OPERATING CONDITIONS OF JOURNAL BEARINGS

UNDER FLUCTUATING LOADS

by

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RÉSUMÉ

At the outset, since no previous practical work was available, it was necessary to evolve apparatus and instruments with which investigation of fluctuating loads on journal bearings could be commenced.

Since the problem is beset by many mechanical difficulties, it seemed expedient to make some simplification, and this was achieved by employing a larger diametral clearance than is commonly found in practice, and also by omitting friction measurement.

The main object was to plot the pressure distribution throughout the bearing, and towards this end a balanced-pressure-piston type of instrument was devised. Transient pressures could be accurately measured whether above or beneath atmospheric pressure, and since the element was placed in the shaft and the apparatus so constructed that the shaft could be traversed axially, the whole area of the bearing could be searched.

Fluctuating loads were applied to the bearing in two directions by spring and lever systems, and sensitive instruments were devised to measure the applied loads./

loads. Displacement of the bearing was measured in two directions, again with an instrument specially developed, and like the other instruments, capable of giving instantaneous readings. All these instruments contain electrical contacts and operate in a low voltage circuit with headset. Simultaneous measurements of pressure distribution, applied load, and attitude were obtained.

Tests were made with a single fluctuating load and also with both loads operating to give a rotating load effect. Investigations were made with shaft stationary, shaft rotating at same speed as load, and with shaft rotating at twice speed of load. Good agreement was obtained between integrated pressure load and measured load.

In the first tests, the bearing was submerged in an oil bath with no other oil supply, but tests were later made with oil supplied under pressure.

It was found that, in general, the bearing does not remain full of oil, but is cavitated. A purely pulsating load produces a large low pressure region on the unloaded side of the bearing in which the pressure approaches absolute zero. A rotating load, which is similar to the normal case of a bearing under steady load, also gives a large low pressure region covering more than half the bearing surface, but in this case the pressure is only slightly beneath atmospheric/

atmospheric pressure with the exception of a narrow region following the pressure peak in which the pressure approaches absolute zero.

The oil film in a submerged bearing is gradually displaced by air which comes out of solution and is trapped in a low pressure region of the bearing, but in some cases a stable condition is reached. When oil is supplied under pressure through an oil hole, cavitation is not eliminated, but a larger quantity of oil is maintained in the bearing and the load carrying capacity thereby increased.

While the load carrying capacity was found to be considerably less than that indicated by theory, it was shown that the critical case predicted by theory does exist; this occurs when the shaft rotates at twice the speed of the load, the oil film supporting no load under these conditions.

1. INTRODUCTION

It has long been appreciated that the plain journal bearing was capable of sustaining shock loads without damage and that in general a bearing of this type can support a fluctuating load of greater magnitude than a steady load. Engine bearings, being subject to inertia and gas pressure loads, are common examples, the gudgeon pin bearings being of particular note since they carry tremendous unidirectional loads with only a small oscillatory rotation. And yet, until recent years, apart from theoretical investigation, no information has appeared upon this aspect of lubrication, all effort having been confined to conditions of steady load.

From the number of searching references to the lack of knowledge of the behaviour of bearings under fluctuating load, made in the General Discussion of Lubrication[7], it is evident that considerable interest is attached to this study. The investigations of Mr.W.J.Harrison[3] and Professor H.W.Swift[6] show that a critical condition exists when a bearing is subject to a load rotating at half the speed of the journal, when no pressure film is formed and metallic contact will result: this is a direct contradiction of the/

the statement of general belief made above, and these conditions may exist in practice. It has been noted by Newkirk and Grobel[5] in their investigation of "Oil Film Whirl" that when a shaft runs at twice its critical speed or at any higher speed, a whirl develops having a frequency equal to the natural frequency of the shaft, and that the motion is one of resonance.

In the constant load field, considerable attention has recently been given to the determination of the boundary conditions of a lubricating film, notably by Dr.A.Cameron[2]. In the above mentioned theoretical analysis* of fluctuating load, this difficulty did not arise and absolute pressures were not evaluated, it being assumed that the bearing was full of oil.

There are already in the field of lubrication many practical difficulties in making measurements; these are increased when fluctuating loads are considered, and the field is greatly extended. Some of the measurements required in an investigation are: film pressure, temperature and thickness, load and friction moment, all of which are transitory, requiring new instrumentation and construction of suitable apparatus. In designing a machine, there is a choice of spring, inertia, or hydraulic loading; it is/

* The basis of this theory is set out in Appendix 1.

is possible to arrange either a pre-determined periodic load or displacement, or to have a compromise in which load and attitude are coupled. In practice, a bearing is required to support a certain load irrespective of its attitude, whether there is metallic contact or not, but in some instances load and attitude are coupled as for example, the main bearings of an engine, by the stiffness of the crankshaft. The case in which attitude is independent of load is not found in practice, but might nevertheless be a convenient arrangement for the purpose of investigation.

This research was directed towards the evolution of suitable apparatus and instruments for the investigation of fluctuating loads, and to a study of the oil film under various conditions.

2, APPARATUS

2.1 GENERAL DESCRIPTION

The apparatus is entirely novel, and was designed with the following considerations in view:

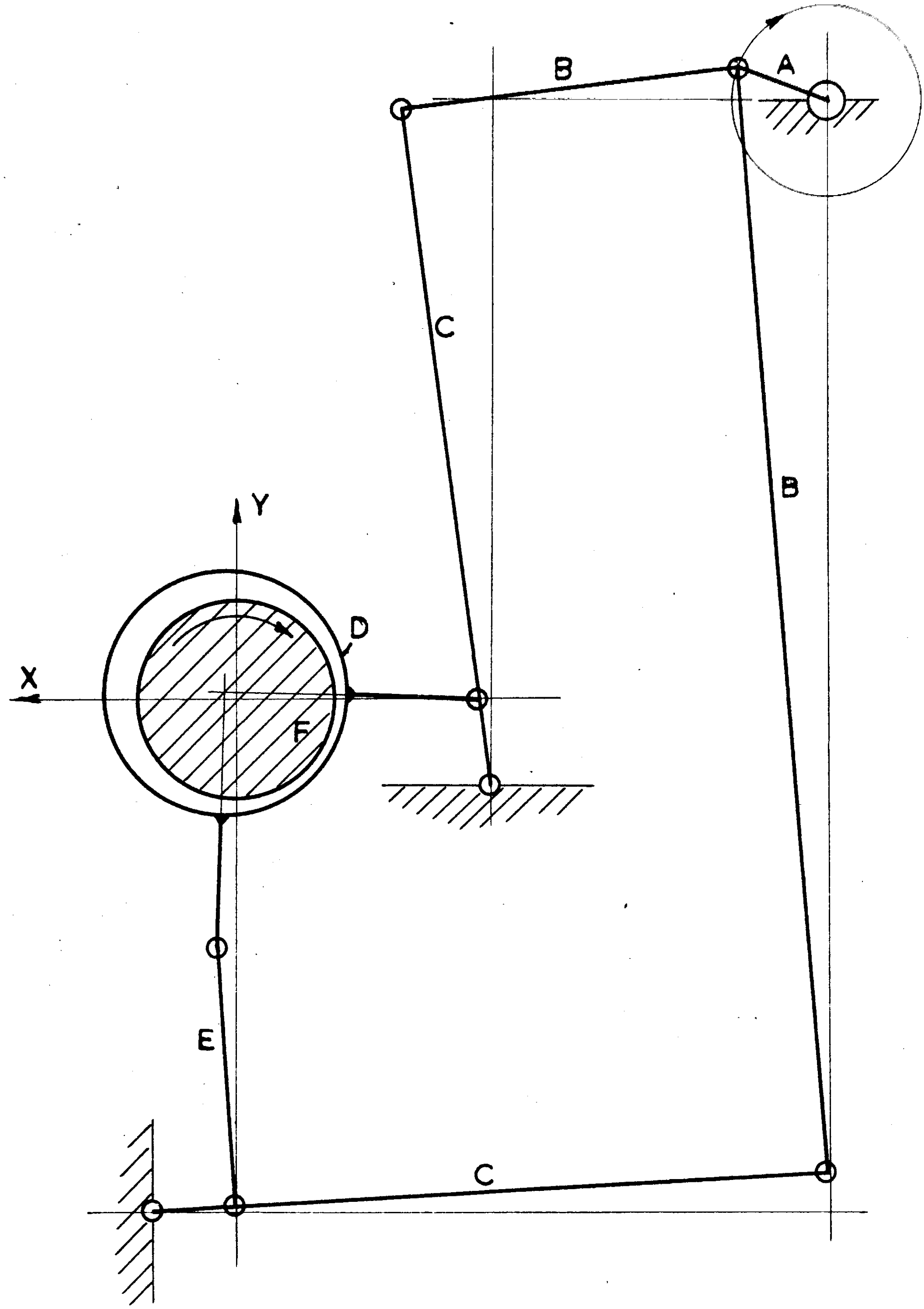
(1) A high degree of adaptibility should be embodied in the design so that a wide range of conditions could be investigated.

(2) As major modification might be necessary in the initial stages, the apparatus should be of simple construction and its size no larger than necessary.

(3) A bearing with abnormally large clearance could be employed with advantage in this first experimental investigation.

As the mechanism is very compact, a diagram has been prepared, see Fig.1, to illustrate the principle of operation. The eccentric A, operates two connecting rods B which are attached to levers C. These levers load the bearing in the horizontal and vertical directions. A link E is introduced between the bearing and the lever of the vertical loading system so that the vertical and horizontal movements will be independent.

The eccentric A is driven from a countershaft by/



LINE DIAGRAM OF MECHANISM

FIG. 1

by chain and sprockets: the journal F is also driven from the countershaft, so that its speed may be fixed in any desired ratio relative to A.

The bearing has a diameter of 1 in. and length of 1.5 in., and a diametral clearance of 0.010 in., thus taking advantage of easy attitude measurement, smaller loads, eliminating temperature effects in the oil film, and permitting the shaft to be traversed axially for pressure measurement. The bearing is submerged in an oil bath, the oil being circulated and maintained at constant temperature.

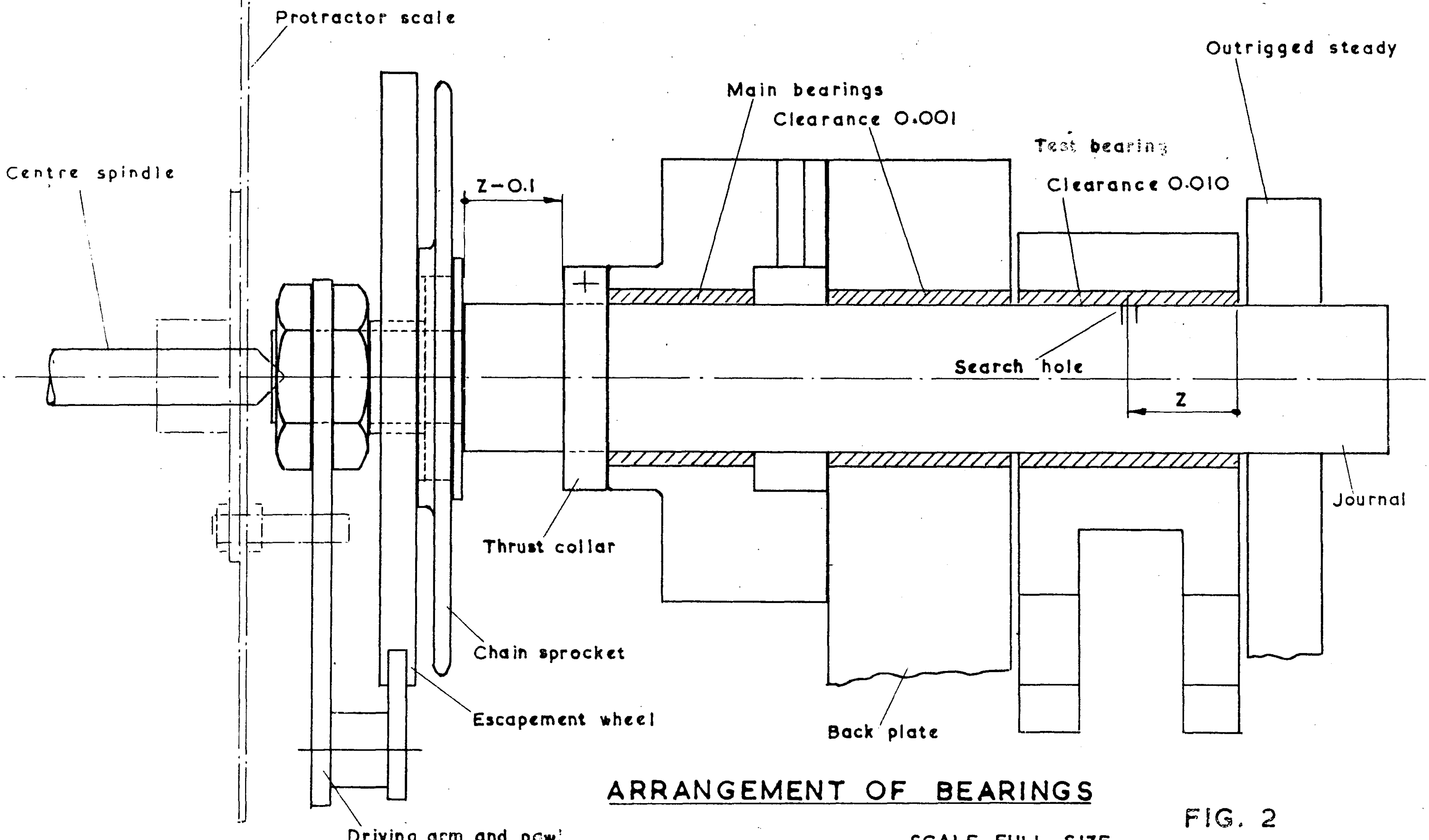
A pressure measuring device is built into the shaft enabling instantaneous values of film pressure to be measured throughout the whole bearing. Provision is also made for measurement of instantaneous load and attitude.

In the following sections, the apparatus is described in detail.

2.2 MECHANICAL ARRANGEMENT

2.21 Journal and bearings

The shaft is of constant diameter throughout its length to permit axial movement, and is made from mild steel and mounted in bronze bushed bearings. The main bearings, mounted in a substantial back plate, are bored to a diametral/



ARRANGEMENT OF BEARINGS

SCALE FULL SIZE

FIG. 2

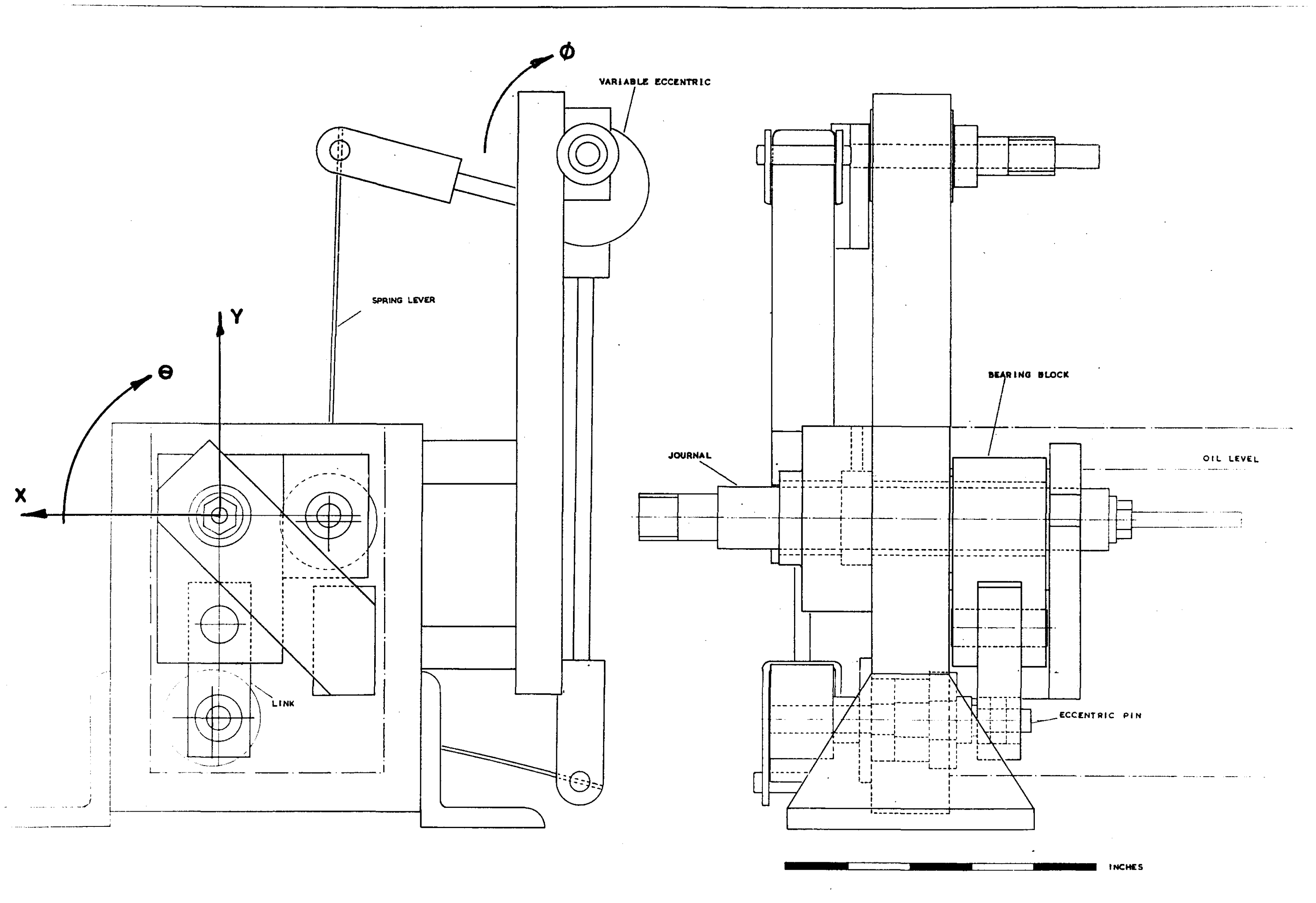
diametral clearance of 0.001 in., ensuring great support compared with the 0.010 in. clearance of the test bearing. In addition, the outer end of the shaft is carried in an outrigged steady, bored to the diameter of the shaft, split and clamped for wear adjustment, and containing three axial oil grooves. The arrangement and relative proportions of the bearings can be seen in Fig.2.

The shaft is located axially by an adjustable thrust collar and also by a conical centre running in the end of the shaft; these are also shown in Fig.2.

2.22 Loading system

A variable throw eccentric*, consisting of two discs clamped together at the centre by a set screw, one disc being fixed eccentrically to the driving shaft, the other carrying the crank pin at the same eccentricity, runs at constant speed and operates two connecting rods; these rods operate the spring levers, see Fig.3. The spring levers are securely clamped to torque spindles which pass through the back plate and terminate in eccentric pins, see Fig.9, ball bearing mountings being employed to reduce friction to a minimum. The torque spindle, eccentric pin, and spring lever is the practical form of the levers C of Fig.1. The/

* The construction of the eccentric is shown in Fig.13.



GENERAL ARRANGEMENT

FIG. 3

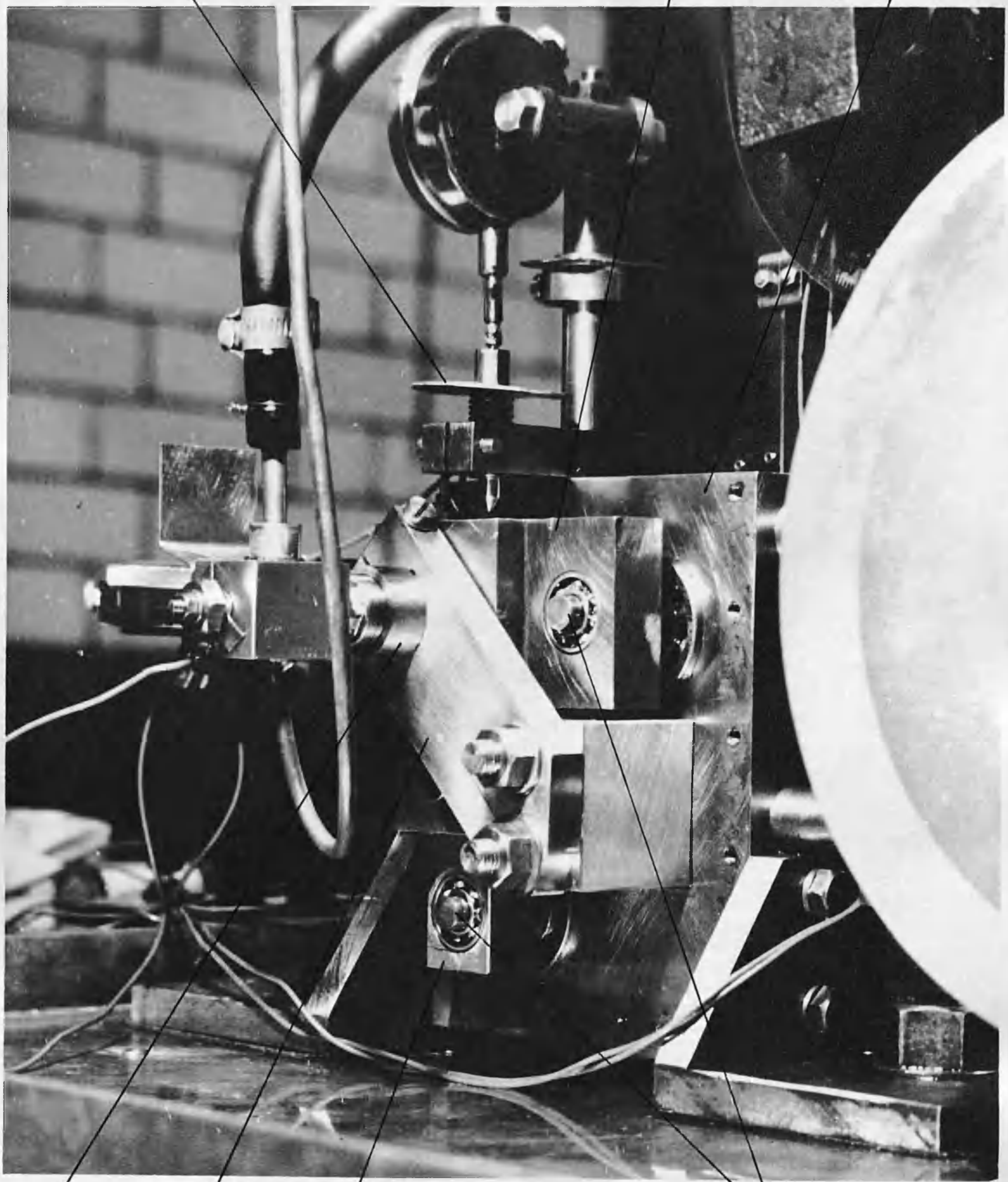


FIG.4

Oil tank removed to show assembly of mechanism

The eccentric pins load the bearing block directly in the horizontal direction and through a link in the vertical direction, a rotating load being produced when both loads are in operation. A three dimensional impression of the assembly may be obtained from Fig.4, which shows the apparatus with oil tank removed.

The inertia effects in the loading system are quite small and are fully treated in Appendix 2.

2.23 Drive

In order to ensure constant speeding throughout, a $\frac{1}{2}$ H.P., 1500 R.P.M. synchronous type motor was employed, providing ample power. It drives a countershaft through a 'V' belt giving a reduction of approximately 3:1. From the countershaft, the variable eccentric and journal are driven at the required speeds by chains and sprockets which maintain a fixed speed ratio. When required, the oil circulating pump is also driven from the countershaft.

To make it possible to search round the journal while in motion, an escapement and tripping mechanism was introduced, the chain sprocket being fixed to an escapement wheel having 36 teeth, the pair being freely mounted on the end of the shaft. An escapement pawl, pivoted upon the driving arm which is locked to the end of the shaft, engages/

engages the escapement wheel and so drives the shaft. The arrangement of this mechanism can be seen in Fig.5. The escapement pawl 1, which engages with the escapement wheel 2, is pivoted upon the driving arm 3, and operated by the bell-crank-lever 4. The lower end of the bell-crank-lever 4 is attached to the fork 5 which runs in a groove in the bush 6. The bush 6 is free to slide along the spindle 7 which has a conical point and is thrust against the end of the main journal. The bush is operated by the bobbin 8. A tension spring 9 is attached to the bush 6 preventing the bush from rotating and at the same time acting as a return spring for the mechanism. Each time the mechanism is tripped, the shaft slips back 10° on its drive.

