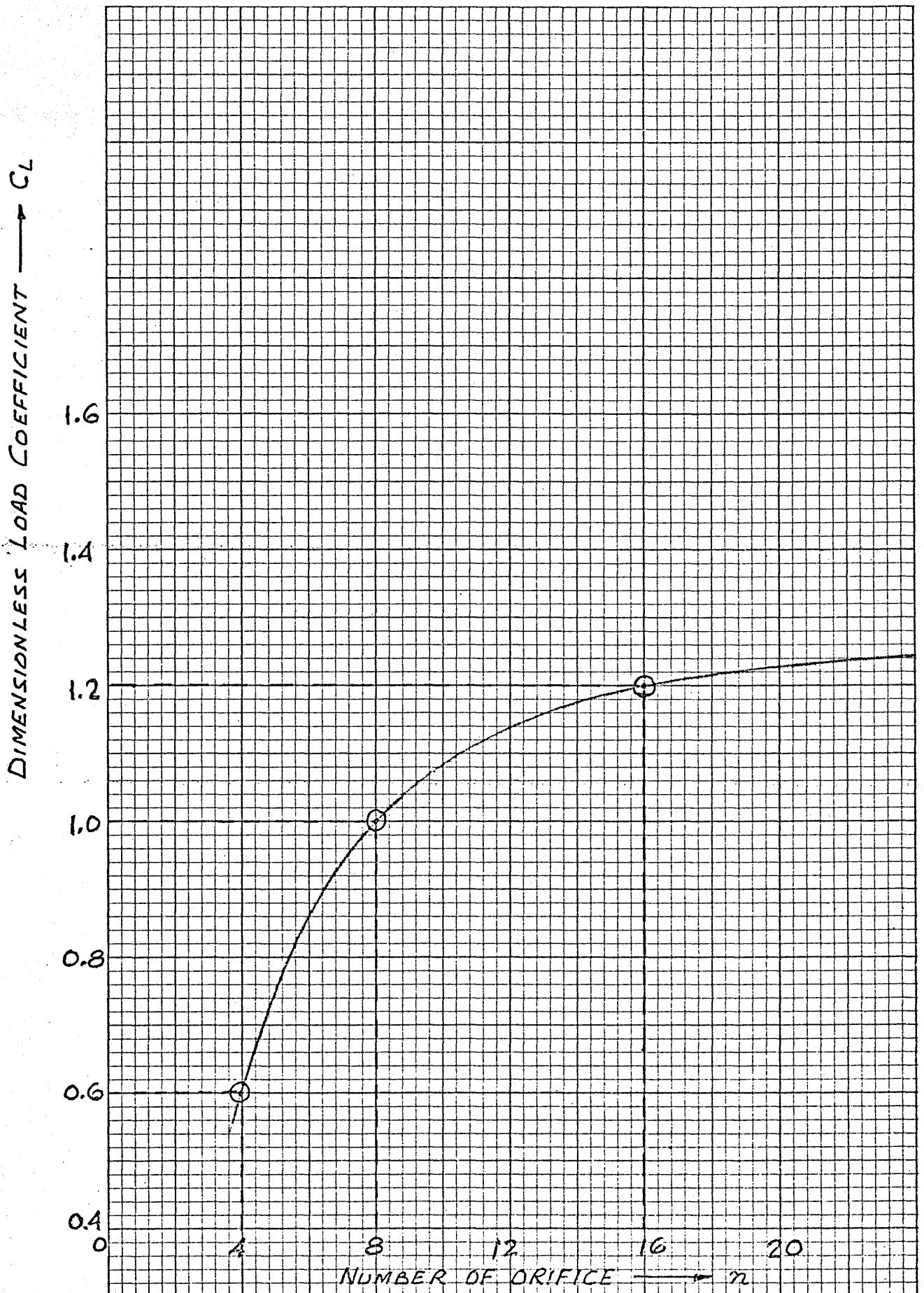


Fig. 7 -- Influence of number of orifices on journal bearing load coefficient.

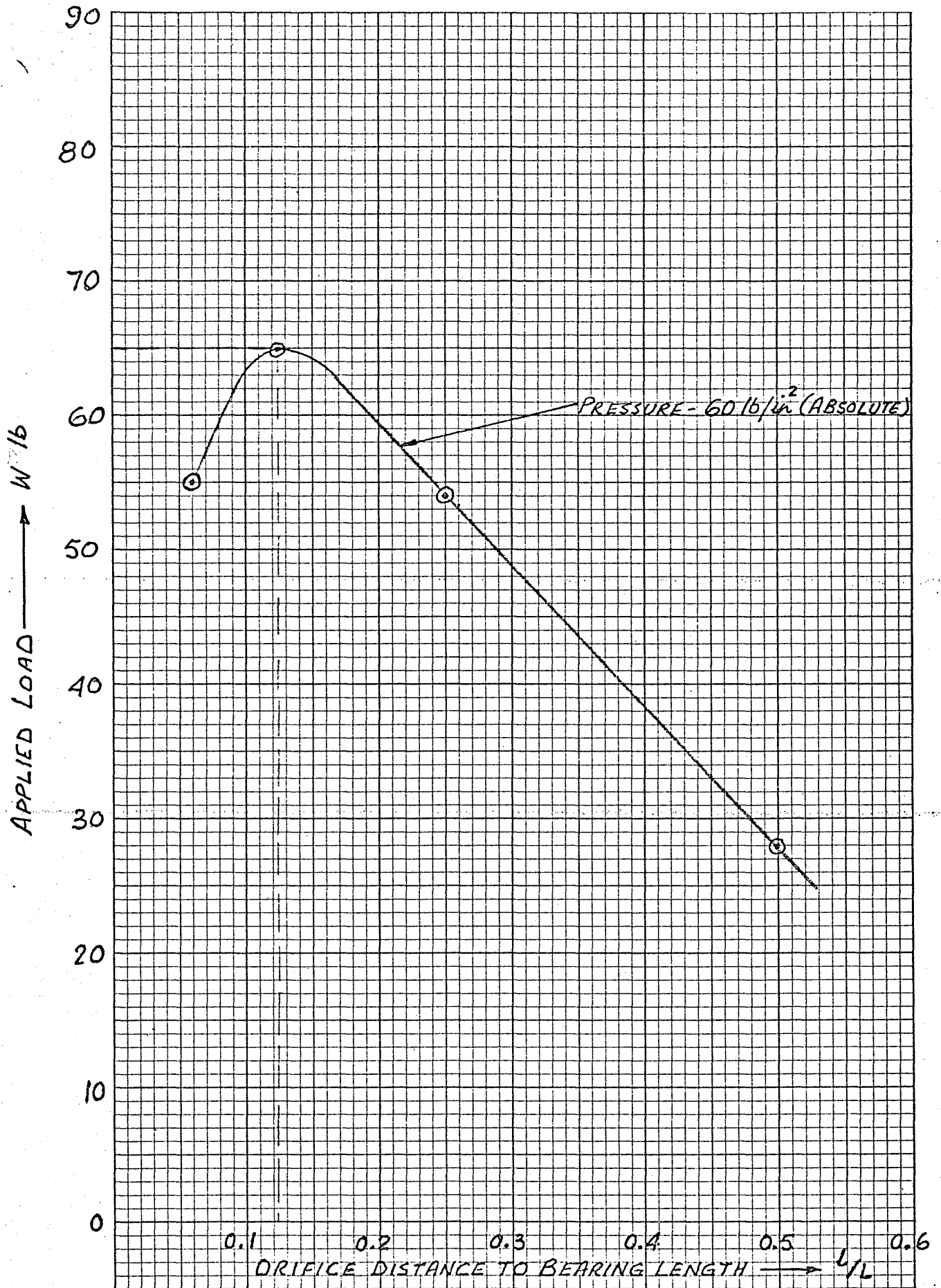


Fig. 8 -- Influence of fractional distance of orifice from bearing ends on load capacity.

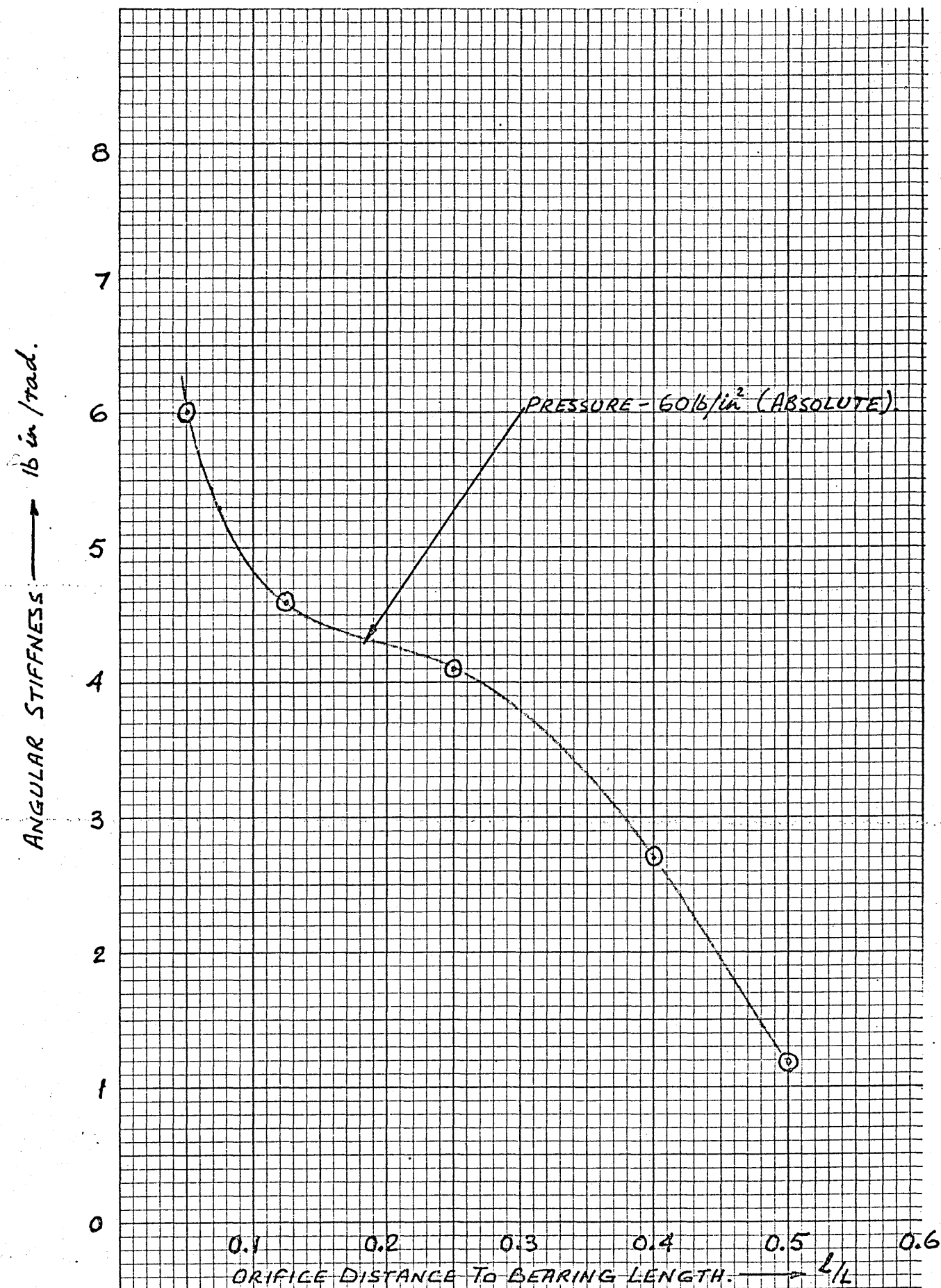


Fig. 9 — Influence of fractional distance of orifice from bearing ends on angular stiffness.

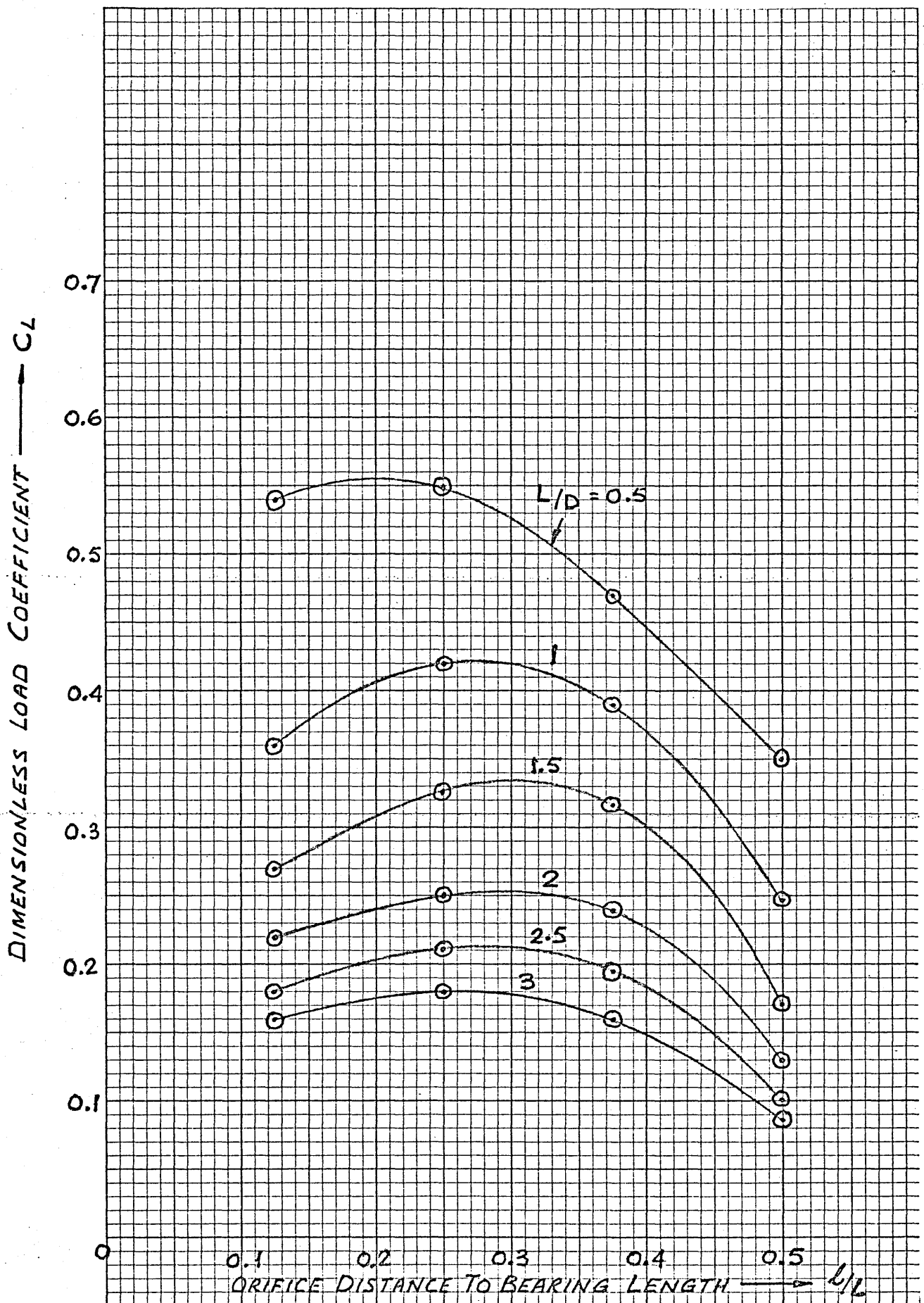
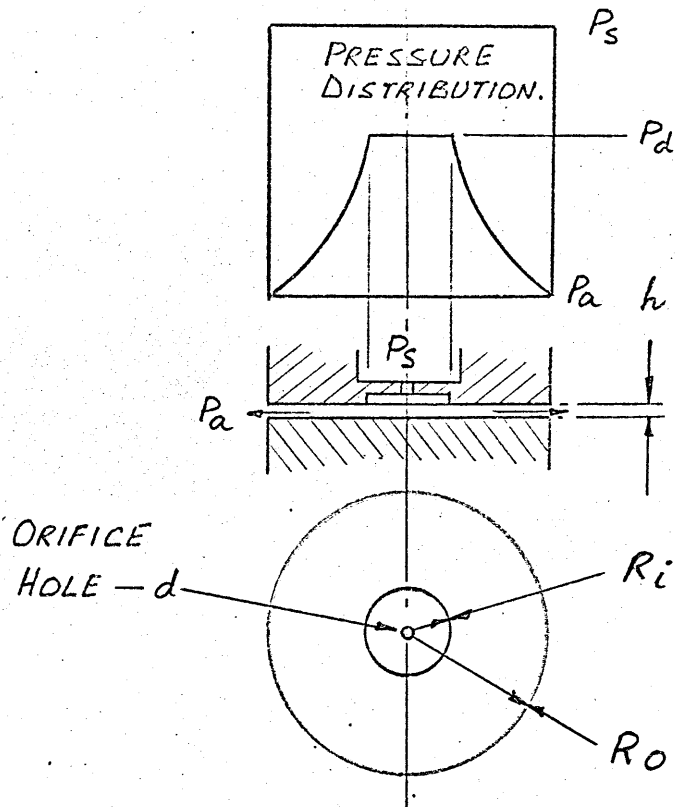
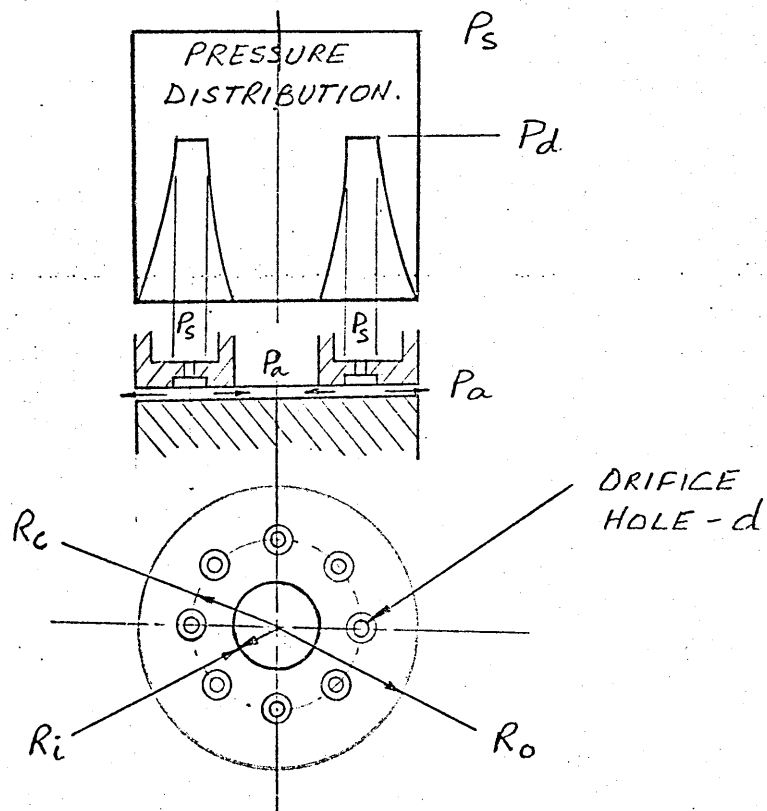


Fig. 10 — Relationship of load coefficient with fractional distance of orifice from bearing ends and length to diameter ratio.



(a) Thrust bearing with central orifice.



(b) Thrust bearing with ring of orifices.

Fig 11 -- Two types of circular thrust bearing with pressure distribution.

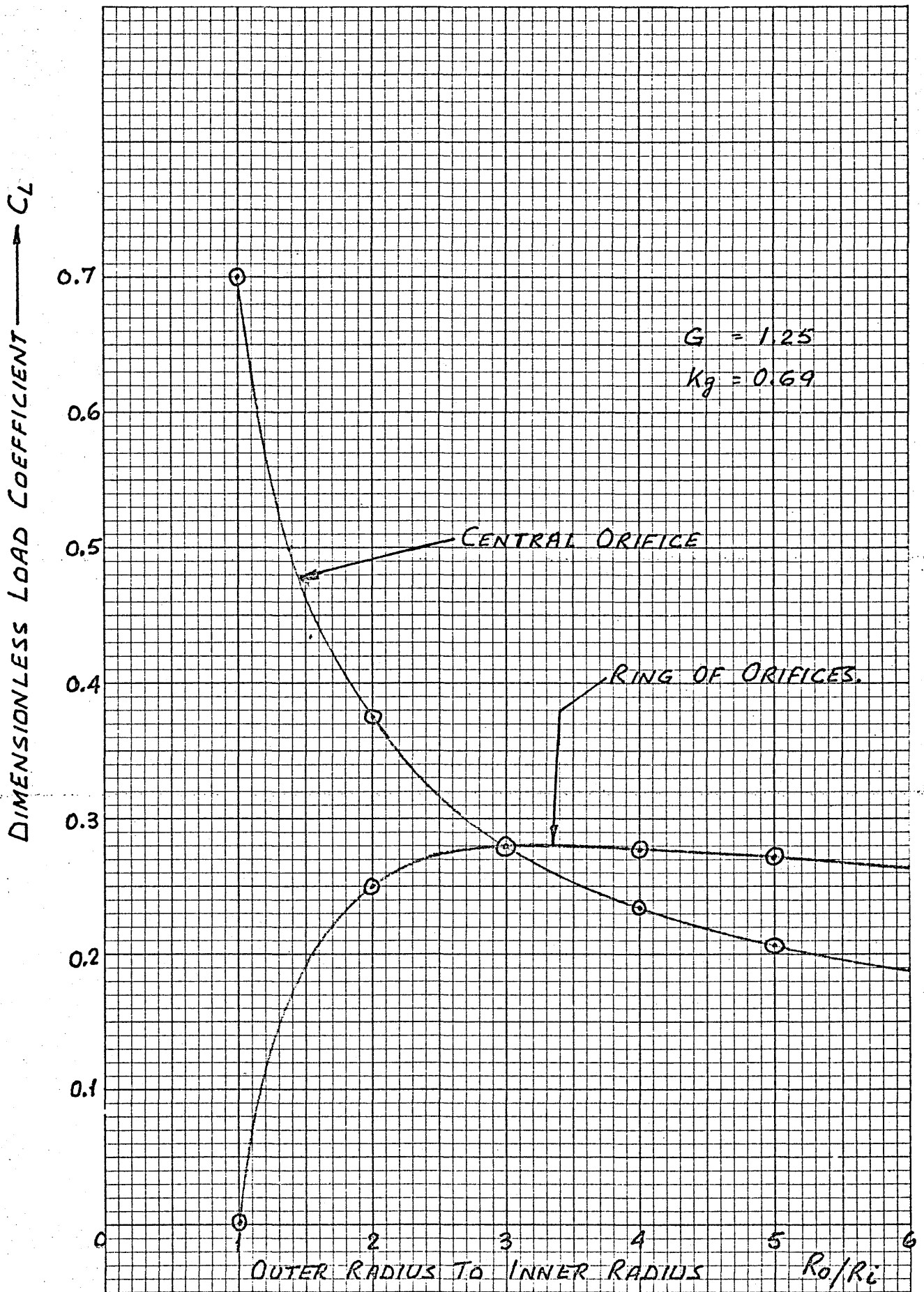
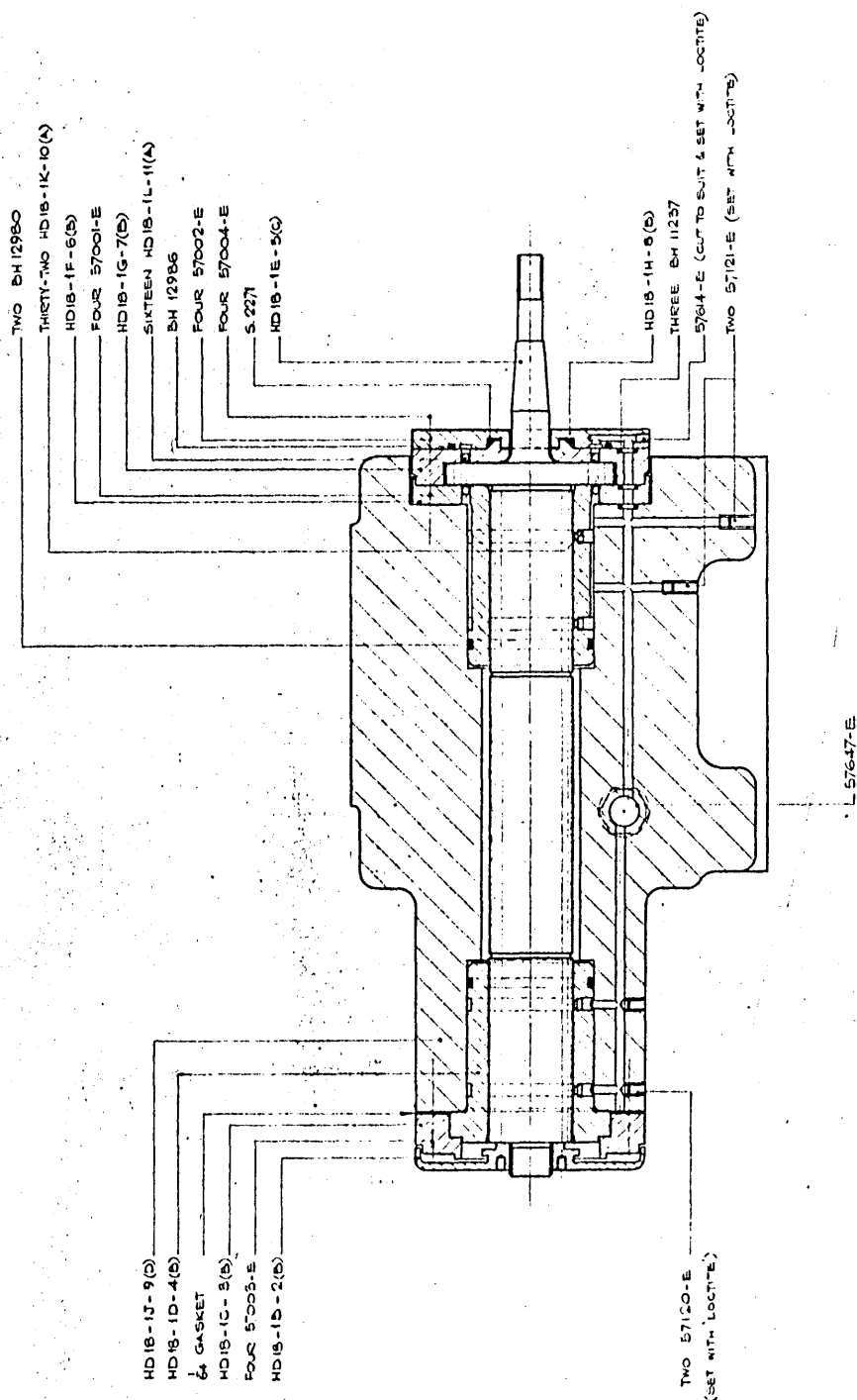


Fig. 12 -- Optimum load coefficient of thrust bearings.

IMPORTANT: THIS IS THREE ANGLE PROJECTION
DO NOT SCALE DIMENSIONS
FROM ALL DIMENSIONS MUST BE TO THE SAME DEGREE
CONFIRMED ALL DIMENSIONS MUST BE TO THE SAME DEGREE



ASSEMBLY NOTES

1. BEARINGS HD18-1D-4(B) & HD18-1F-6(B) ARE TO BE FITTED & FINISHED LAPPED UNTIL THE BORES LE WITHIN THE SIZE TOLERANCE OF 1/2500 1/2500 DIA.
2. BEFORE INSERTING THE SPINDLE HD18-1B-2(B) IT IS ESSENTIAL THAT THE WHEELHEAD BODY, SPINDLE & BEARINGS ARE CLEANED BY A HIGH FREQUENCY PROCESS & THAT THEY ARE NOT TOUCHED BY HAND AT ANY TIME DURING THE ASSEMBLY PROCEDURE. THIS IS EXTREMELY IMPORTANT.
3. UNDER NO CIRCUMSTANCE MUST ANY PART COME IN CONTACT WITH OIL, GREASE, OR WATER.
4. THE SPINDLE MUST NOT BE ROTATED ONCE IT HAS BEEN ASSEMBLED INTO THE BEARINGS.

Part no - HD18-1-1D
Material - ---

Fig. 13 -- General assembly of externally pressurized air-bearing for internal grinder.