In tests with shaft stationary, the chain is removed and a protractor scale, mounted on spindle 7, Fig.5, engaged by a pin to the driving arm 3. This arrangement is shown in Fig.6. The pointer, against which the protractor scale is read, is shown more clearly in Fig.10. It is clamped to the vertical column carrying the dial gauge, and its edge, against which the scale is read, is set parallel to the main shaft so that it can be employed for all axial positions of the shaft./

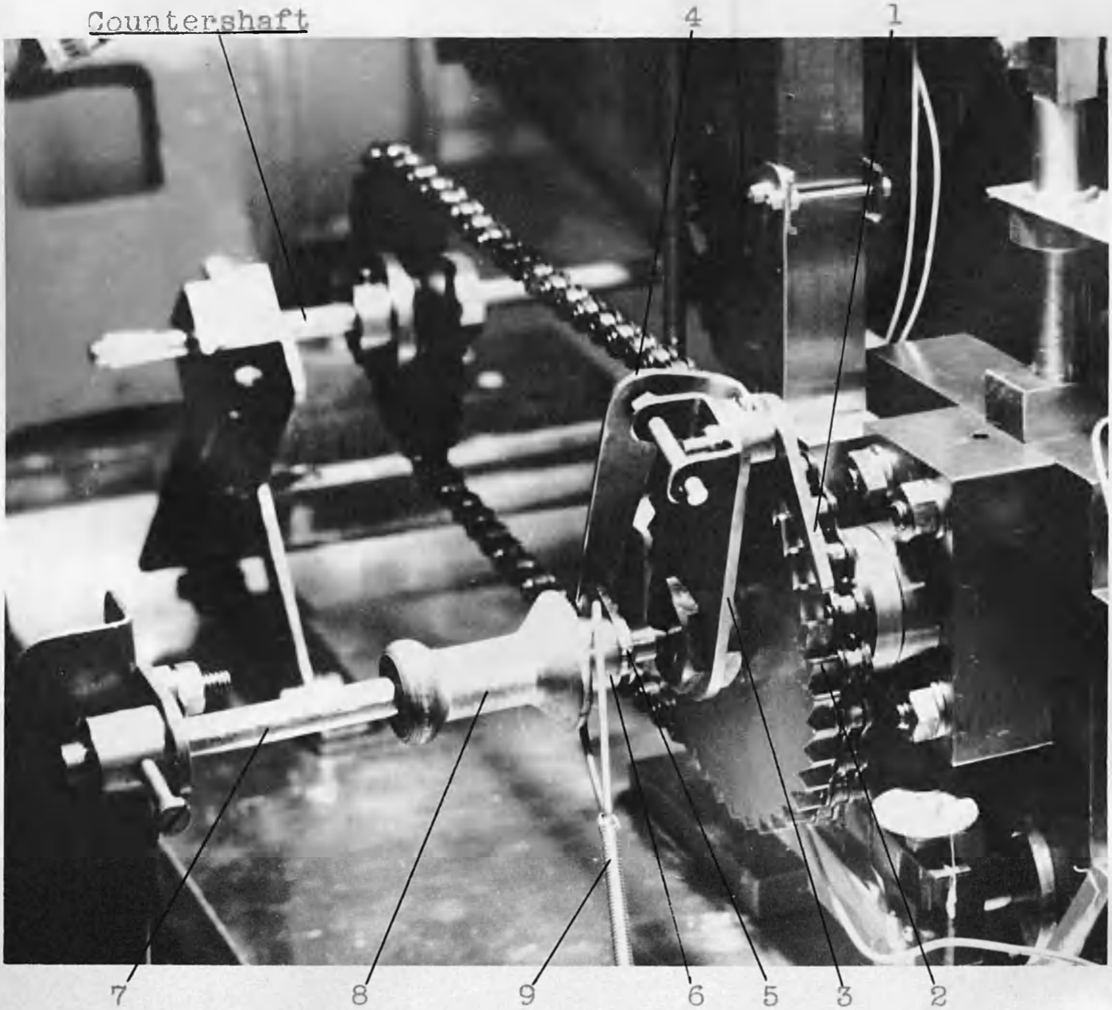


FIG.5

Escapement and Tripping Mechanism

1 escapement pawl

2 escapement wheel

3 driving arm

4 bell crank lever

5 fork

6 bush

7 centre spindle

8 bobbin

9 return spring.

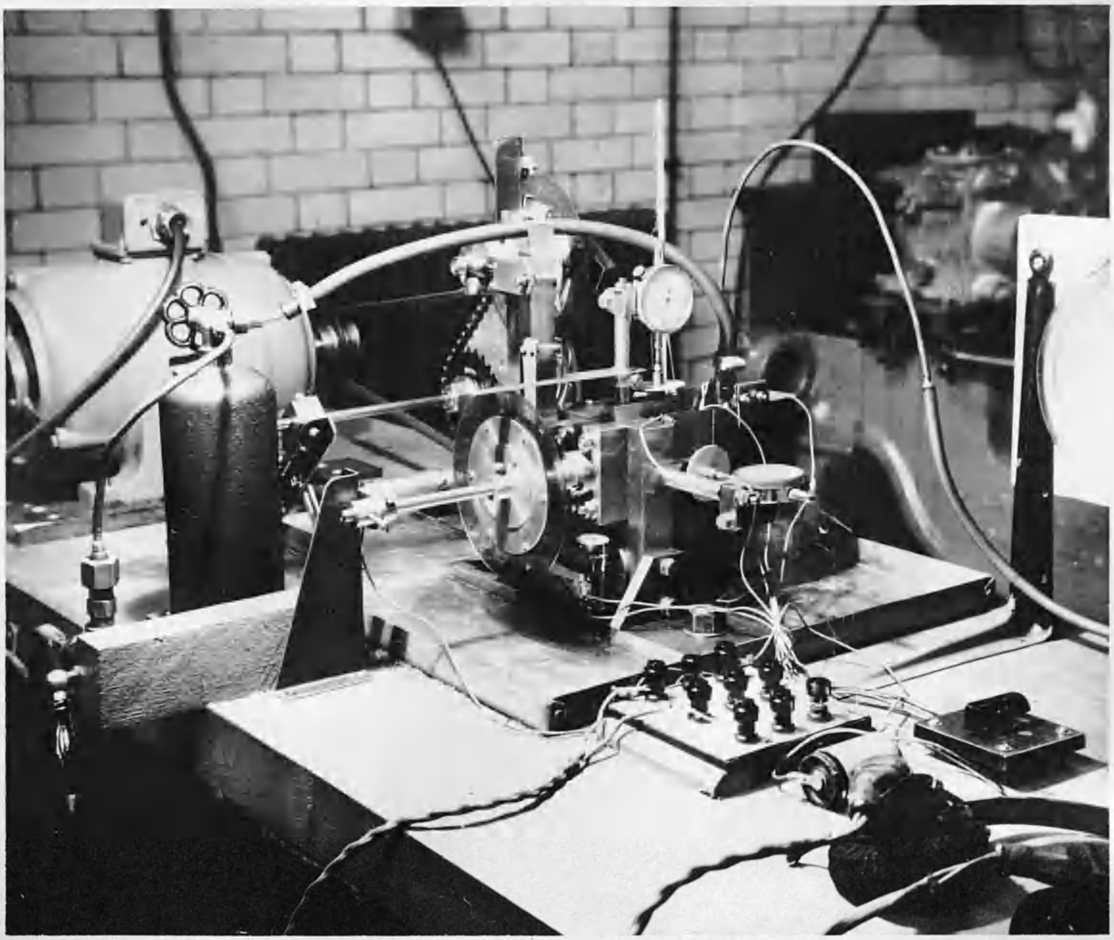


FIG.6

General view showing arrangement of apparatus. The shaft drive has been removed and protractor fitted for tests with stationary shaft.

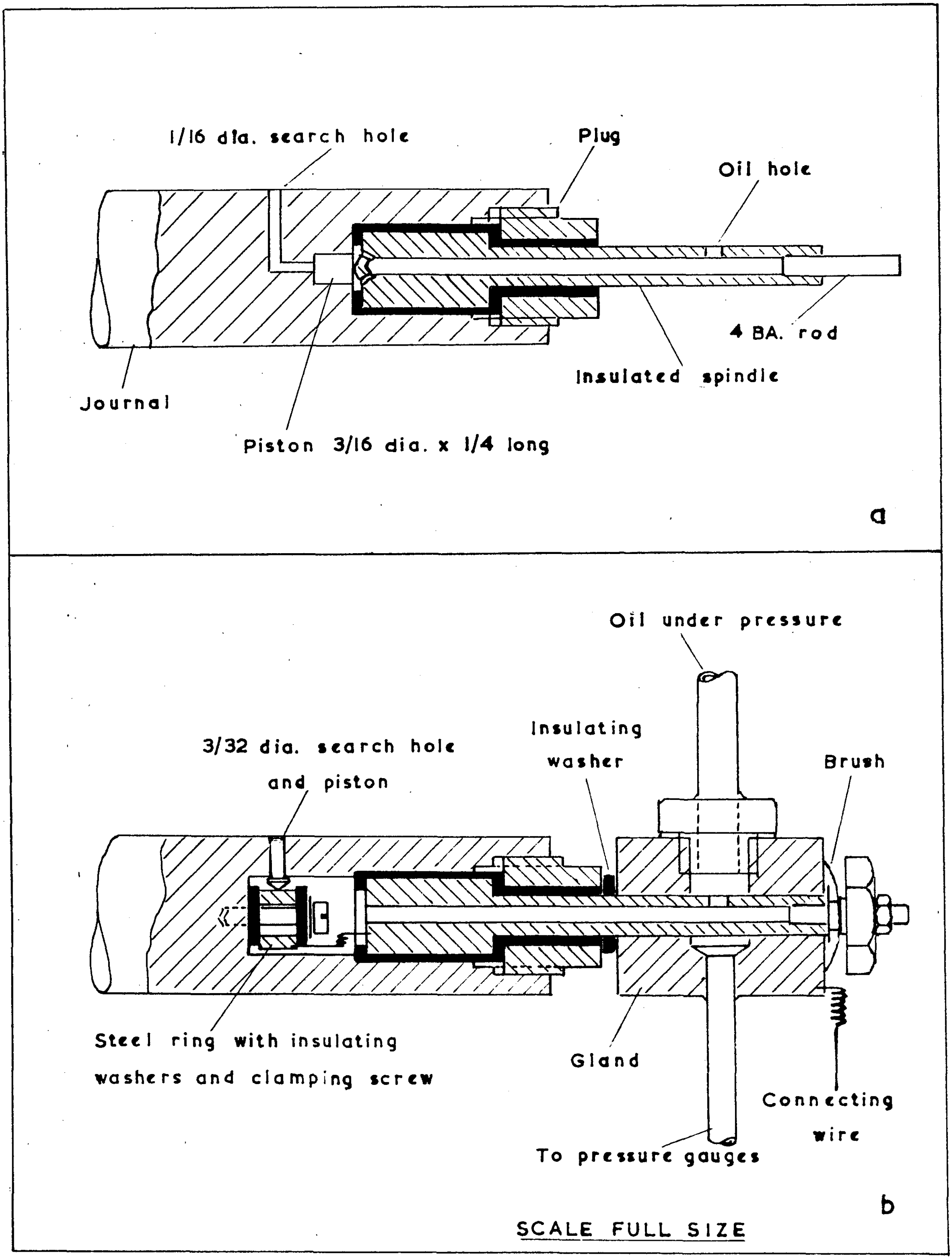
shaft.

2.3 BALANCED PRESSURE PISTON

2.31 Construction and development

While devised and developed independently, the instrument presented here is quite similar to that used by Buske and Rolli[1]. It consists essentially of a small free moving piston, one end of which is exposed to the oil film pressure at a particular point in the bearing while the other end is subject to a constant back pressure which may be varied at will, and according to the pressure difference it moves and makes or breaks an electrical circuit. This instrument has the advantage of simple construction and high sensitivity; the gap may be reduced to a very small amount so that very little oil is affected by the operation of the instrument.

During preliminary tests with the original form of this instrument shown in Fig.7a, completely unsatisfactory results were obtained on account of cavitation of the oil film. After some investigation it was deduced that the oil in the search hole frothed when subject to low pressure, so evacuating the oil and resulting in a considerable phase lag making the instrument useless. The remedy was to remove the oil space as completely as possible and the instrument/



BALANCED PRESSURE PISTON

FIG. 7

instrument was rebuilt as shown in Fig.7b. In this form, good agreement was obtained between the applied load and integrated pressure load.

The search hole was lapped to remove ridges and the piston carefully machined and lapped to fit with negligible friction so that it could be operated by extremely small pressures.

2.32 <u>Correction for centrifugal force</u>

Since the piston lies transversely in the journal, a centrifugal force will act upon it when the journal is in rotation. This force will act with or against the back pressure according as the back pressure is greater or less than atmospheric pressure, that is:

Film pressure = gauge pressure $\pm$ centrifugal pressure,

according as the gauge pressure, corrected for instrument error, reads pressure or vacumm.

The piston is made of steel (0.283 lb. per $in.^3$), and is 3/32 in. diameter and 5/16 in. long; radius to centre of gravity is 11/32 in.

Area of cross-section	0.00691 in^2.
Volume	0.00216 in^3.
Weight	0.000611 lb.

Centrifugal force acting on piston at 530 R.P.M. /

R.P.M. is therefore 0.00163 lb., and so the centrifugal pressure at this speed is 0.236 lb. per in^2.

The effect of centrifugal force may therefore be neglected at this speed.

2.33 Time lag in making contact

In the present design, a small time lag occurs when recording rising pressures due to the time required for the piston to travel the length of the gap; no such lag occurs when the pressure is falling as the piston moves and breaks contact immediately the film pressure is less than the back pressure.

Since the velocity of the piston is small, inertia forces are possibly the main factor; the quantity of oil required is small and therefore easily gathered from the oil film.

Assuming that the pressure increases uniformly with time, i.e. $\dot{p}$ = a constant, so that, $p = \dot{p}t$, the accelerating force F is equal to $a\dot{p}t$, where 'a' is the cross-sectional area of the piston and 't' is the time from rest. The motion of the piston may then be expressed by:

$$M\ddot{x} = a\dot{p}t,$$

where M is mass of piston and 'x' is distance of travel/

travel from rest. Integrating twice,

$$Mx = a\dot{p}t^3/6,$$

and hence, $$t = (6Mx/a\dot{p})^{1/3}.$$

't' may be regarded as the time lag, while 'x' is the gap.

In this form, the time lag is directly proportional to the cube root of mass and gap, and inversely proportional to the cube root of cross-sectional area and pressure gradient. The instrument is therefore shown to be suitable from a practical point of view as, for example, if the gap were increased eight times the time lag would only be doubled.

The range of pressure gradient in the following experiments is approximately 5000 to 500 lb. per in^2. per sec. and the table beneath is computed for this range, for a gap of 0.001 in. The gap was set to this amount by placing a piece of mica between the contacts during assembly.

$\dot{p}$	t-secs.	deg. at 530 R.P.M.
5000	.00065	2.07
4000	.0007	2.225
3000	.000771	2.45
2000	.000882	2.805
1000	.001111	3.53
500	.0014	4.45

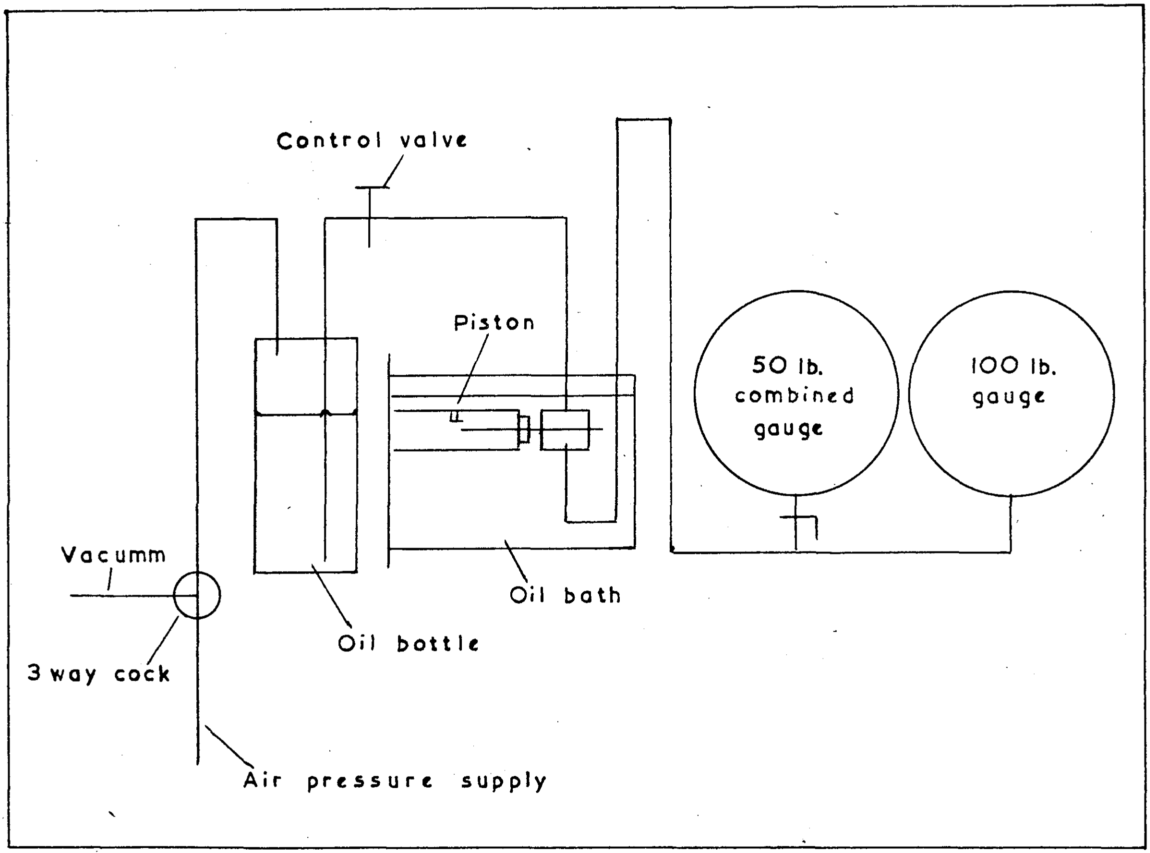
2.34 Pressure system

The back pressure for the balanced pressure piston is supplied from a bottle containing oil under air pressure, see Fig.8. It is led through a control valve and gland to the insulated spindle and so to the back of the piston, see detailed sketch Fig.7b. Adjustment of the valve gives the required balancing pressure. The gland consists of a block of brass reamed to fit the insulated spindle. It is bored out in the centre so that the incoming oil can communicate with the oil passage in the spindle, and has a pipe leading away from the bottom to the pressure gauges: this tapping is arranged as near as possible to the piston and is unaffected by the small leakage from the gland. It is important that the pipes should not be too small as oil froths under low pressure and results in sluggish movement of the gauges.

The oil in the bottle may be subjected to vacumm from a water-vacumm pump, pressure or vacumm being selected by a three-way cock. Since the gland is submerged in oil it is equally satisfactory for vacumm pressures.

2.4 LOAD INDICATORS

The spring levers are not rigid, but bend slightly/



BALANCED PRESSURE SYSTEM

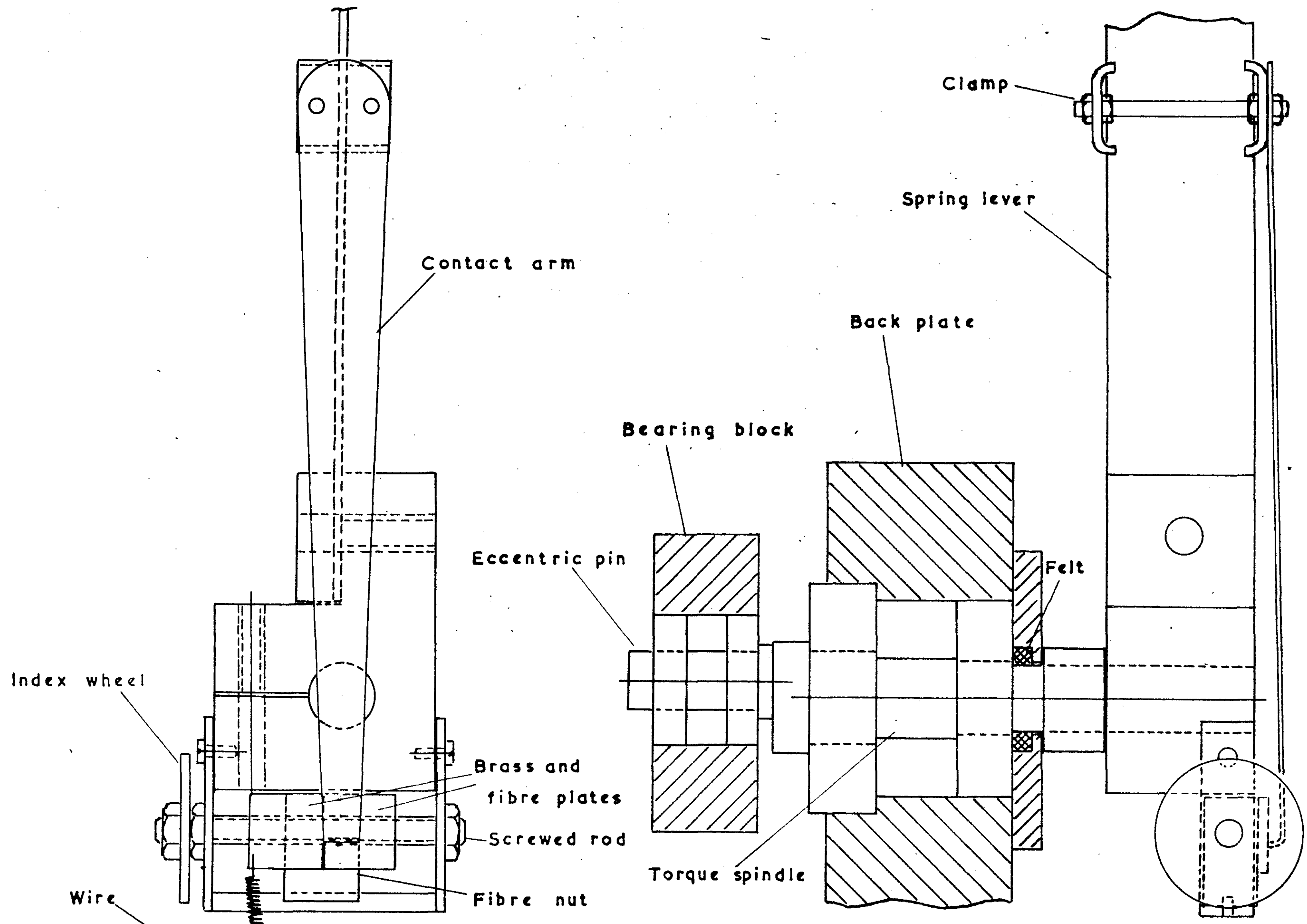
FIG. 8

slightly under load. An instrument was devised to measure the curvature at the fixed end of the spring, so indicating the applied load.

The construction of the instrument can be followed from Figs.9&10. A contact arm is clamped at a short distance from the fixed end of the spring by means of plates having 'V' notches cut in their inturned edges. When the spring flexes under load, the point of the arm moves over two plates, one of brass the other of fibre, mounted side by side upon a fibre nut; the nut runs upon a screwed rod operated by an index wheel thereby enabling the position of the plates to be adjusted. With increasing load the point of the contact arm moves over the fibre plate to make contact with the brass plate. The index wheel is divided into ten divisions with numbers increasing with increasing positive load, complete turns of the wheel being counted starting from an arbitrary figure, e.g. 5 turns or 50 divisions.

A load indicator of this type is incorporated in both horizontal and vertical loading systems: the instrument is robust and yet highly sensitive and gives a straight line calibration.

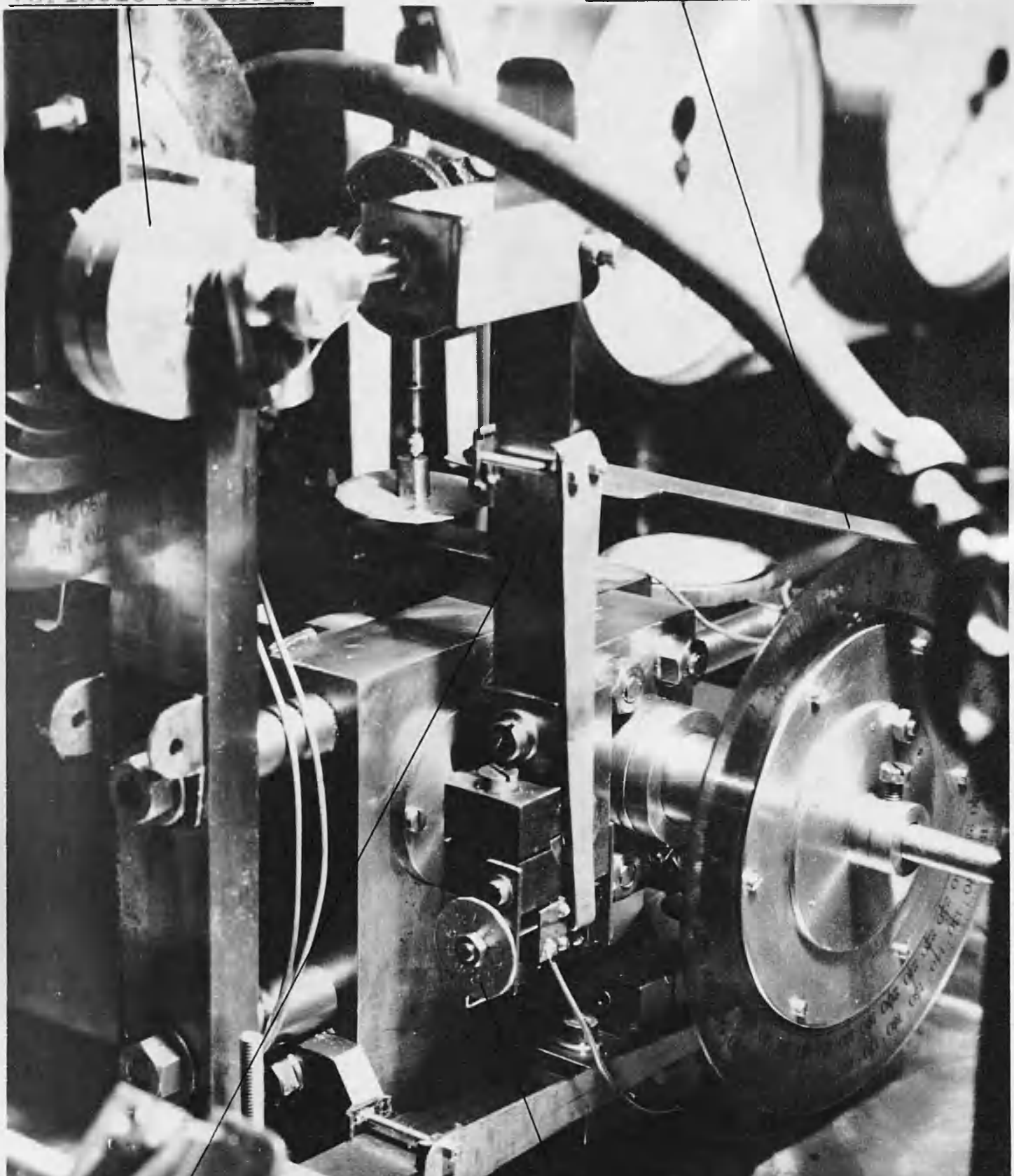
When the shaft is being driven, it is impossible to reach the index wheel of the X-indicator as it lies behind the chain drive; it is then operated by means of/



LOAD INDICATORS SCALE FULL SIZE FIG. 9

Variable eccentric

Pointer for angle θ



Spring lever

Load indicator

FIG.10

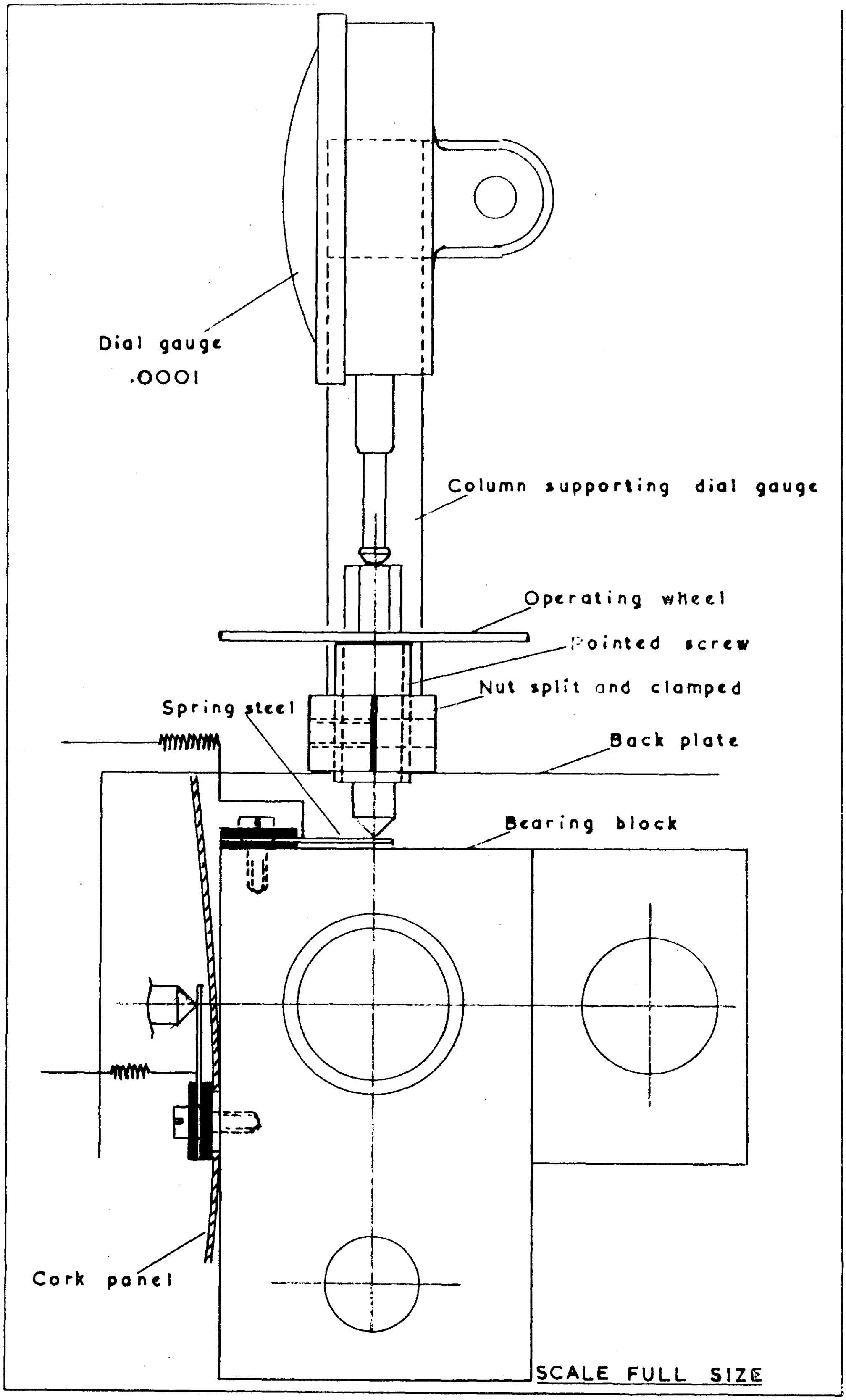
of a long rod with a two-pronged fork on its end, the prongs fitting into two holes drilled diametrally in the index wheel.

2.5 DISPLACEMENT INDICATORS

In the design of the machine, care was taken to ensure that the movement of the journal relative to the back plate would be small compared with the clearance of the test bearing, so that it would be sufficient to measure the movement of the bearing relative to the back plate.

The displacement indicators are constructed as follows: a piece of spring steel $\frac{3}{4}$ in. long by 3/16 in. wide and 0.017 in. thick, is clamped as a cantilever to the bearing block, but insulated from it, see Fig.11; a pointed screwed member mounted in a rigid attachment to the back plate makes contact with the spring completing an electrical circuit; a dial gauge rests upon the top face of the screw, which has been carefully machined true with the thread.

As the bearing moves up and down the spring makes and breaks contact with the point of the screw, and provided the natural frequency of the spring is high it can be assumed that contact is made and broken when the spring is in the unstrained position. The magnitude of the error is/



DISPLACEMENT INDICATORS

FIG. 11

is dealt with in Appendix 3. The dial gauge therefore indicates the displacement of the bearing at the instant when contact is made or broken. Two such instruments are employed, the attitude being derived from the combination.

Provided the contacts are clean and free from oil, as it is essential that electrical contact be made with very light pressure, these instruments read accurately to $\pm$ 0.0001 in., remaining stable over a long period. The stiffness of the spring is small and has negligible effect upon the loading of the bearing.

The contacts of the horizontal displacement indicator are kept free of oil by arranging a cork panel in the side of the oil tank, the spring of the indicator being clamped through a square hole cut in the centre of the panel, while the sides and bottom of the panel fit into guide slots in the side of the tank. The cork has the necessary flexibility to allow free movement and at the same time it effectively seals the oil.

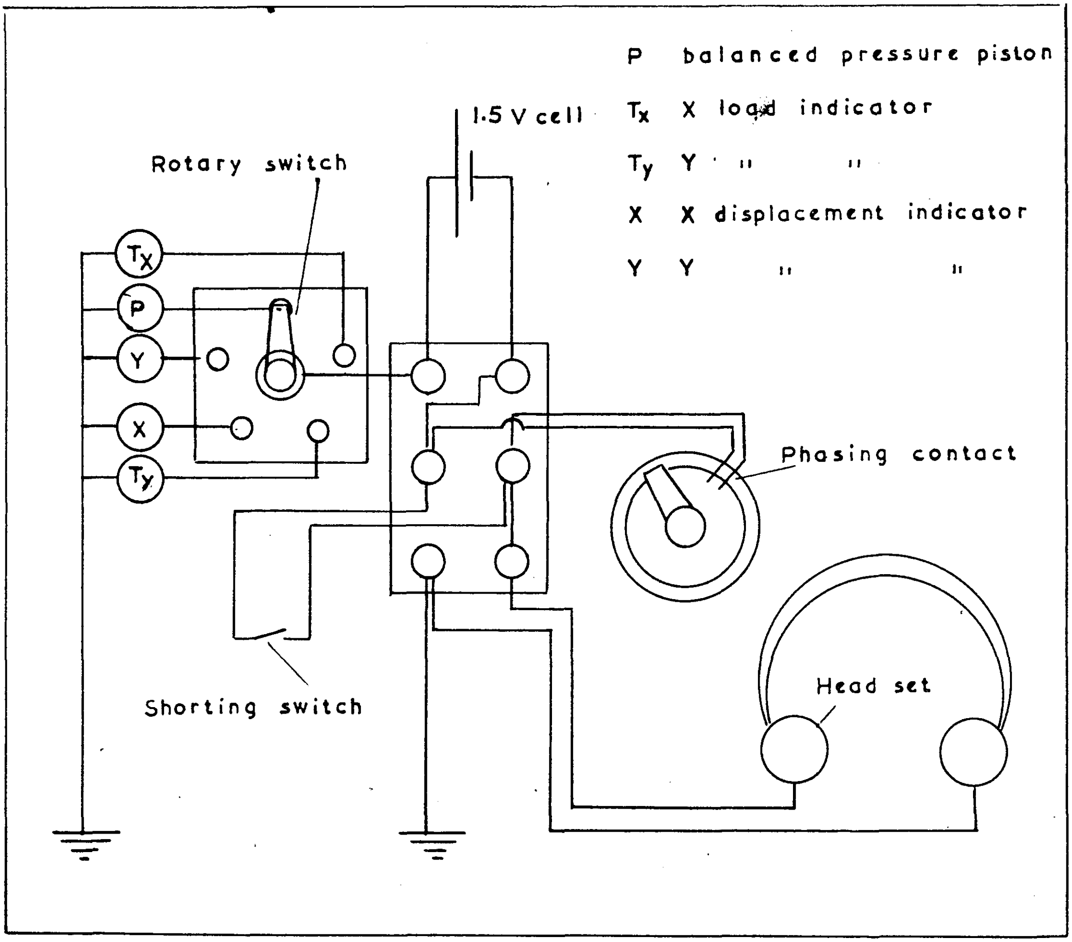
2.6 ELECTRICAL CIRCUIT

All five instruments, balanced pressure piston, horizontal and vertical load indicators and displacement indicators, are placed in parallel in a 1.5 V. circuit with/

with headset and selected by a five point rotary switch; the low voltage avoids damage by arcing at the contacts and the headset provides a sensitive, simple and reliable means of indicating electrical contact, see circuit diagram Fig.12. For demonstration purposes, the headset may be replaced by an amplifier and loudspeaker.

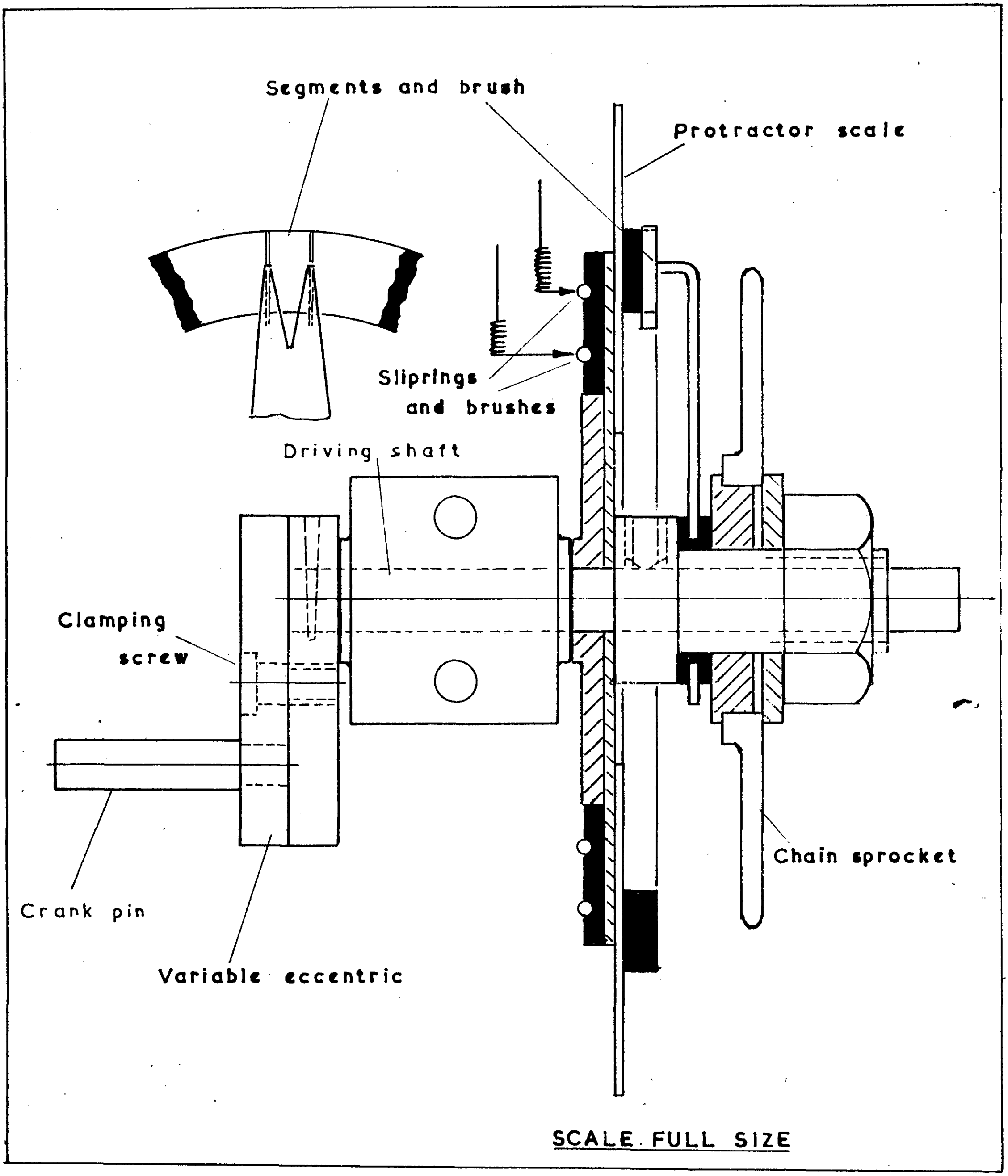
Included in the circuit is a phasing contact, detailed in Fig.13, consisting of a circular protractor scale carrying a fibre ring with two segments inserted, the whole being freely mounted upon the variable eccentric shaft and provided with a clamp so that it may be fixed in any desired position. An arm, rotating with the shaft but insulated from it, carries a brush which short-circuits the segments over a small arc of rotation thus closing the circuit for an instant in each rotation of the shaft. To allow continuous rotation of the scale, the current is carried to the segments through slip rings and brushes.

With variable loads, the pressure distribution, load and displacement change continuously. By setting the phasing contact to a particular value, a complete picture of pressure distribution, load and displacement may be obtained from the instruments for that particular instant in the load cycle. The angular setting of the scale serves to mark the phase of the load. The scale is also a time base since the/



ELECTRICAL CIRCUIT DIAGRAM

FIG. 12



PHASING CONTACT

FIG. 13

the shaft runs at constant speed.

2.7 OIL CIRCULATION

Since the test bearing is lightly loaded, has a large clearance and runs at moderate speed, very little heat is generated, and therefore the temperature gradient in the oil bath is small. In tests with stationary shaft, the oil requires initial warming and stirring, thereafter remaining at constant and uniform temperature: a 15 W. heating element is employed with a variable transformer to control the temperature of the oil bath.

With shaft rotating, there is a good circulation of oil but a little heat is generated in the main bearings. Adequate temperature control is achieved by use of the heating element and a water jacket surrounding the oil bath.

When oil is supplied under pressure, as in tests made with oil sprayed upon the ends of the bearing or in tests made with oil pumped directly into the bearing, an oil circulating system is employed. Oil is drawn from the bottom of the oil tank by a small force pump of $\frac{3}{8}$ in. bore and stroke driven by the countershaft, and forced through a coil of copper pipe contained in a water bath and thence to the bearing. The water bath gives accurate control of the supply temperature. For spray application, a jet pipe/

pipe is employed, see Fig.14a, having two jets, the distance between them being slightly greater than the width of the bearing block, so that the oil is sprayed down upon the journal at each side of the bearing. The jet pipe is fixed slightly above the bearing block, butting against the back plate. Oil temperature is measured by placing a thermometer between the bearing block and the back plate where it lies in the stream from the jet.

2.8 PRESSURE FEED

In the last experiments made, oil was pumped directly into the bearing through a 0.10 in. diameter oil hole at $\theta = 135^{\circ}$ in the central plane of the bearing, using the force pump and circulating system described in §2.7 above. The bearing block was drilled and tapped to take the $\frac{3}{8}$ in. bolt which has communicating oil passages drilled in it and carries the special union detailed in Fig.14b. Oil is led to the union by a length of flexible pipe so that movement of the bearing remains unrestricted. A spring loaded relief valve is incorporated in the union enabling the peak pressure to be adjusted.

This system gives an injection of oil once per cycle; the timing may be varied as required. Oil supply/

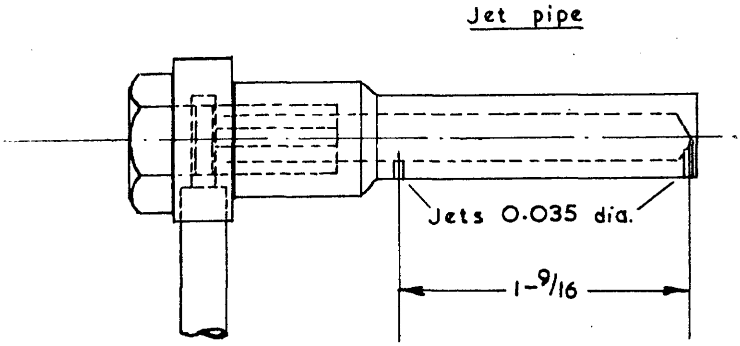
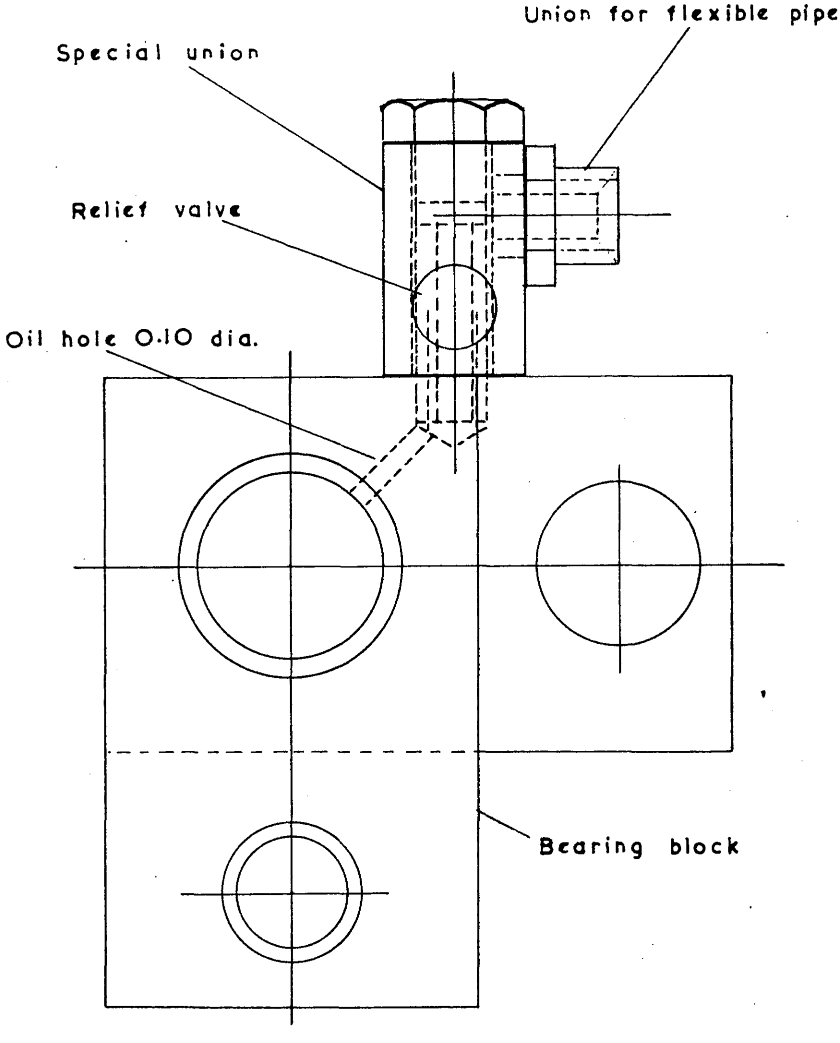


FIG. 14 a



JET AND PRESSURE FEED

SCALE FULL SIZE

FIG. 14 b

supply temperature is measured by placing a thermometer in the oil escaping from the relief valve.

3. EXPERIMENTAL PROCEDURE

3.1 CALIBRATIONS

3.11 Pressure gauges

Two pressure gauges were employed, a 50 lb./30 in. combined gauge and a 100 lb. gauge. These were calibrated by the usual gauge-tester employing dead weights, the vacumm scale being calibrated against a mercury column. All pressure readings were corrected before plotting.

3.12 Load indicators

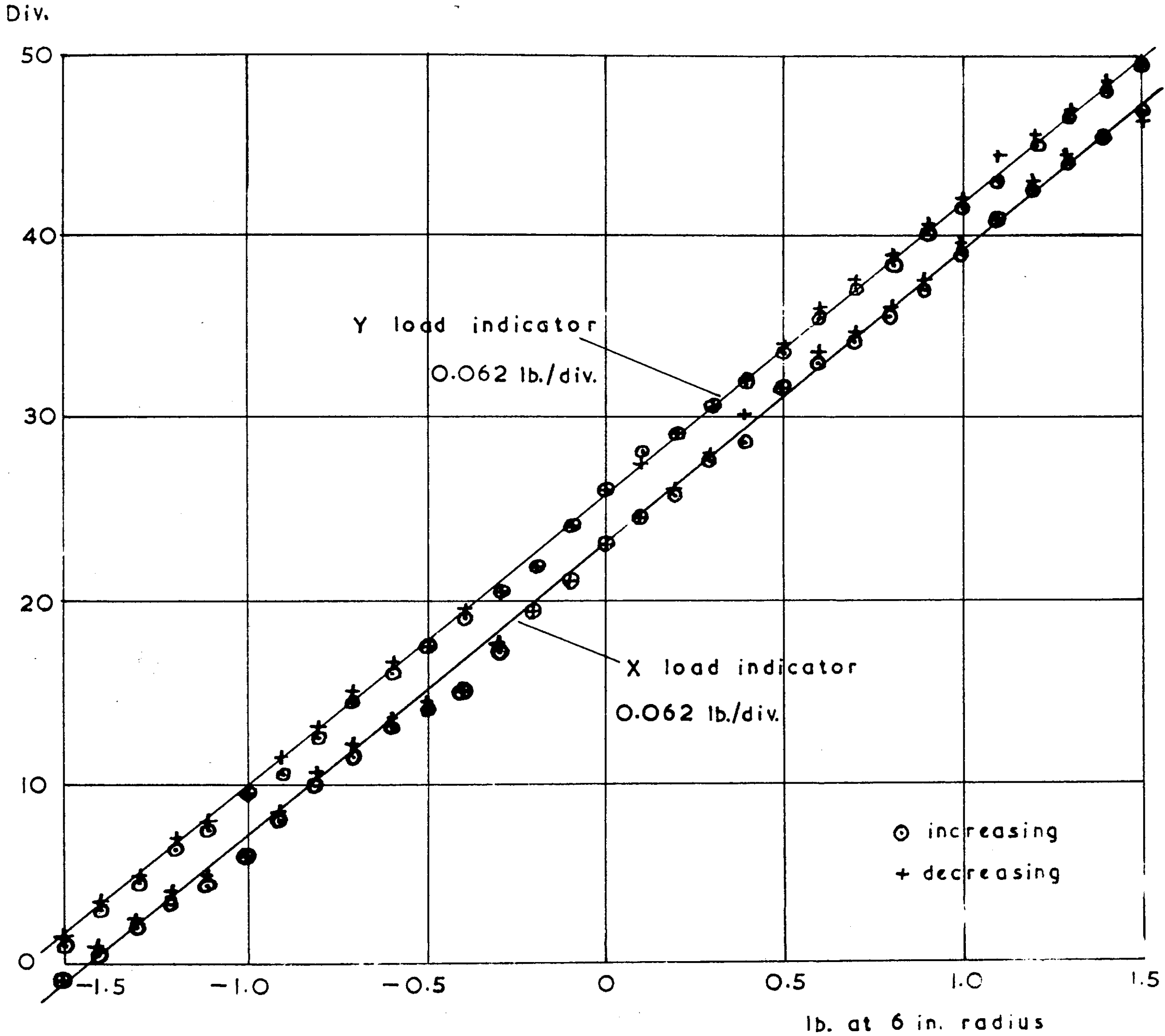
The product of torque in the torque spindle, i.e. load in connecting rod stirrup multiplied by radius 6 in., and angular displacement of spindle, is equivalent to the load applied to the bearing times the corresponding displacement of the bearing. This statement may be transposed, so that, load applied to bearing is equal to the product of load applied at stirrup and magnification ratio, i.e. displacement of stirrup to displacement of bearing, assuming that the spring lever is rigid./

rigid.

In operation, the spring lever is strained alternately in both directions and consequently the load indicator requires to be calibrated over the working range of positive and negative load. For this purpose the unit was removed from the machine and clamped upon its back to a table; a horizontal cord was attached at a radius of 6 in., passing over a pulley and carrying weights, first to one side of the lever then to the other for positive and negative loads. readings were taken at 0.1 lb. intervals up to 1.5 lb. with load increasing and decreasing. The electrical circuit was engaged, with the phasing contact short-circuited, and the index wheel was read to the nearest half division. Both units gave quite similar calibrations, see Fig.15, obeying a straight line elastic law, any slight departure being presumed due to inaccuracies in the screwed rods and guides. The actual numerical values read from the index wheel are quite arbitrary and liable to be altered by careless handling, but the rate is absolutely stable.