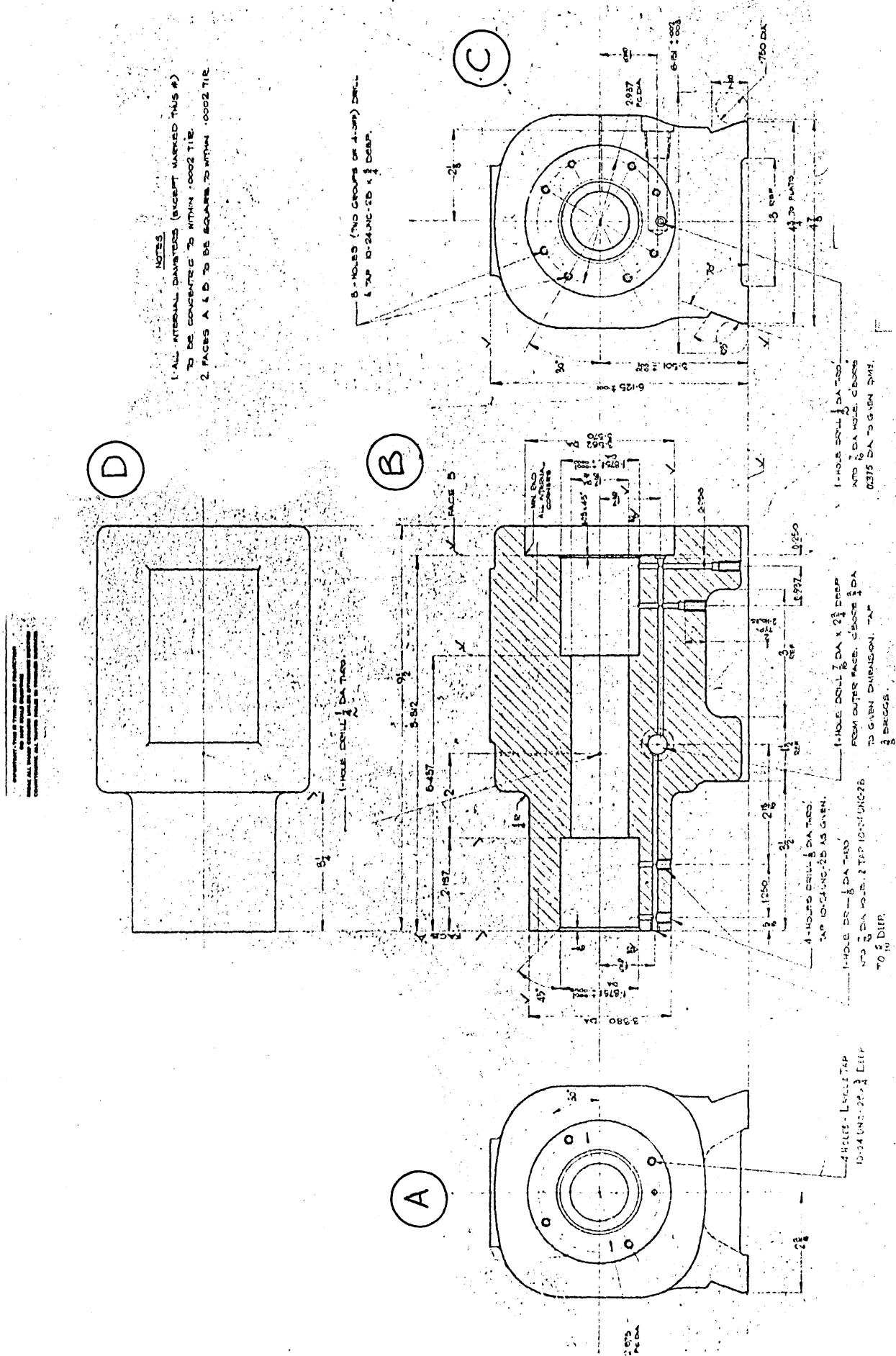


Fig. 14 -- Head casting (Body).

Part no - HD18-1J-9D
Material- Cast Iron.

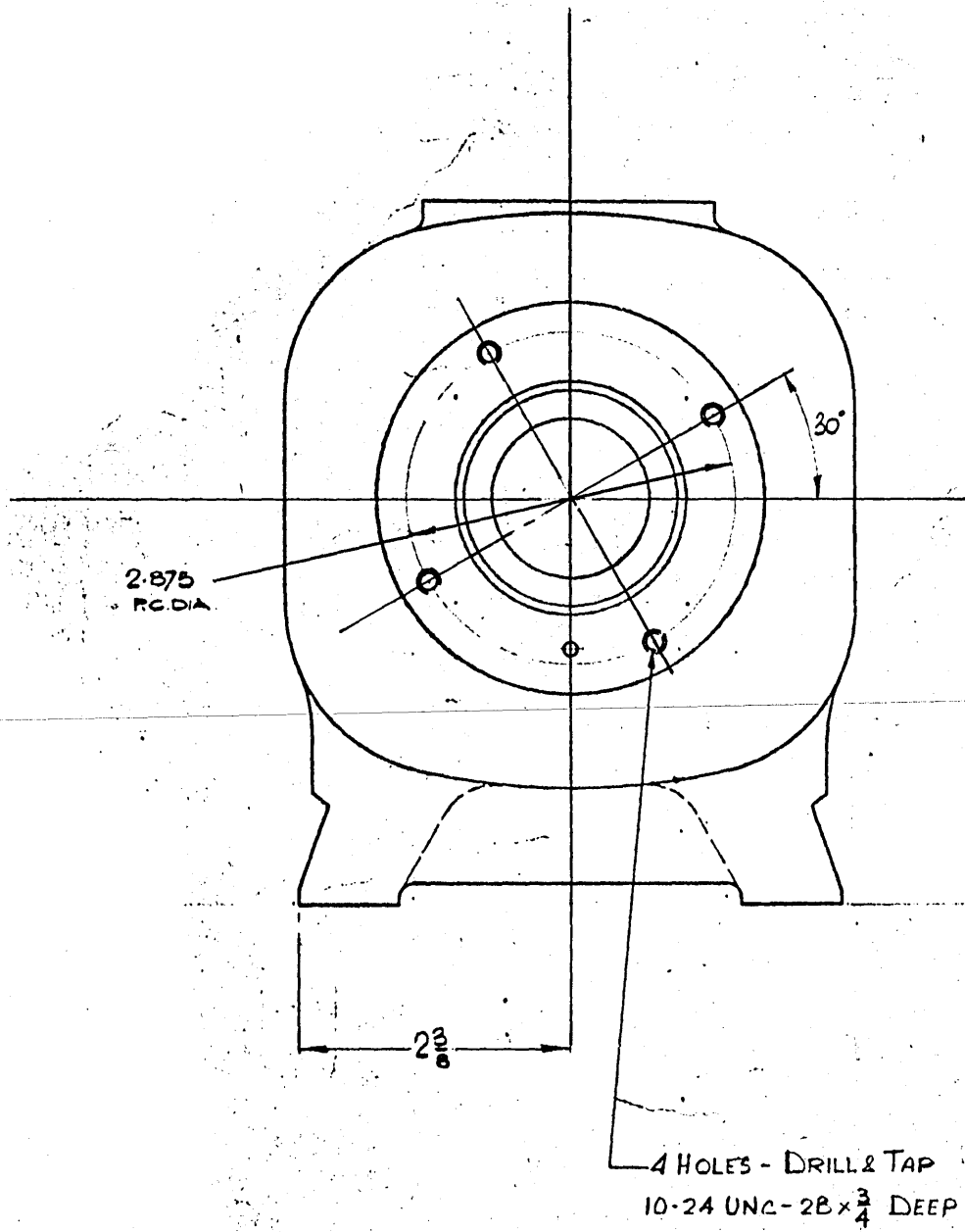
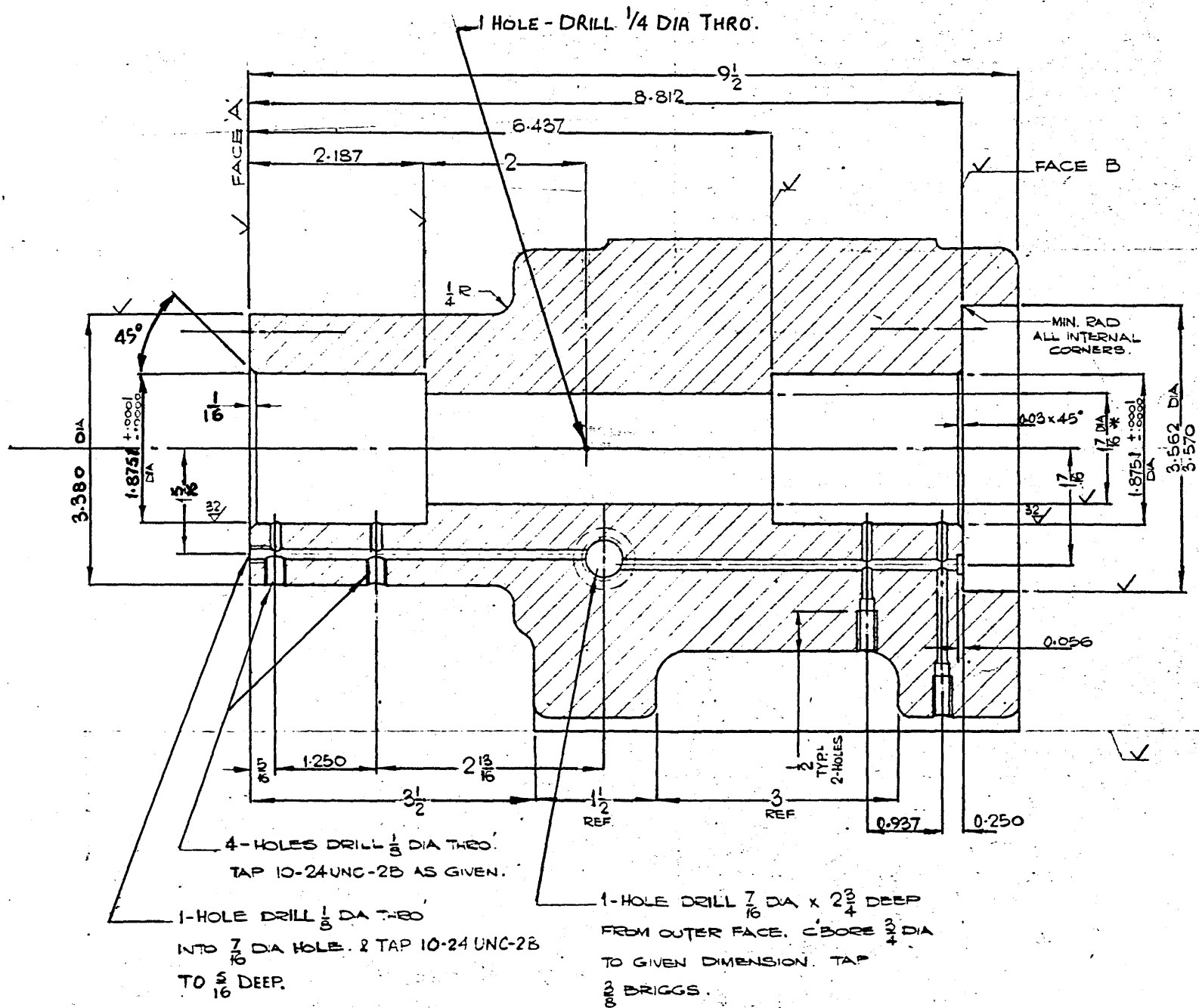
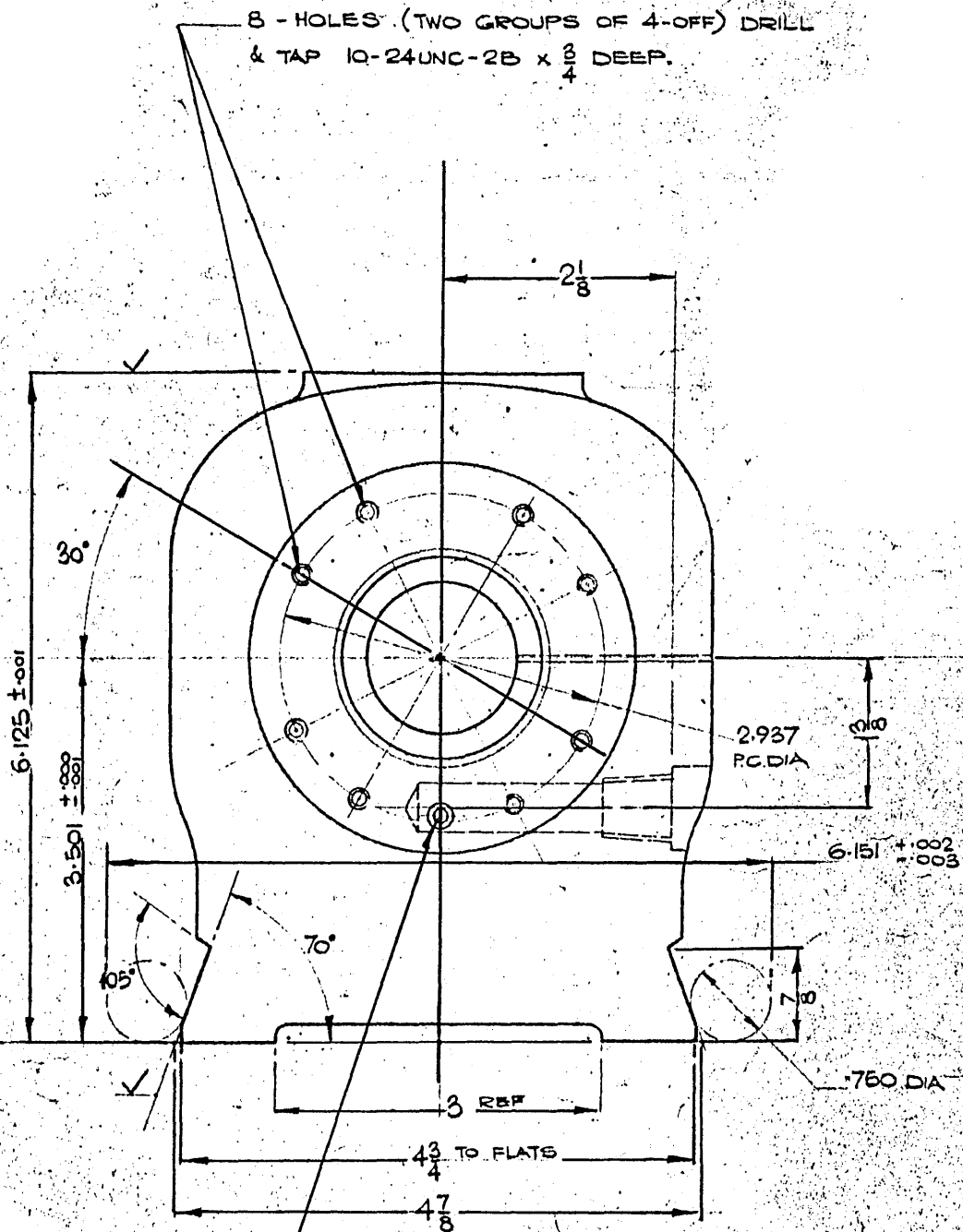


Fig. 14A -- View 'A' of head casting.

Fig. 14B—View 'B' of head casting.





1 HOLE DRILL 1/4 DIA THRO
INTO 7/16 DIA HOLE. C' BORE
0.375 DIA TO GIVEN DIMNS.

Fig. 14C -- View 'C' of head casting.

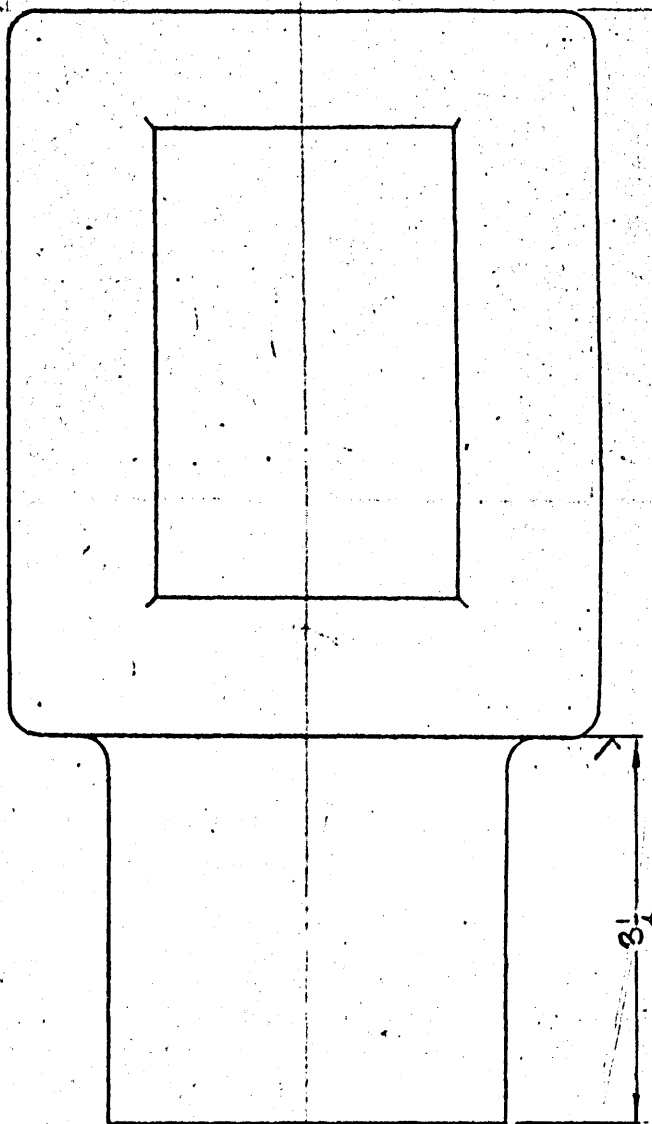
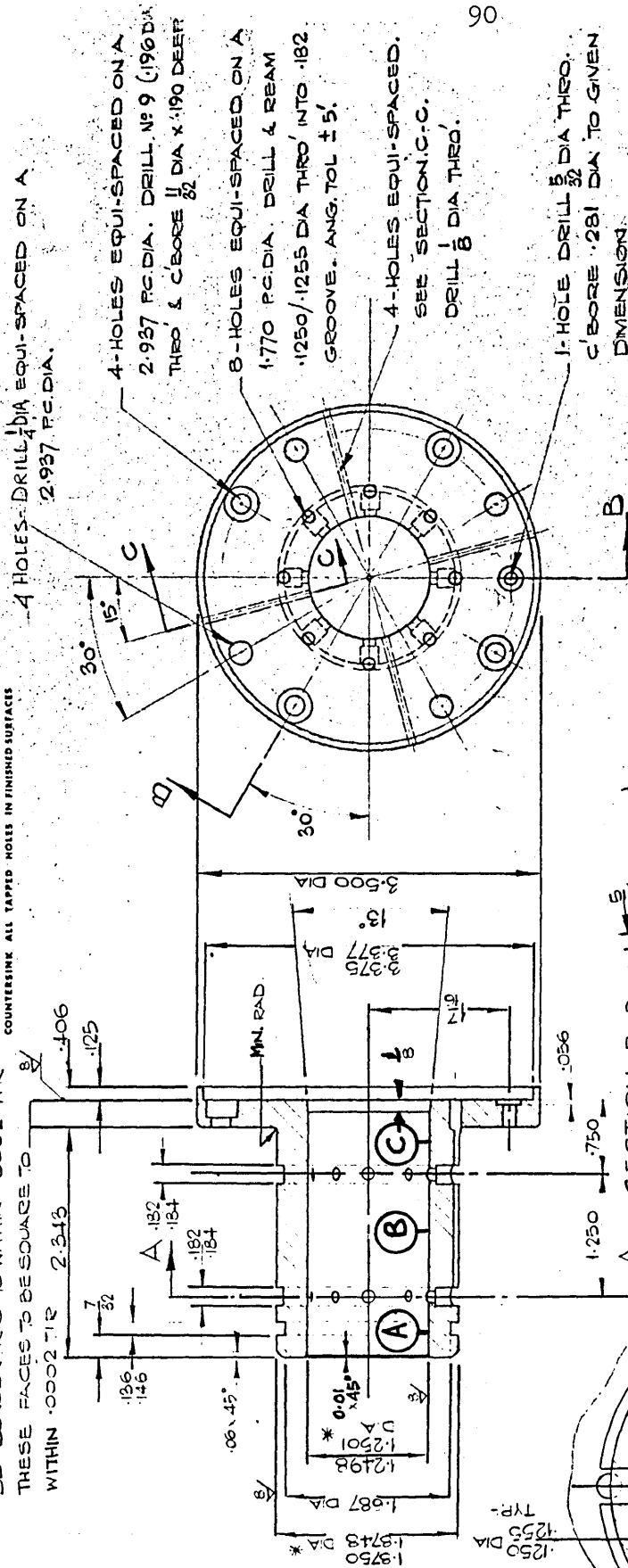


Fig. 14D -- View 'D' of head casting.

DIAMETERS MARKED THUS * TO
BE CONCENTRIC TO WITHIN .0002 TIR
THESE FACES TO BE SQUARE TO
WITHIN .0002 TIR 2.313



(A), (B), (C), REFERS TO FIG. 35.

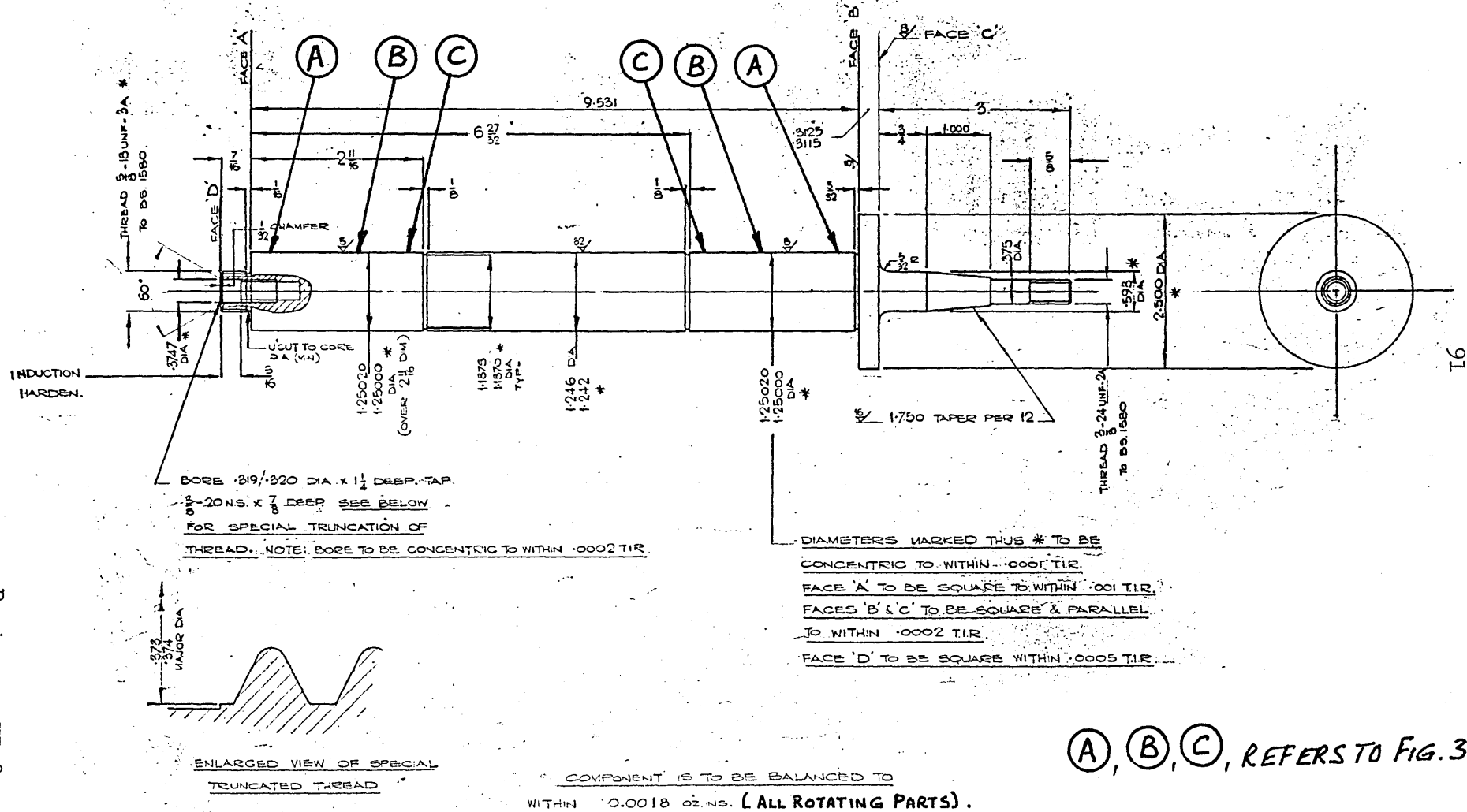
SECTION C-C
SCALE: 2/1

SECTION A-A **SCALE: 2:1**

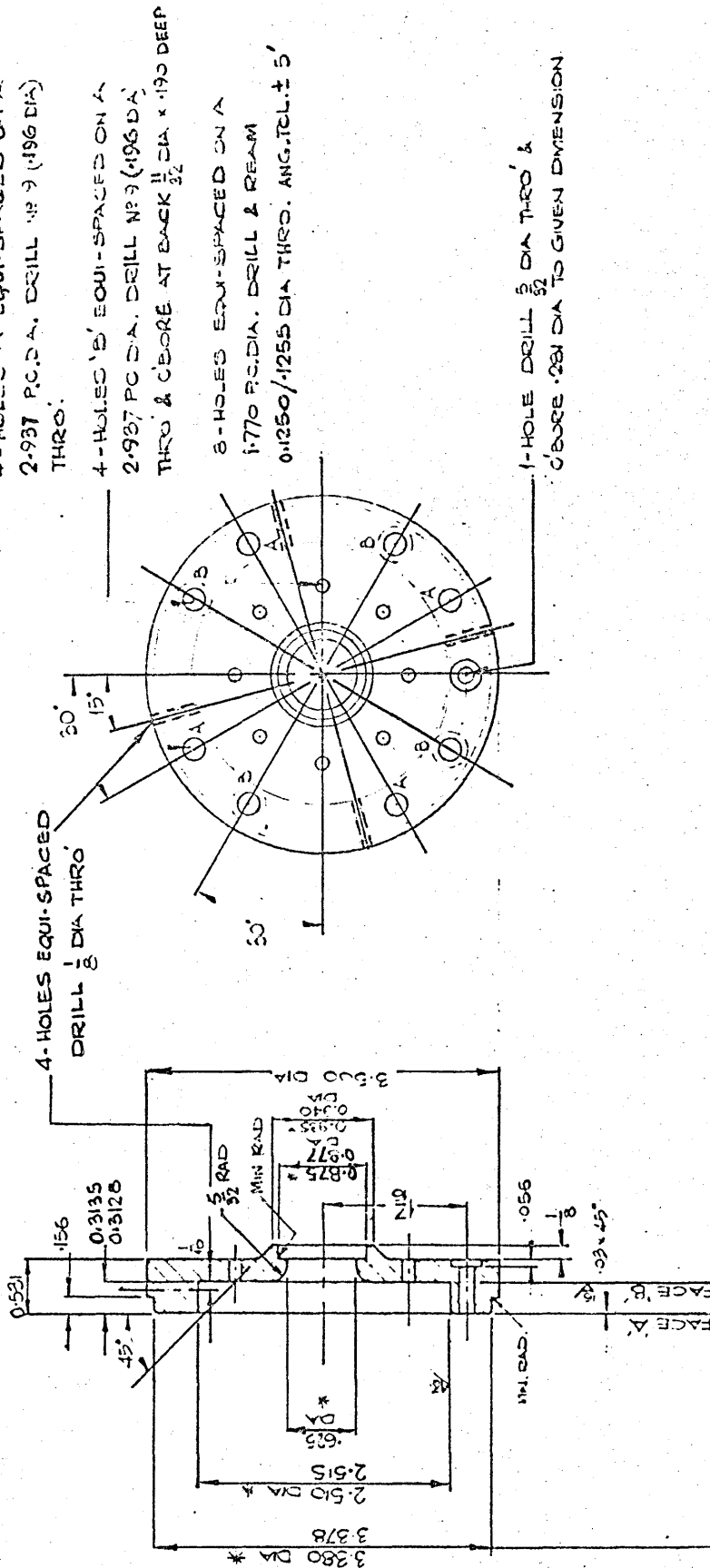
Part no.- HD18-1F-6B
Material- Phos.Bronze.

Fig. 16 -- Rear journal bearing
and front thrust bearing.

Fig. 17 -- Spindle.



Part no.- HD18-1E-5C
Material- Steel BS970
EN 19C.



DIAMETERS MARKED THUS * TO BE
CONCENTRIC TO WITHIN .001 IN
FACES 'A' & 'B' TO BE SQUARE &
PARALLEL TO WITHIN .001 IN 10

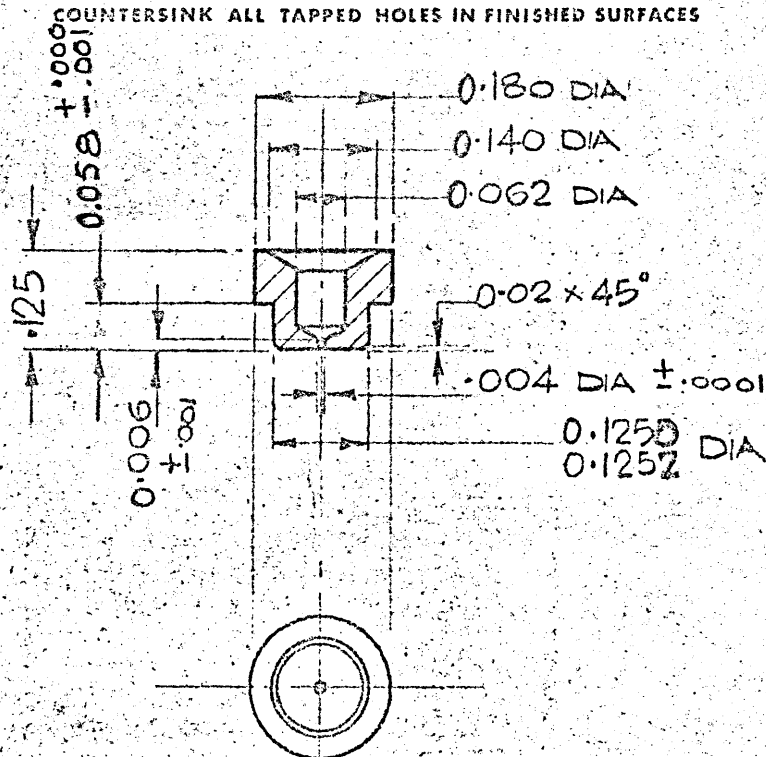
Fig. 19 -- Rear thrust bearing.