3.13 Magnification ratios

To determine the ratio of displacement of connecting rod stirrup to displacement of bearing, a dial gauge was placed upon one of the spring levers at measured radius,/



CALIBRATION OF LOAD INDICATORS

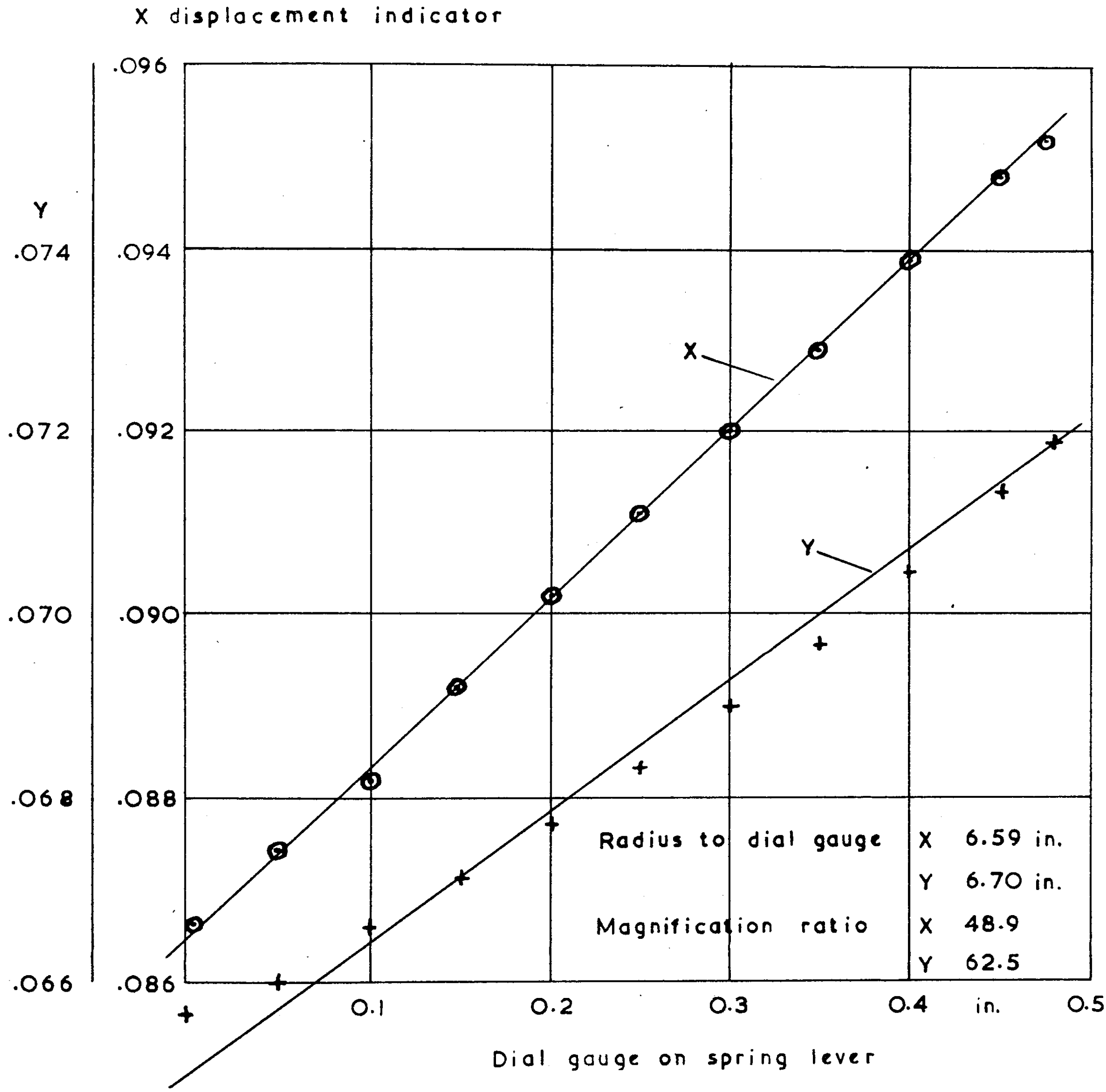
FIG. 15

radius, the connecting rod and stirrup having been removed, while the position of the bearing was measured by the corresponding displacement indicator; the other lever was set so that the bearing was in the centre of its clearance. Readings were taken over the centre part of the clearance by setting the spring lever at various points in its travel. The results are plotted in Fig.16, from which the ratio is calculated and then corrected to a radius of 6 in. These ratios should have been equal to 60 by design, but evidently machining errors have occurred.

3.14 Attitude of bearing

By removing the connecting rods from the spring levers, the limits of travel of the bearing in the horizontal and vertical directions may be recorded from the displacement indicators. This operation is carried out with the phasing contact short-circuited and the machine at rest. From these readings, the attitude of the bearing may subsequently be derived.

This calibration was always made immediately before and after a test, since dial gauges are easily disturbed by accident. With this calibration, it is a simple matter to set the machine so that the motion of the bearing takes place accurately in the centre of the clearance, or/



CALIBRATION OF MAGNIFICATION RATIOS

FIG. 16

or in any other desired position, and to ensure that metallic contact does not occur.

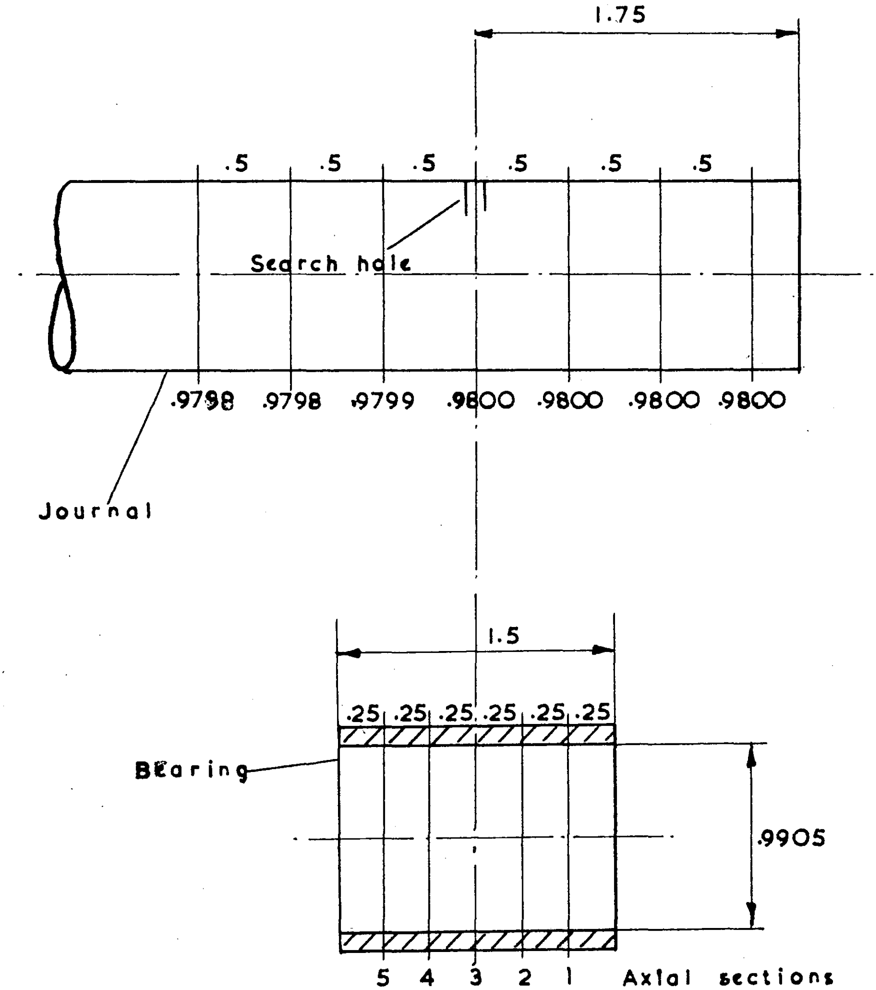
The travel of the bearing, measured in this manner, is usually slightly greater than the diametral clearance, presumably due to movement of the shaft in the main bearings.

3.15 Dimensions of journal and bearing

Although the design clearance of the test bearing was specified 0.010 in., this was checked by measurement. It was found that, as regards ovality and taper, both shaft and bearing were accurately machined, but the diametral clearance was measured as 0.0106 in., see Fig.17. This value is used in the calculations of theoretical performance.

3.16 Viscosity of oil

Since the bearing carries only light loads at moderate speeds and has a large clearance, there can be little variation of temperature throughout the oil film, and the oil film temperature should be close to that of the surrounding bath. In the tests presented, the oil temperature was maintained at 70°F., the corresponding viscosity being 2.9 poise. Measurements were made with a Redwood No.1 Viscometer and hydrometer over the range $70-140^{\circ}$F., see/



DIMENSIONS OF JOURNAL AND BEARING

FIG. 17

see Fig.18, and viscosity estimated by the formula:

Viscosity = Specific gravity(0.0026 Red. - 1.715/Red.) poise.

3.17 Speed of countershaft

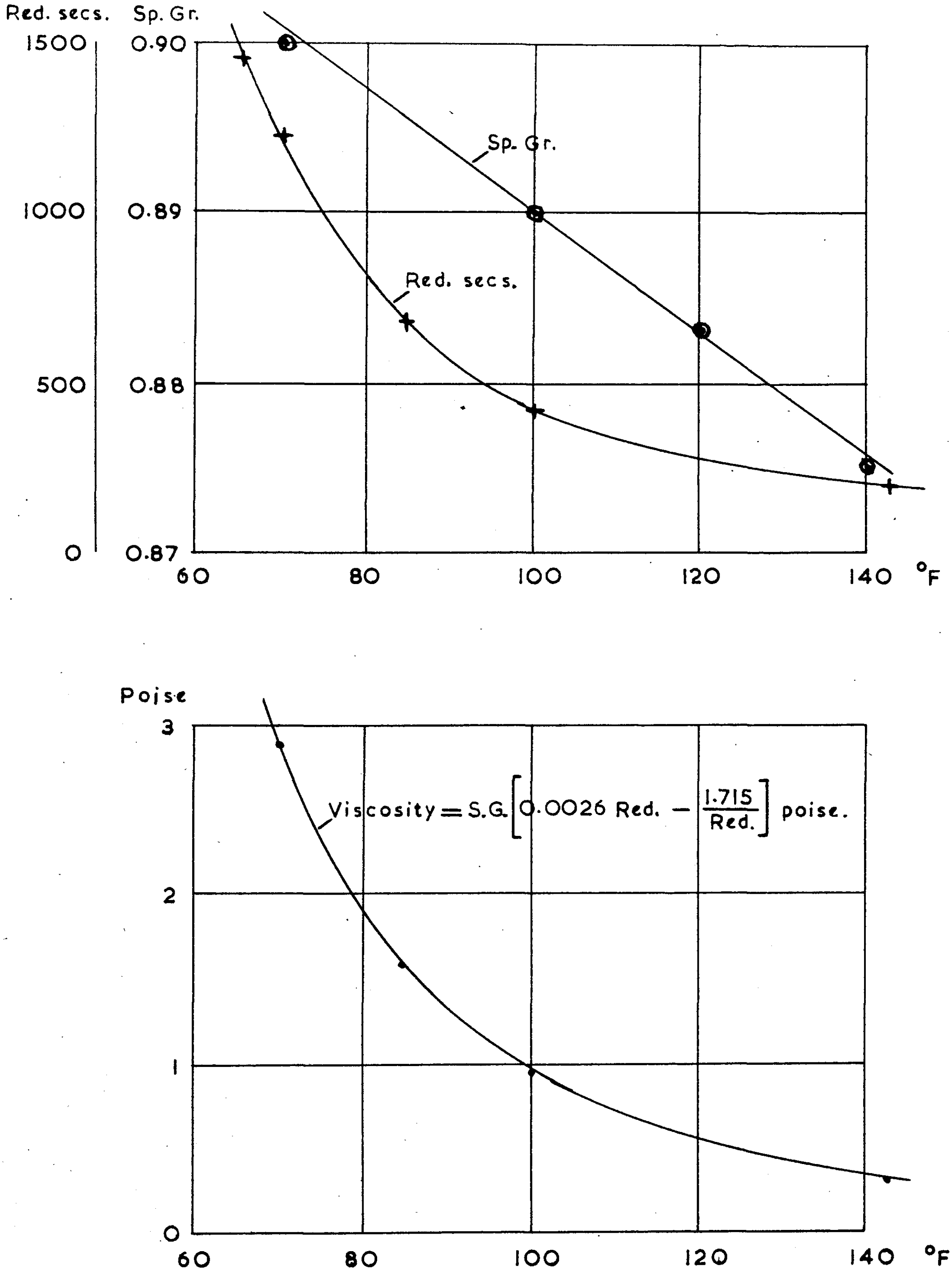
The speed of the countershaft, measured by a revolution-counter at a time when the mains frequency was exactly 50 c/s., was 530 R.P.M. Apart from small frequency changes, the motor being of synchronous type, this speed is invariable.

3.2 COORDINATES

3.21 Load and displacement

The bearing is loaded in the horizontal and vertical directions, denoted X and Y, the positive directions being indicated in Fig.3. The load indicators are arranged so that, with increasing load, the contact arm moves from the fibre plate to the brass plate; also, the index wheels are numbered 0 to 9, increasing with positive load.

The displacement of the bearing is also measured in directions X and Y and the arrangement is such that the dial gauge readings increase with displacement in the positive directions./



VISCOSITY OF OIL

FIG. 18

directions.

3.22 Load phase

Since the spring levers are operated by the variable eccentric which runs at constant speed, the angular position of the eccentric, shown by the scale on the phasing contact is an indication of the phase of the load. However, due to the flexing of the spring lever and other components, the load is not in phase with the eccentric.

The direction of rotation of the variable eccentric is shown by an arrow in Fig.3, and while the numerical value of the scale reading of the phasing contact is arbitrary, it indicates advance of load phase in this direction.

3.23 Angular position of search hole

The direction of rotation of the shaft is shown by an arrow in Fig.3, the angular position of the search hole, θ, being measured in this direction from the X direction except in the case of the first test, §4.1, in which θ is measured from the Y direction.

The driving arm 3, Fig.5, is securely locked to the end of the shaft, and one of its edges was used as a datum plane in angular measurements. The shaft was set in 'V' blocks upon a surface table and rotated to the position where the centre of the search hole was at the same height/

height as the centre of the shaft, i.e. search hole in horizontal (+X) position; the inclination of the datum plane was then measured. At any future time, when the datum plane is set horizontal by means of a small spirit-level, the zero error of the search hole, positive or negative, is therefore known. From this point, the procedure differs according to whether the shaft is being driven or not:

(a) Shaft stationary - With the protractor scale placed upon the centre spindle with pin engaged in the driving arm, see Fig.2, the shaft was rotated till the datum plane was horizontal and the scale reading noted. The zero error, described above, was then subtracted from this reading to obtain the error by which all scale readings must be corrected to obtain the true angle θ. The shaft and protractor rotate together and the scale reading of the protractor increases as θ is increased.

(b) Shaft rotating - With the pawl engaging tooth No.0/36 of the escapement wheel, the shaft was rotated till the datum plane was horizontal; next, the zero error, measured on the surface table, divided by the speed ratio S, ratio of shaft speed to load speed, was set on the phasing contact scale. The variable eccentric was then rotated till the brush short-circuited the segments. The chain sprockets were then made secure upon their shafts. Accordingly, the/

the true angle θ may be determined from the formula:

$$\theta = \phi . S + 10.n,$$

where, θ is angle of search hole from X position,

ϕ is scale reading of phasing contact,

S is speed ratio, journal speed to load speed,

and n is tooth number of escapement wheel, the teeth being numbered in the direction of rotation, each tooth being 10°.

3.24 Axial location of search hole

The axial position of the search hole from the right hand side of the bearing is denoted Z, see Fig.2. This value can be determined by measuring the distance between the thrust collar and the shoulder of the shaft against which the escapement wheel runs, which by calibration was found to be Z-0.1 in.

Pressure readings were taken at five axial sections of the bearing 0.25 in. apart, numbered 1 to 5, see Fig.17; a set of gauges was made to enable these settings to be quickly and accurately obtained. When making tests with shaft rotating, it is necessary to shift the driving sprocket on the countershaft by a corresponding amount, bringing it/

it into alignment; the angular settings must then be rechecked.

3.3 PRESSURE PLOTTING

The balanced pressure piston operates as follows: when the back pressure is less than the film pressure, the piston is in its innermost position, making contact and completing the electrical circuit, so that, in the headset, a signal is received once in every rotation of the variable eccentric when the brush short-circuits the segments of the phasing contact. By increasing the back pressure till the signal is received intermittently, indicating that the pressures are balanced, a reading is obtained from the pressure gauges; if the back pressure is too great, no signal will be received.

Pressures may be measured at any desired instant of the load cycle by setting the phasing contact, and by rotating the shaft, or, in tests with rotating shaft, by using the tripping mechanism, pressure readings may be taken at points round the bearing; adjustment of the thrust collar enables readings to be taken at any axial section of the bearing. Thus it was possible to examine the whole pressure field of the bearing at any instant of the load cycle./

cycle.

3.4 LOAD AND ATTITUDE

All five instruments are arranged in such a manner that when selected in circuit by the rotary switch, a continuous signal indicates that the reading shown on the instrument is numerically too low, except ofcourse in the case of the vacumm scale of the pressure gauge, and must be increased to obtain balance.

On this principle, readings were made of load and displacement at chosen points in the load cycle, e.g. setting the phasing contact at $\emptyset = 0,20,40^{o}$,etc. successively and selecting the instruments in turn by means of the rotary switch.

In some cases the shaft was removed and load and displacement readings repeated. The numerical value of zero load was determined with the machine at rest.

/

rest.

4. RESULTS

4.1 FLUCTUATING LOAD - SHAFT STATIONARY

4.11 Conditions of test

This, the initial test, was made with the bearing oscillating horizontally, and the journal stationary, simulating a gudgeon pin bearing. The Y spring was firmly clamped, the connecting rod having been removed, so that the bearing was fixed in the centre of its vertical clearance; the X spring was adjusted so that the bearing oscillated horizontally in the centre of its clearance. The variable eccentric was geared in the ratio 1:1 to the countershaft which runs at 530 R.P.M.

The oil bath was maintained at a temperature of 70°F. by means of the 15 W. heating element. Little heat is generated by the bearing under these conditions and consequently the bath was at constant temperature throughout, so that, after the oil had been stirred and warmed to the correct temperature, no further circulation was required./

required.

4.12 Instability

It was found that the value of peak film pressure fluctuated over a range, and upon investigation, a cycle of events was observed; commencing with a small but dense stream of exceedingly small bubbles, see Fig.19, too small to be discerned by the naked eye and only with difficulty by a magnifier, this stream, which might issue from either side of the bearing, gradually lessened and at the same time the bubbles became slightly larger, until finally a large bubble appeared having a diameter of perhaps $\frac{1}{8}$ in. Correspondingly, the peak film pressure was observed to fall from an initial value of say 35 lb. per in^2. to a value of 30 lb. per in^2., returning to 35 again immediately the big bubble appeared, the period of the cycle being 2 minutes. Similar fluctuations were observed in the load indicators.

In a few minutes, these small bubbles either dissolve in the oil or burst upon the surface, so that the oil does not become saturated with bubbles.

4.13 Readings

Where a fluctuation of pressure occurred, the highest value was recorded.

Pressure readings were taken at 10° intervals/

intervals round the journal for 12 points in the load cycle, ϕ = 0,30,60, etc. to 330, and at five axial sections of the bearing. This gives 180 readings at each load point, and a total of 2160. These readings were plotted after being corrected for instrument error and inches Hg. converted to lb. per in^2., see Figs.21 to 32. Note that the angle θ in this particular test is measured from the Y-axis.

Load and displacement readings were also taken at ϕ = 0,20,40,etc., and repeated with journal removed, to show the amount of friction and the lag in displacement caused by the oil film; these results are plotted in Fig.20.

It should be noted that all the curves in Fig.20 are plotted to the same base, ϕ, and that Figs.21 to 32 which follow, show the pressure distribution at ϕ = 0,30,60,etc. to 330. The numbers 1 to 5 refer to the axial sections of the bearing shown in Fig.17.

4.14 Balance of measured load and pressure load

The pressure load upon the shaft was obtained by Simpson's Rule from the integral

$$\int_0^{l}\int_0^{2\pi} r.p.\sin\theta \, d\theta \, dl$$

where,/

where, r is radius of journal,

p is film pressure,

θ is measured from Y-axis,

and l is length of bearing,

and is plotted in Fig.20 for comparison with the measured load (from the load indicators).

The discrepancy between measured and pressure load may be due to the fact that measured load includes friction which cannot be determined with accuracy; with journal removed the friction amounts to 4 lb., reversing with direction of motion, and it is to be expected that the friction will be greater when the system is loaded. The small time lag of the balanced pressure piston, which only occurs when pressure is increasing, must also be taken into consideration. Taking account of these facts, the two measurements are found to be in close agreement.

4.15 Low pressure region

The film pressure in the unloaded half of the bearing is beneath atmospheric pressure and sometimes approaches absolute zero. The lowest pressures were found just after the direction of motion reverses.

Owing to frothing of the oil under low pressure/

pressure and consequent expansion, it requires a considerable time to record pressures beneath 26 in. Hg.

4.16 Cavitation of the oil film

It was found that, at two periods of the load cycle, the film pressure was less than atmospheric throughout the whole bearing, see Figs.24 & 30. This indicates that oil is flowing into the bearing at all points, and therefore it follows that a cavity must exist in the bearing into which the oil can flow.

Further, in examining the pressure distribution, see Figs.25 & 31, it was found that when the pressure begins to rise on the loaded side of the bearing, the curves show a dent in the middle where previously the lowest pressure occurred. Oil must flow into the low pressure area from the surrounding region of higher pressure. This indicates the continued existence of a cavity in the bearing and shows that the cavity is of considerable size./

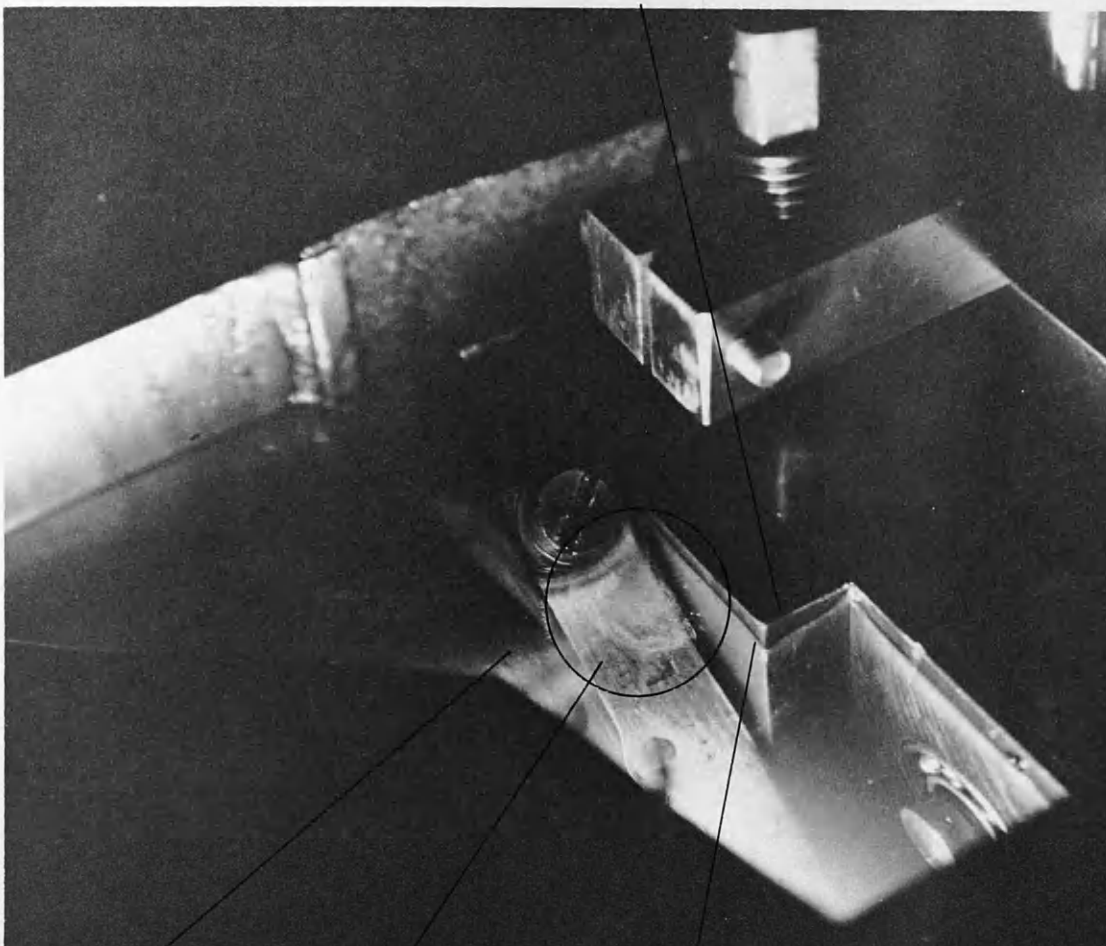
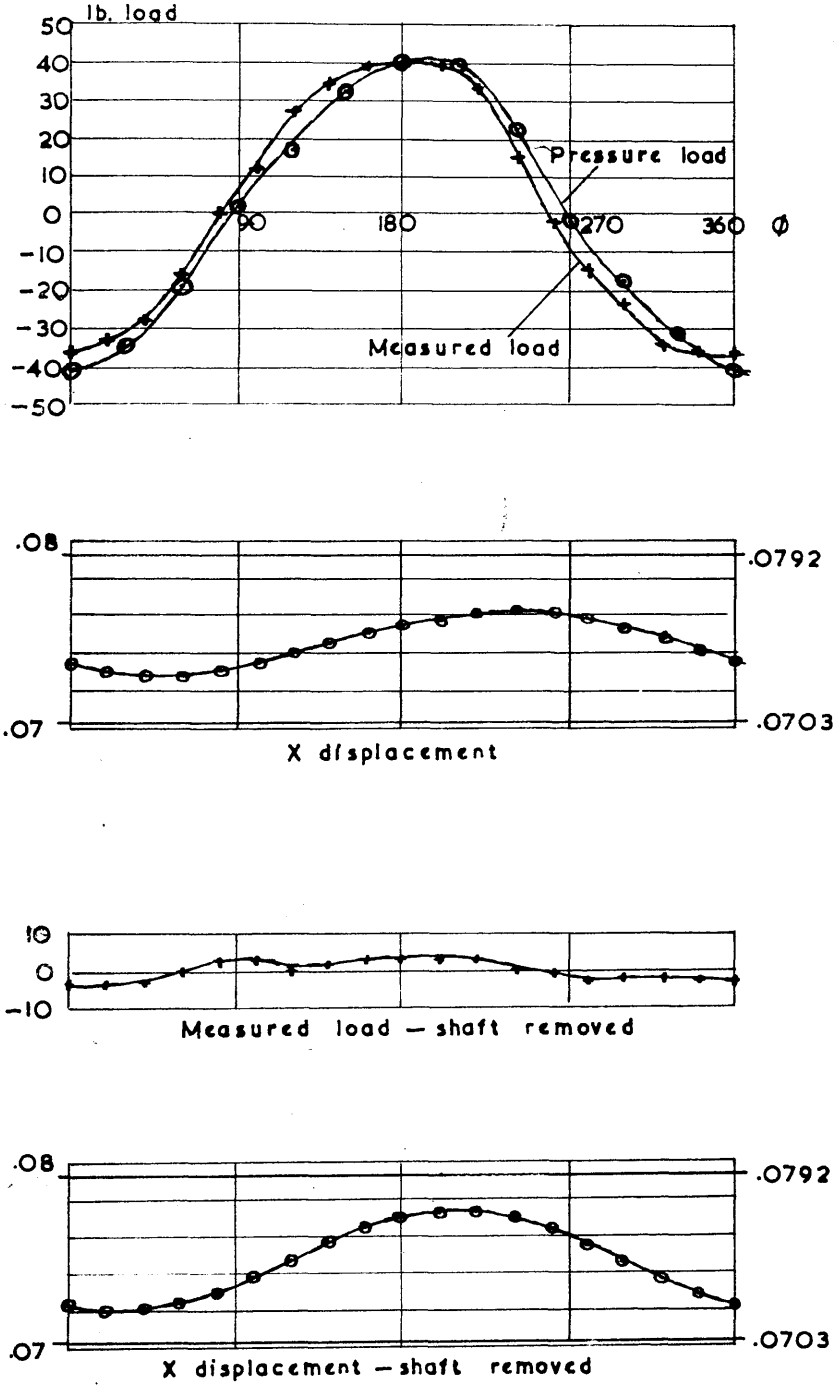


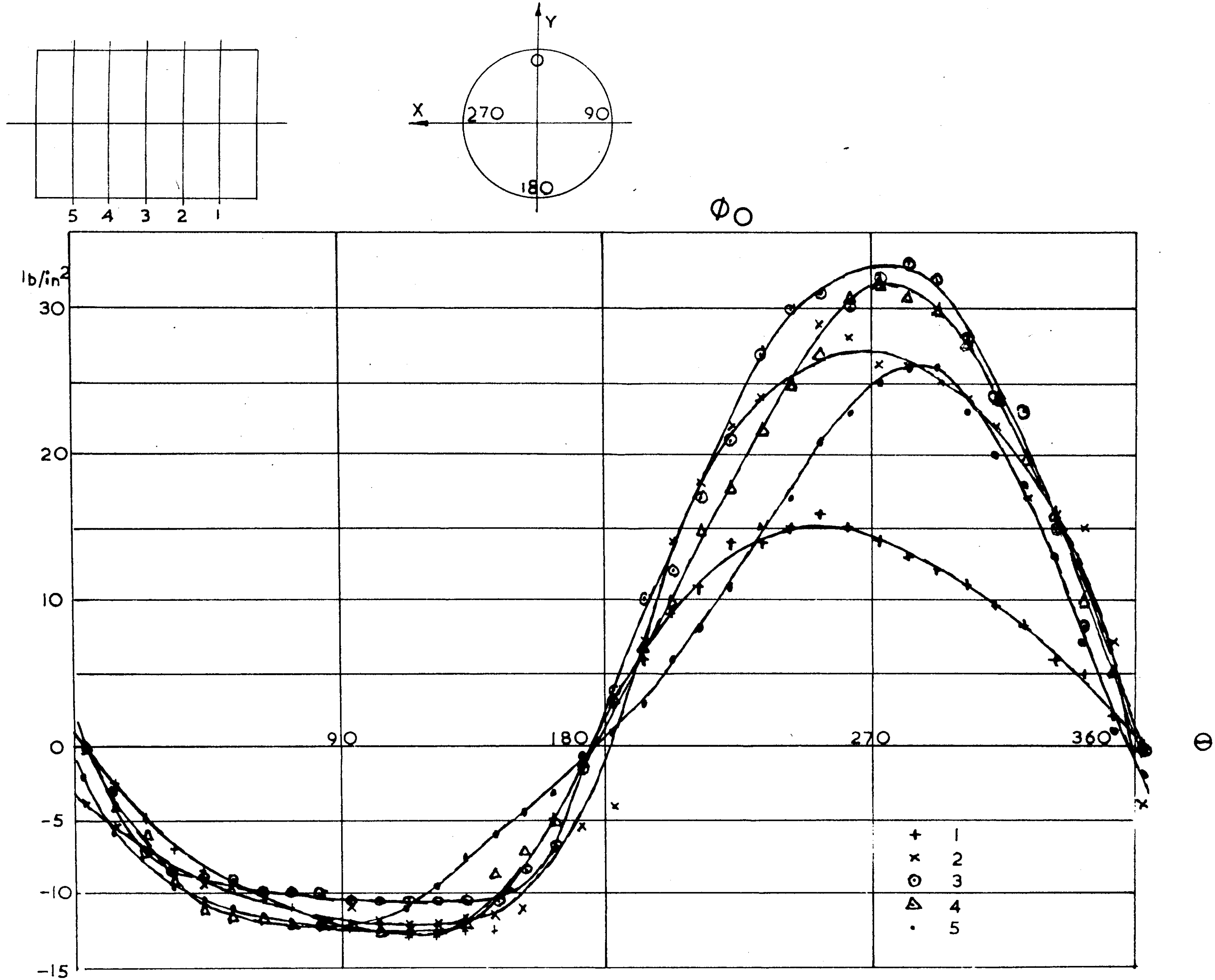
FIG.19

A cloud of extremely small cavitation bubbles coming from the bearing under conditions of fluctuating load and stationary shaft.



FLUCTUATING LOAD — STATIONARY SHAFT

FIG. 20



5 4 3 2 1
Y
X
0
90
180
270
ϕ_O
lb/in²
30
20
10
0
-5
-10
-15
90
180
270
360
Θ
+ 1
× 2
⊙ 3
△ 4
• 5