IMPORTANT - THIS IS THIRD ANGLE PROJECTION

DO NOT SCALE DRAWINGS

BREAK ALL SHARP CORNERS UNLESS OTHERWISE SPECIFIED

UNDERSINK ALL TAPPED HOLES IN FINISHED SURFACES

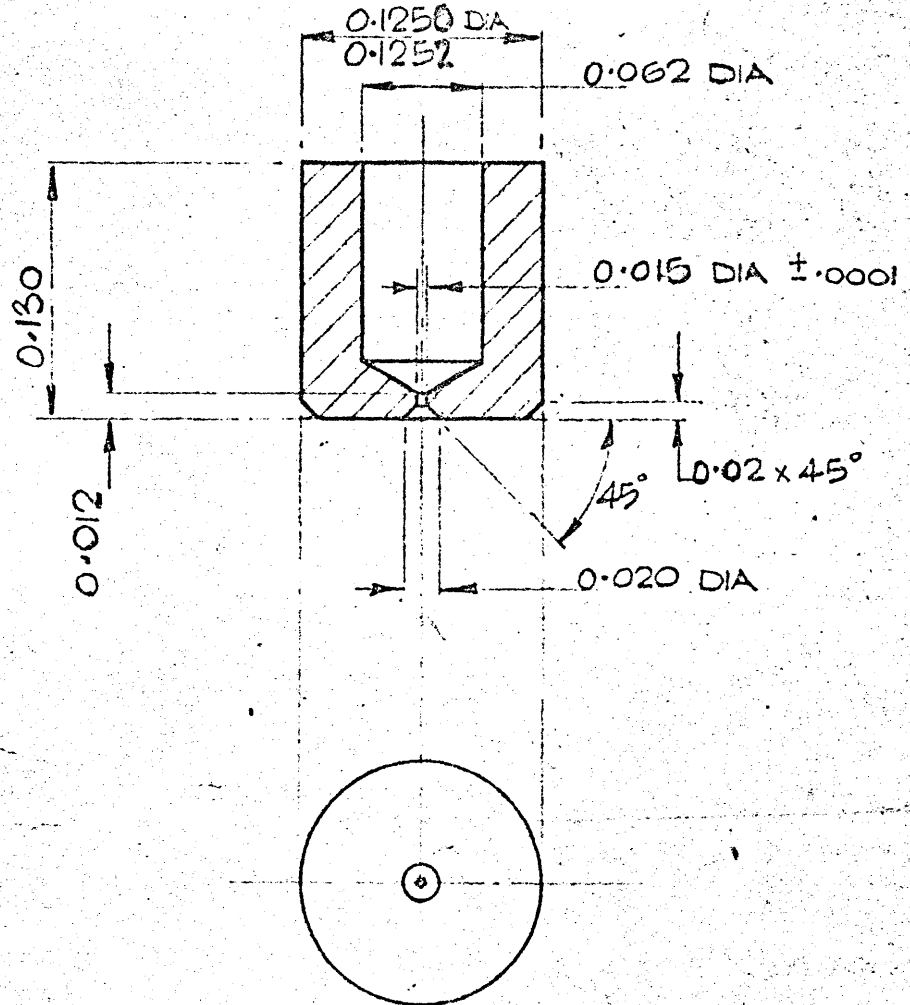


ACTUAL SIZE

Part no.- HD18-1K-10A
Material- Phos.Bronze.

Fig. 20 -- Orifice (Journal Bearing).

IMPORTANT - THIS IS THIRD ANGLE PROJECTION
DO NOT SCALE DRAWINGS
BREAK ALL SHARP CORNERS UNLESS OTHERWISE SPECIFIED
COUNTERSINK ALL TAPPED HOLES IN FINISHED SURFACES



ACTUAL SIZE.

Part no.- HD18-1L-11A
Material- Phos.Bronze.

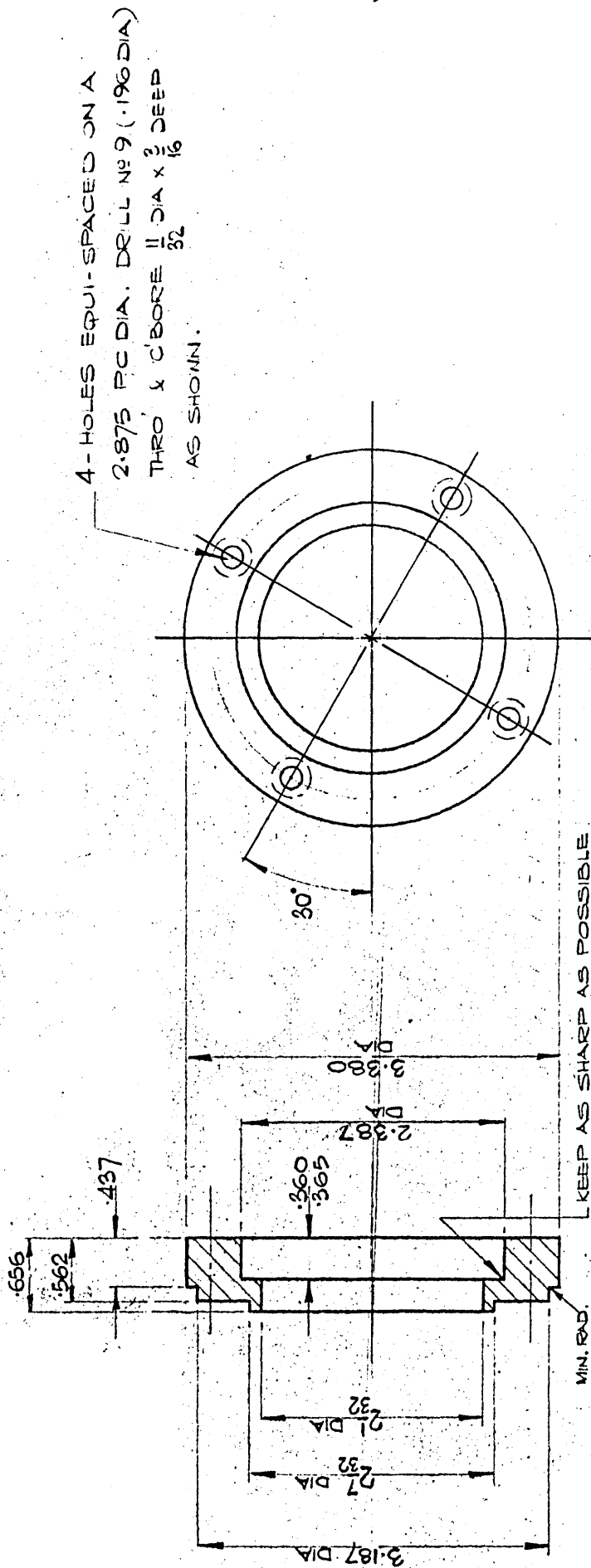
Fig. 21 -- Orifice (Thrust Bearing).

IMPORTANT - THIS IS THIRD ANGLE PROJECTION

DO NOT SCALE DRAWINGS

BREAK ALL SHARP CORNERS UNLESS OTHERWISE SPECIFIED

COUNTERSINK ALL TAPPED HOLES IN FINISHED SURFACES



Part no.-- HD18-1C-3B
Material- Mild Steel
BS970 EN1A.

Fig. 22 -- Front cover.

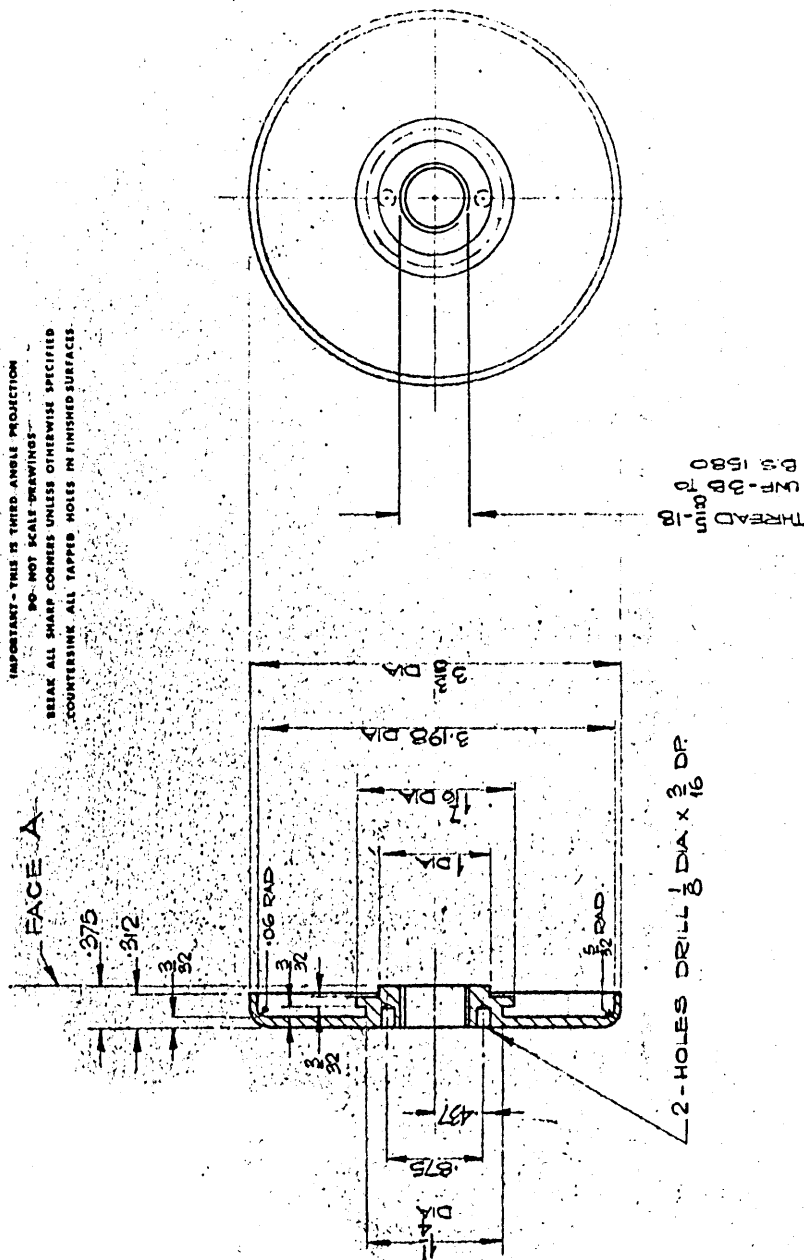
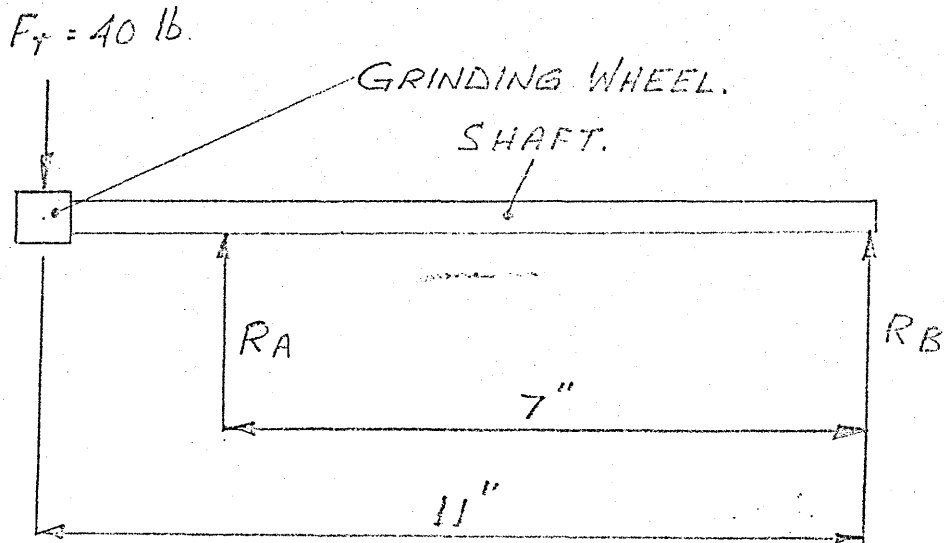


Fig. 23 -- Slinger.



TAKING MOMENT ABOUT R_B .

$$R_A \cong 62.5 \text{ lb. } \uparrow$$

$$R_B \cong 22.5 \text{ lb. } \downarrow$$

R_A & R_B LOAD ACTING ON BEARING CENTRES.

APPROXIMATE VALUE OF DISTANCE TAKEN.

Fig. 24 -- Load acting on journal bearings.

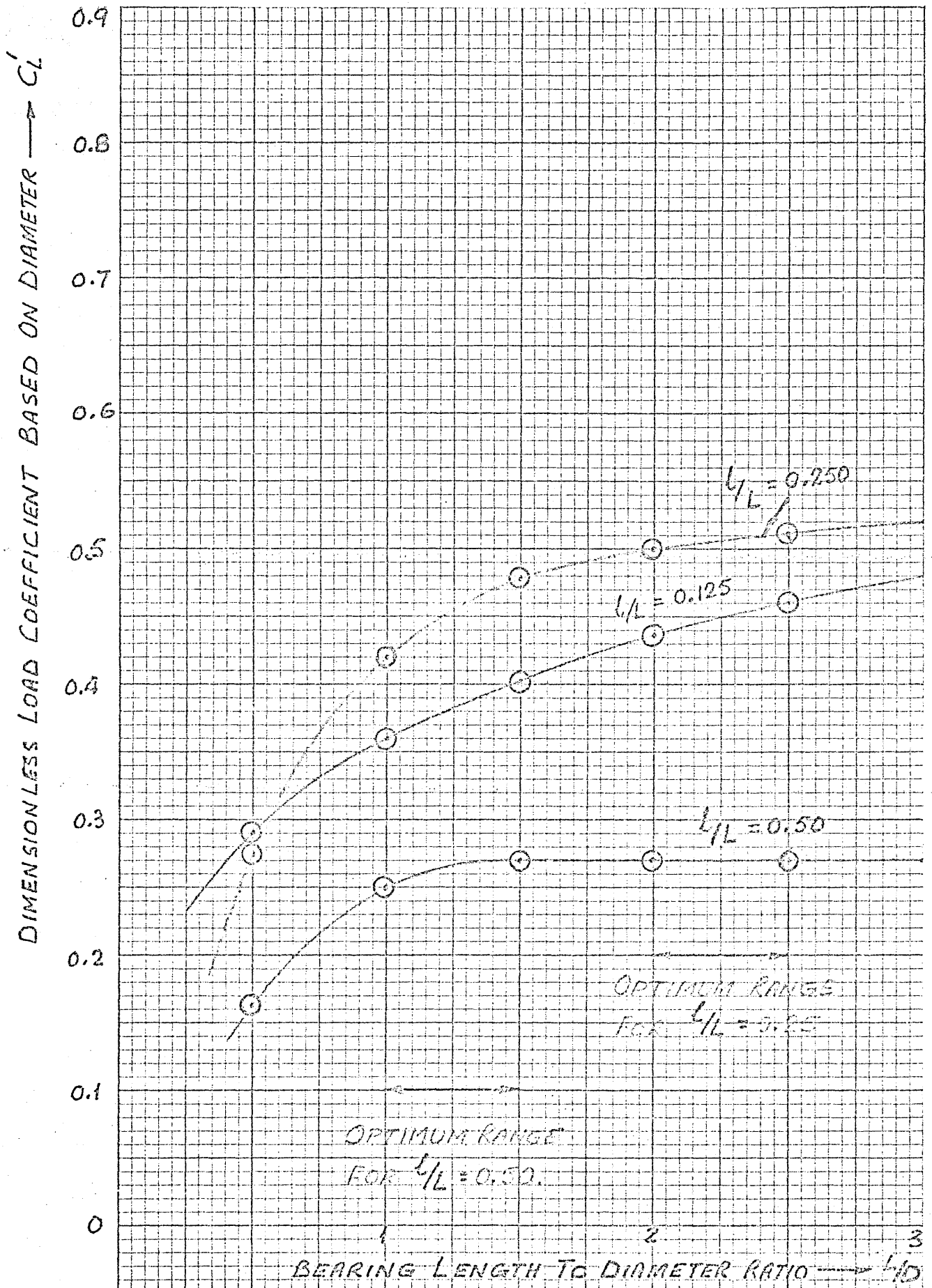


Fig. 25 -- Influence of length to diameter ratio on journal bearing load capacity.

Gas	Temperature °C	F_g ft. x 10^6	Recessed jets	Plain jets
Air	15	5.66	1.00	1.00
Air	300	9.20	0.66	0.44
CO ₂	15	3.44	1.28	1.64
H ₂	15	10.30	0.74	0.55
He	15	16.90	0.59	0.35
H ₂ O	100	5.85	0.98	0.97

Fig. 26 -- Influence of gas properties on orifice diameters, for ambient pressure.
(After Shires, (15)).

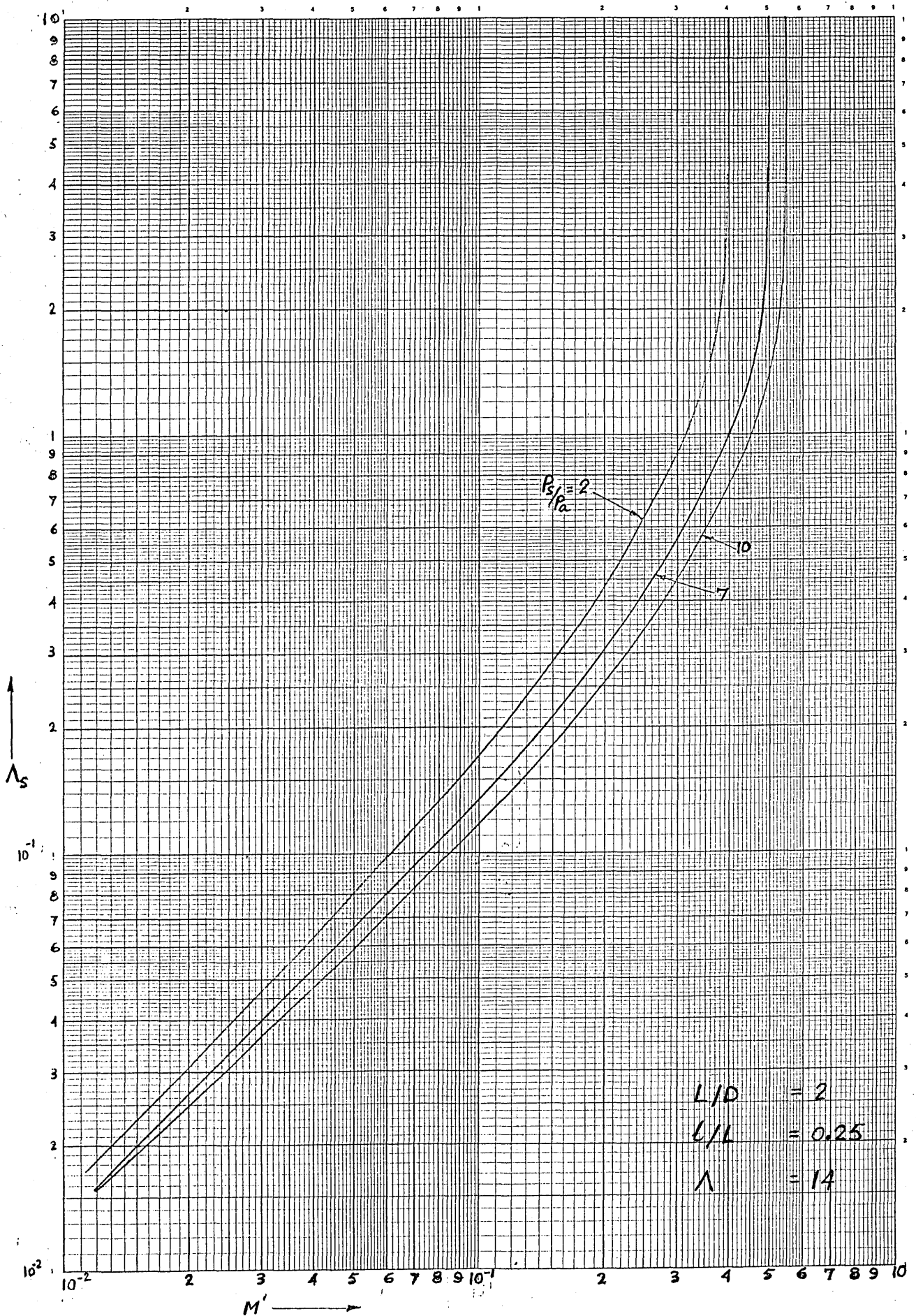


Fig. 27 -- Hydrostatic journal bearing, restrictor coefficient, Λ_s , versus flow factor, M' .

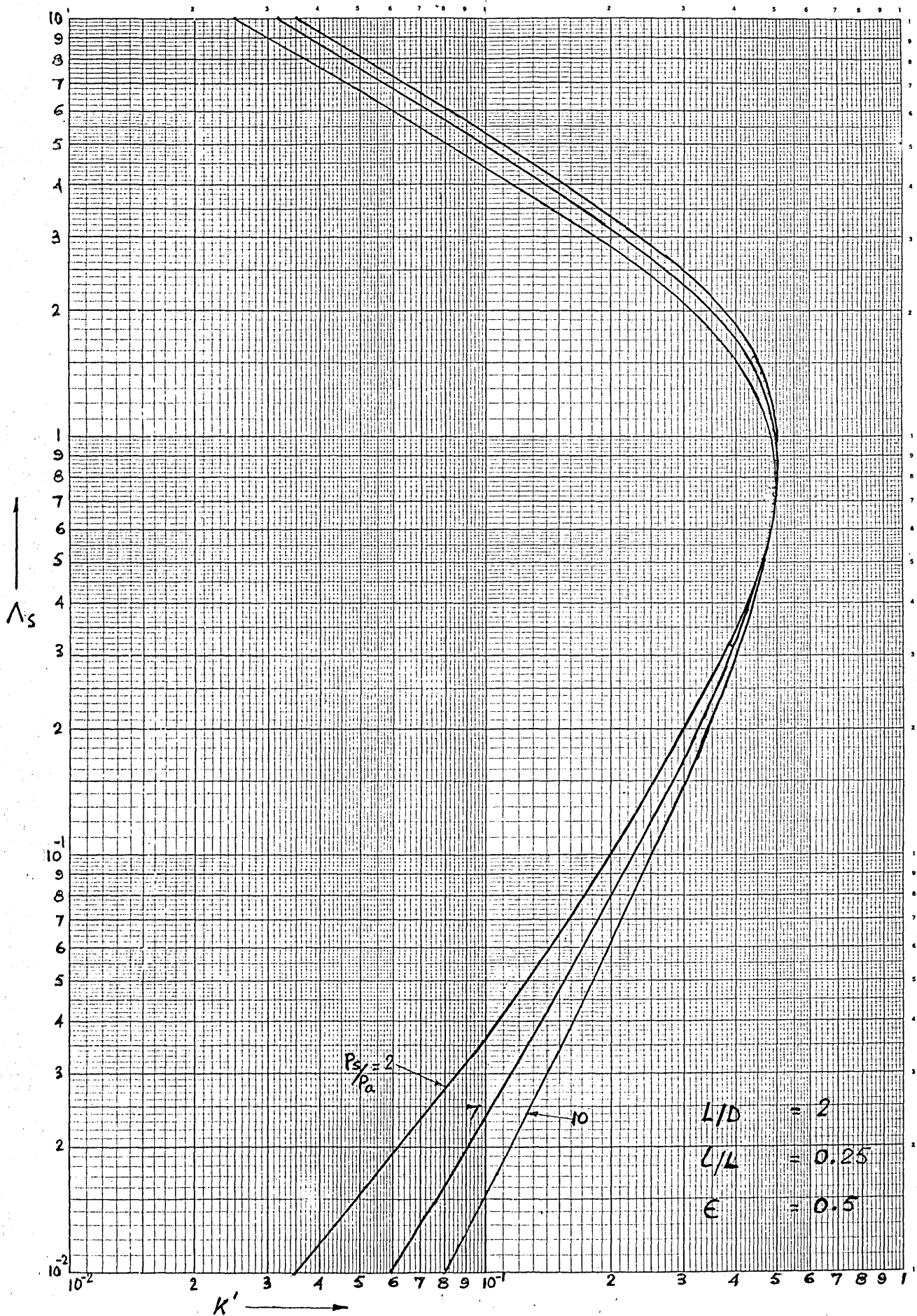


Fig. 28 — Hydrostatic journal bearing restrictor coefficient, Λ_s , versus load or stiffness parameter, K' .

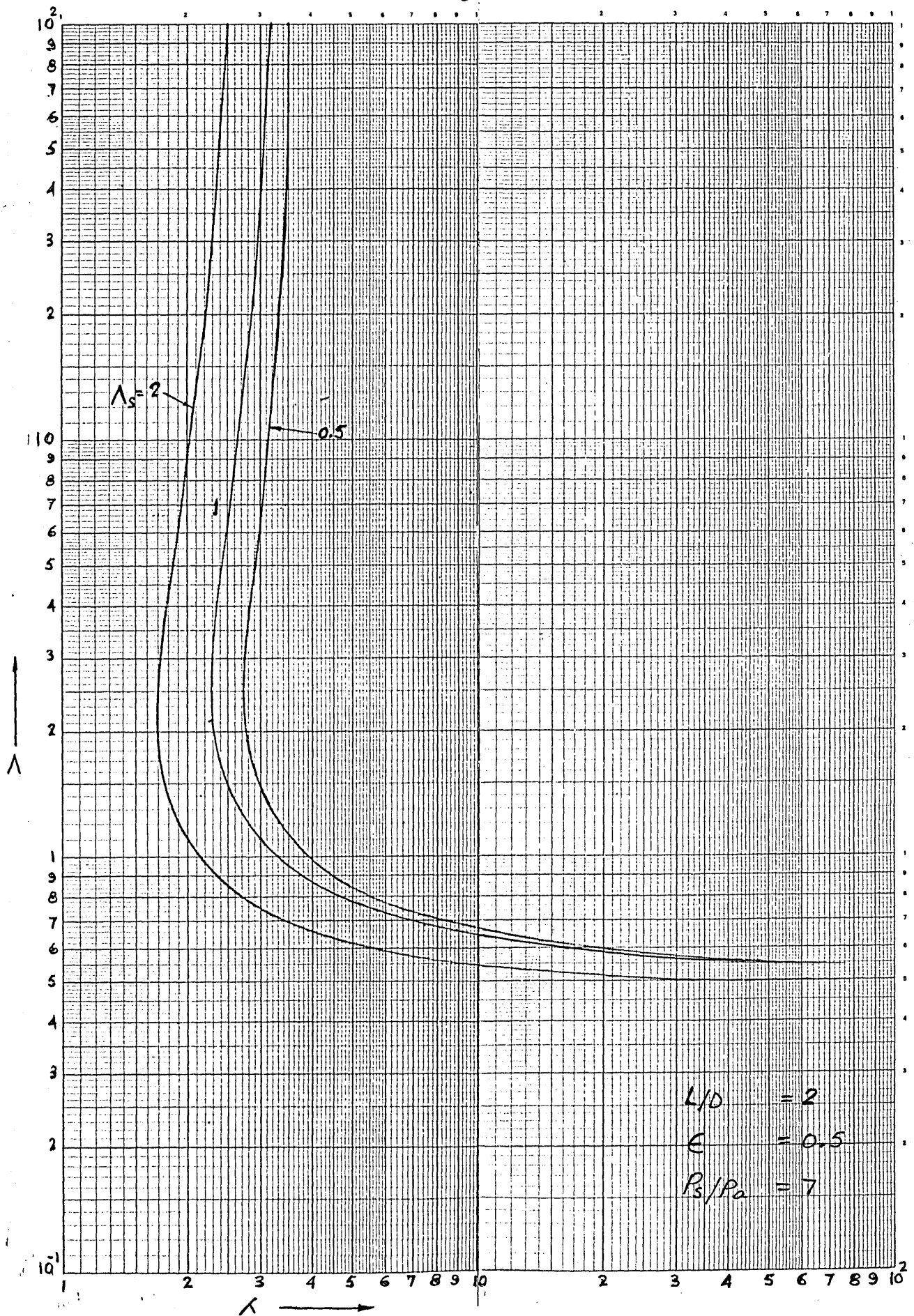
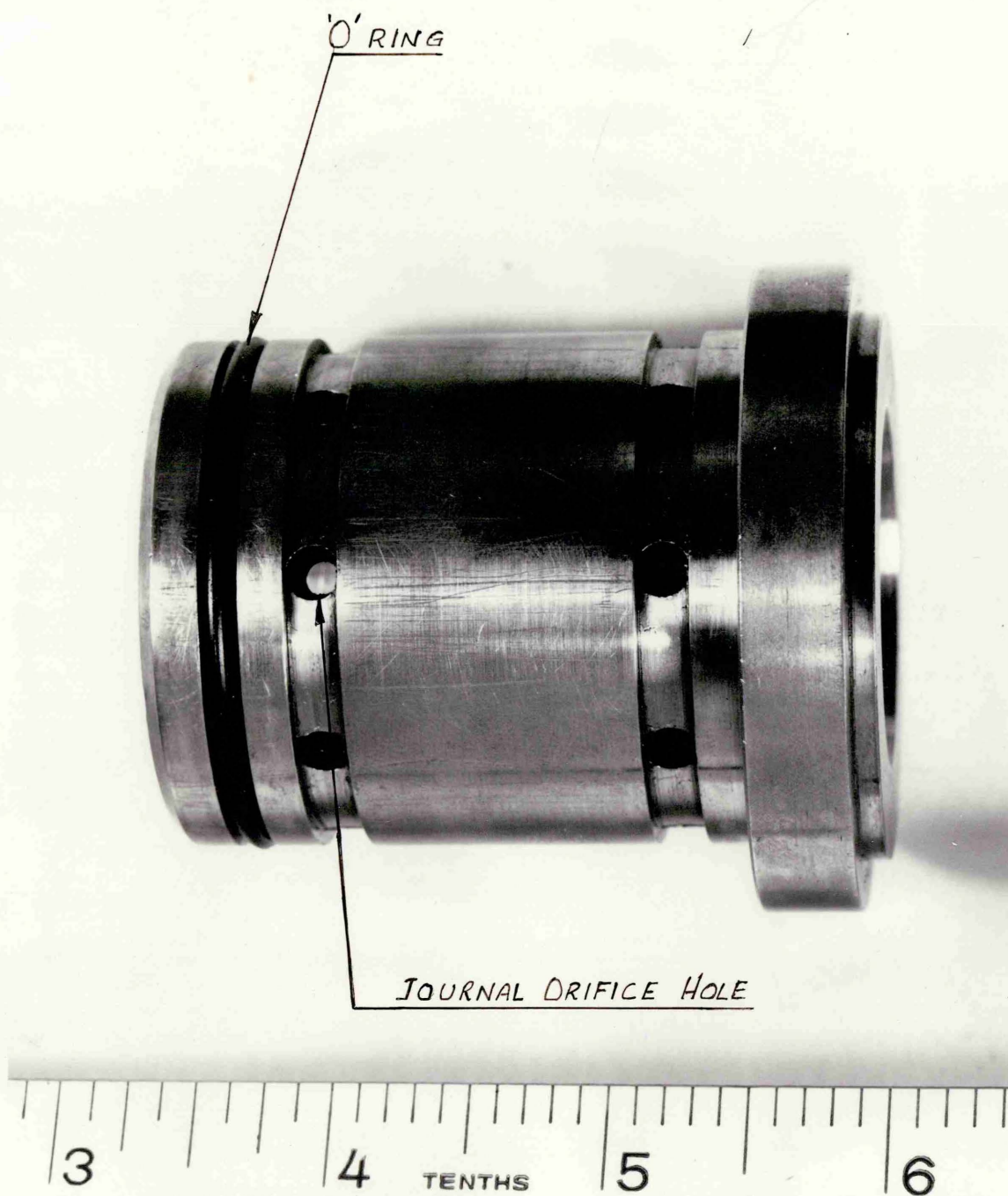


Fig. 29 — Hydrostatic journal bearing, compressibility number, Λ , versus instability threshold parameter, λ .