FIG. 21

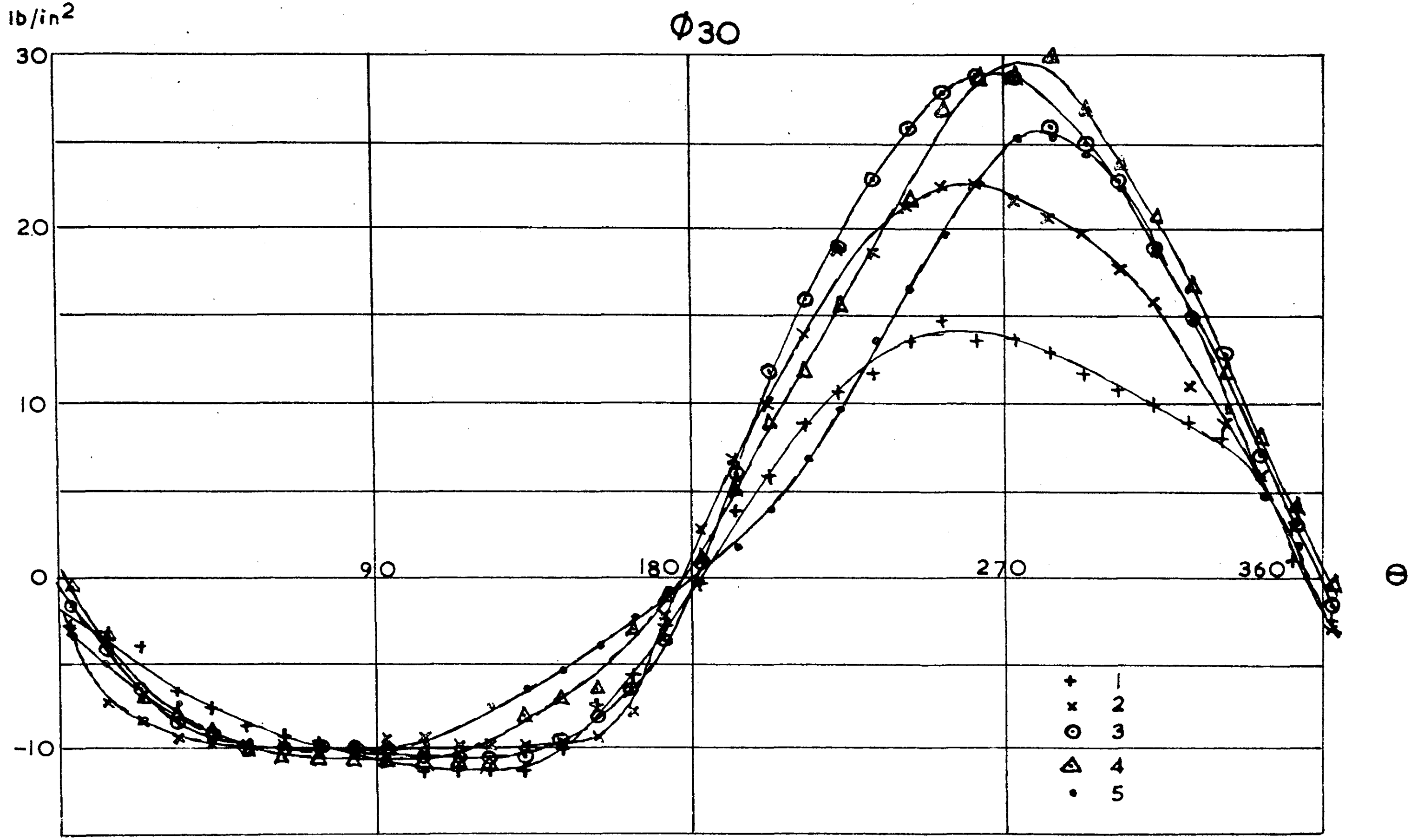


FIG. 22

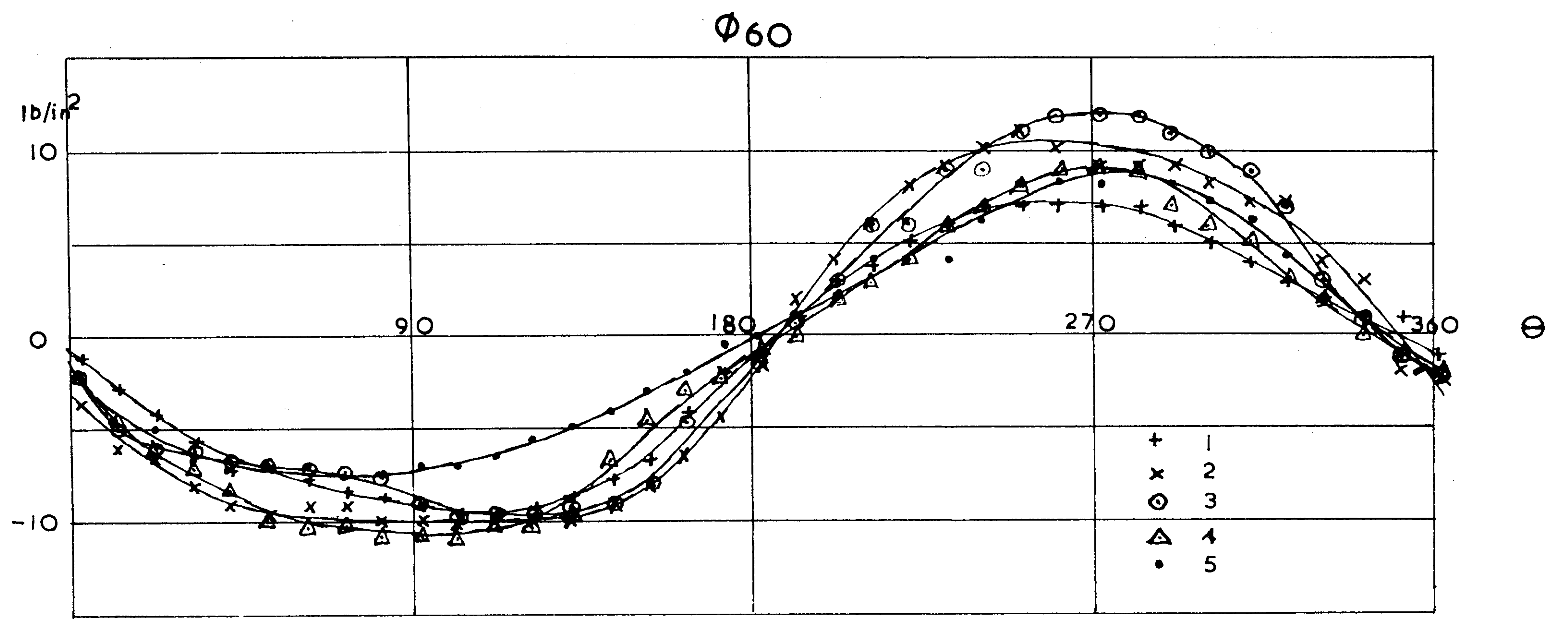


FIG. 23

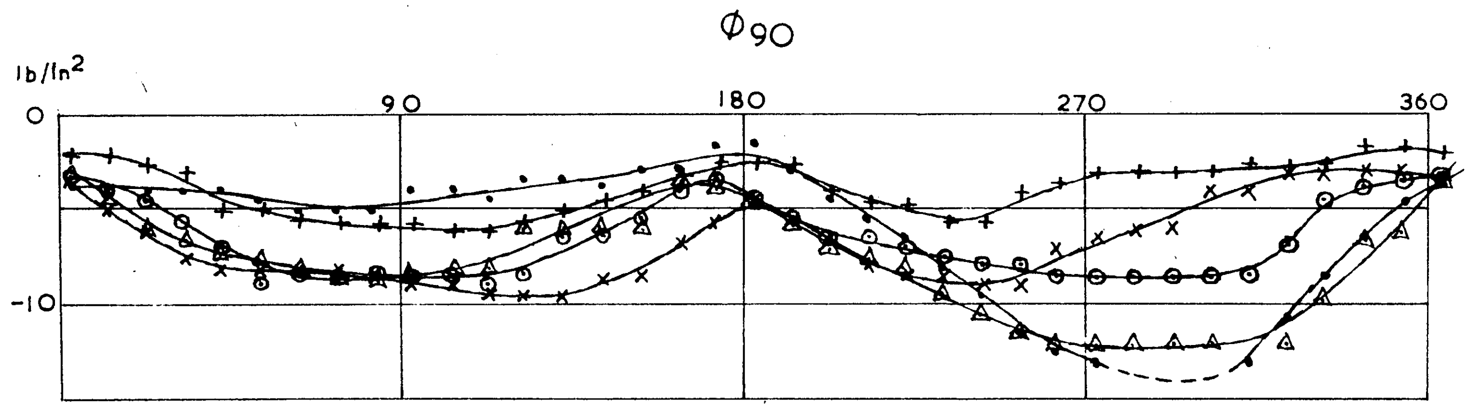

ϕ_{90}
lb/in^2
0
-10
90
180
270
360
Θ

FIG. 24

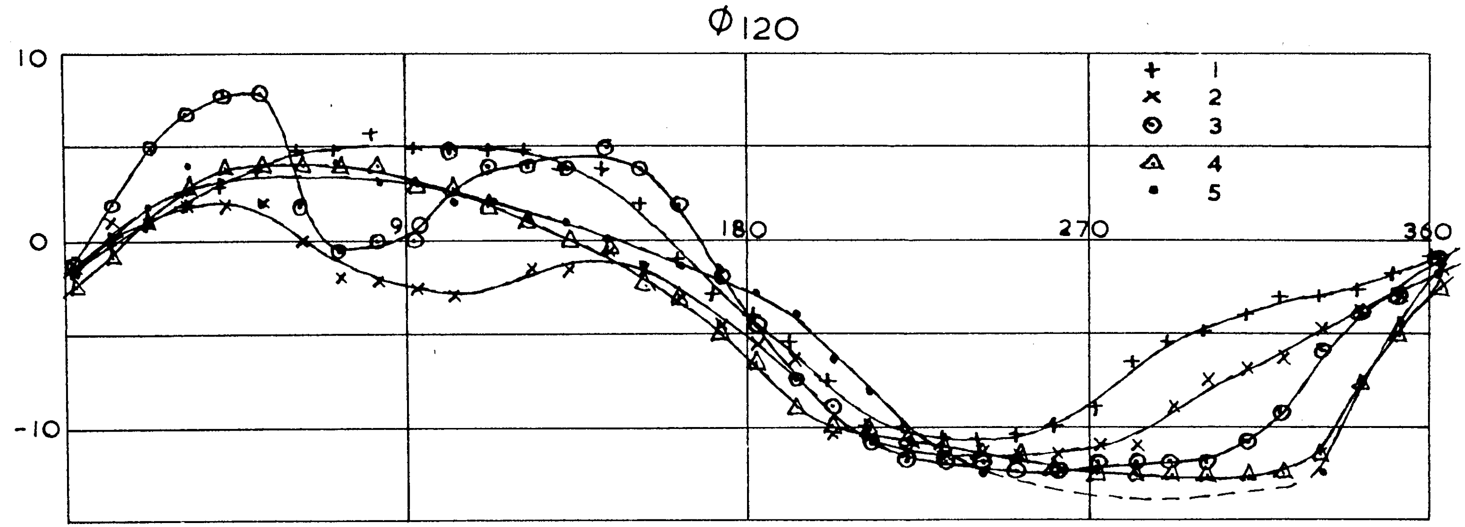

ϕ_{120}
10
0
-10
90
180
270
360
Θ
+ 1
× 2
⊙ 3
△ 4
• 5

FIG. 25

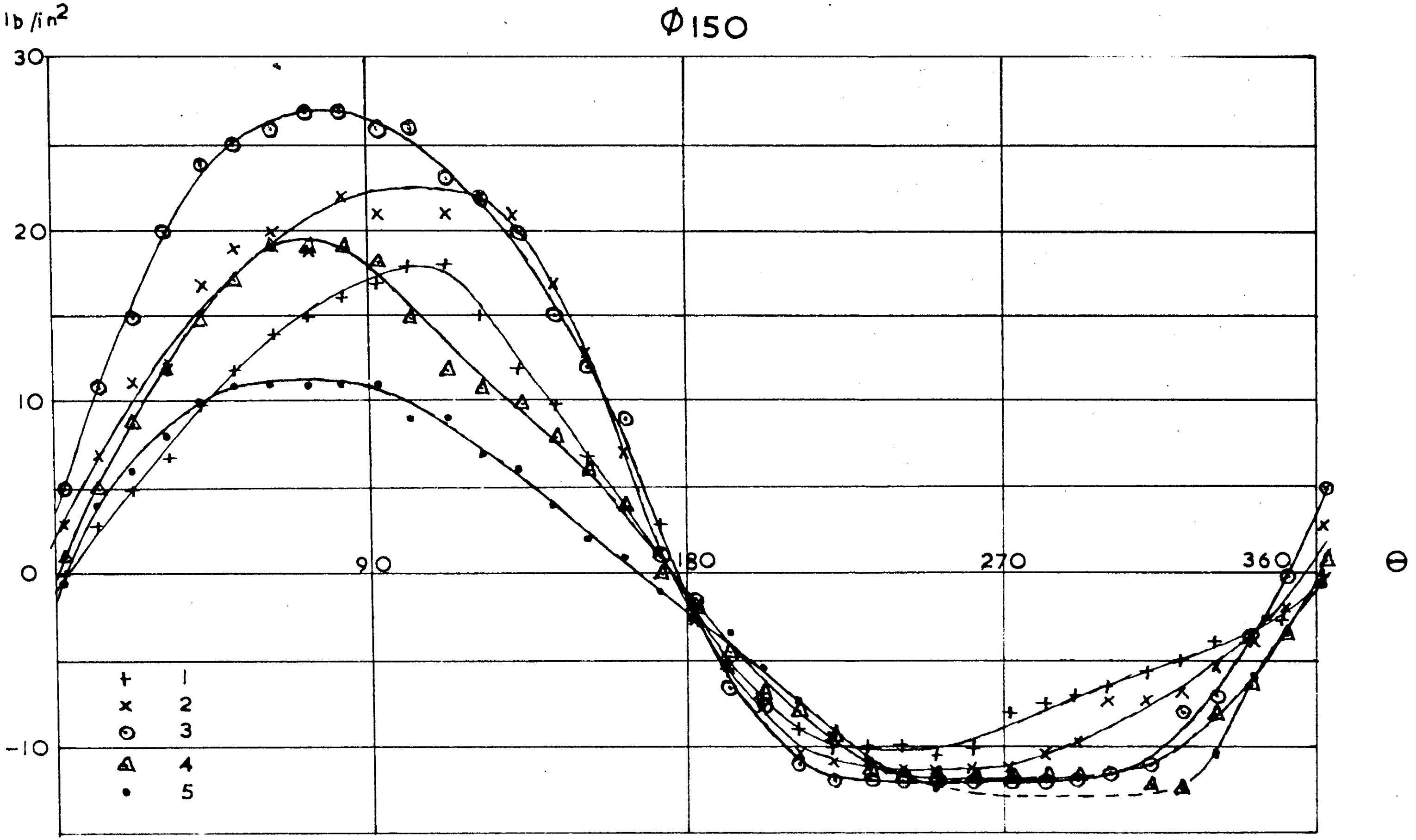


FIG. 26

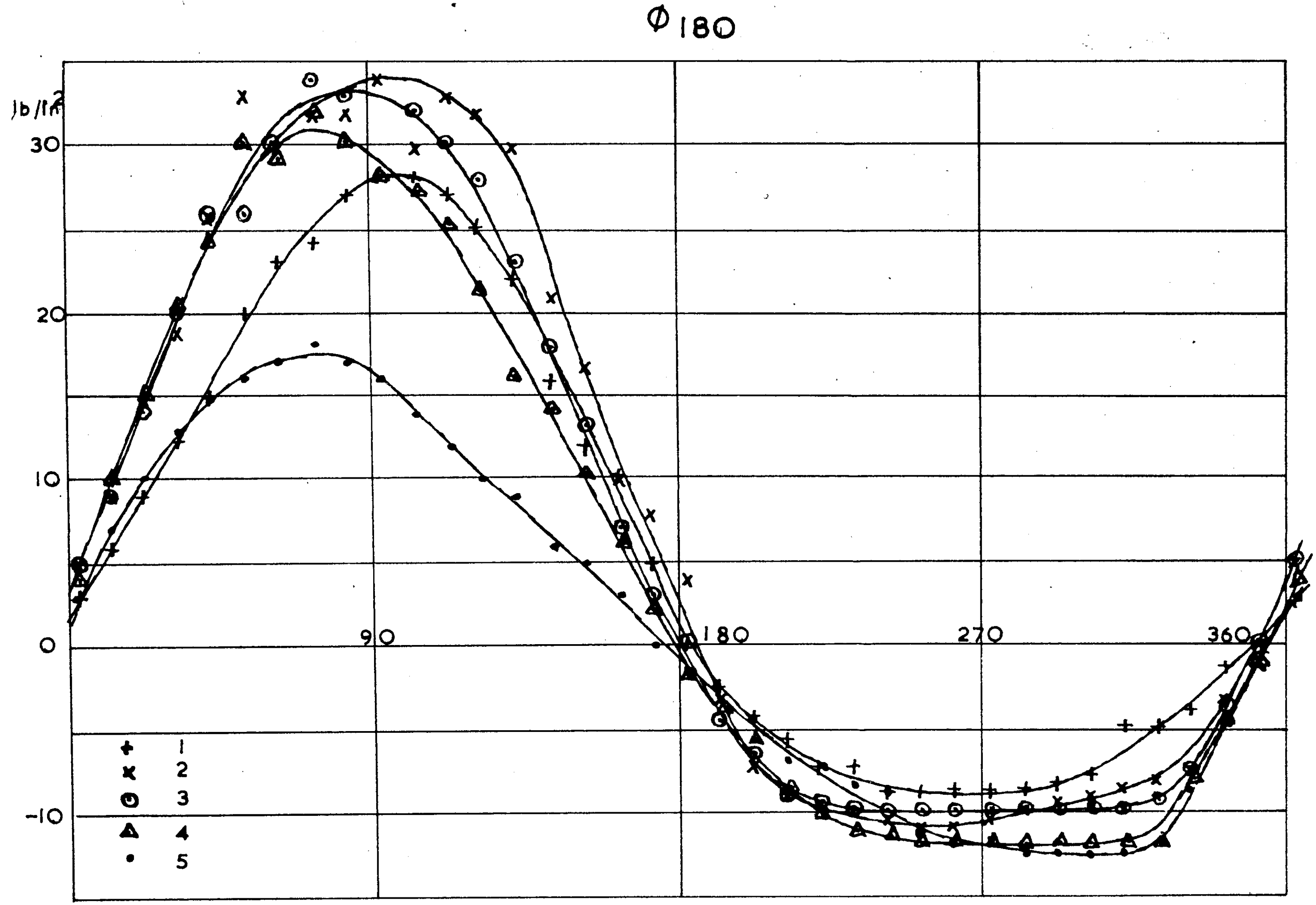


FIG. 27

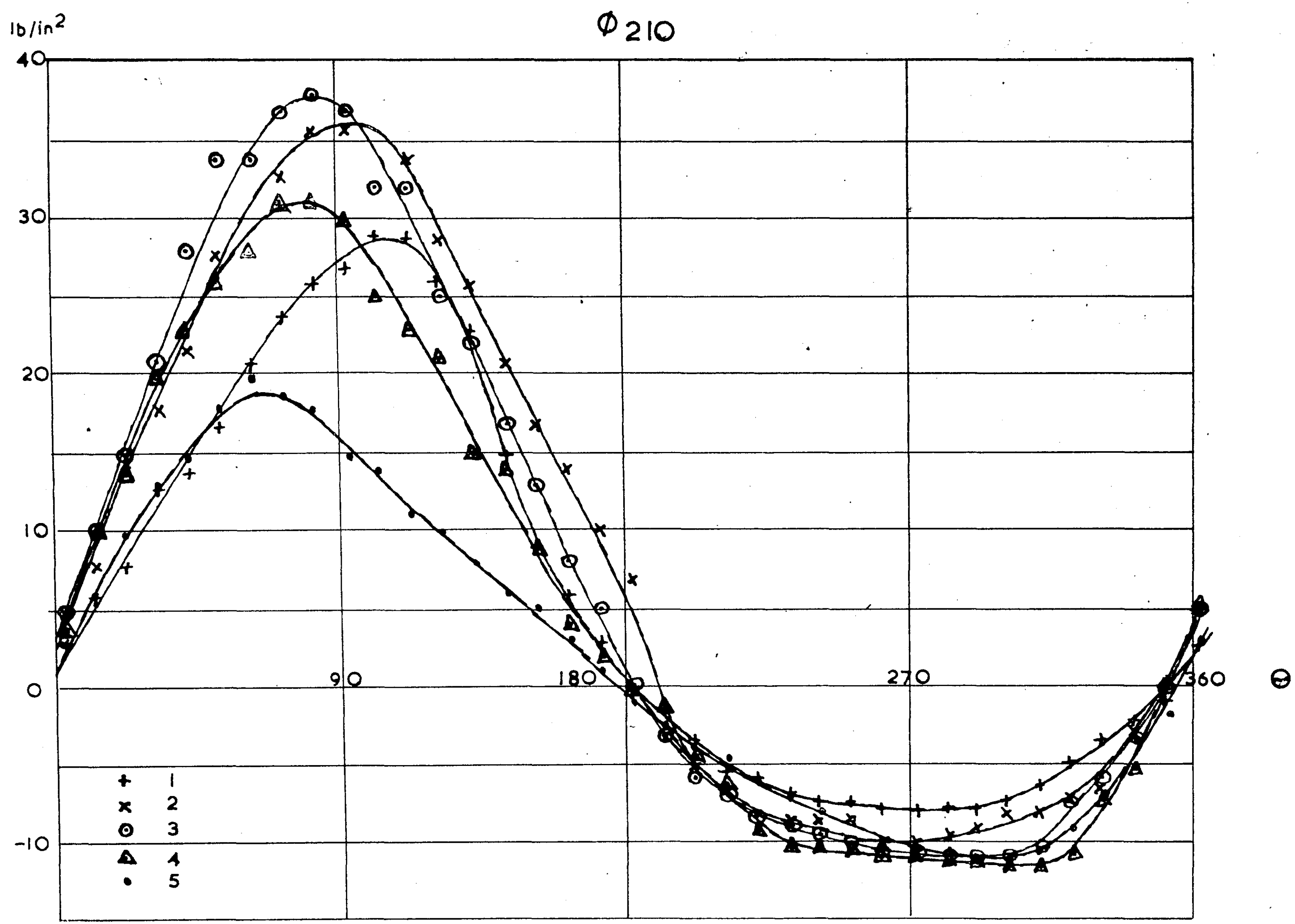


FIG. 28

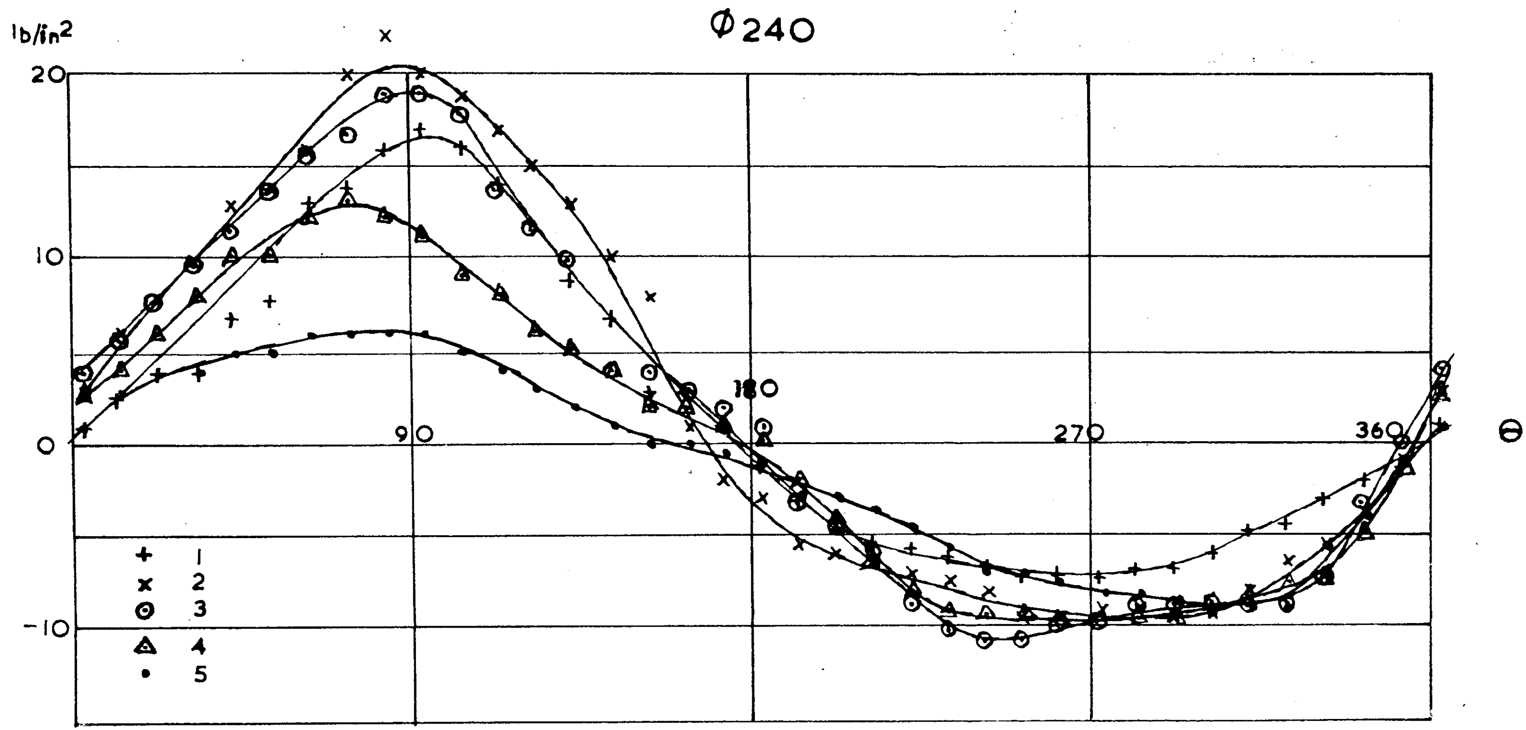
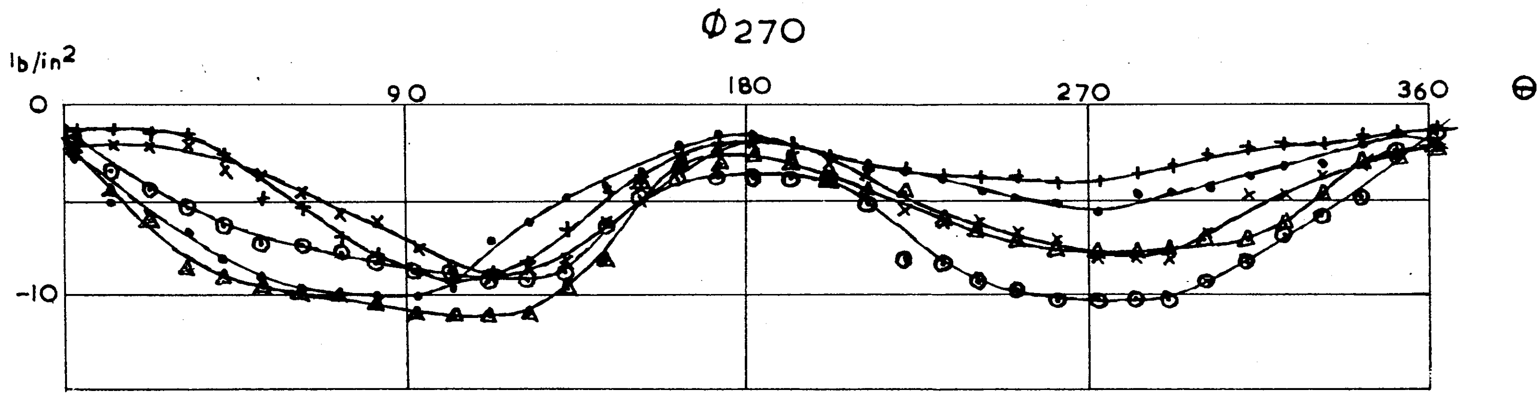
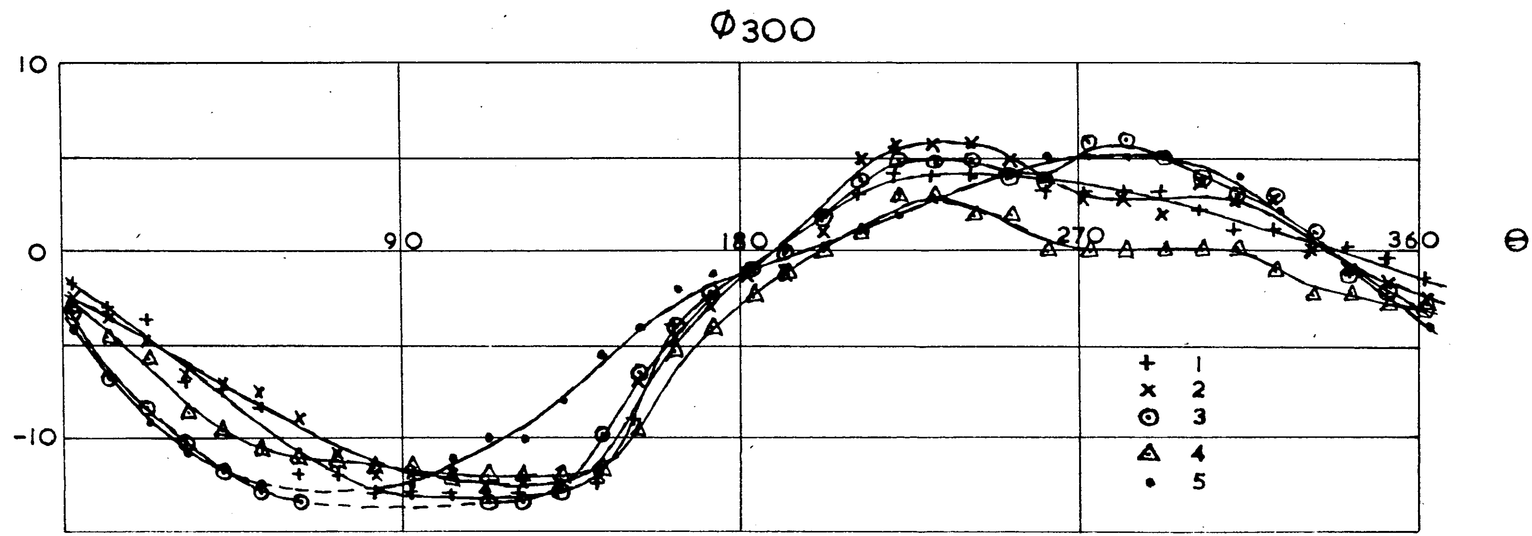


FIG. 29



Φ270
lb/in²
0
-10
90
180
270
360
Θ

FIG. 30



Φ300
10
0
-10
90
180
270
360
Θ
+ 1
x 2
⊙ 3
Δ 4
• 5

FIG. 31

ϕ_{330}

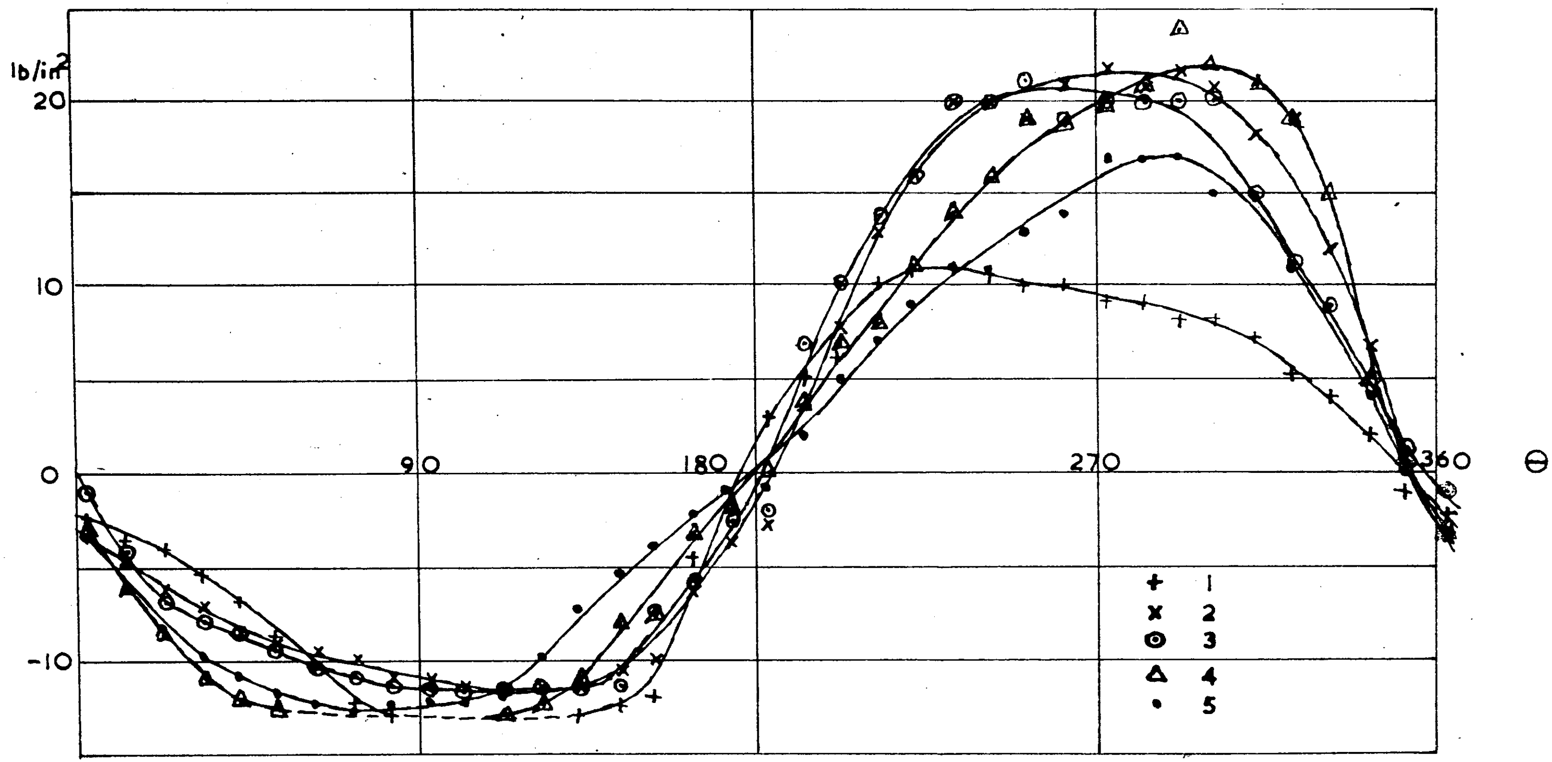


FIG. 32

size.

4.2 ROTATING LOAD - SHAFT STATIONARY

4.21 Conditions of test

This test was made with the bearing moving round (not rotating) the journal, the journal itself remaining stationary. This action drives the oil round the bearing and a pressure film is formed, although neither journal nor bearing rotate. The movement was achieved by employing both horizontal and vertical loads, adjusted so that the bearing moved in the centre of its clearance. Use was made of the displacement indicators to obtain this setting.

The oil bath temperature was again maintained at 70°F.

4.22 Drop in film pressure

It was noted that the peak film pressure decreased with time, the oil temperature remaining constant. A prolonged test was therefore made to investigate the rate of fall, see Fig.33. The pressure decreased gradually and stabilized after three hours running, thereafter remaining constant. At the end of this test, when the machine was stopped, a stream of bubbles, perhaps 1/16 in. diameter/

diameter judging by the 1/16 in. gap between the bearing block and the steady bearing, issued from the bearing in rapid succession; 53 bubbles were counted in all, indicating that the bearing was only 2/3 full of oil. On restarting, the initial pressure was recovered. Similar tests showed that if the machine was stopped for only a few seconds and restarted before any bubbles had escaped, then the film pressure remained unaltered, whereas if some bubbles were allowed to escape, recovery of initial film pressure was only partial.

During these tests, no bubbles whatsoever came from the bearing till the film pressure had stabilized, when a bubble would appear every few minutes.

4.23 Readings

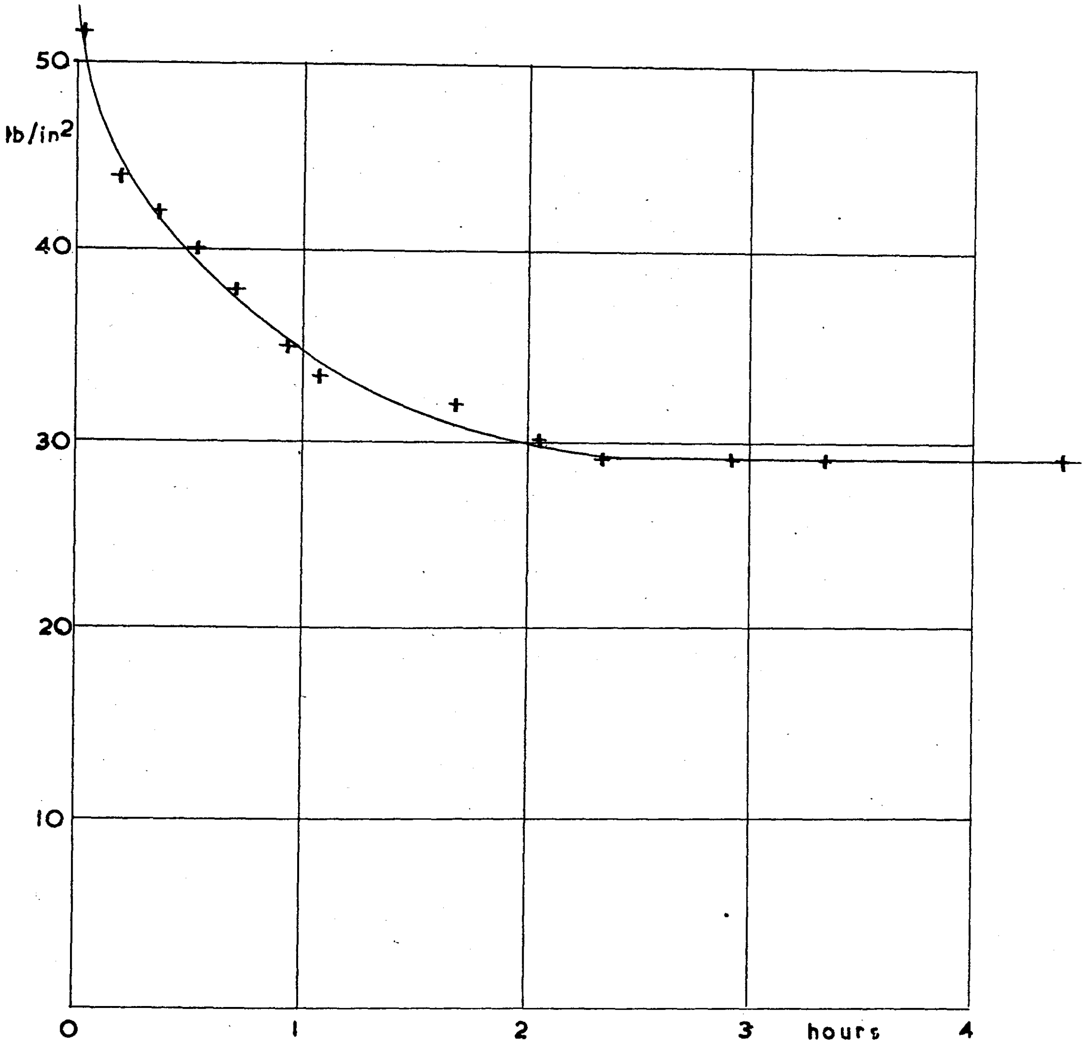
After four hours running to allow conditions to stabilize, pressure readings were taken at 10° intervals round the journal in axial section 3 of the bearing, for three equally spaced points in the load cycle, see Figs. 35,36&37. Loads and displacements were also recorded, see Fig.34. All the curves in this figure are plotted to the same base $\emptyset$, which serves as a time base since the/

the machine runs at constant speed.

4.24 Pressure distribution

In order to indicate the axial pressure distribution, results are presented in Fig.38, from another test under similar conditions. While these readings may not have been completely stable, they show that the pressure film occurs on a fairly narrow strip of the bearing and that the larger part of the bearing is a region of constant pressure slightly beneath atmospheric pressure.