★ A W FABER ★

Fig. 30 — Front journal bearing.

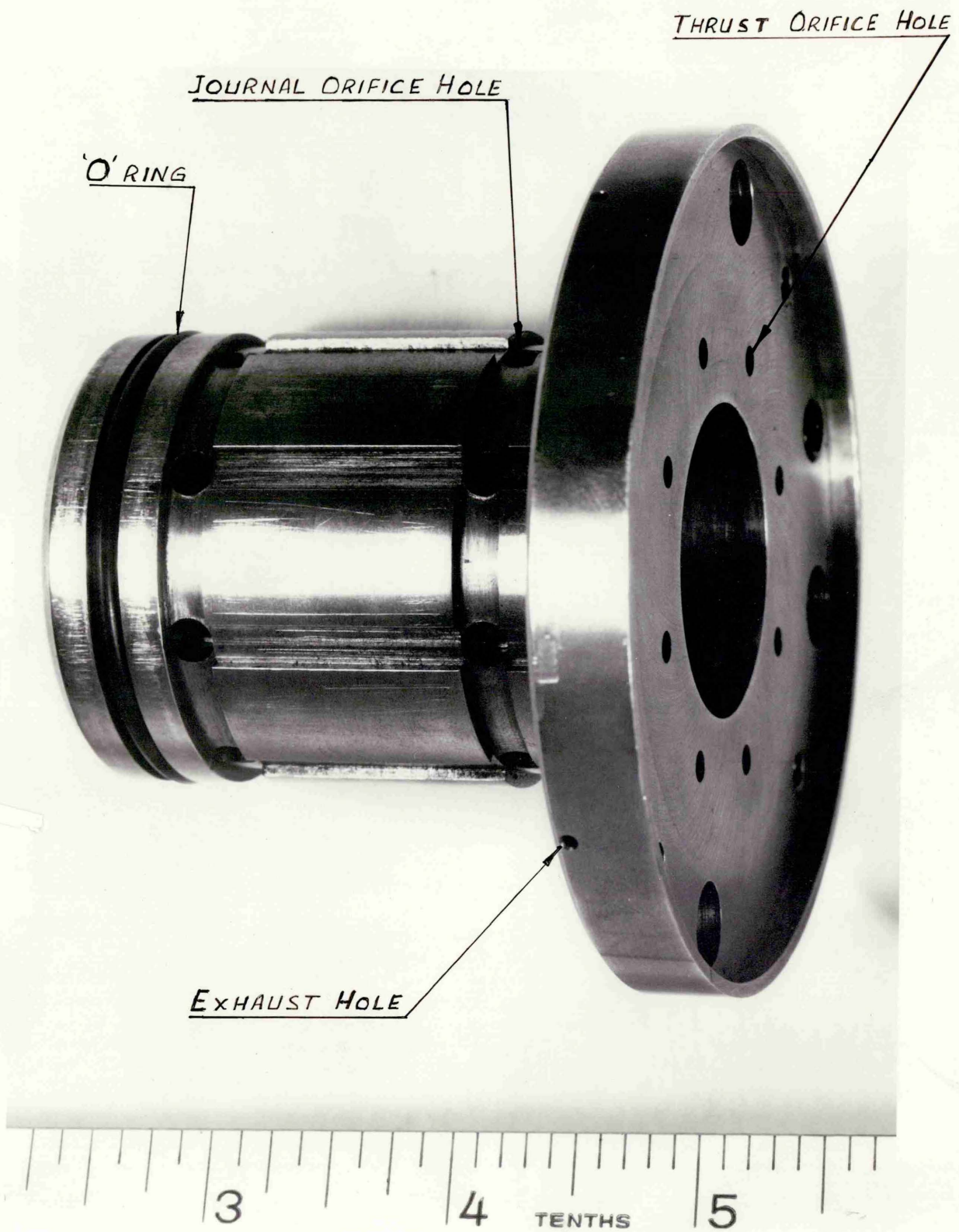
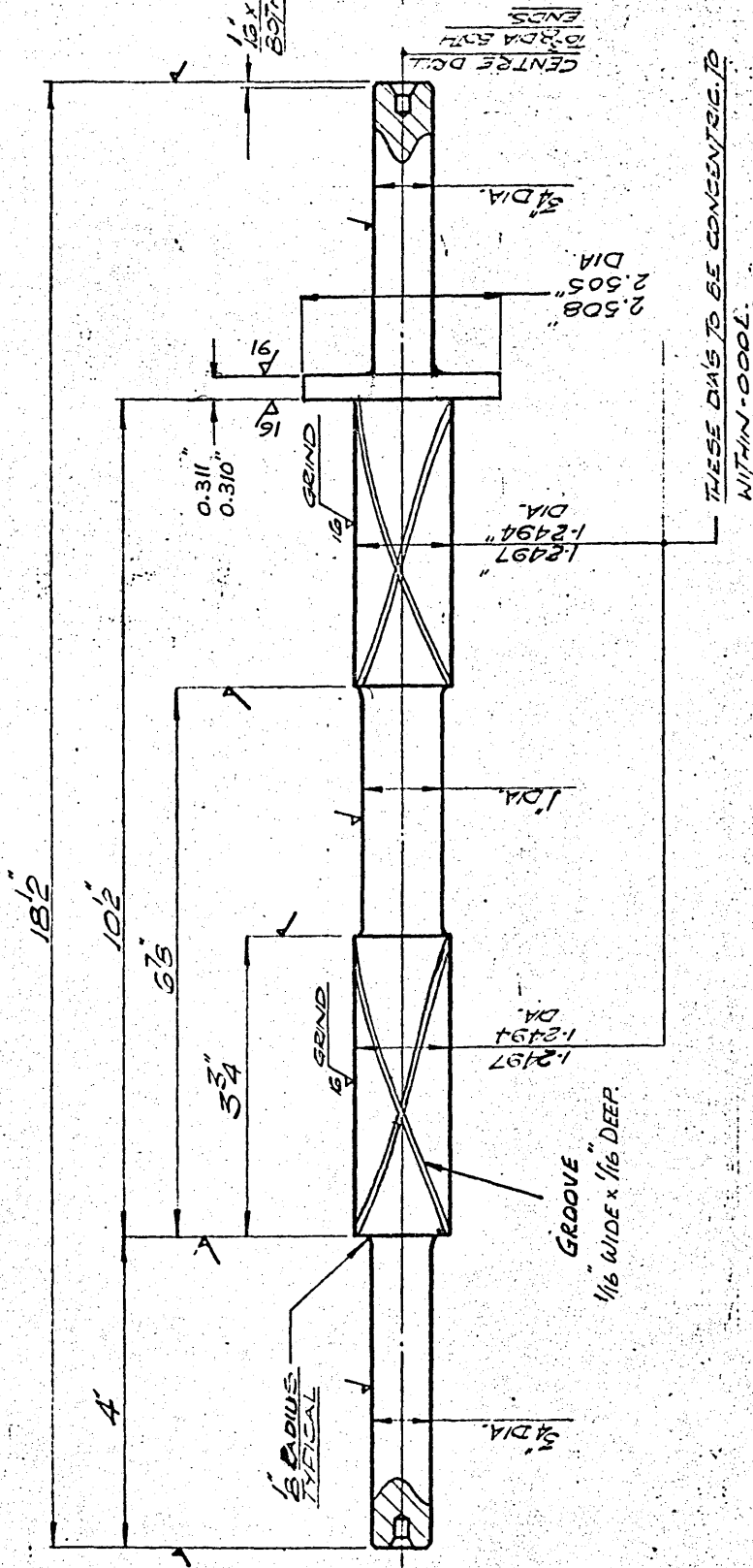


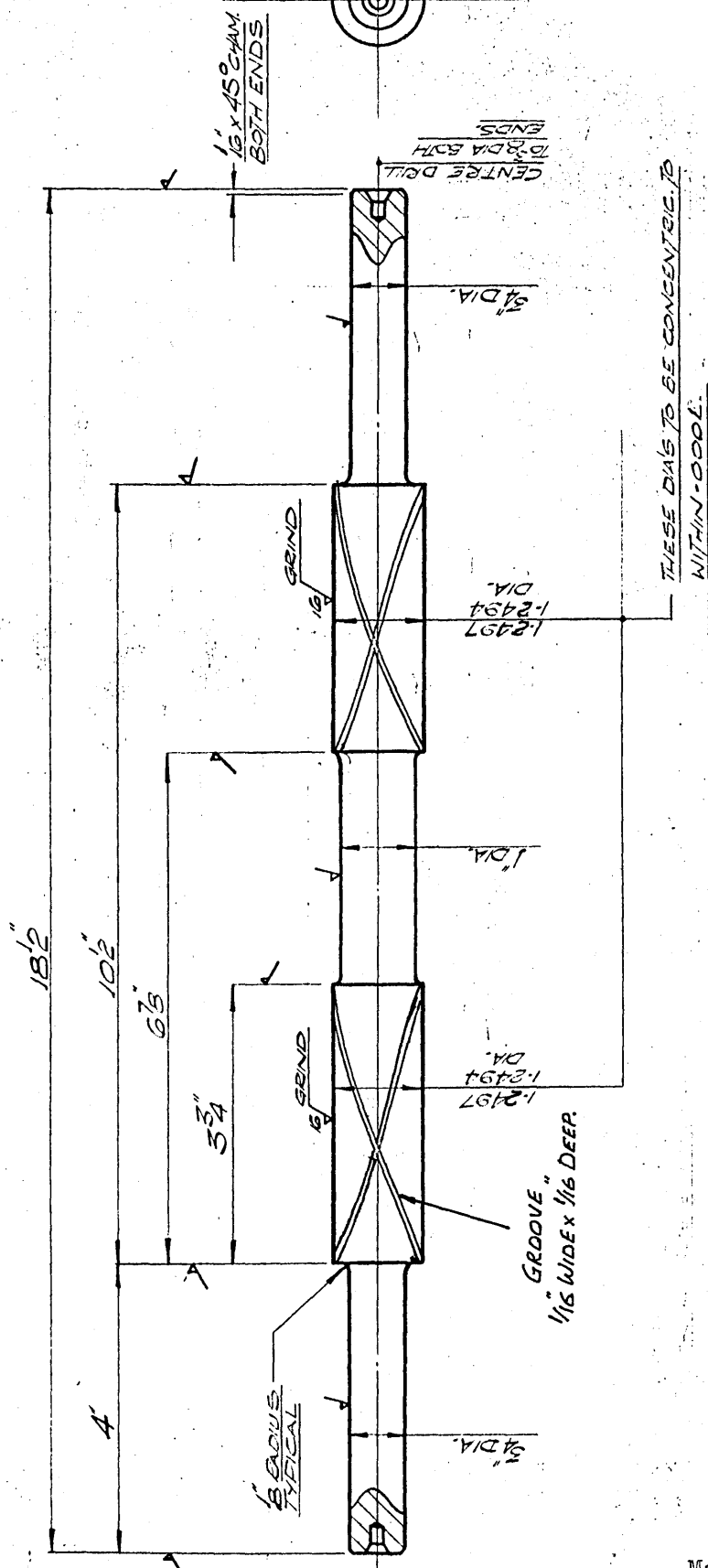
Fig. 31 -- Rear journal bearing
& front thrust bearing.

IMPORTANT: THIS IS A PRELIMINARY DRAWING.
 IT IS NOT TO BE USED FOR FABRICATION.
 MAKE ALL DIMENSIONS AS SHOWN UNLESS OTHERWISE SPECIFIED.
 CONDITIONS: ALL TAPED HOLES IN FINISHED SURFACES.



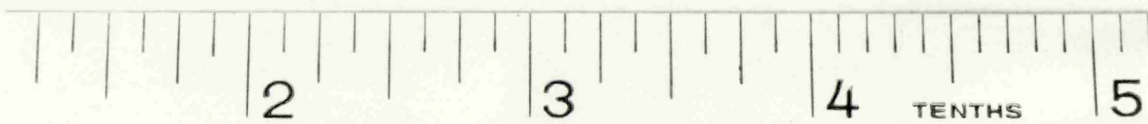
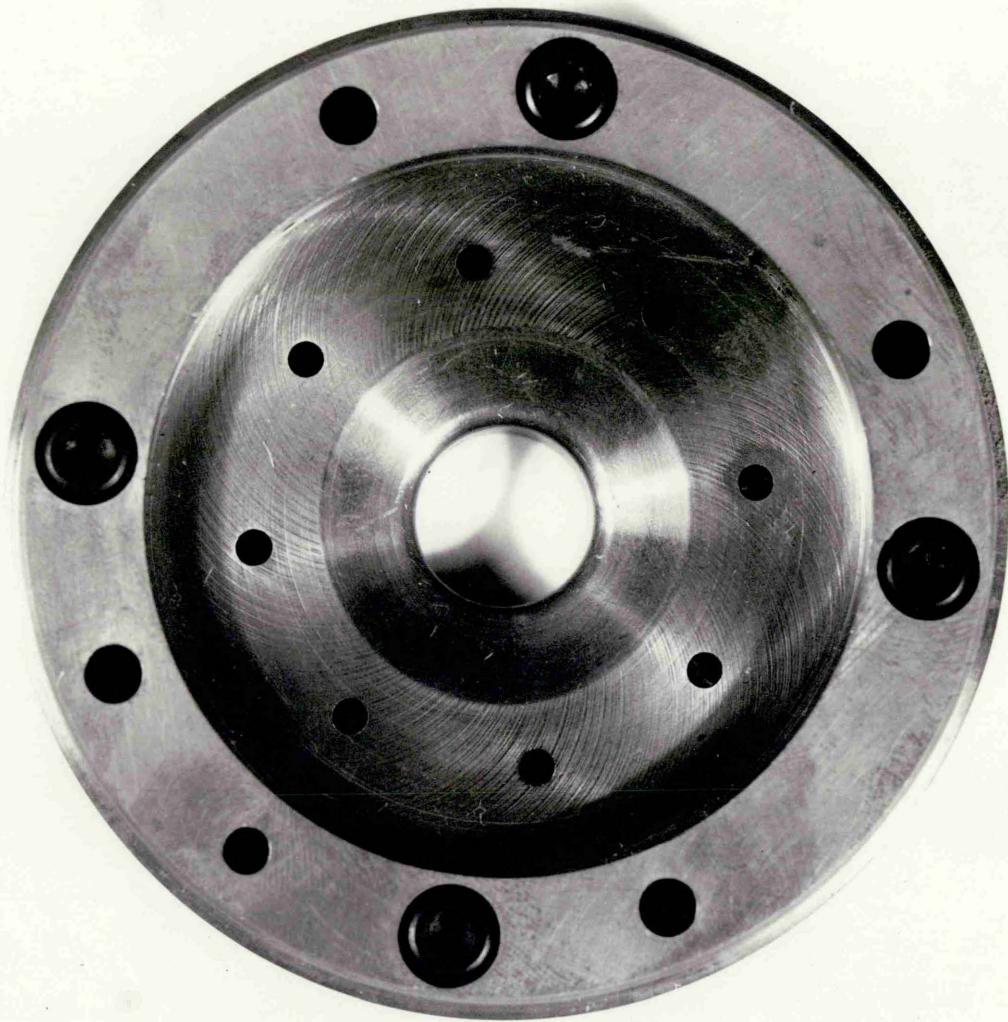
Material - Meehnite.

Fig. 32 -- Lap with thrust pad.



Material - Meehanite.

Fig. 33 -- Lap without thrust pad.



55 A

Fig. 34 -- Rear thrust bearing
before lapping.

THE COLLEGE OF AERONAUTICS

Experimental Record

Experiment: Measurements of the inside
diameters of journal bearings.

Name: K. GHOSH

Date: 20/7/69.

Remarks: Readings on three positions A, B & C
and also another three positions 90° to the
previous point. 1) Rear journal bearing. (see fig. 16).

POSITIONS →	A	90° TO A	B	90° TO B	C	90° TO C			
READINGS									
↓	1.25030	1.25030	1.25031	1.25030	1.25030	1.25030			
↓	1.25030	1.25030	1.25031	1.25030	1.25032	1.25031			
in.	1.25030	1.25030	1.25030	1.25031	1.25032	1.25032			
	Roundness - 3.5 μ"								
	Surface finish - 1.5 μ"								
	2) <u>Front journal bearing (see fig. 15).</u>								
	1.25029	1.25030	1.25030	1.25031	1.25031	1.25030			
	1.25029	1.25030	1.25031	1.25030	1.25031	1.25030			
	1.25030	1.25030	1.25031	1.25031	1.25031	1.25030			
	Roundness - 4.0 μ"								
	Surface finish - 2.0 μ"								
	3) <u>Front thrust bearing.</u>								
	Squareness - 20 μ" for 1 1/4" length.								
	Surface finish - 8 μ"								
	4) <u>Rear thrust bearing.</u>								
	Squareness - 25 μ" for 1 1/4" length.								
	Surface finish - 8 μ"								

Fig. 35 -- Measurements of journal bearings.

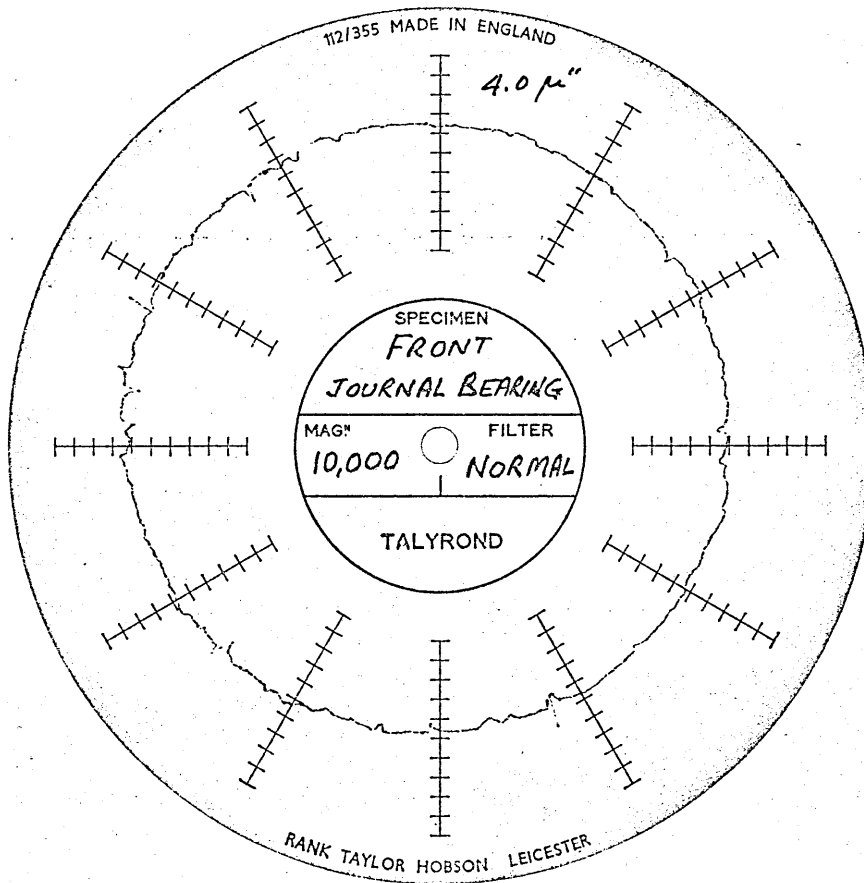
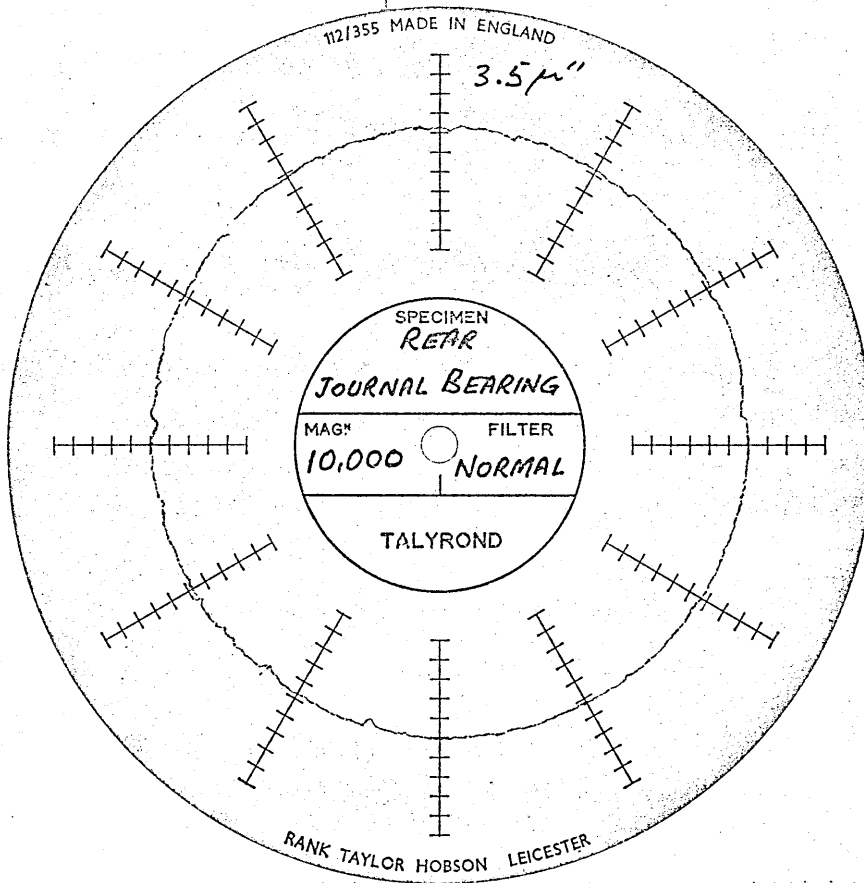


Fig. 36 -- Talyrond traces showing roundness accuracy of journal bearings.

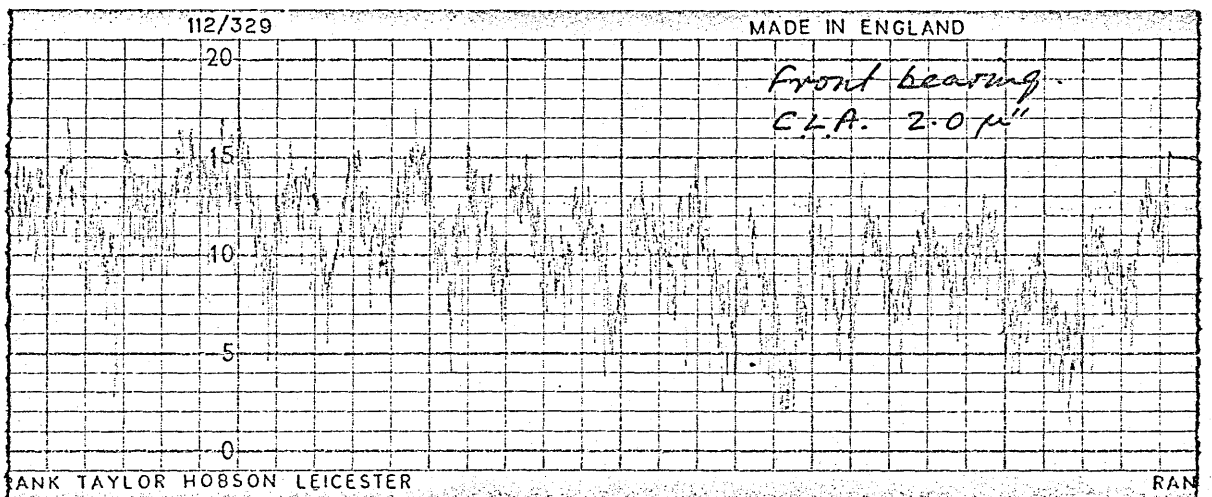
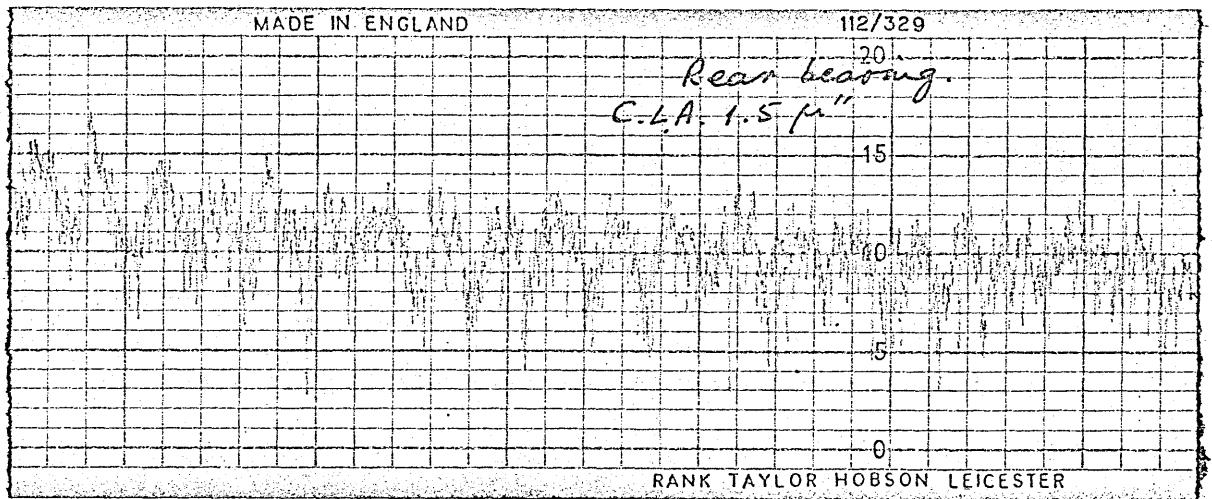


Fig. 37 — Talysurf traces showing surface finish of journal bearings.

THE COLLEGE OF AERONAUTICS

Experimental Record

Experiment: Measurements of the shaft journal diameters.

Name: K. GHOSH

Date: 20/7/69.

Remarks: Reading on three positions A, B & C and also another three positions 90° to the previous point.

1) Rear end.

POSITIONS →	A	90° To A	B	90° To B	C	90° To C
READINGS						
↓	1.24929	1.24930	1.24929	1.24929	1.24931	1.24930
	1.24930	1.24930	1.24930	1.24930	1.24931	1.24931
in	1.24930	1.24930	1.24930	1.24930	2.24931	1.24931

Roundness - $7.5 \mu''$

Surface finish - $3.5 \mu''$

2) Front end

1.24930	1.24931	1.24930	1.24931	1.24929	1.24930
1.24931	1.24930	1.24931	1.24930	1.24929	1.24929
1.24930	1.24930	1.24931	1.24930	1.24930	1.24930

Roundness - $8.0 \mu''$

Surface finish - $4.0 \mu''$

3) Thrust Pad.

Squareness - $20 \mu''$

Roundness

Surface finish - $10 \mu''$

Fig. 38 — Measurements of shaft diameters.

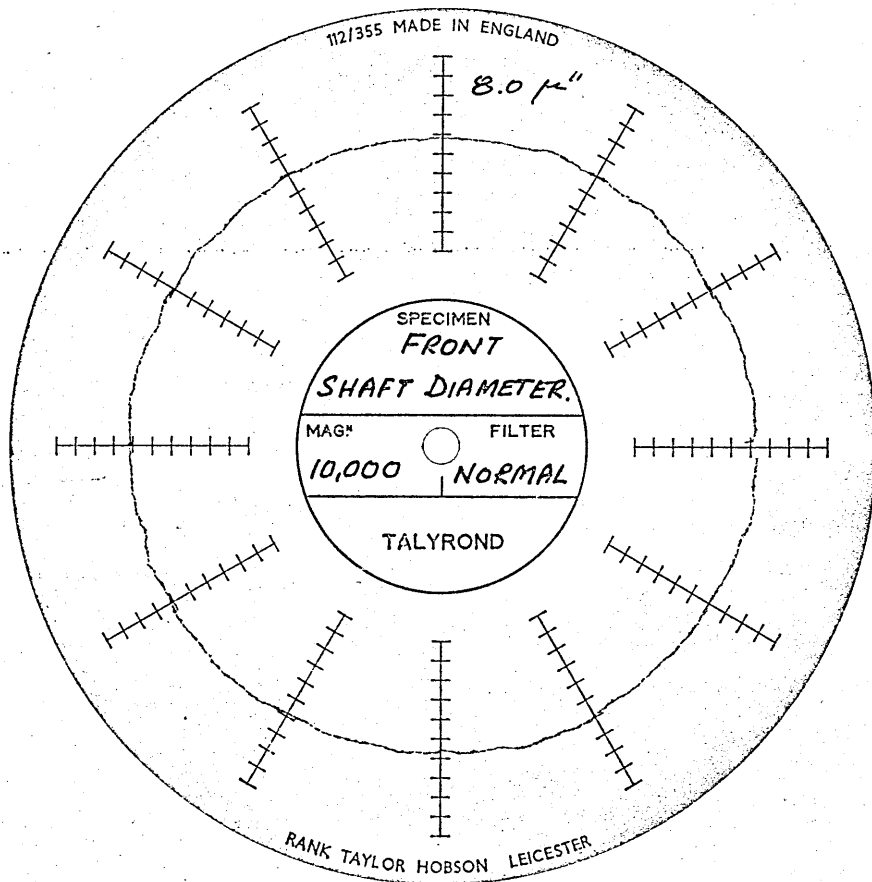
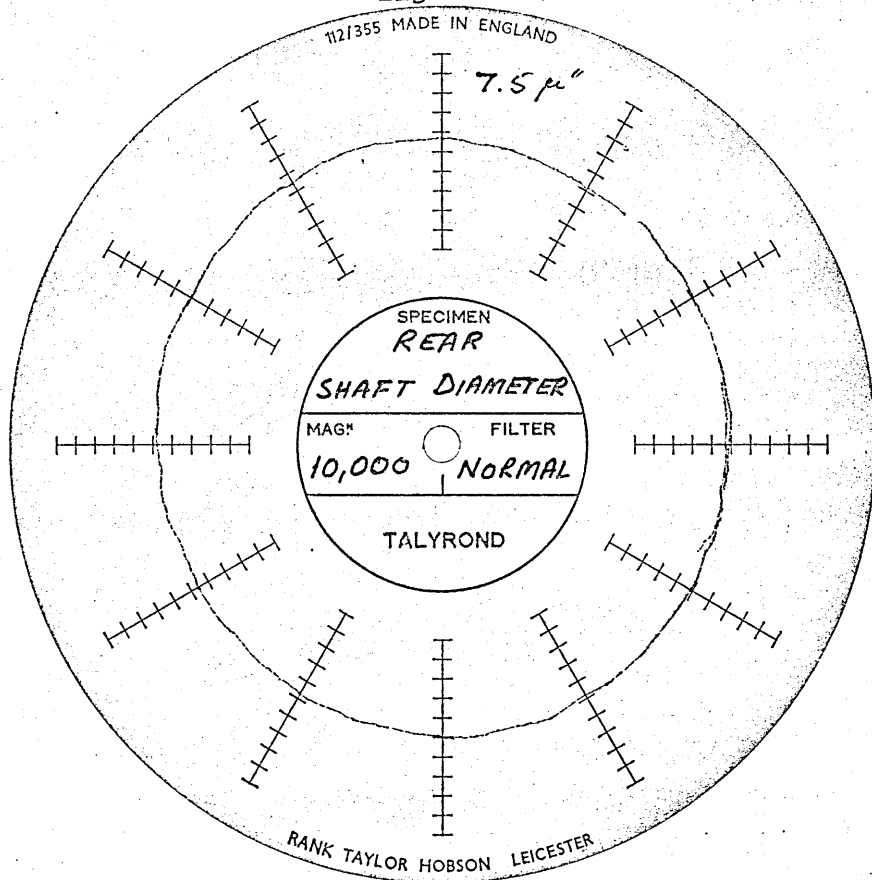


Fig. 39 -- Talyrond traces showing roundness accuracy of shaft journal diameters.

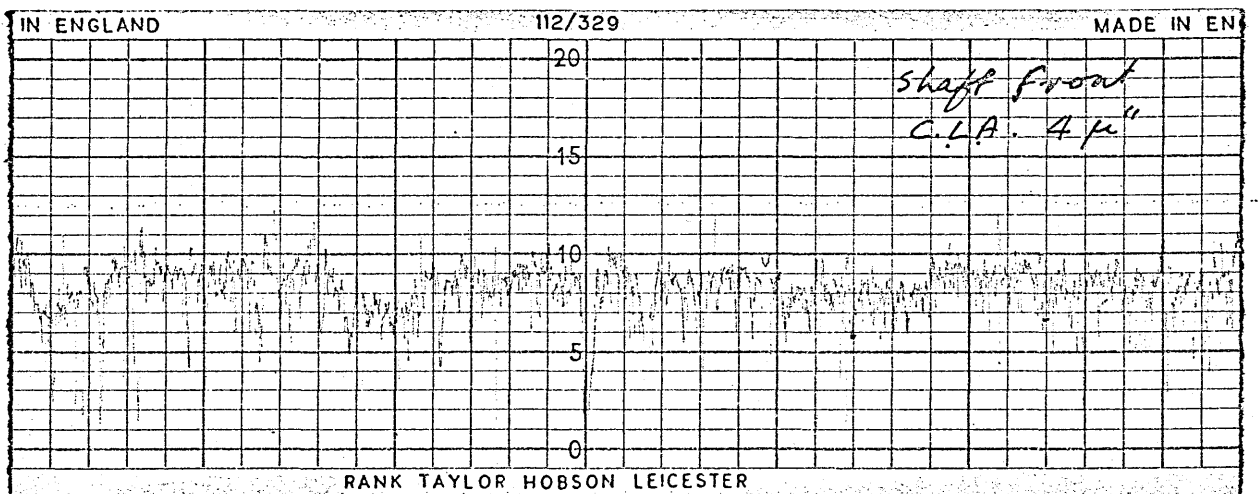
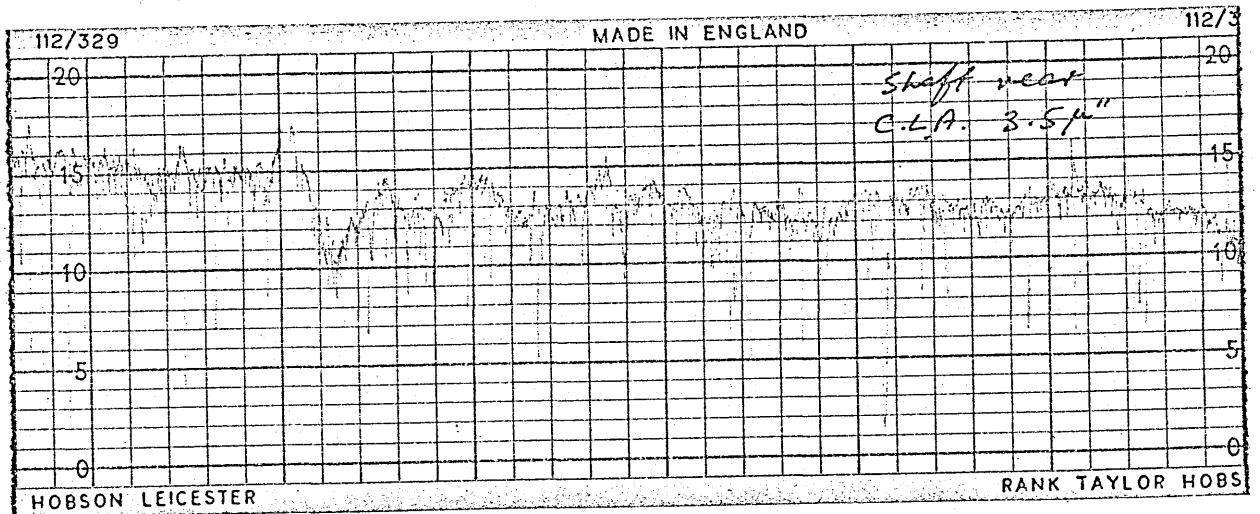


Fig. 40 -- Talysurf traces showing surface finish of shaft journal diameters.

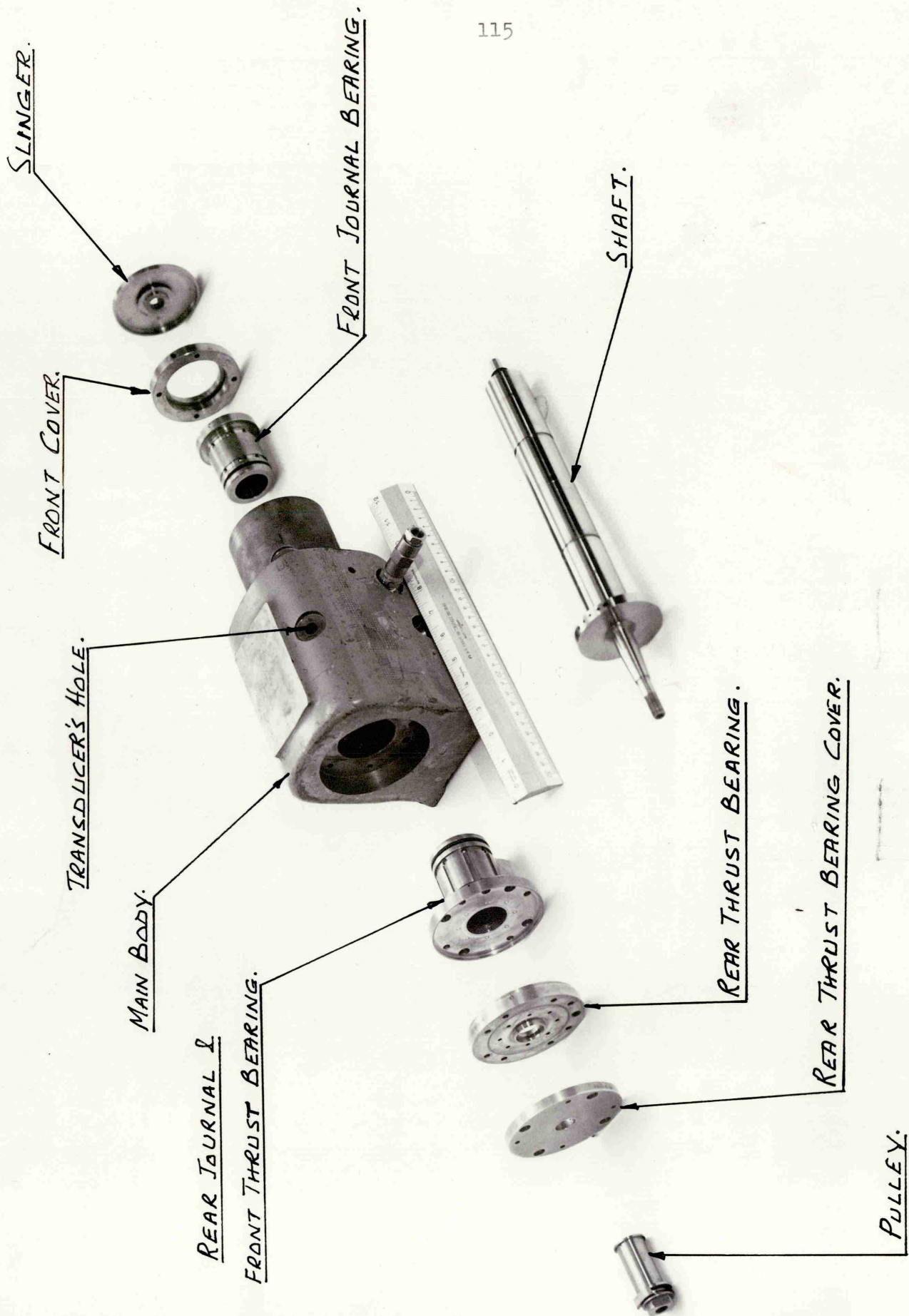
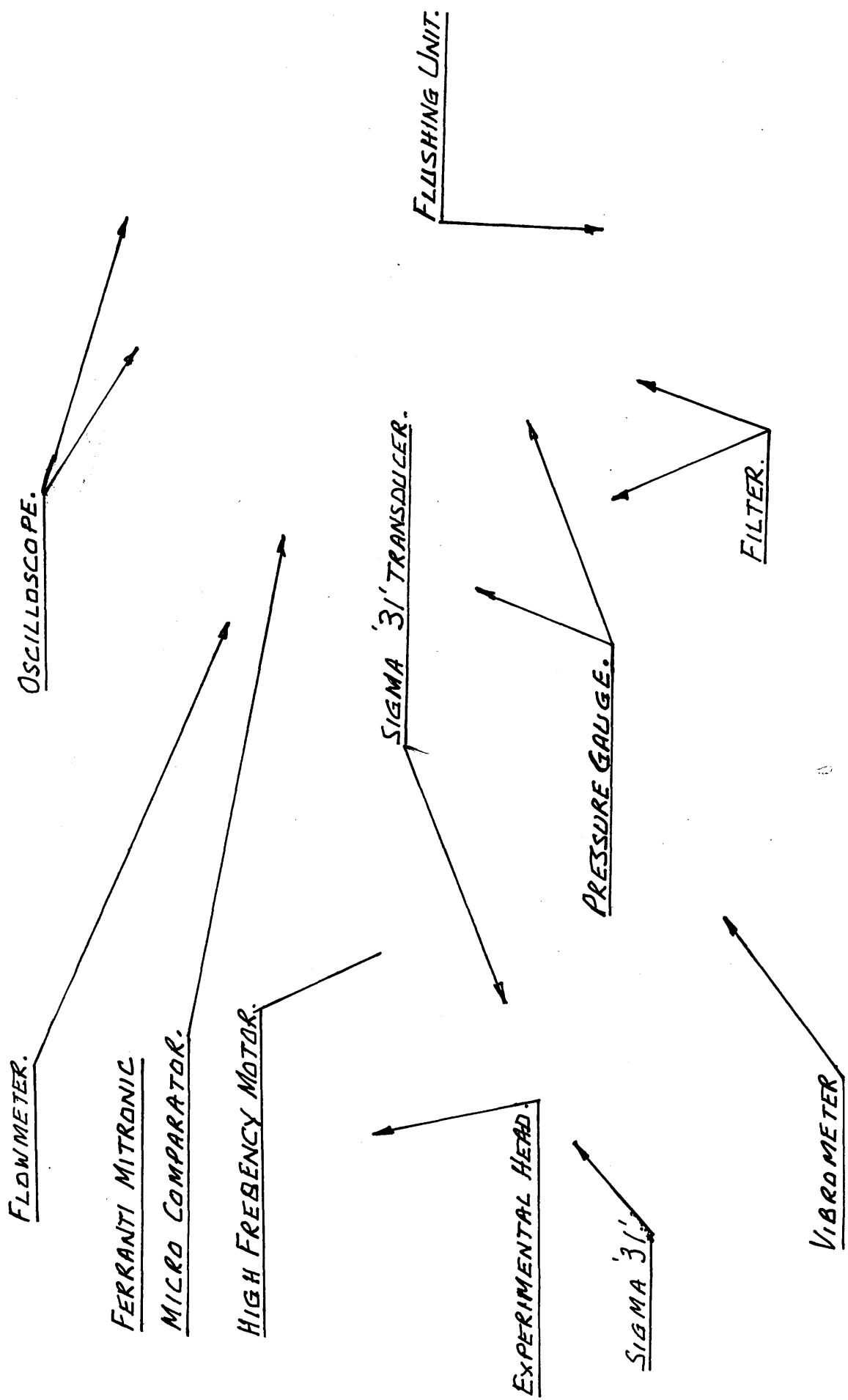
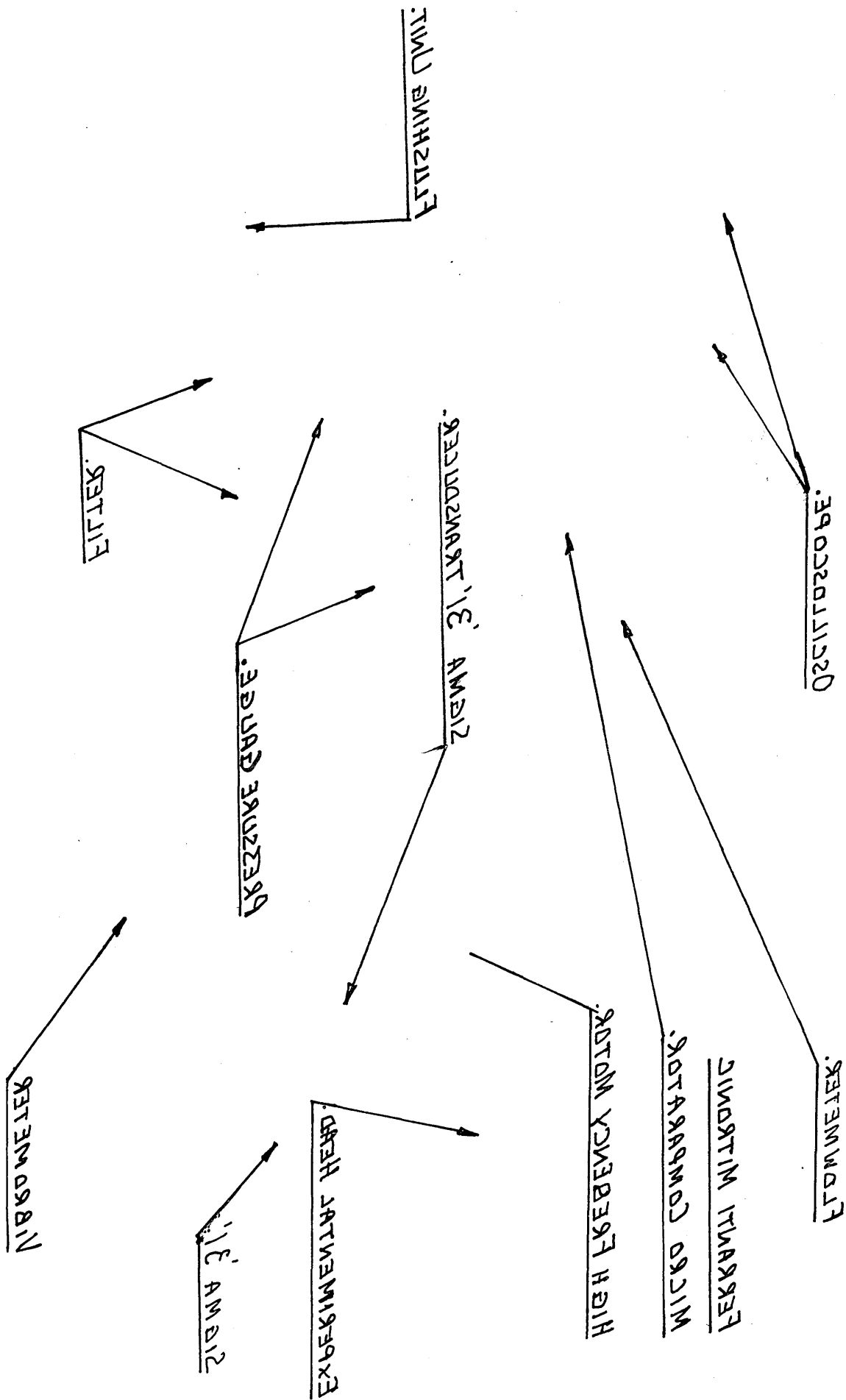


Fig. 41 — Complete arrangement showing the components.





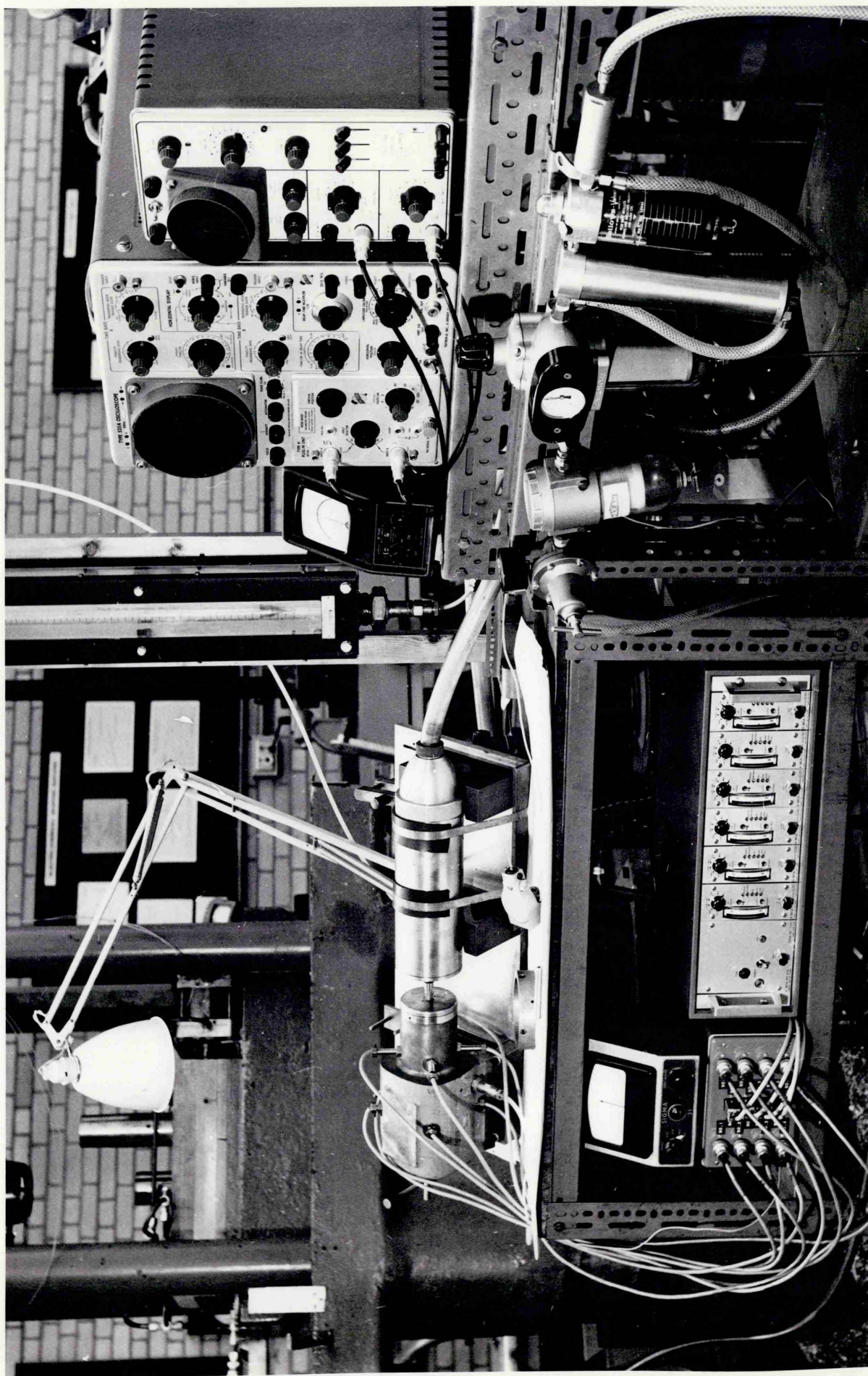


Fig. 42 -- First test rig with measuring equipments.

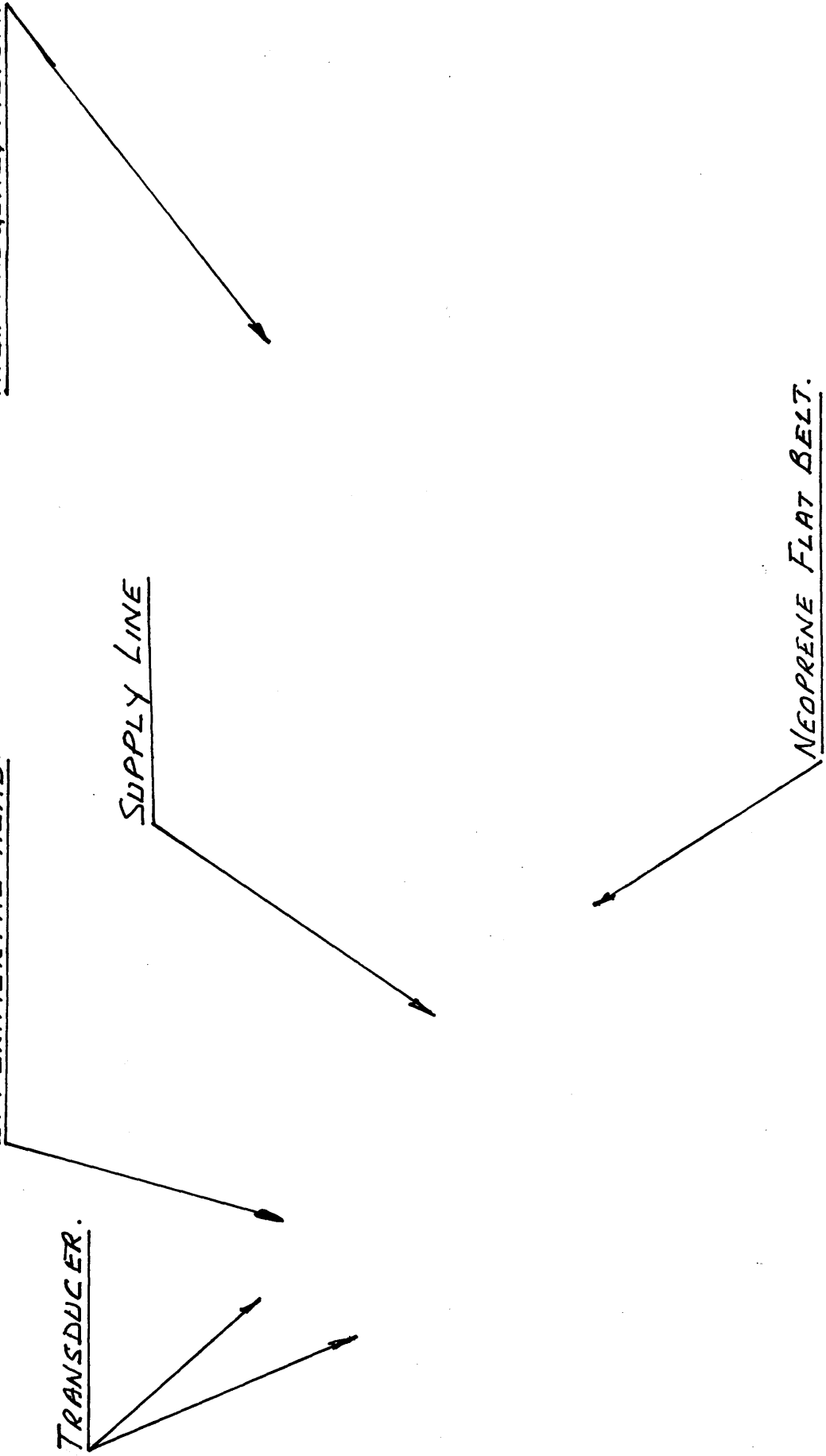
EXPERIMENTAL HEAD.

HIGH FREQUENCY MOTOR.