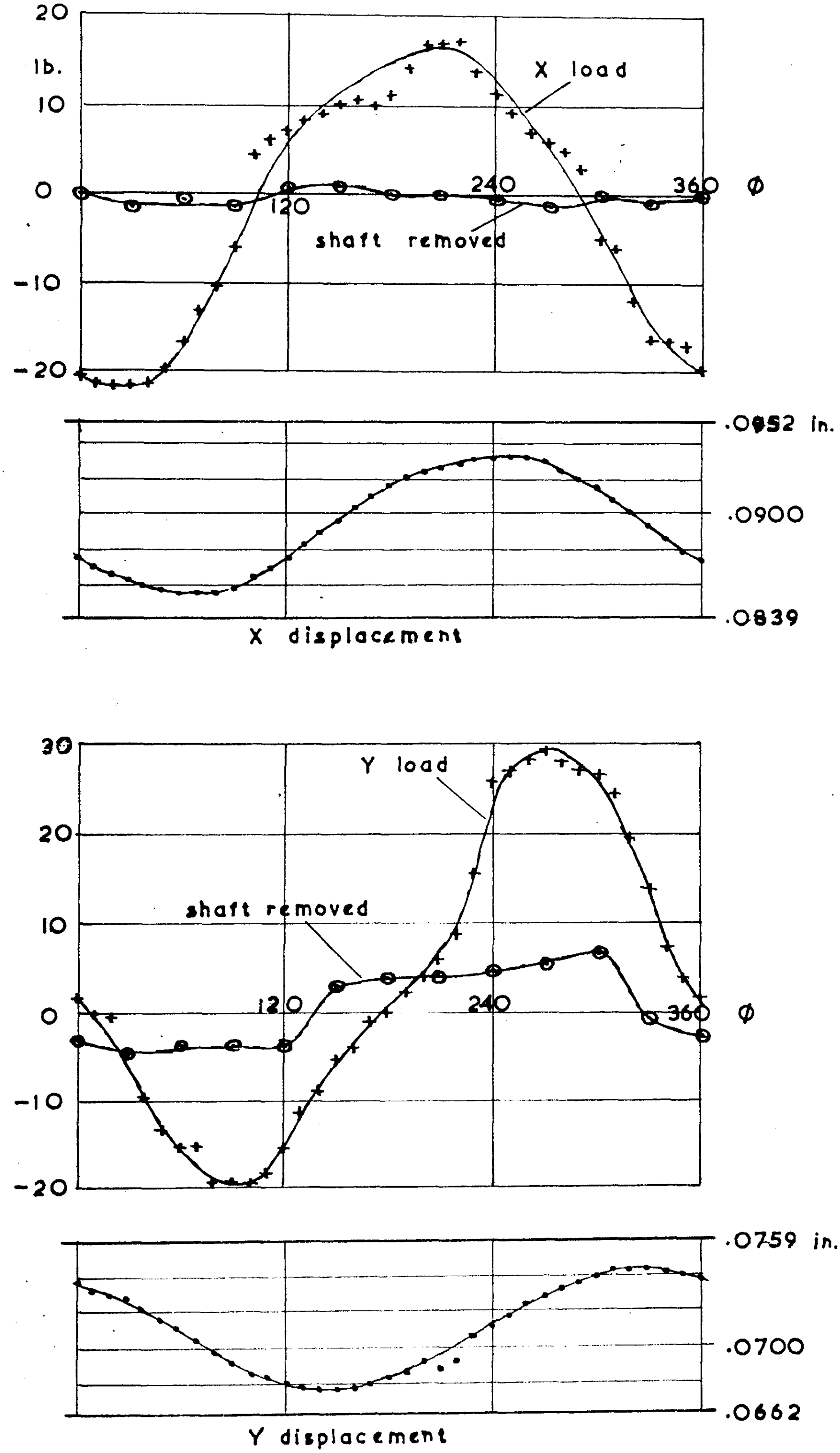
From Figs.35,36&37, it is seen that the film pressure is considerable at the point of minimum film thickness, and that the pressure peak is immediately followed by a narrow region of very low pressure./



DROP IN FILM PRESSURE

ROTATING LOAD — SHAFT STATIONARY

FIG. 33



ROTATING LOAD — SHAFT STATIONARY

FIG. 34

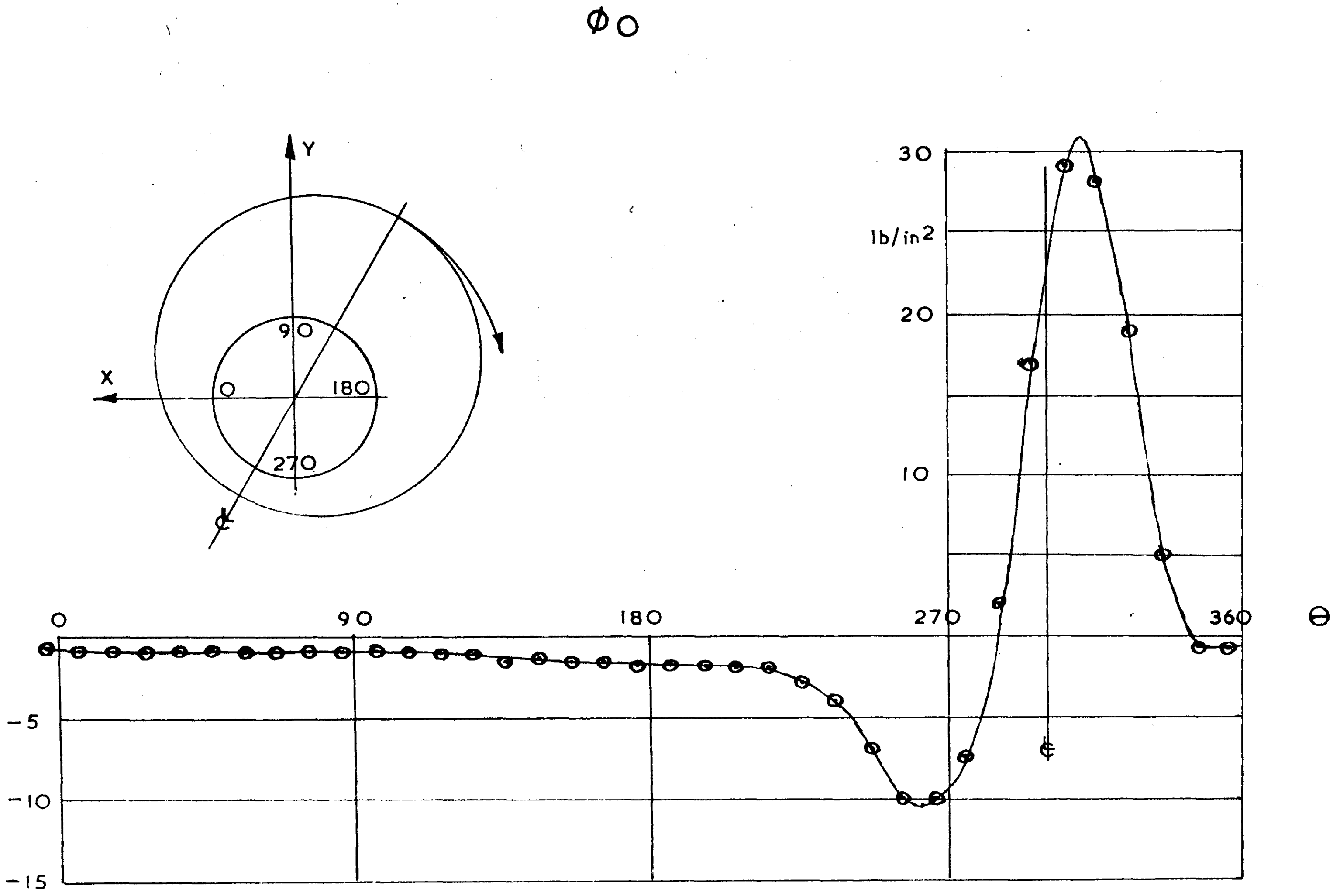


FIG. 35

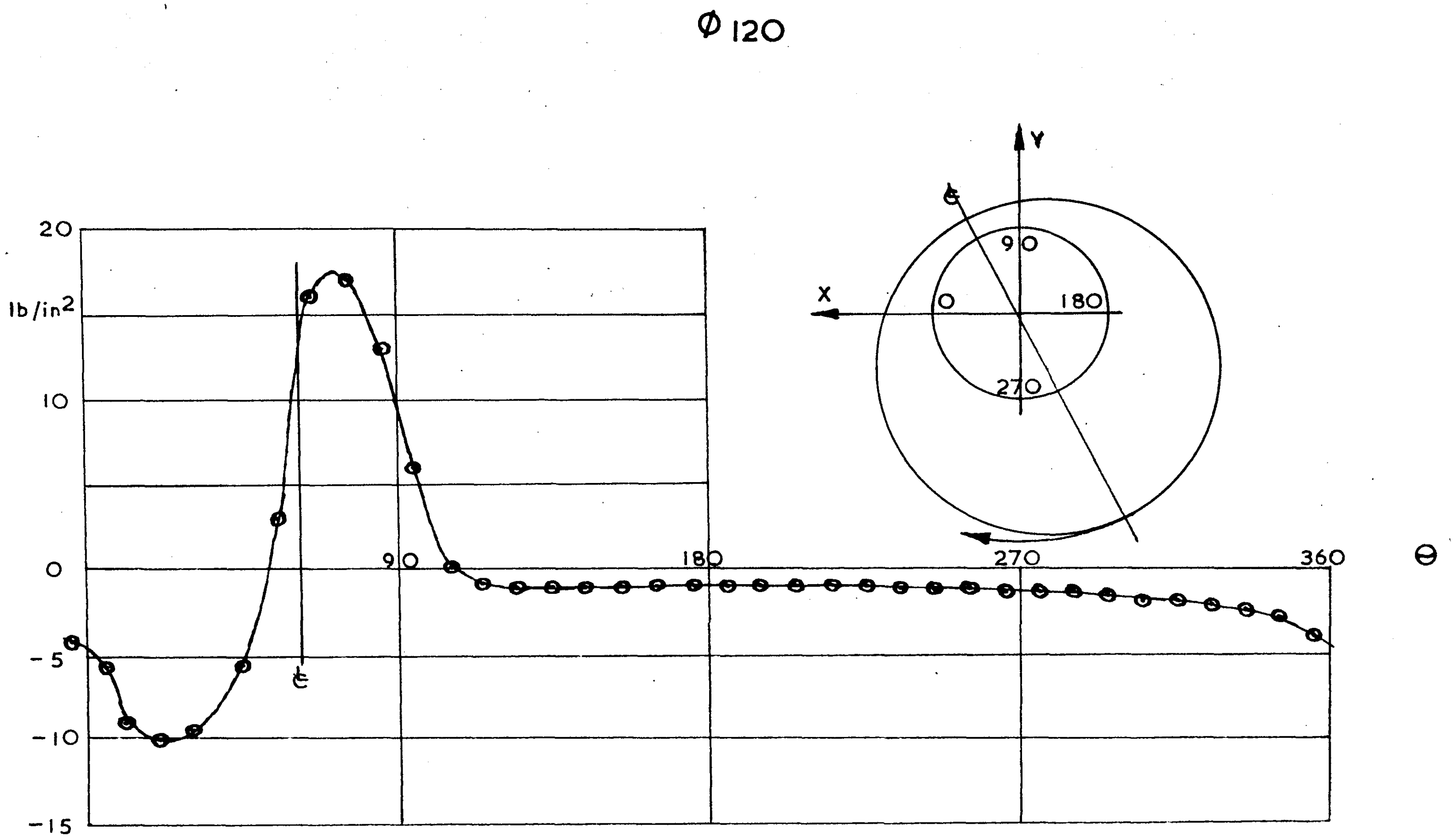


FIG. 36

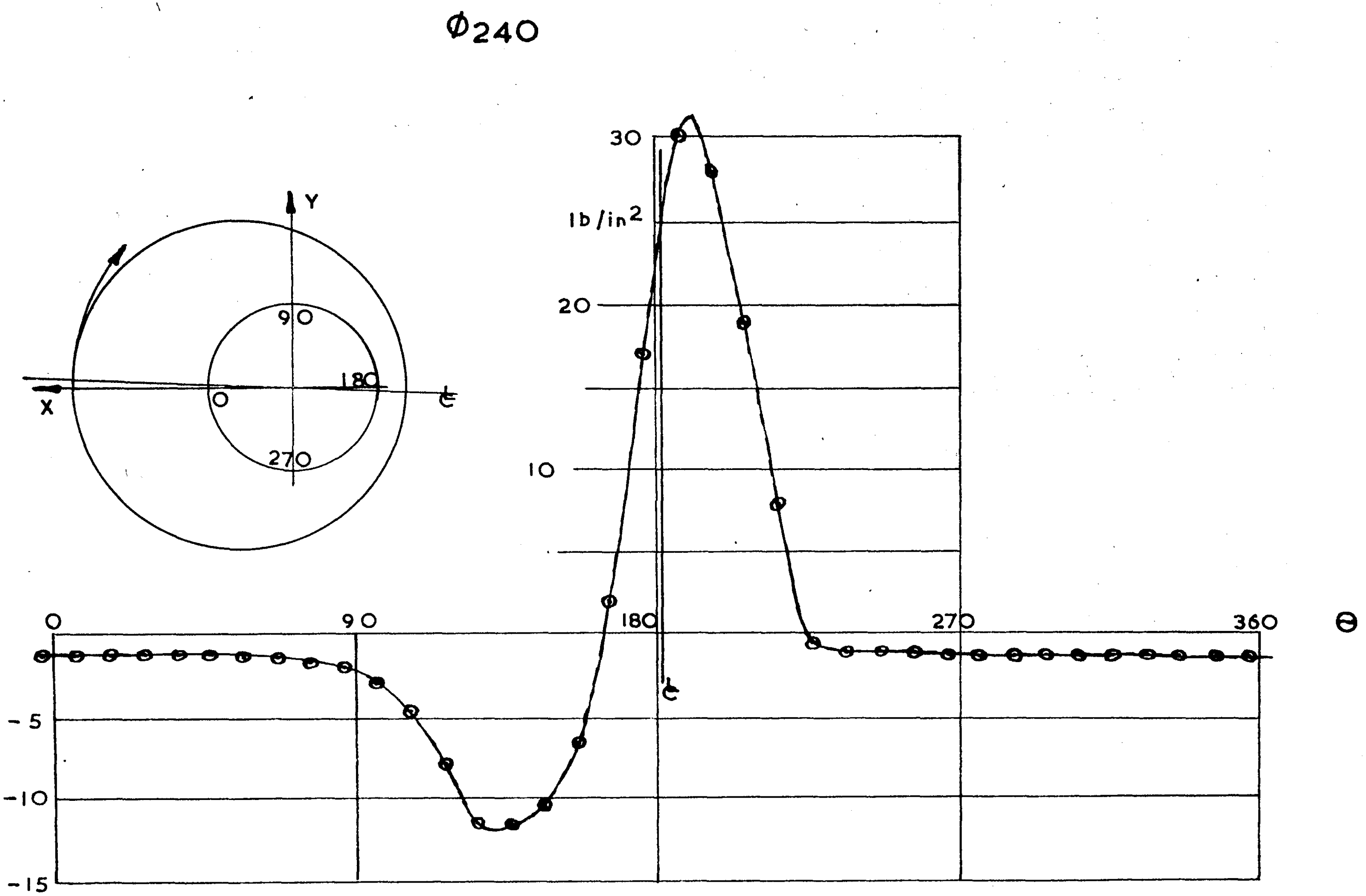


FIG. 37

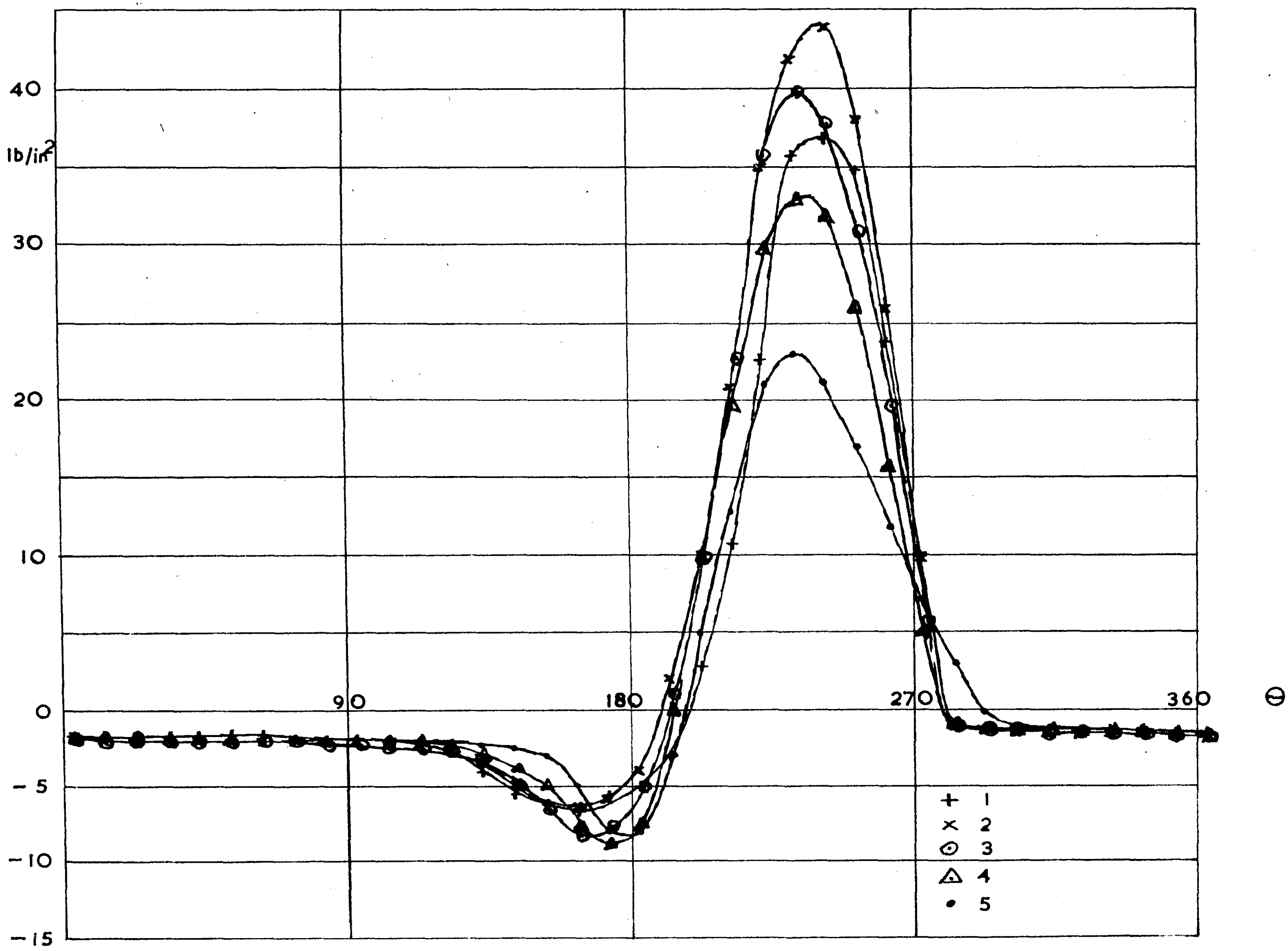


FIG. 38

pressure.

4.3 ROTATING LOAD - SHAFT ROTATING 1:1

This test was made with the bearing moving round the journal as previously, but with the journal also rotating. Both journal and variable eccentric were driven from the countershaft in the ratio 1:1, so that they ran at the same speed.

Under these conditions, the journal is stationary and the bearing rotates backwards, with reference to the line of centres, i.e. the line passing through the centres of both journal and bearing. The line of centres ofcourse rotates.

This picture is quite similar to the normal case of a journal under steady load, in which the bearing is stationary with reference to the line of centres, but the journal rotates.

The drop in film pressure was now much more rapid, descending to almost zero in two hours running. As in the case of the test with shaft stationary, it was found that pressure could be recovered by stopping the machine and allowing air bubbles to escape from the bearing. While running, a continuous stream of small bubbles came from/

from the bearing, see Fig.39. The bubbles were of various sizes, the largest being easily discerned. In the course of two hours running the oil bath became saturated with these small bubbles, but it was shown that this aeration had no appreciable effect upon the pressure drop, as tests made with severely aerated oil gave similar results to those made with fresh oil.

In a series of short runs, the pressure distribution was examined and was found to be quite similar to that of the earlier test with stationary shaft. Typical pressure curves are shown in Fig.40; these were taken at three equally spaced points in the load cycle, in axial section 3 of the bearing./

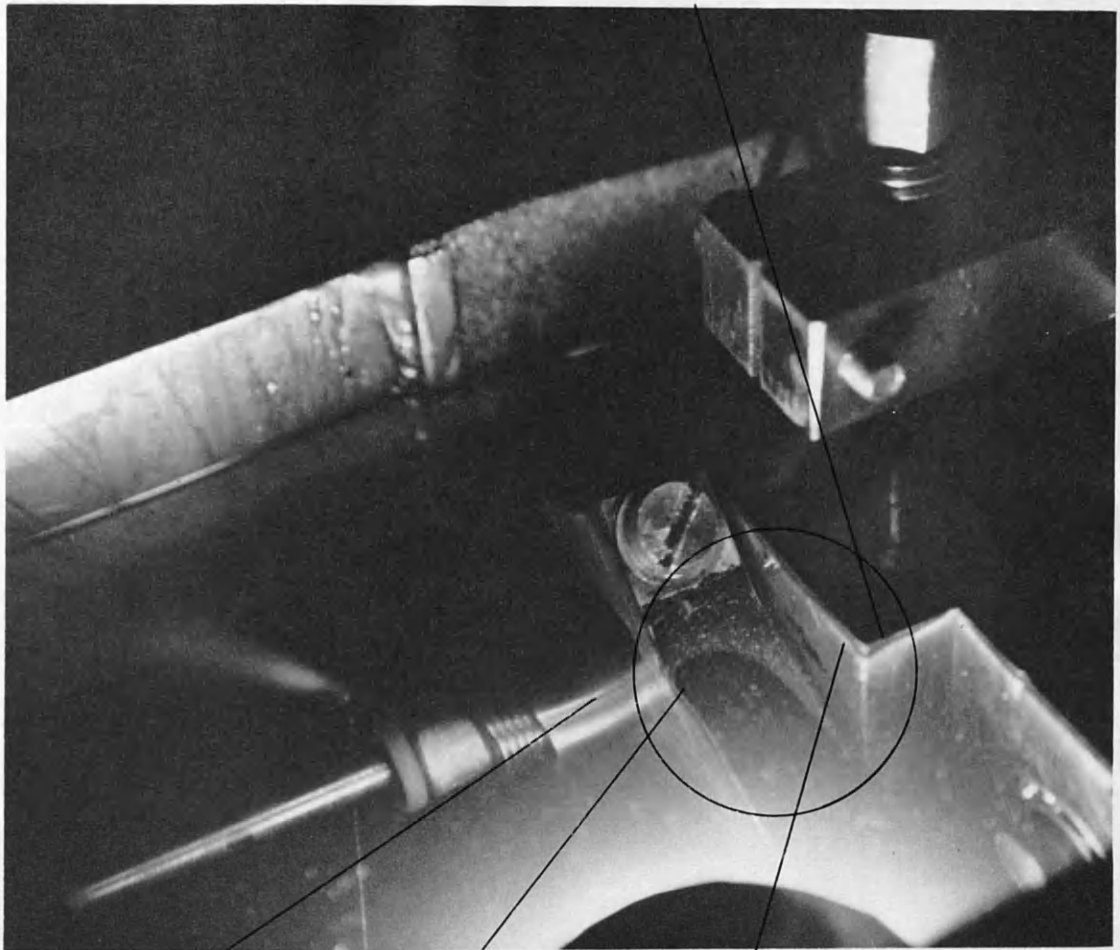
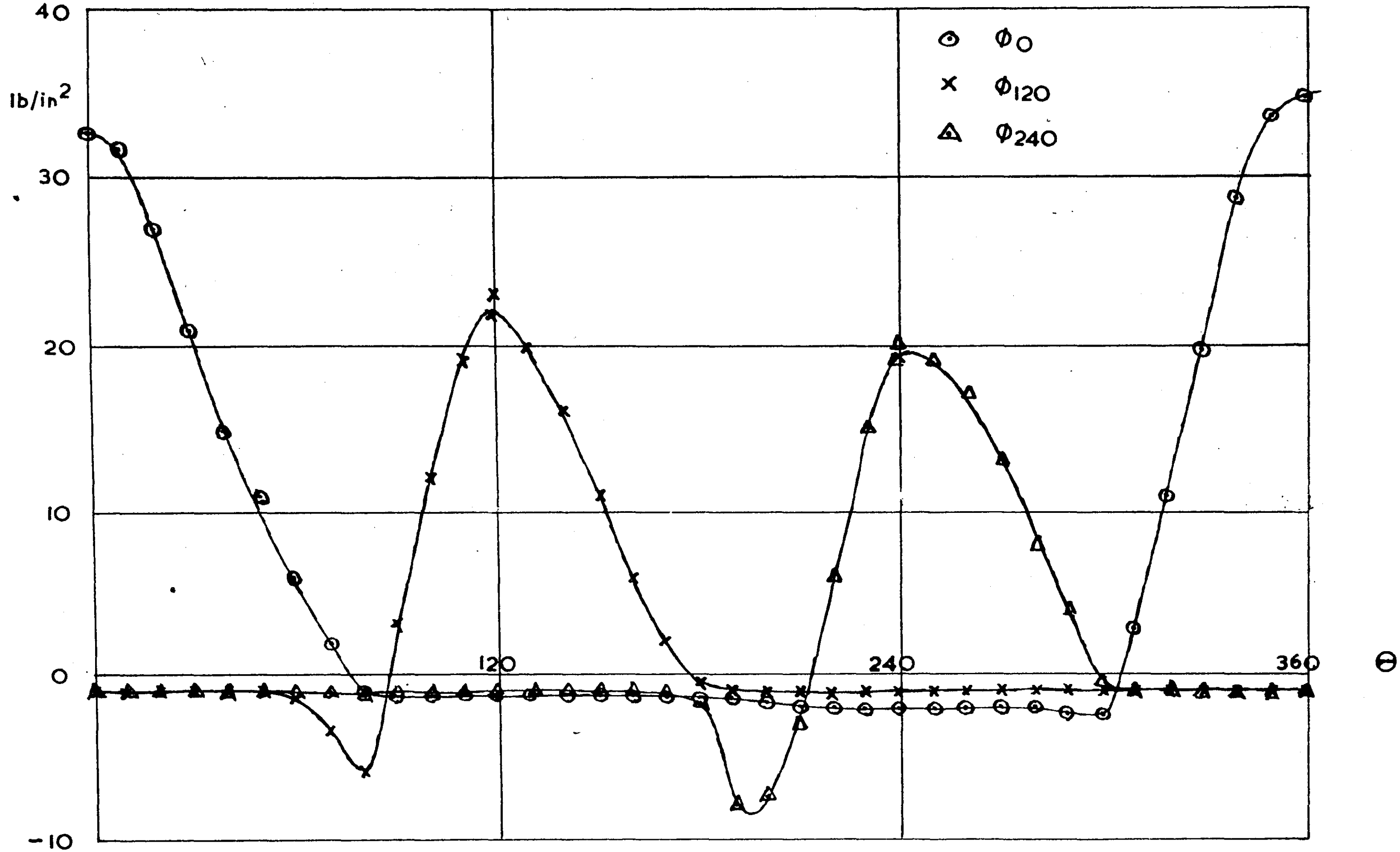


FIG.39

When the shaft is rotating, a stream of bubbles issues from the bearing. The size of the bubbles varies, the largest being quite easily visible. The stream is drawn into the vortex produced by the rotating shaft, and circulated throughout the oil bath. This photograph was taken with rotating load.



ROTATING LOAD — SHAFT ROTATING 1:1

FIG. 40

bearing.

4.4 FLUCTUATING LOAD - SHAFT ROTATING 1:1

A horizontal fluctuating load was now applied to the bearing, with the journal running. These conditions are somewhat similar to those existing in a big-end bearing.

In order to free the bearing in the vertical direction, the Y spring lever was unclamped from its torque spindle and removed. The X, or horizontal loading system remained in operation.