SUPPLY LINE

NEOPRENE FLAT BELT.

TRANSDUCER.



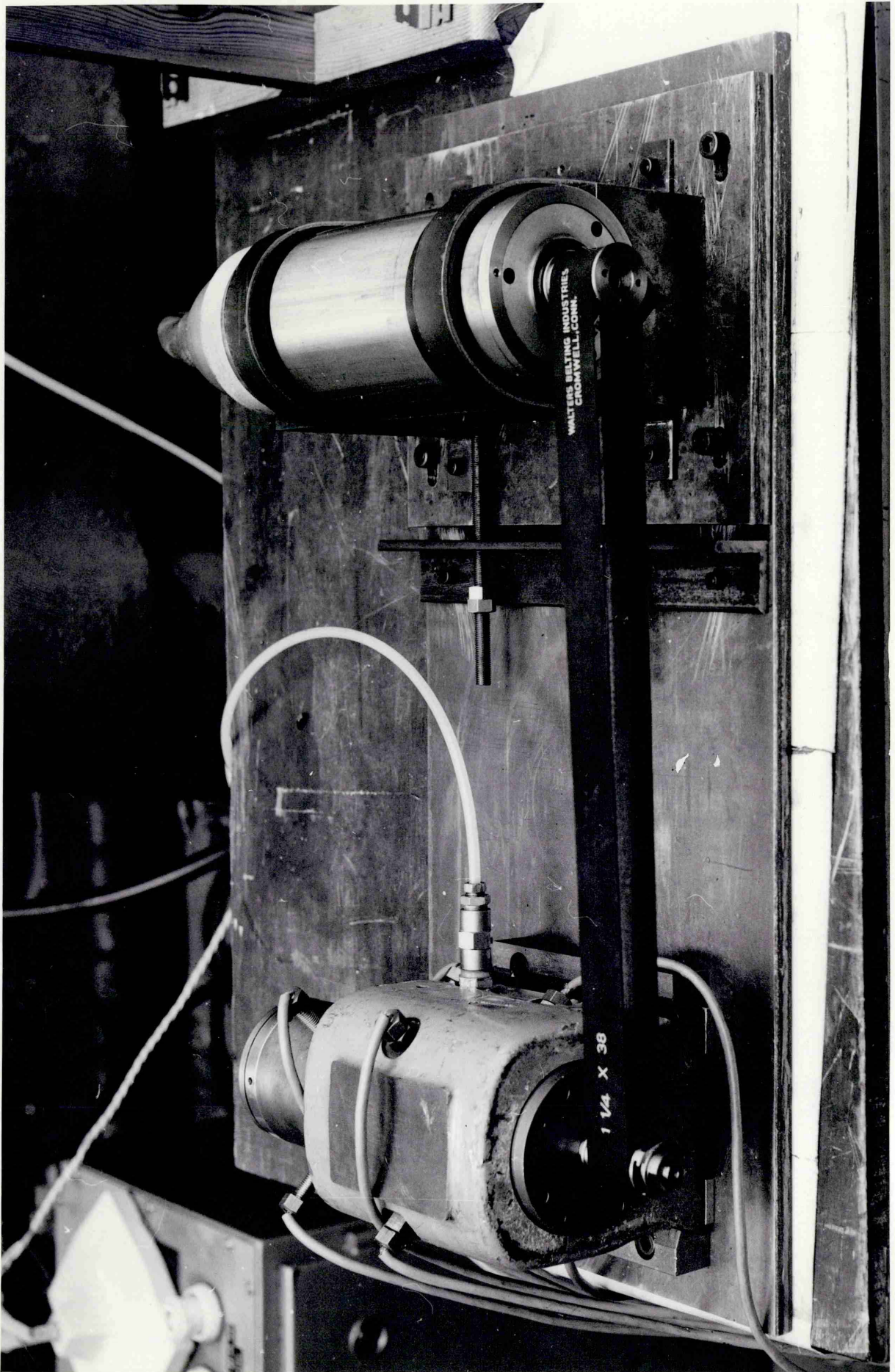


Fig.43 -- Second test rig showing driving arrangement.

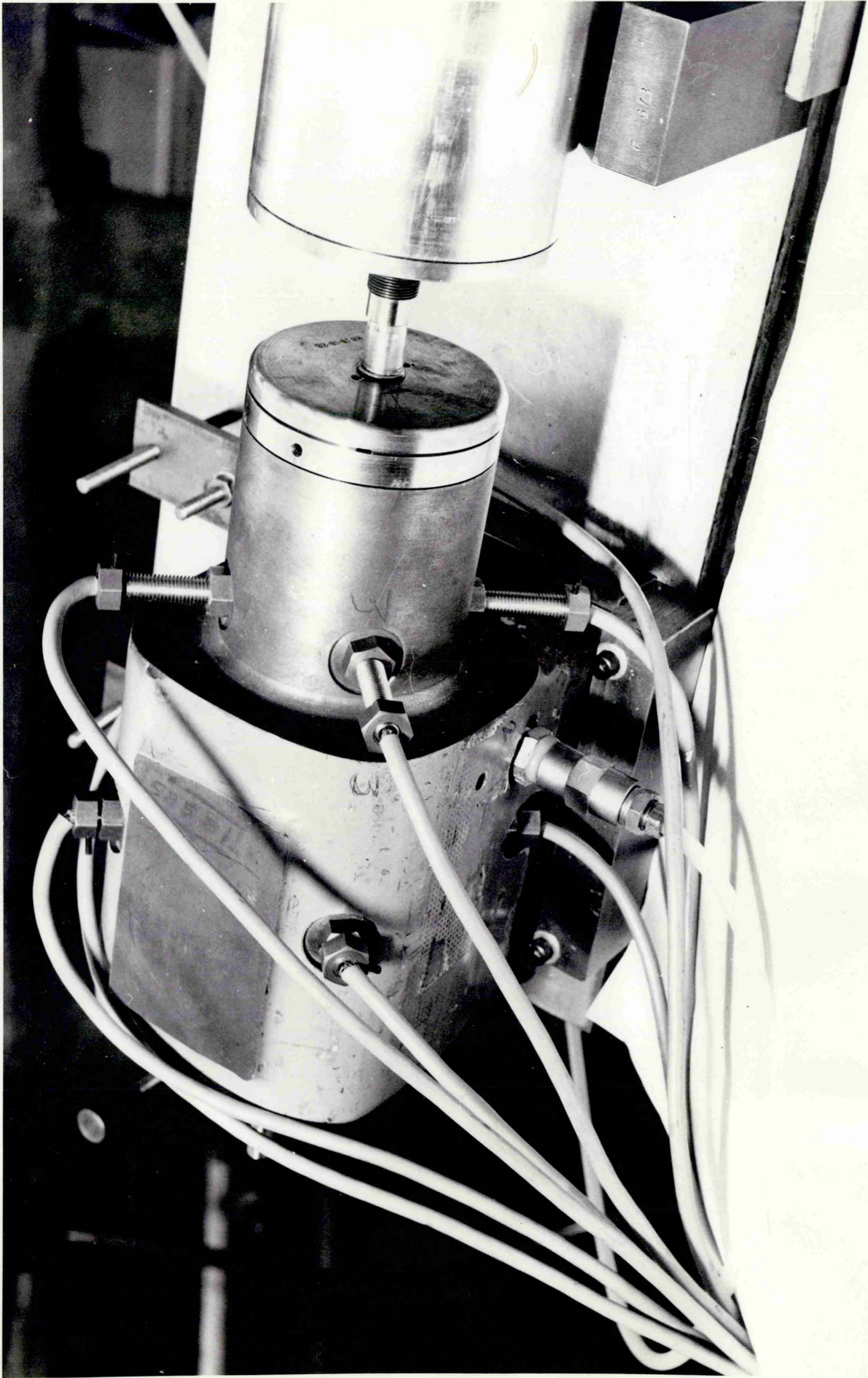
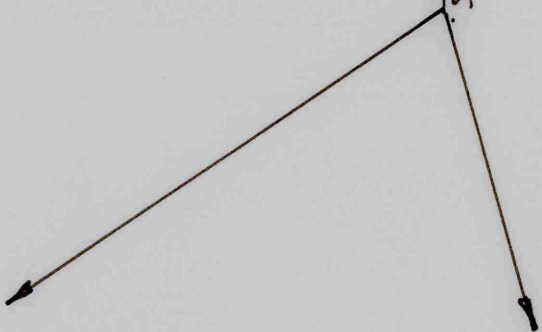
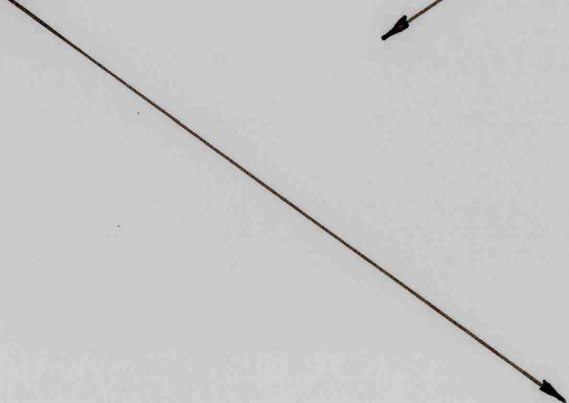


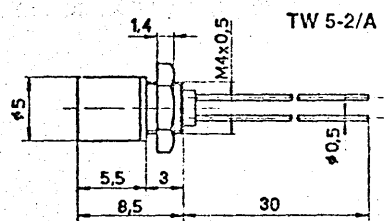
Fig. 44 -- Showing transducers position and driving arrangements.

PAIR OF TRANSDUCERS.

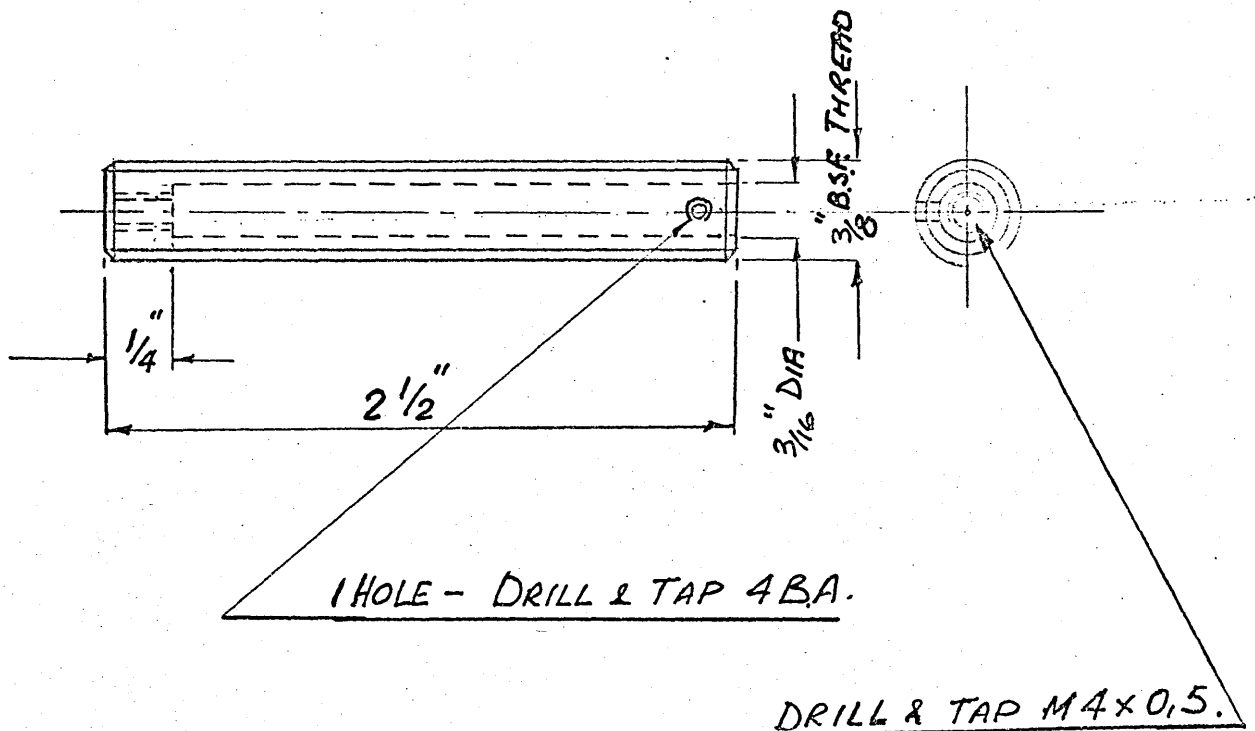


FLEXIBLE COUPLING.





a) Transducer.



b) Transducer holder.

MATERIAL - BRASS.

Fig.45. - Sketches of transducer and holder.

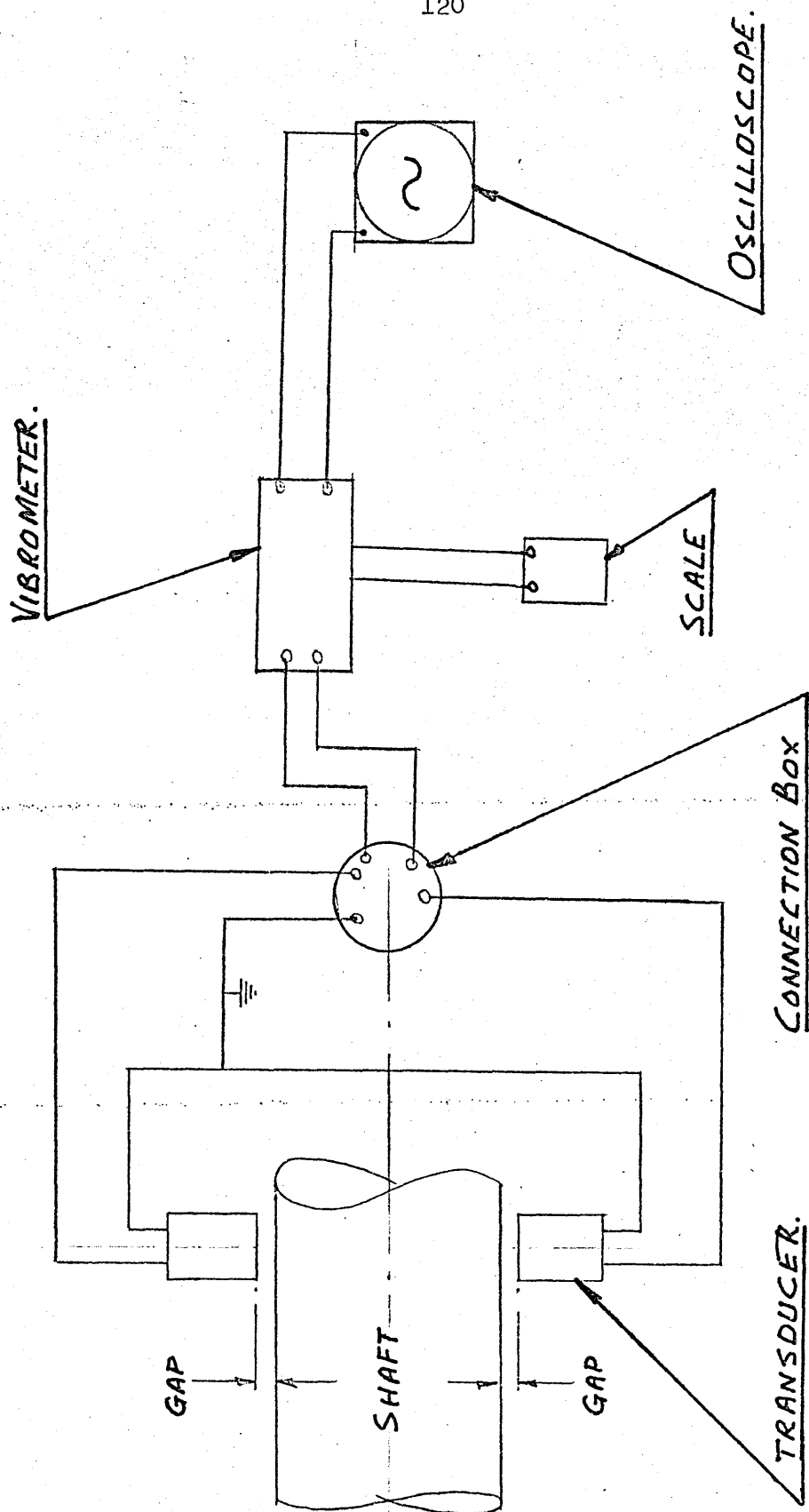
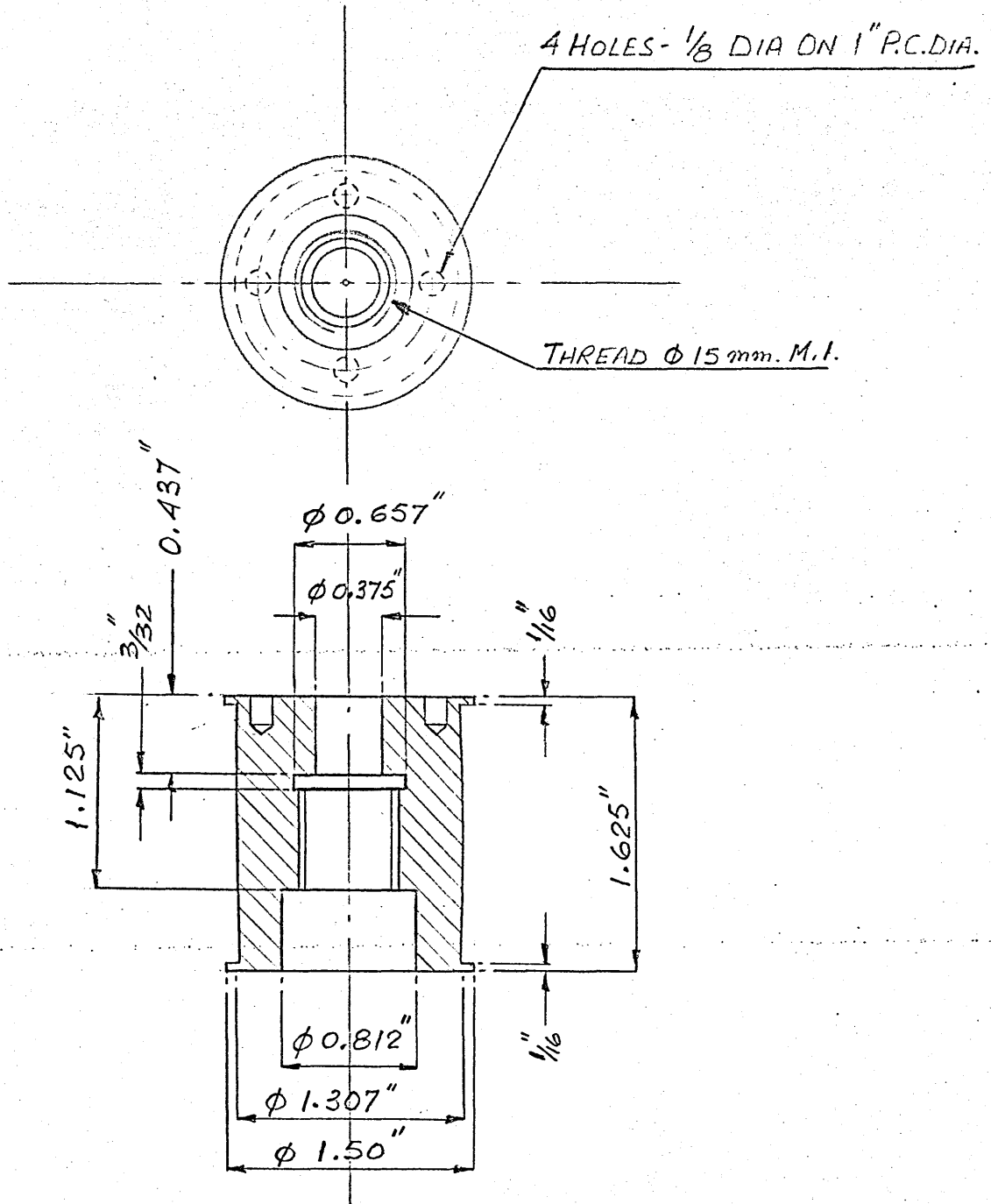


Fig. 45c — Circuit drawing shown schematically.

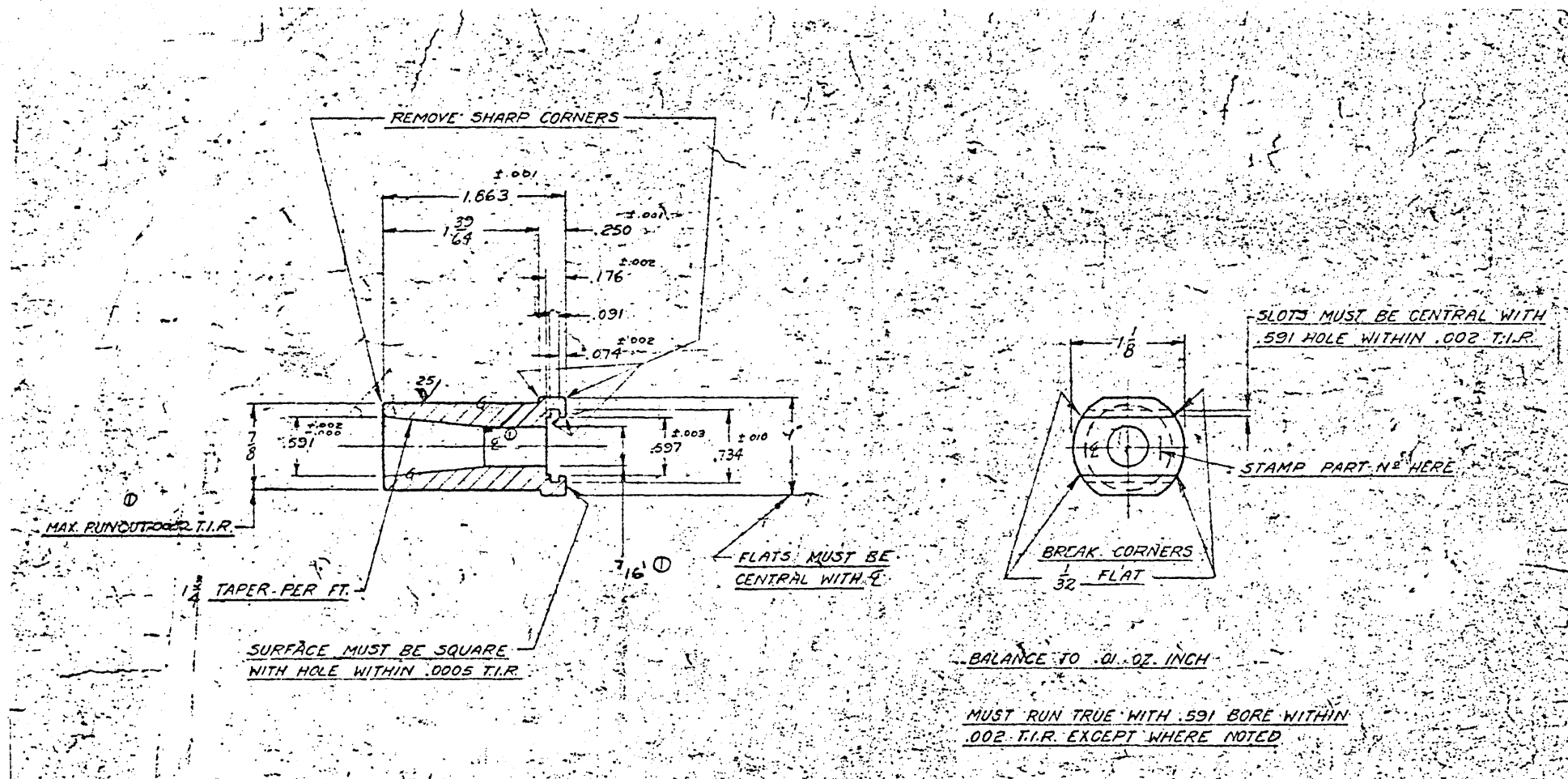


MAT- MILD STEEL.

GENERAL TOLERANCE - ± 0.005 "

Fig. 46 --- Driving pulley.

Fig. 47 - Pulley for experimental head.



EXPERIMENTAL RECORD

NAME K. GHOSH

DATE 15/8/69

Deflection $\rightarrow e. in \times 10^{-4}$															
Pressure lb/in ²	60	60	60	Average		80	80	80	Average		100	100	100	Average	
Load	0	0	0	0	0	0	0	0	0		0	0	0	0	
5	0.21	0.21	0.21	0.21		0.18	0.18	0.18	0.18		0.14	0.14	0.14	0.14	
10	0.41	0.42	0.43	0.42		0.35	0.36	0.34	0.35		0.28	0.28	0.28	0.28	
15	0.64	0.65	0.66	0.65		0.55	0.55	0.55	0.55		0.46	0.45	0.44	0.45	
20	0.87	0.86	0.88	0.87		0.75	0.76	0.74	0.75		0.62	0.62	0.62	0.62	
25	1.15	1.15	1.15	1.10		1.00	1.00	1.00	1.00		0.81	0.82	0.83	0.82	
30	1.42	1.41	1.43	1.42		1.26	1.25	1.24	1.25		1.00	1.00	1.00	1.00	
35	1.70	1.70	1.70	1.70		1.45	1.45	1.45	1.45		1.25	1.25	1.25	1.25	
40	2.04	2.05	2.06	2.05		1.71	1.73	1.72	1.72		1.50	1.50	1.50	1.50	
45	2.4	2.4	2.4	2.4		2.1	2.0	2.05	2.05		1.75	1.76	1.74	1.75	
50	2.79	2.81	2.8	2.8		2.28	2.3	2.32	2.3		2.00	2.00	2.00	2.00	
STIFFNESS:- AT LOADING POINT AT BEARING CENTRE															
Supply Pressure 800 lb/in ² -						233,200	lb/in				289,000 lb/in.				
" " 80 lb/in ² -						190,000	lb/in				235,000 lb/in.				
" " 60 lb/in ² -						146,000	lb/in.				181,000 lb/in.				

Fig. 48 --- Results of load capacity and stiffness tests of front and rear journal bearings. continued---

The College of Aeronautics

EXPERIMENTAL RECORD

EXPERIMENT: Load capacity & Stiffness (Radial)

NAME K. GHOSH.

REMARKS: Load applied on the shaft at 2 1/4" from bearing centre.
Static condition.

DATE 15/8/69.

REAR JOURNAL BEARING.

Deflection $\rightarrow$ e in $\times 10^{-4}$															
Pressure $lb/in^2 \rightarrow$		60	60	60	Average		80	80	80	Average		100	100	100	Average
LOAD		0	0	0	0		0	0	0	0		0	0	0	0
	5	0.21	0.22	0.23	0.22		0.18	0.18	0.18	0.18		0.14	0.15	0.16	0.15
	10	0.44	0.45	0.46	0.45		0.37	0.36	0.38	0.37		0.29	0.3	0.31	0.3
lbs	15	0.64	0.65	0.66	0.65		0.58	0.58	0.58	0.58		0.5	0.5	0.5	0.5
	20	0.92	0.92	0.92	0.92		0.81	0.8	0.79	0.8		0.71	0.7	0.69	0.7
	25	1.2	1.2	1.2	1.20		1.04	1.05	1.06	1.05		0.91	0.9	0.89	0.9
	30	1.5	1.51	1.49	1.50		1.29	1.30	1.31	1.30		1.14	1.15	1.16	1.15
	35	1.81	1.80	1.79	1.80		1.56	1.57	1.55	1.56		1.34	1.35	1.36	1.35
	40	2.14	2.15	2.16	2.15		1.85	1.85	1.85	1.85		1.6	1.6	1.6	1.6
	45	2.52	2.53	2.54	2.53		2.14	2.15	2.16	2.15		1.91	1.9	1.89	1.9
	50	2.84	2.85	2.86	2.85		2.5	2.5	2.5	2.5		2.14	2.16	2.15	2.15
STIFFNESS :-															
						AT LOADING POINT					AT BEARING CENTRE				
Supply Pressure						100 lb/in^2 -					203,000 lb/in .				
"						80 lb/in^2					183,000 lb/in				
"						60 lb/in^2					140,000 lb/in				

Fig. 48 — Results of load capacity and stiffness tests of front and rear journal bearings.

EXPERIMENTAL RECORD

NAME K. GHOSH.

DATE 16/8/69

Clearance Gap $\rightarrow h$ in $\times 10^{-4}$																	
Pressure $16/\text{in}^2 \rightarrow$	60	60	60	Average		80	80	80	Average		100	100	100	Average			
Load $\downarrow$	0	0	0	0	0	0	0	0	0		0	0	0	0			
lbs	5	6.7	6.7	6.7	6.7	6.74	6.73	6.72	6.73		6.77	6.77	6.77	6.77			
	10	6.41	6.4	6.39	6.4	6.47	6.47	6.47	6.47		6.54	6.55	6.56	6.54			
	15	6.07	6.08	6.09	6.08	6.22	6.23	6.21	6.22		6.31	6.32	6.33	6.32			
	20	5.79	5.8	5.81	5.8	5.94	5.95	5.96	5.95		6.1	6.1	6.1	6.1			
	25	5.49	5.5	5.51	5.5	5.71	5.69	5.7	5.7		5.85	5.84	5.86	5.85			
	30	5.2	5.2	5.2	5.2	5.46	5.45	5.44	5.45		5.63	5.61	5.62	5.62			
	35	4.76	4.77	4.78	4.77	5.2	5.2	5.2	5.2		5.4	5.4	5.4	5.4			
	40	4.56	4.57	4.58	4.57	4.96	4.94	4.95	4.95		5.14	5.15	5.16	5.15			
	45	4.29	4.31	4.3	4.3	4.67	4.67	4.67	4.67		4.96	4.95	4.94	4.95			
	50	4.0	4.0	4.0	4.0	4.41	4.42	4.43	4.42		4.72	4.72	4.72	4.72			

STIFFNESS:-

Supply Pressure $100 \text{ lb/in}^2 - 219,000 \text{ lb/in.}$

"	"	80 lb/in ²	- 194,000 lb/in.
---	---	-----------------------	------------------

n	n	60 lb/in ² - 169,000 lb/in.
---	---	--

Fig. 49 --- Results of load capacity and stiffness tests of front and rear thrust bearings.

continued

The College of Aeronautics

EXPERIMENTAL RECORD

EXPERIMENT: Load capacity & Stiffness (Axial)

NAME K. GHOSH.

REMARKS: Static condition.

DATE 16/8/69

REAR THRUST BEARING.

Clearance gap $\rightarrow$ h in $\times 10^{-4}$														
Pressure lb/in ²	$\rightarrow$ 60	60	60	Average	80	80	80	Average	100	100	100	Average		
Load	0	0	0	0	0	0	0	0	0	0	0	0		
$\downarrow$	5	6.68	6.69	6.67	6.68	6.72	6.73	6.74	6.73	6.78	6.78	6.78	6.78	
lbs	10	6.4	6.4	6.4	6.4	6.50	6.50	6.47	6.49	6.56	6.55	6.54	6.55	
	15	6.10	6.12	6.11	6.11	6.23	6.23	6.23	6.23	6.31	6.32	6.33	6.32	
	20	5.79	5.80	5.81	5.80	6.0	6.0	6.0	6.0	6.09	6.09	6.12	6.1	
	25	5.54	5.55	5.56	5.55	5.69	5.71	5.7	5.7	5.9	5.9	5.9	5.9	
	30	5.23	5.23	5.23	5.23	5.46	5.47	5.48	5.47	5.64	5.65	5.66	5.65	
	35	4.96	4.94	4.95	4.95	5.2	5.2	5.2	5.2	5.47	5.46	5.45	5.46	
	40	4.66	4.65	4.64	4.65	4.96	4.96	4.99	4.97	5.24	5.25	5.26	5.25	
	45	4.40	4.40	4.40	4.40	4.75	4.75	4.75	4.75	5.0	5.0	5.0	5.0	
	50	4.06	4.07	4.08	4.07	4.47	4.48	4.49	4.48	4.74	4.74	4.77	4.75	
STIFFNESS:-														
Supply Pressure	100 lb/in ²	— 227,000 lb/in												
"	"	80 lb/in ²	— 200,000 lb/in											
"	"	60 lb/in ²	— 173,000 lb/in.											

Fig. 49 — Results of load capacity and stiffness tests of front and rear thrust bearings.

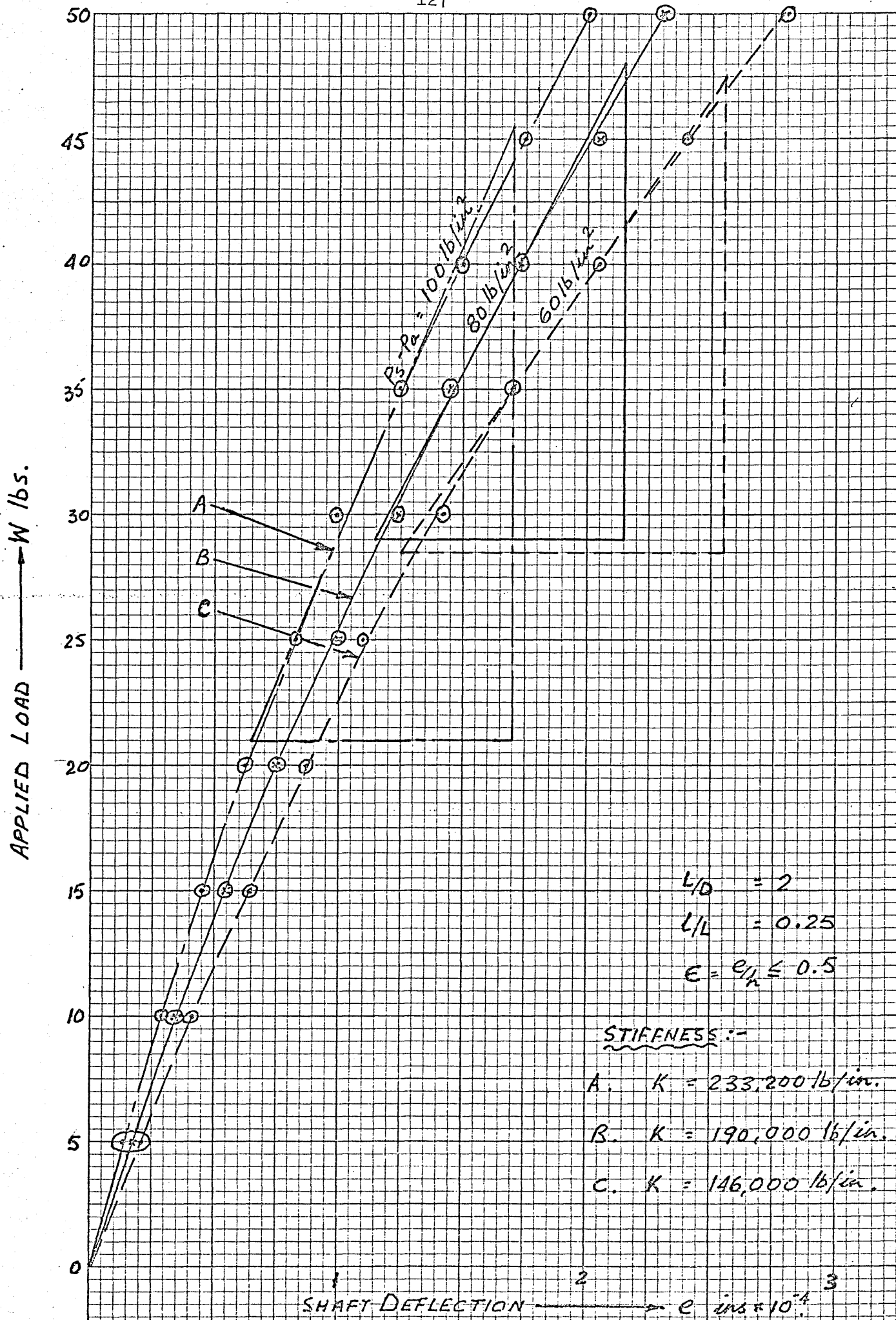


Fig. 50 — Relationship between applied load and shaft deflections, (front position).

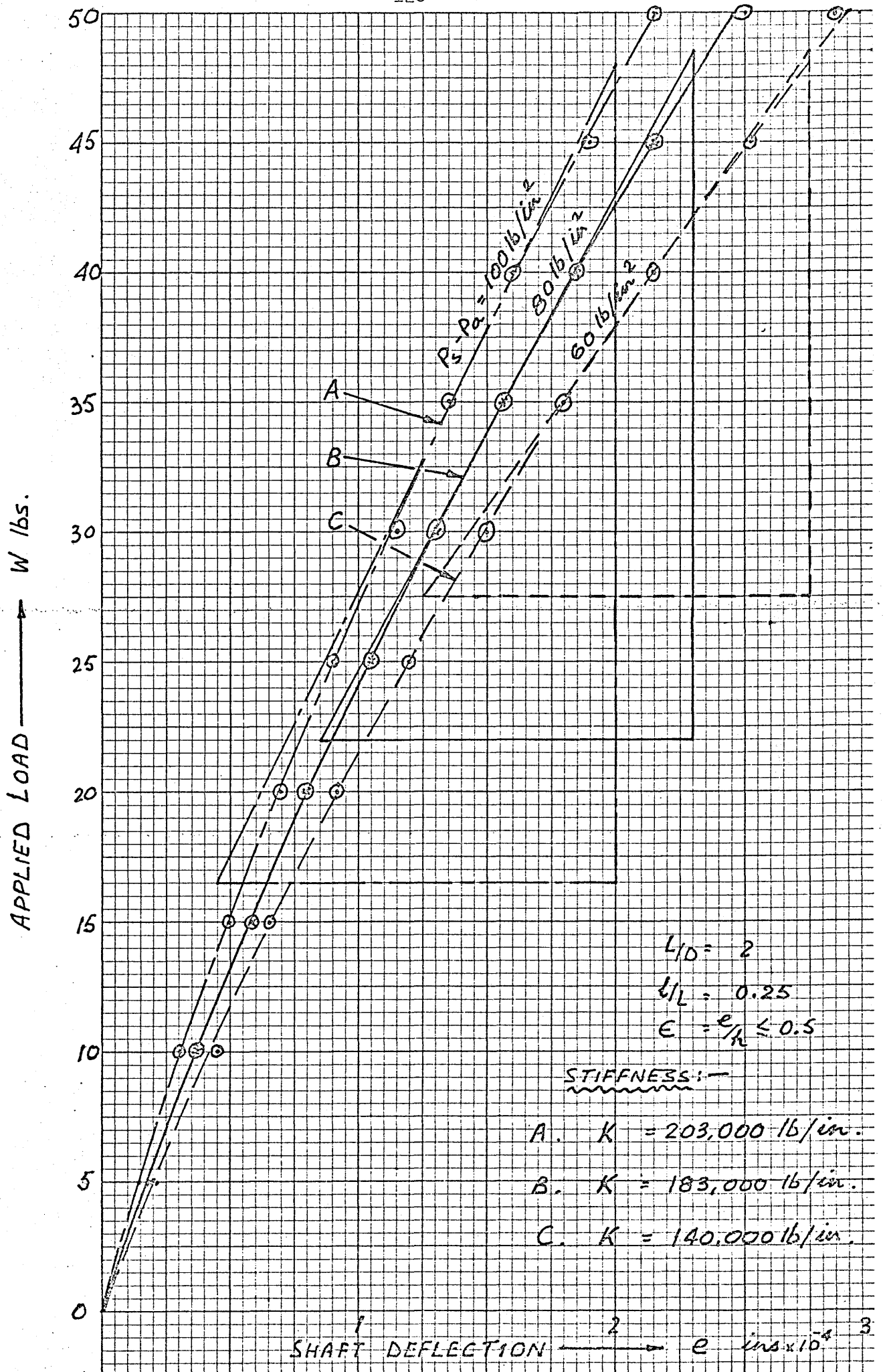


Fig. 51 — Relationship between applied load and shaft deflections, (rear position).

APPLIED LOAD $\longrightarrow$ W lbs.

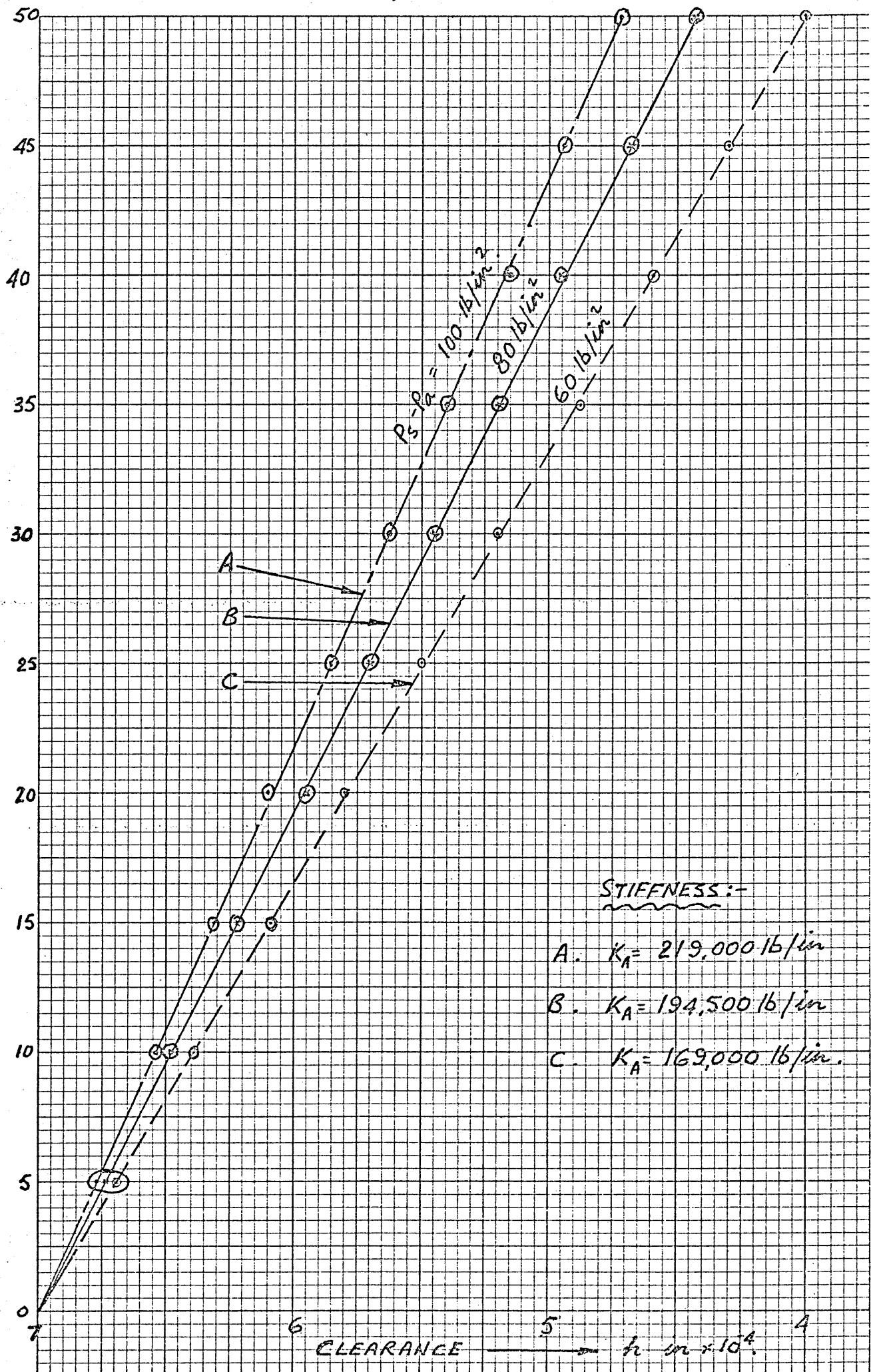


Fig. 52 — Relationship between applied load and clearance of front thrust bearing.

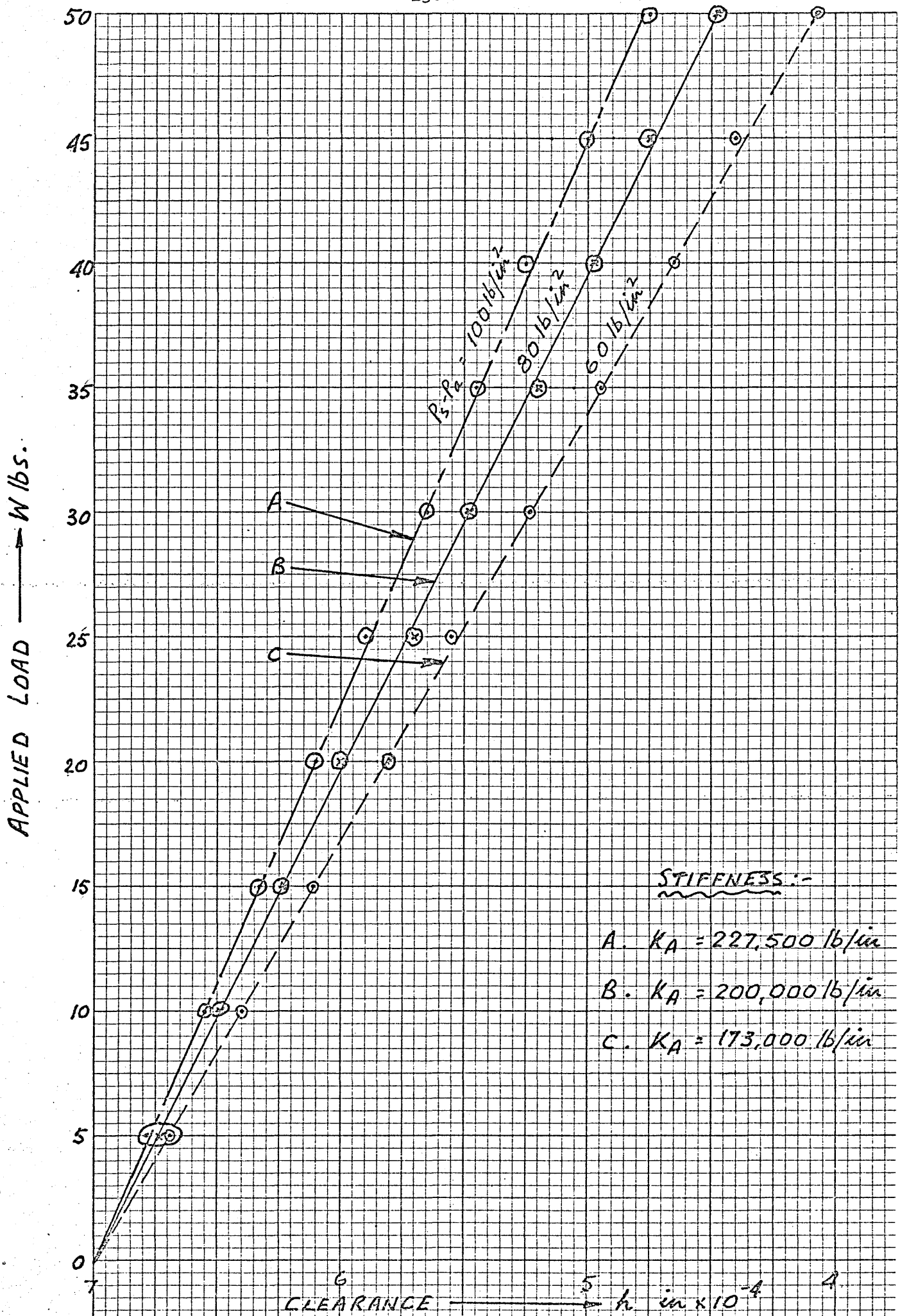


Fig. 53 -- Relationship between applied load and clearance of rear thrust bearing.

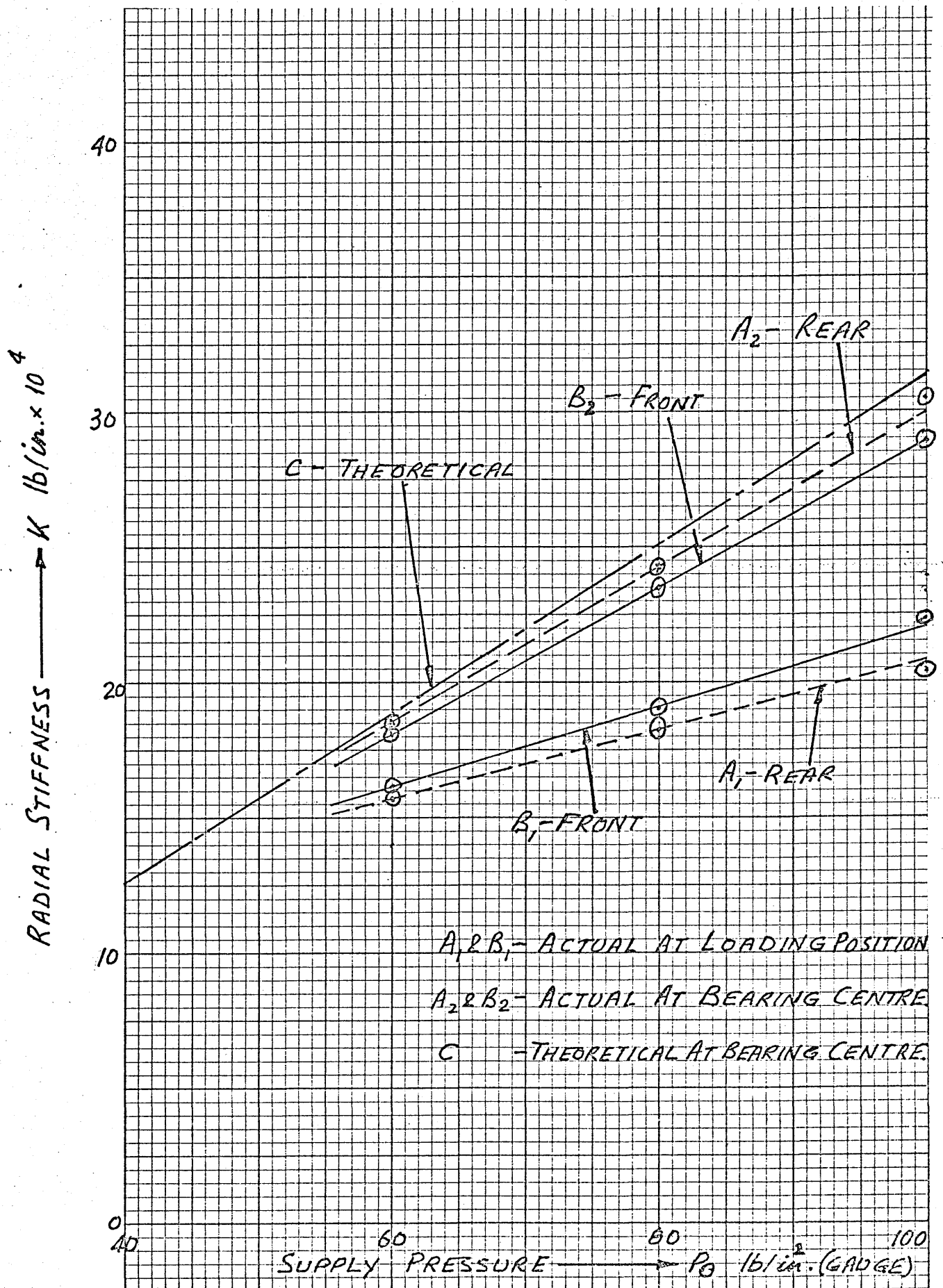


Fig. 54 -- Relationship between radial stiffness and supply pressure of journal bearings.

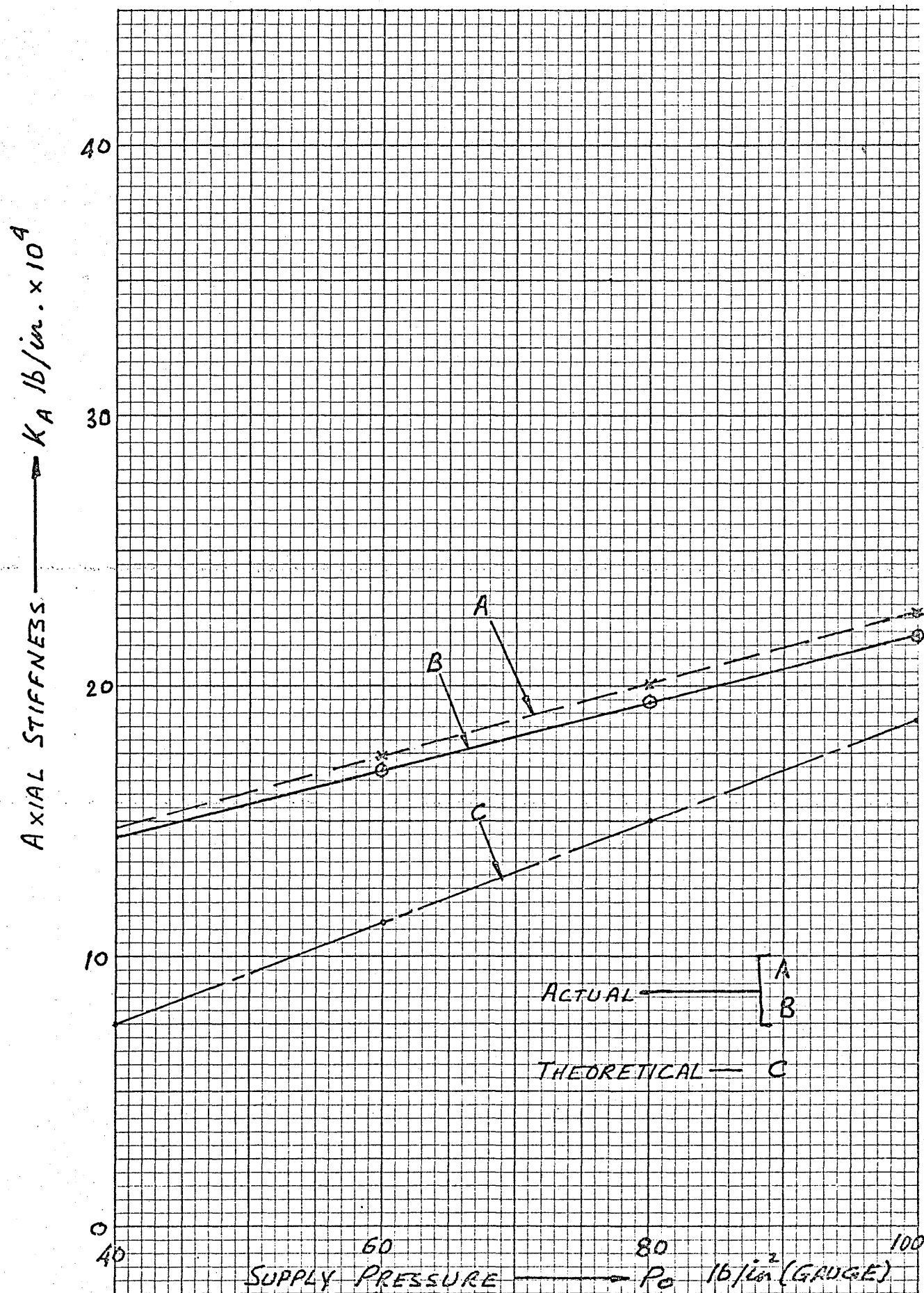


Fig. 55 — Relationship between axial stiffness and supply pressure of thrust bearings.

Experiment: Volumetric flow measurement

Date: 20/0/69

First test without load, variable pressure
Second test with 50lbs load at end, variable pressure
Third test without load rotational condition, constant pressure

Fig. 56 -- Results of volumetric flow measurements.

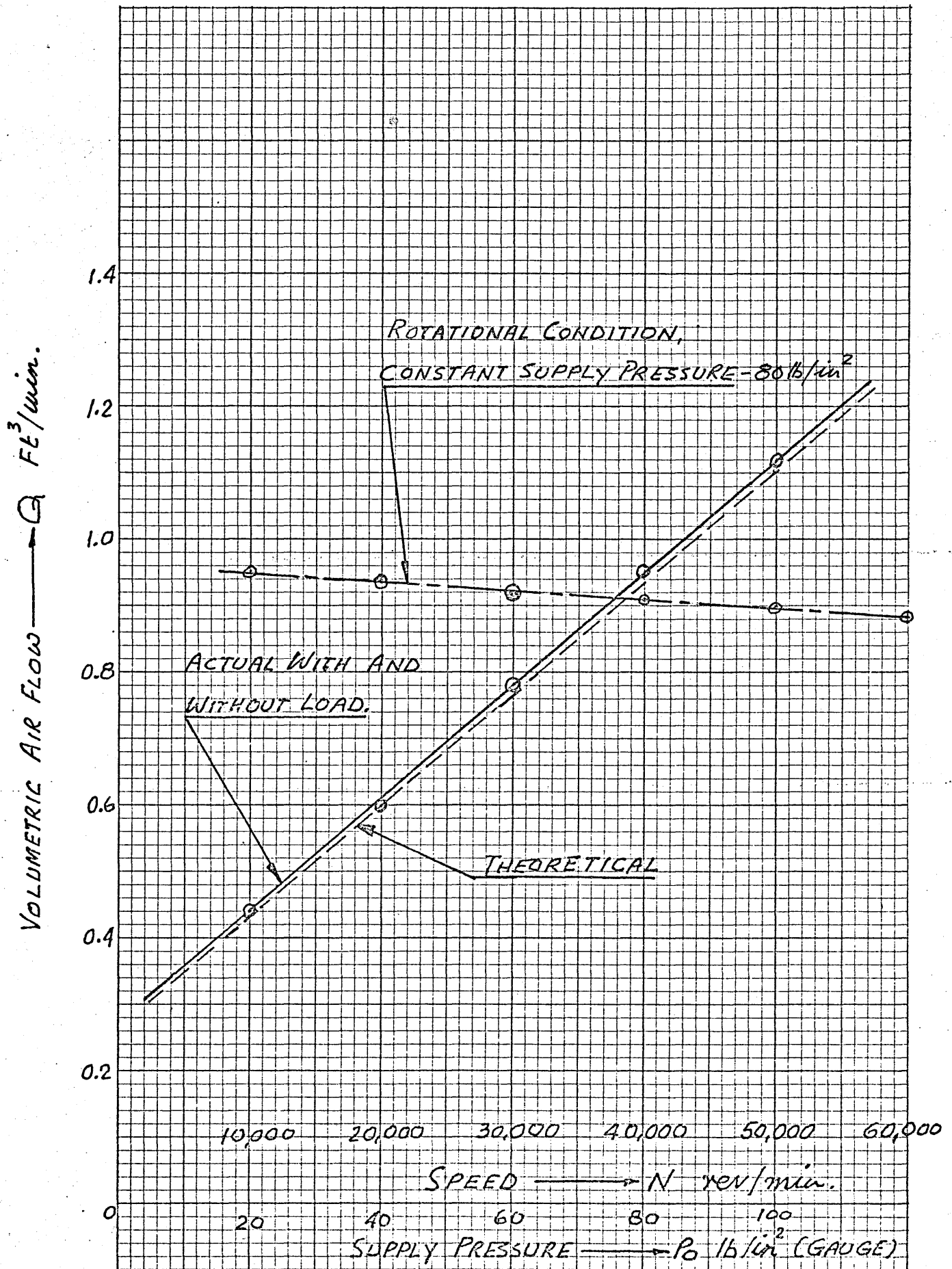


Fig. 57 — Relationship between air flow and supply pressure and speed.

Experiment: Free running test.

Name: K. GHOSH

Amplitude against speed.

Date: 25/9/69

Remarks:

Constant supply pressure
 $P_0 = 80 \text{ lb/in}^2$

$A_1 = \text{Front bearing.}$

$A_2 = \text{Rear bearing}$

Frequency		Amplitude (eccentricity) $\rightarrow$					in $\times 10^{-4}$			
C/sec										
D	0	0	0	0	0	0	0	0	0	0
R	100	0.02	0.02	0.03	0.01	0.03	0.03	0.03	0.03	0.03
E	200	0.07	0.06	0.08	0.07	0.11	0.09	0.1	0.1	0.1
T	300	0.15	0.15	0.17	0.15	0.18	0.17	0.18	0.18	0.18
D R I V E	400	0.25	0.25	0.25	0.25	0.36	0.35	0.34	0.35	0.35
	500	0.08	0.08	0.08	0.08	0.14	0.15	0.17	0.15	0.15
	600	0.26	0.27	0.28	0.27	0.42	0.42	0.42	0.42	0.42
	700	0.17	0.17	0.17	0.17	0.27	0.25	0.25	0.26	0.26
	800	0.07	0.06	0.08	0.07	0.1	0.11	0.1	0.09	0.09
		B ₁ = Front bearing					B ₂ = Rear bearing			
B	0	0	0	0	0	0	0	0	0	0
E	100	0.1	0.09	0.09	0.08	0.08	0.09	0.07	0.08	0.08
L	200	0.9	0.91	0.89	0.9	0.7	0.7	0.7	0.7	0.7
T	300	0.15	0.15	0.15	0.15	1.25	1.25	1.25	1.25	1.25
	400	0.49	0.51	0.5	0.5	0.99	1.01	1.0	1.0	1.0
D	500	0.27	0.27	0.26	0.28	0.75	0.76	0.74	0.74	0.74
R	600	0.1	0.1	0.1	0.1	1.19	1.21	1.19	1.21	1.21
I	700	0.6	0.59	0.6	0.61	1.75	1.75	1.75	1.75	1.75
V	800	1.11	1.09	1.1	1.1	1.5	1.49	1.51	1.5	1.5
E	900	0.78	0.78	0.78	0.78	1.25	1.24	1.26	1.25	1.25
	1000	0.44	0.45	0.46	0.45	1.6	1.6	1.6	1.6	1.6
	1100	0.72	0.72	0.72	0.72	2.1	2.1	2.1	2.1	2.1
	1200	1.16	1.18	1.17	1.17	2.71	2.69	2.7	2.7	2.7
	1250	1.50	1.49	1.50	1.51	3.0	3.01	3.00	3.00	3.00

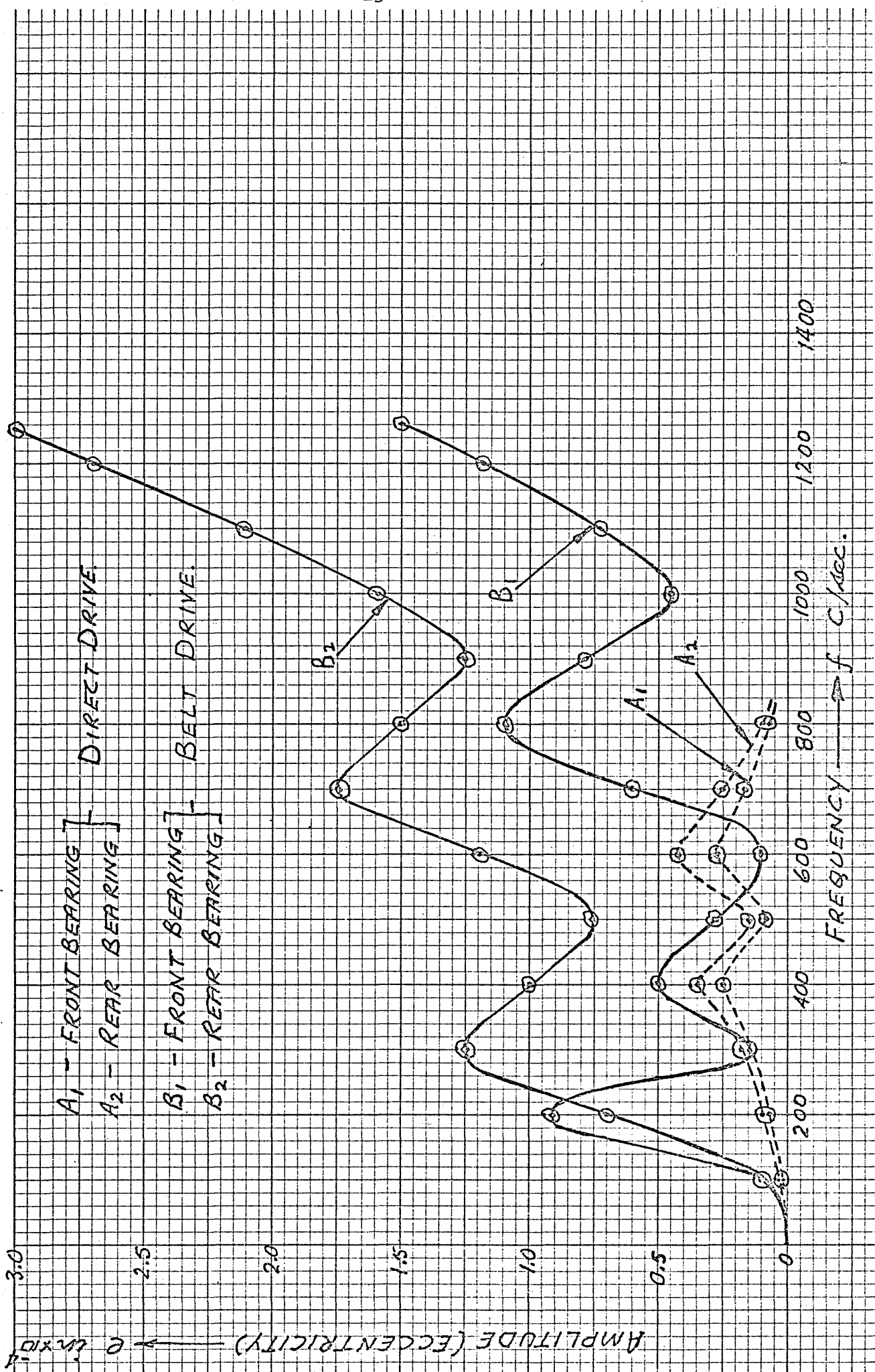


Fig. 59 — Relationship of amplitude (eccentricity) against frequency for direct and belt drive.

THE COLLEGE OF AERONAUTICS

Experimental Record

Experiment: Belt Slip Name: K. GHOSH
Driven speed against Driving Speed Date: 26/9/69
 Remarks: Constant Pressure 80 lb/in².

measured frequency f C/sec	Driven speed frequency N rev/min	Driving speed N rev/min	Average rev/min	Belt Slip %
50	3000	3000	3000	0
100	6000	6000	6000	0
150	9000	9000	9000	0
200	12000	12000	12000	0
250	15000	15000	15000	0
300	18000	18100	17900	0
350	21000	21400	21600	
400	24000	24500	24500	
450	27000	27000	27000	0
500	30000	29800	31200	0
550	33000	33000	33000	0
600	36000	36100	35900	0
650	39000	39600	39400	1 1/4
700	42000	42500	42500	1 1/4
750	45000	46000	46000	2 1/4
800	48000	48900	48200	0.5
850	51000	52000	52000	2
900	54000	54600	54400	1
950	57000	57500	57500	1
1000	60000	61000	61000	1 1/2
1050	63000	63800	64200	1 1/2
1100	66000	67600	67400	2
1150	69000	70500	70500	2
1200	72000	73800	74200	2 1/2
1250	75000	77000	77000	2 1/2

Fig. 60 -- Results of belt slip measurements.

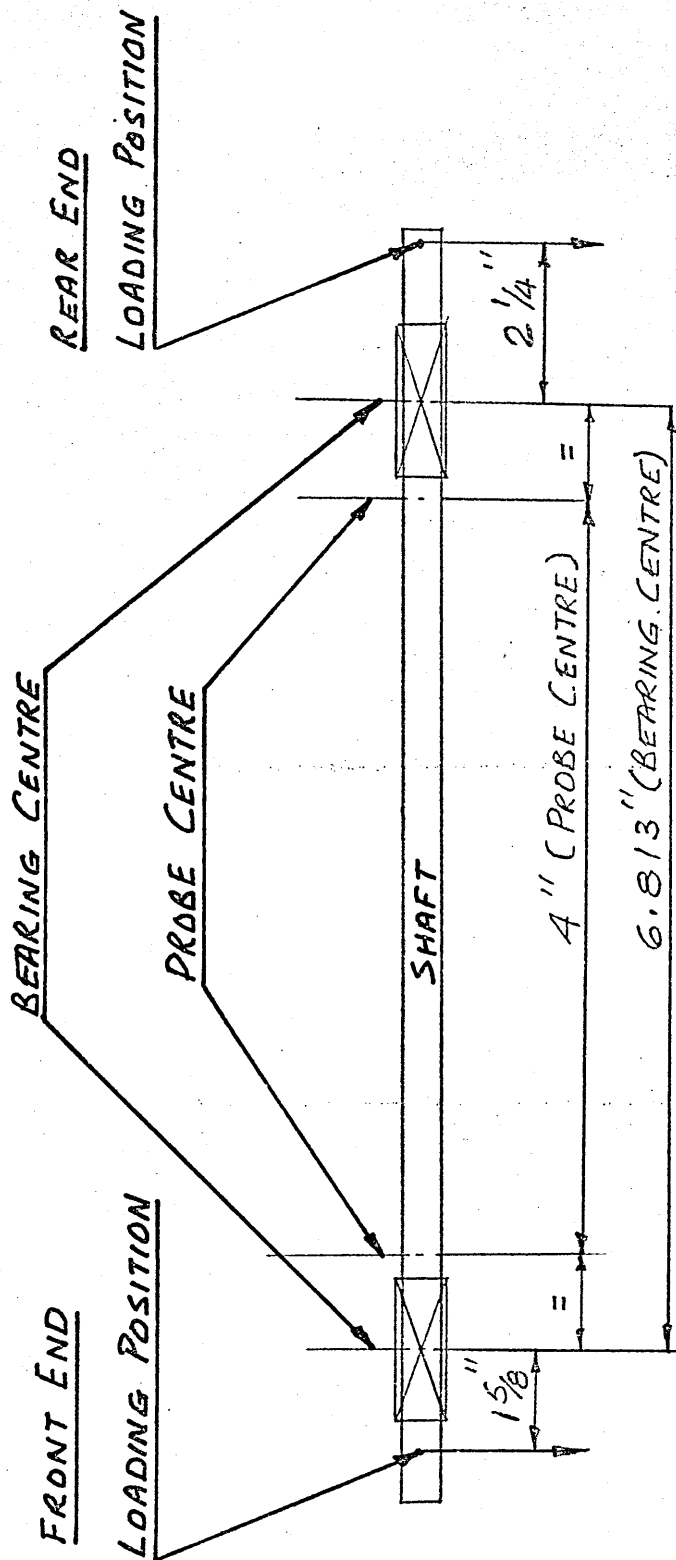


Fig. 61 -- Sketch of bearing centre, probe centre and loading positions.

9.1. REFERENCES.

1. Hirn, G. 'Study of the principle phenomena shown by friction and of various methods of determining the viscosity of lubricants'. Bulletin Soc. Industr. Mulhouse 26. No. 129 (1854) 188.
2. Kingsbury, A. 'Experiments with an air lubricated bearing'. J. Amer. Soc. Nav. Engrs. 9(1897) 267.
3. Wunsch, H.L. 'Air-bearing applications to machine tools and measuring instruments'. N.E.L. Report No.390. 1969.
4. Sternlicht, B. 'Gas lubrication - achievements, problems and possibilities'. Gas Bearing Symposium. Southampton University. 1963.
5. Muller, P.M. 'Air-turbine driven spindles for internal grinding machines'. Product Engineering. 23(3), 1952. Page 160 - 163.
6. Sheinberg, S.A. 'Internal grinding spindles with air lubricated bearings'. Machines and Tooling. 33(3), 1962. Page 8 - 12.
7. Lehmann, R., Wiener, A. and Rogal, R. 'Einiges über theorie und anwendung von luftlagern'. Maschinenbautechnik. 11(1). 1962. Page 24 - 27.
8. Shires, G.L. 'The viscid flow of air in a narrow slot'. N.G.T.E. Memo No. M46, Dec. 1948.
9. Shires, G.L. 'On a type of air lubricated journal bearing'. N.G.T.E. Report No. R61, 1949.
10. Gross, W.A. 'Gas film lubrication'. John Wiley. New York.

11. Constantinescu, V.N. 'An approximate method for the analysis of externally pressurized gas lubricated journal bearings'. Gas Bearing Symposium Southampton University. 1967.
12. Dudgeon, F.H. and Lowe, I.R.G. 'A theoretical analysis of hydrostatic gas journal bearings'. Canadian N.R.C. Report MT 54. 1965.
13. Robinson, C.H. and Sterry, F. 'The static strength of pressure fed gas journal bearings'. A.E.R.E. Report, Sept. 1958.
14. Pinkus, O. and Sternlicht B. 'Hydrodynamic gas bearings'. Chap-5. Theory of hydrodynamic lubrication 1961. New York. McGraw-Hill.
15. Shires, G.L. 'The design of externally pressurized bearings'. Gas lubricated Bearings. Butterworths London 1964.
16. Shires, G.L. and Pantall, D. 'Aerostatic jacking of a vented aerodynamic journal bearing'. I.Mech. E., Lubrication and wear Group Convention. Bournemouth. '1963'
17. Henrich, G. 'The theory of the externally pressurized bearing with compressible lubricant'. First International Symposium on gas-lubricated bearings 1959. Washington D.C.
18. Gross, W.A. 'Investigation of whirl in externally pressurized air-lubricated journal bearings'. J. Basic Engineering March '62.
19. Powell, J.W. 'Hydrodynamic effects in hydrostatic gas bearings'. Engineer, July 1962.
20. Fuller, D.D. 'Theory and practice of lubrication for engineers'. John Wiley & Sons. New York.

21. Geary, P.J. 'Fluid film bearing. A survey of design construction and use'. B.S.I.R.A. 1962.
22. Licht, L., Fuller, D.D. and Sternlicht, B. 'Self-excited vibration of an air lubricated thrust bearing'. Trans A.S.M.E. 80. NO.22(1958) 411.
23. Holster, P.L. 'Reliable and easy to handle design formulae for externally pressurized gas thrust and journal bearings'. Gas Bearing Symposium. Southampton University. 1967.
24. Lund, J.W. 'A theoretical analysis of whirl instabilities and Pneumatic hammer for a rigid rotor in pressurized gas journal bearings'. Trans. A.S.M.E. Lub. Technology. April 1967.
25. Allen, D.S. and Stokes, P.J. 'A Study for gas lubricated hydrostatic journal bearings'. U.K.A.E.A.- D.E.G. Report (1959b).
26. Powell, J.W., Moye, M.H. and Dwight, P.R. 'Fundamental theory and experiments on hydrostatic air bearings'. Proc. of I.Mech.E. Lubrication and Wear Convention. May. 1963.
27. Vohr, J.H. and Lund, J.W. 'Journal bearings'. Design Notes presented jointly by Mechanical Technology Inc., and Rensselaer Polytechnic Institute. U.S.A. 1968.
28. Fischer, G.K., Cherubim, J.L. and Decker, O. 'Some static and dynamic characteristics of high speed shaft systems operating with gas lubricated bearings'. Proc. of the First International Symposium on Gas Lubricated Bearings. Washington D.C. 1959.
29. Lund, J.W. 'Hybrid gas journal bearing instability'. M.T.I. 63TR60, Dec. 1963. U.S.A.F. Contract No. AF- 33(657)- 10694.

30. Rieger, N.F. and Wilcock, D.F. 'A simplified approach to the design of hydrostatic gas bearings'. Machine Design. March - 31, 1966.
31. Sternlicht, B. and Rieger, N.F. 'Rotor stability'. Proceedings of Inst. Mech. Engineers. 1967 - 1968, Volume - 182, Pt.3A.
32. Lewis, P., Eusepi, M. and Morris, J.L. 'Design criteria for externally pressurized gas bearings for operation at high temperature and high speeds'. Transaction of A.S.L.E. - 9, 1966.
33. Stansfield, F.M. 'Machine tool spindles'. M.T.I.R.A. Lecture Note of College of Aeronautics, Cranfield.
34. Ausman, J.S. and Wildman, M. 'How to design hydrodynamic gas bearings'. Product Engineering. Nov.- 25, 1957.
35. Pantall, D. 'The design of aerodynamic bearings'. Gas Lubricated Bearings. Ed. Grassam and Powell. Butterworths. London. 1964.
36. Lund, J.W. 'Gas bearing design methods'. Volume 2, Mechanical Technology Inc., and Rensseler Polytechnic Institute's Course Notes. June. 1966.
37. Powell, J.W. 'Unbalance whirl of rotors supported in gas journal bearings'. The Engineer, July 26, 1963.
38. Whitly, S. 'Gas bearing research in the United Kingdom Atomic Energy Authority'. Proceedings of the First International Symposium on Gas Lubricated Bearings. Washington D.C. 1959.
39. Timoshenko, S. 'Vibration problems in engineering'. D. Van Nostrand Company. New York.
40. Rieger, N.F. 'An experimental investigation of the pressurized air lubricated journal bearing in the high speed range'. Proceedings of Inst. Mech. Engineers. Volume - 173, 1959.

41. Powell, J.W. 'Dynamic characteristics of the externally pressurized bearings'. Gas Lubricated Bearings. Ed. Grassam and Powell. Butterworths. London. 1964.
42. Powell, J.W. 'The design and development of an externally pressurized bearings'. Gas Bearing Symposium. Southampton University. 1965.
43. Powell, J.W. 'Gas behaviour and load capacity of hydrodynamic gas journal bearings'. Inst. Mech. Engineers. Lubrication and Wear Group Convention. Paper-11, 1963.
44. Taylor, G.I. 'Stability of a viscous liquid contained between two rotating cylinders'. Phil. Tran. Roy. Soc. Volume - 223, Page 289 - 343.
45. Wilcock, D.F. 'The hydrodynamic pocket bearing'. Transaction of A.S.M.E. - 77, Page 311 - 319.
46. Petroff, N. 'Friction in machines and the effect of the lubricant', (Russian). Engineering Journal. St. Petersburg. No. 1 - 4, Page 71.

10.1. BIBLIOGRAPHY.

1. Ford, G.W.K.,
Harris, D.M. and
Pantall, D. 'Principle and application of
hydrodynamic type gas bearings'.
Proc. I. Mech. Engineers. 171,
No. 2(1957) 93.
2. Boyd, J., and
Raimondi, A.A. 'Applying bearing theory to the
analysis and design of journal
bearings'.
Trans. A.S.M.E. of Applied Mechanics.
Vol. 18, No. 3, September, 1951.
3. Drescher, H. 'Plain bearings with air lubrication'.
Ministry of Supply translation,
T.I.B./T 4309, August 1954.
4. Grinnell, S.K., and
Richardson, H.H. 'Design study of a hydrostatic gas
bearing with inherent orifice
compensation'.
A.S.M.E. Paper No. 55-A-177, 1955.
5. Taniguchi, O.,
Tamura, A., and
Ono, K. 'Experimental study of whirl
instability for externally
pressurized air journal bearings'.
Bulletin of Japan Society of
Mechanical Engineers. Vol. 11,
No. 43, 1968.
6. Shires, G.L. 'The design of pressurized gas
bearings'.
Tribology. Vol. 1, No. 4,
November 1968.
7. Brix, V.H. 'Synchronous whirling of shafts
in plain (gas) bearings'.
U.K.A.E.A., IGR-R/CA-176, 1956.
8. Wildman, M. 'Experiments on gas lubricated
bearings'.
Amer. Soc. mech. Engrs,
Paper No. 56-LUB-8.
9. Raimondi, A.A. 'A numerical solution for the gas
lubricated full journal bearing of
finite length'.
Trans. Amer. Soc. lub. Engrs. 4,
No. 1 (1961) 131.

10. Sasaki, T.
Mori, H. and others 'Air-lubricated bearings for spinning spindles'.
Trans. Japan Soc. mech. Engrs. 21.
No. 102 (1966).
11. Montgomery, A.G.
and Sterry, F. 'A simple air bearing rotor for very high rotational speeds'.
A.E.R.E., E.D./R 1671 (1956).
12. Sixsmith, H. 'Air bearing for high rotational speed'.
Machinery 90 June (1957).
13. Richardson, H.H. 'Static and dynamic characteristics of compensated gas bearings'.
Memo. No. RM 7401-1, Mas Inst.
Tech. Cambridge, Mass. April (1957),
Trans. Amer. Soc. mech. Engrs 80,
No. 7 (1958) 1503.
14. Sasaki, T.,
Mori, H. and others. 'Survey of gas-lubricated bearing research in Japan with recent developments in the study of externally pressurized bearings'.
First Int. Symp. Gas-lubricated Bearings, Washington D.C. Oct. (1959) 266.
15. Fox, G.R. and
Sneck, H.J. 'Orifice flows in externally-pressurized gas bearings'.
First Int. Symp. Gas-lubricated bearings, Washington, D.C. Oct. (1959) 482.
16. Laub, J.H. 'Evaluation of externally-pressurized gas pivot bearings for instruments'.
First Int. Symp. Gas-lubricated Bearings, Washington, D.C. Oct. (1959) 435.
17. Laub, J.H. 'Hydrostatic gas bearings'.
Trans. Amer. Soc. mech. Engrs,
J. Basic Engng 82 Ser. D, No. 2
(1960) 276.
18. Allen, D.S.
Stokes, P.J.
Whitley, S. 'The performance of externally pressurized bearings using simple orifice restrictors'.
Trans. Amer. Soc. lub. Engrs 4,
No. 1 (1961) 181.

19. Laub, J.H. 'Externally pressurized journal gas bearings'.
Trans. Amer. Soc. lub. Engrs 4,
No. 1 (1961) 156.
20. Powell, J.W.
Tully, N. 'Whirl instability of externally
pressurized gas journal bearings'.
Engineer, Lond. 213, No. 553 (1962)
1121.
21. Mori, H. 'A theory of an externally pressurized
circular thrust gas bearing with
consideration of the effects of
lubricant inertia'.
Amer. Soc. mech. Engrs paper
62-LUB 13.
22. Allais, D.C. 'The design of externally pressurized
gas thrust bearings for maximum
stiffness and stability'.
Trans. Amer. Soc. lub. Engrs 5,
No. 1 (1962) 245.
23. Gross, W.A. 'Investigation of whirl in externally
pressurized air lubricated journal
bearings'.
Trans. A.S.M.E., March 1962.
24. Tang, I.C., and
Gross, W.A. 'Analysis and design of externally
pressurized gas bearings'.
Trans. Amer. Soc. lub. Engrs 5, 1962.
25. Licht, L., and
Elrod, H. 'A study of the stability of
externally pressurized gas bearings'.
Journal of Applied Mechanics, Vol. 27
1960.
26. Sheinberg, S.A. 'Vibrational resistance of gas
lubricated bearings'.
Russian Engineering Journal. No. 6,
1961.
27. Katto, Y., and
Soda, N. 'Theory of lubrication by compressible
fluid with special reference to air
bearing'.
Proceeding Second Japan National
Applied Mechanics. 1952.
28. Larson, R.H., and
Richardson, H.H. 'A preliminary study of whirl
instability for pressurized gas
bearings'.
Journal of Basic Engineering. 84,
1962.

29. Sheinberg, S.A. 'Internal grinding spindles with air lubricated bearings'. E.N.I.M.S. 1959.
30. Cherubim, J. 'Problems associated with the application of gas lubricated bearings'. Amer. Soc. Mech. Engrs Paper. Spring Lubrication Symposium, 1963.
31. Sasaki, T., and Mori, H. 'On the characteristics of air bearing'. Mem. Fac. Engng, Kyoto University. Vol. 13, No. 1, 1951.
32. Sugimoto, Y. 'The theory of air bearing'. Trans. Japan Society Mechanical Engineers. Vol. 17, No. 63, 1951.
33. Shires, G.L., and Dee, C.W. 'Pressurized fluid bearings with inlet slots'. Gas Bearing Symposium. Southampton University. 1967.
34. Licht, L. 'Stability of externally pressurized gas journal bearings'. Franklin Institute. Report No.1-A2049-8 October 1959.
35. Lund, J.W. 'Calculation of stiffness and damping properties of gas bearings'. Trans. A.S.M.E. 68-LubS-19. October 1968.
36. Makarov, V.A. 'The design of aerostatic thrust bearings'. Russian Engineering Journal. Vol. XLVII. No. 9. 1967.
37. Raimondi, A.A., and Boyd, J. 'An analysis of orifice compensated hydrostatic journal bearings'. A.S.M.E. Paper No. 54 - Lub - 17.
38. Mori, H., and Mori, A. 'On the stabilizing methods of externally pressurized thrust gas bearings'. Trans. A.S.M.E. Paper No. 66 - Lub/5. July 1967.
39. Woodrow, J. 'Critical speeds of rotor in gas journal bearing'. U.K.A.E.A., A.E.R.E., E/R663, 1951.

40. Sasaki, T., and Mori, H. 'Theory of journal air bearing'. Trans. Japan Society Mechanical Engineers. Vol.19, No.6. 1953. Translated by Jackson, D., U.K.A.E.A. IGRL - T/CA - 86, 1959.
41. Lund, J.W. 'The stability of an elastic rotor in journal bearings with flexible, damped support'. Journal of Applied Mechanic, Trans. A.S.M.E., Series- E87, 911.
42. Whitley, S., and Betts, C. 'Study of gas lubricated, hydrodynamic, full journal bearings'. Brit. J. Appl. Phys. Vol. 10, 1959.
43. Tondl, A. 'The motion of a journal in a bearing in the unstable region of equilibrium position of the centre of the journal'. Ninth International Congress of Applied Mechanics. 1957.
44. Whitley, S., Bowhill, A.J., and McEwan, Miss P. 'Half speed whirl and load capacity of hydrodynamic gas journal bearings'. Proc. Instn. Mech. Engrs. Vol. 176, No. 22. 1962.
45. Cunningham, R.E., Fleming, D.P., and Anderson, W.J. 'Experiments on rotating externally pressurized air journal bearings. Part 1 - Load capacity and stiffness'. National Aeronautics and Space Administration. No. TN D5191, Washington D.C. 1969.
46. Castelli, V., and Elrod, H.G. 'Solution of the stability problem for 360 deg. self acting, gas lubricated bearings'. J. Basic Engineering, Trans. A.S.M.E. Series D, 87. 1965.
47. Fischer, G.K., Cherubim, J.L., and Fuller, D.D. 'Some instabilities and operating characteristics of high speed gas lubricated journal bearings'. A.S.M.E. Paper No. 58-A-231, 1958.
48. Sternlicht, B., and Winn, L.W. 'On the load capacity and stability of rotors in self acting gas lubricated plain cylindrical journal bearings'. J. Basic Engineering, Trans. A.S.M.E. Paper No. 62-LUB-8. 1963.

11.1. FUTURE WORK.

This investigation indicates that further work to be carried out. Some research work has been recommended for the future; it is desirable to investigate this theoretically and experimentally in more detail.

The effects of belt whipping and tension on the air bearing system could be further investigated and to avoid this problem it is desirable that some consideration is given to drive the spindle by an air turbine.

The complete cycles of grinding tests could be carried out and their results should be analysed. Moreover, the test results could be compared with a conventional grinding head regarding their general performance, surface finish, roundness and wheel wear, etc.

The unbalance effect of the shaft deserves detailed investigation, especially the advantage when balanced in its own air bearings or in a pair of slave bearings.

Accurate experimental data is scarce on the verification of stability theory. It would be desirable therefore, to use the same shaft and bearings with different clearances and orifice hole diameters and conduct experiments to verify the theory.

The trend in internal grinders is to operate at higher speeds, which introduces the hybrid effect; this deserves considerably more theoretical and experimental study.