It was found that a stream of bubbles came from the bearing, stronger than before, the bubbles also being of larger size; also, the film pressure fell more rapidly making it impossible to take systematic readings. By searching the pressure field in a series of short runs, the indications were that the distribution had features of two cases previously examined. A large area of constant pressure was exhibited as in the test with shaft and load rotating, and also, at two periods in the load cycle, the pressure was beneath atmospheric throughout the whole bearing as in the first test with fluctuating load and stationary shaft. Regions of very low pressure were also found in similar positions to those recorded in §4.1 i.e. just after/

after the direction of motion reverses.

It was noted that the bubbles, on entering the vortex of the shaft, travelled inwards till they reached the surface of the shaft, and it was thought that this might occur alongside the bearing; also there was a possibility of warm oil being retained in the confined space on each side of the bearing, and also a possibility of heat generated in the main bearings travelling along the shaft to the test bearing and contributing to the drop in film pressure.

Wooden partitions were built into the oil tank to guide a supply of fresh oil, from a gravity tank, up past each end of the test bearing, the oil being drained away thereafter. No improvement was noted with this arrangement.

Further, an oil circulating system, described in §2.7, was constructed. Oil was sprayed from jets upon the journal at each side of the bearing so that warm oil and air bubbles would be washed away, but no success was achieved in arresting the fall of film pressure.

Finally, an additional pipe was led from the jet pipe to feed the main bearings with oil under pressure, as it had been noted that the main bearings were slightly starved of oil and might be generating heat which would travel along the shaft and affect the test bearing. This/

This device also was unsuccessful.

Evidently, the drop in film pressure is due to the growth of the cavity inside the bearing, and little else; it is remarkable that the rate of growth should be so small.

4.5 ROTATING LOAD - SHAFT ROTATING 2:1

According to theory, a critical condition occurs when the load is rotating at half the speed of the shaft, the film being unable to support any load whatsoever in these circumstances.

In order to obtain experimental verification, the speed of the variable eccentric was reduced in the ratio 2:1, while the shaft was driven at its normal speed of 530 R.P.M. After making adjustments to ensure that the bearing moved in the centre of its clearance, the film pressure was examined in the central plane of the bearing. The total range of pressure was found to be +3 to -4 lb. per in^2. Readings were then taken of loads and displacements, see Figs.41 to 44. A continuous stream of bubbles came from the bearing similar to those shown in Fig.39, but no bubbles appeared after the machine was stopped, indicating that there was no cavity of any size in the bearing./

bearing.

The chain drive to the shaft was then removed, and a comparative test made with load rotating and shaft stationary. Peak film pressures of 29, 19 and 30 lb. per in^2. were then obtained at three points of the load cycle, i.e. ϕ = 0,120,&240, making a most convincing demonstration. Readings of loads and displacements, see Figs.41 to 44, showed that the loads had increased and the attitude (or eccentricity) decreased. No bubbles appeared while the machine was running but a number of bubbles came from the bearing when the machine was stopped.

Further, the shaft was removed and measurements made of loads and displacements. It will be seen from Figs.41 to 44, that these are almost identical with the readings taken when the shaft was running, showing that in fact the oil film carries no load under these conditions./

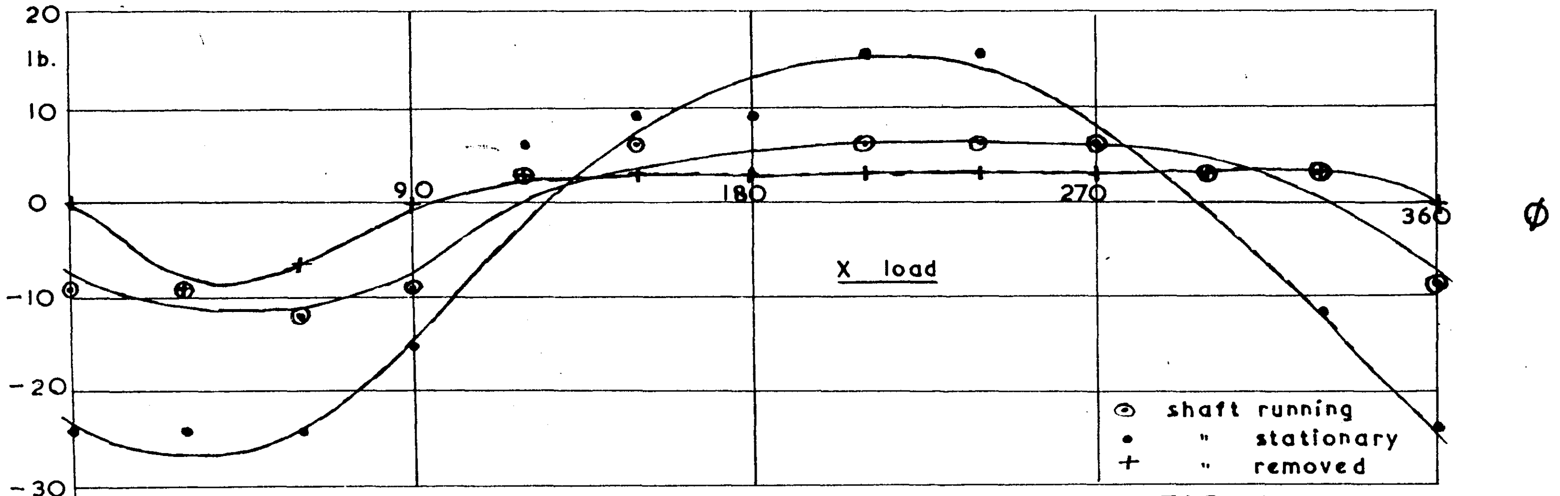
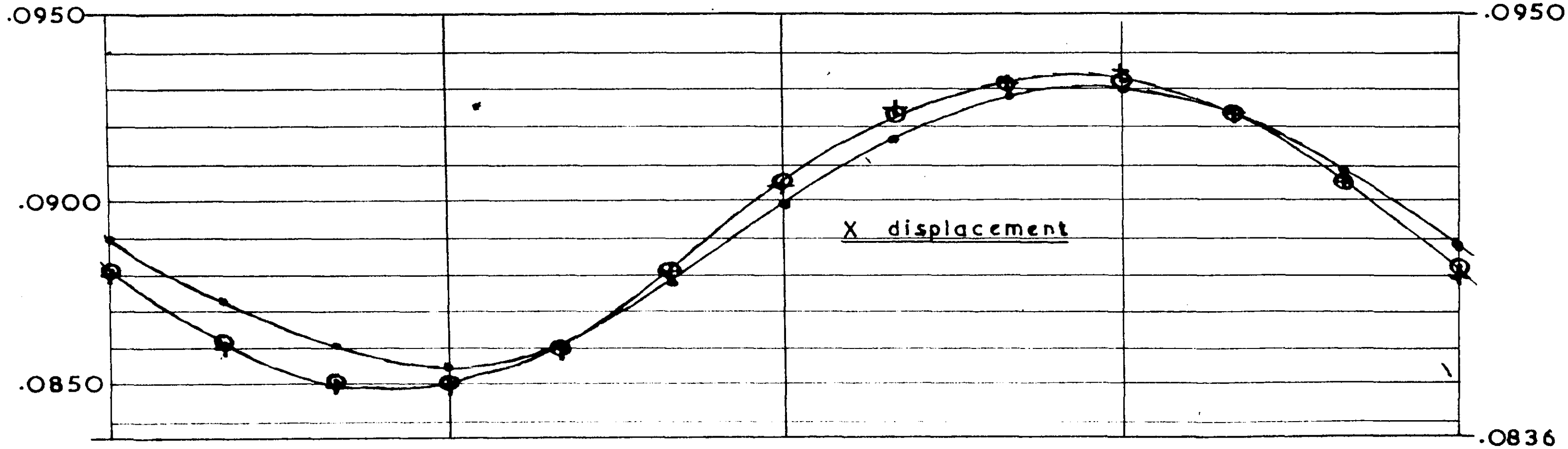


FIG. 41



ROTATING LOAD – SHAFT ROTATING 2:1

FIG. 42

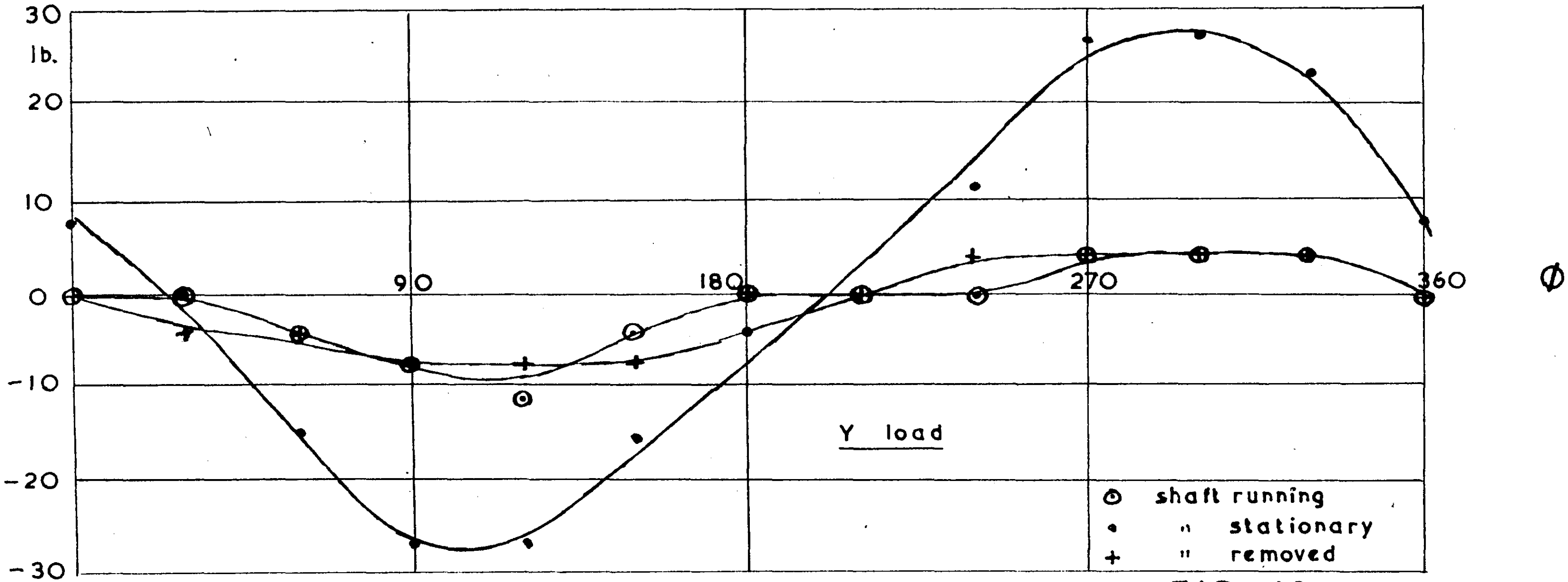

30
lb.
20
10
0
-10
-20
-30
90
180
270
360
Ø
Y load
⊙ shaft running
• " stationary
+ " removed

FIG. 43

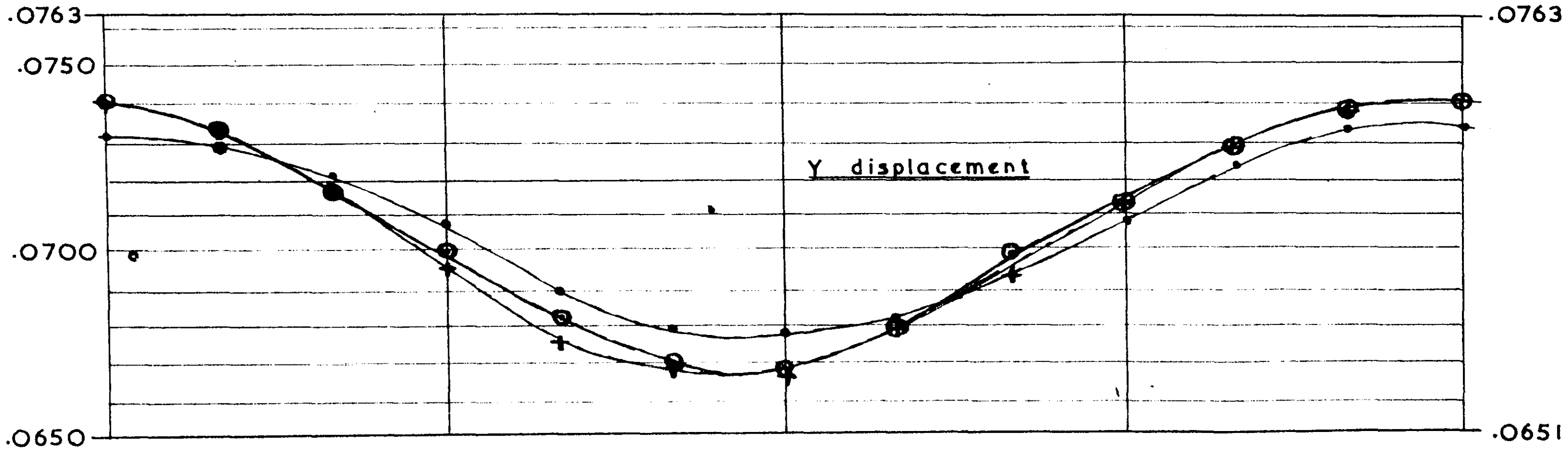

.0763
.0750
.0700
.0650
.0763
.0651
Y displacement

FIG. 44

conditions.

4.6 SHAFT RUNNING CENTRALLY IN BEARING

This experiment, simulating the Petroff bearing, was carried out to make a further study of cavitation. The bearing was adjusted to its central position, no loads or displacements being applied, and the shaft run at its normal speed. Closer adjustment was made by examining the pressure distribution round the shaft, and finally the total range of pressure was reduced to, 0 to -2 lb. per in^2. It was found that, under these conditions, a stream of bubbles came from the bearing, see Fig.45, quite as strong as in earlier cases, and yet apparently produced purely by the shearing of the film.

To make a closer inspection of this phenomenon, a glass test-tube was obtained having an internal diameter of 0.895 in.; a brass cylinder, 1.5 in. long, 0.880 in. diameter, fitted with a length of screwed rod, was spun at 600 R.P.M. inside the test-tube, containing a quantity of oil. A cloud of small bubbles soon formed in the free oil above and beneath the cylinder, and when the cylinder was stopped, small bubbles could be seen in the oil film between the cylinder and the tube. The bubbles in the free/

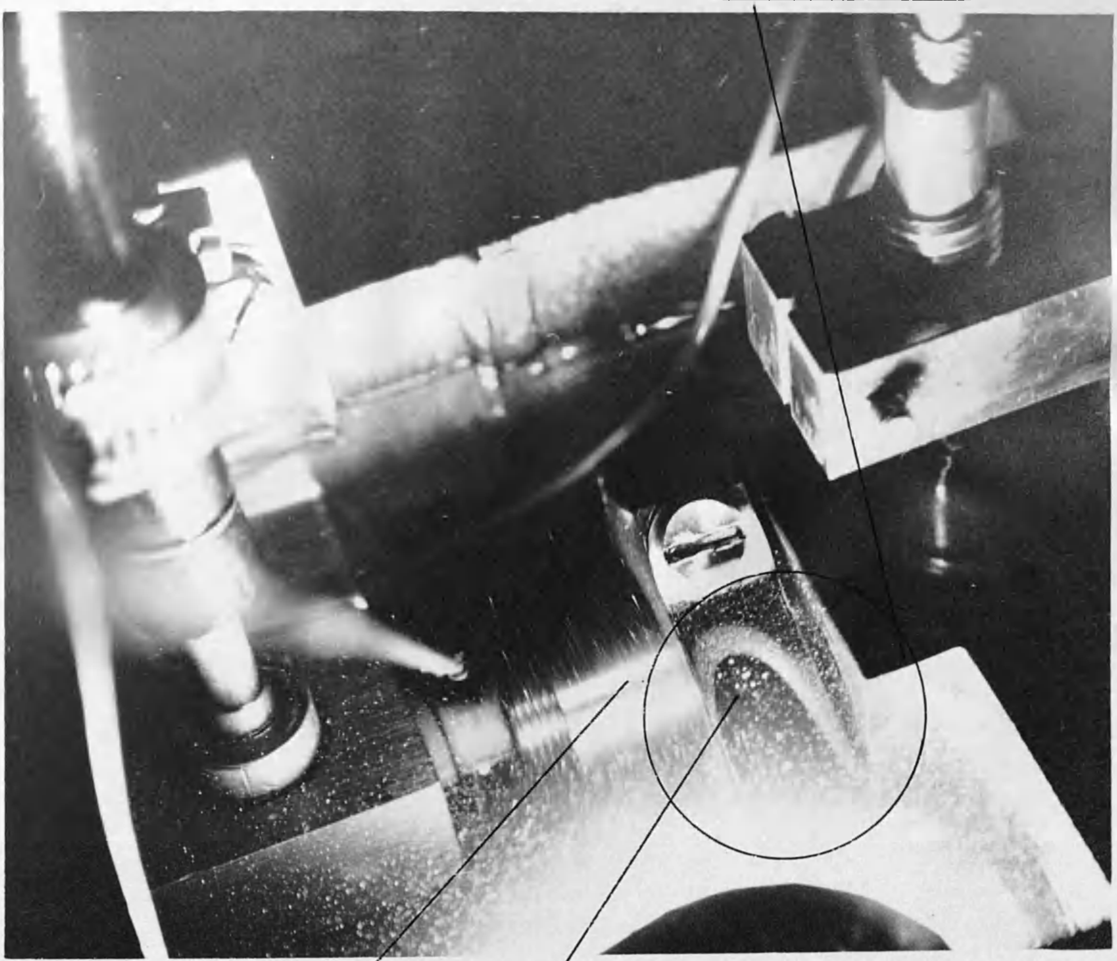
free oil were much more dispersed than those found in the bearing machine, where they are confined to a narrow channel.

While it is possible that the air bubbles are formed in very local low pressure regions, caused by irregularities of the surfaces, it is suggested that the viscous shear of the oil is sufficient to bring at least some air out of solution.

The test-tube apparatus provided a simple means of examining the behaviour of an oil film, and is to be recommended; it was placed in a vertical drill.

It was noted that if the test-tube was loaded in one direction, the oil film would break, a number of long shaped circumferential bubbles forming side by side, with oil streaming between them./

Oil surface level with top of bearing block



Shaft   Steady

FIG.45

A stream of bubbles issuing from the bearing when the shaft is running centrally in its clearance. The bubbles only become visible when a light is placed beneath the surface of the oil.

them.

4.7 FLUCTUATING LOAD - SHAFT ROTATING 1:1 WITH PRESSURE FEED

In the earlier test under similar conditions, recorded in §4.4 above, when only simple oil bath lubrication was employed, it was found that the film pressure diminished rapidly as the quantity of air, trapped in the bearing, grew and so reduced the quantity of oil in the bearing. The force pump and oil circulating system, previously used, was therefore employed to pump oil into an oil hole, 0.10 in. diameter, which was drilled into the bearing as shown in Fig.14b. This arrangement gives the bearing a charge of oil once per cycle, and it was hoped thereby to achieve steady conditions.

4.71 Oil supply

The force pump is driven by an eccentric on the countershaft and therefore gives a periodic charge of oil which may be set to occur at any desired point of the load cycle. For this test, it was arranged that the unloaded side of the bearing would be charged with oil, and that the charge would have reached its peak value before maximum/

maximum load occurred on that side. Oil entering at $\theta = 135$ is carried round towards $\theta = 180$ by the rotation of the shaft, ensuring that this side of the bearing has a good supply of oil. The oil bath was maintained at its normal level.

The actual pressure of the oil supply was measured by the balanced pressure piston in the following manner: the spring levers were removed so that the bearing was completely free, and also, the drive to the shaft was removed. The shaft was then rotated to the position where the piston faced the oil hole, and, with the machine running, the oil relief valve was adjusted to give a maximum pressure of approximately 40 lb. per in^2. The value of the supply pressure throughout the cycle, was obtained by use of the phasing contact in the usual manner, and is plotted in Fig.46. The base ϕ, of this graph, is common throughout this test and the test with rotating load which follows.

4.72 Conditions of test

As in the earlier test with this type of loading, the bearing was loaded in the horizontal direction only, and freed in the vertical direction by removing the Y spring lever. This should give zero vertical load, apart from friction and the weight of the bearing block. Although the bearing is loaded only in the horizontal direction, it/

it must move slightly in the vertical direction due to rotation of the shaft.

The oil supply was maintained at a temperature of 70°F. by means of the water bath through which the oil is circulated, and the shaft and variable eccentric, geared in the ratio 1:1 to the countershaft, run at the normal speed of 530 R.P.M.

4.73 Cavitation bubbles

A continuous stream of bubbles of varying size was again observed, and on stopping the machine a few big bubbles emerged from the bearing. The film pressure readings were remarkably stable and it was possible to read quite precisely to $\pm\frac{1}{2}$ lb. per in^2. There was no noticeable drop in film pressure from starting.

It was found that no bubbles came from the bearing when the chain drives to the shaft and variable eccentric were removed, only the oil pump then remaining in operation, proving that the bubbles were not produced by the oil pump. Also, when a lamp was placed in the oil flowing from the relief valve, only a very small number of bubbles could be seen.

4.74 <u>Readings</u>

Film pressure readings were taken at 10^{o} intervals round the shaft at each of the five axial sections of the bearing and for 12 load phase points, ϕ = 0,30, etc. to 330, see Figs.48 to 59.

The X-load and X&Y displacements were also recorded, and are shown in Fig.47 in which the pressure loads are also plotted for comparison. The pressure loads in directions X and Y were obtained by Simpson's Rule from the integrals:

$$\text{X-load} = -\int_0^{\ell}\int_0^{2\pi} r.p.\cos\theta \, d\theta \, d\ell,$$

$$\text{Y-load} = -\int_0^{\ell}\int_0^{2\pi} r.p.\sin\theta \, d\theta \, d\ell,$$

where r is radius of journal,

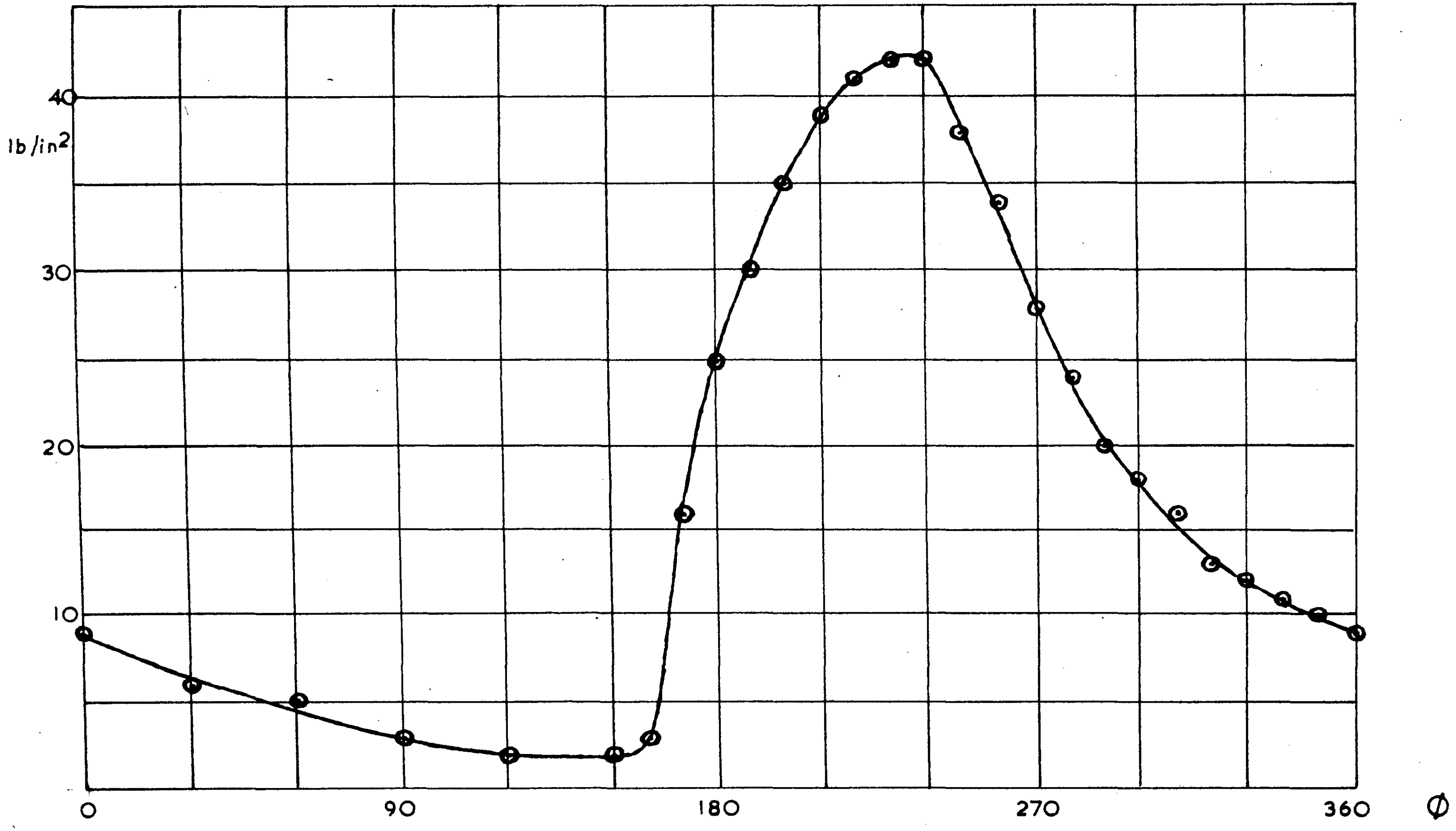
p is film pressure,

θ is measured from X-axis,

and ℓ is length of bearing.

In this case it is not practical to run the machine with shaft removed as the bearing is not located in the vertical direction.

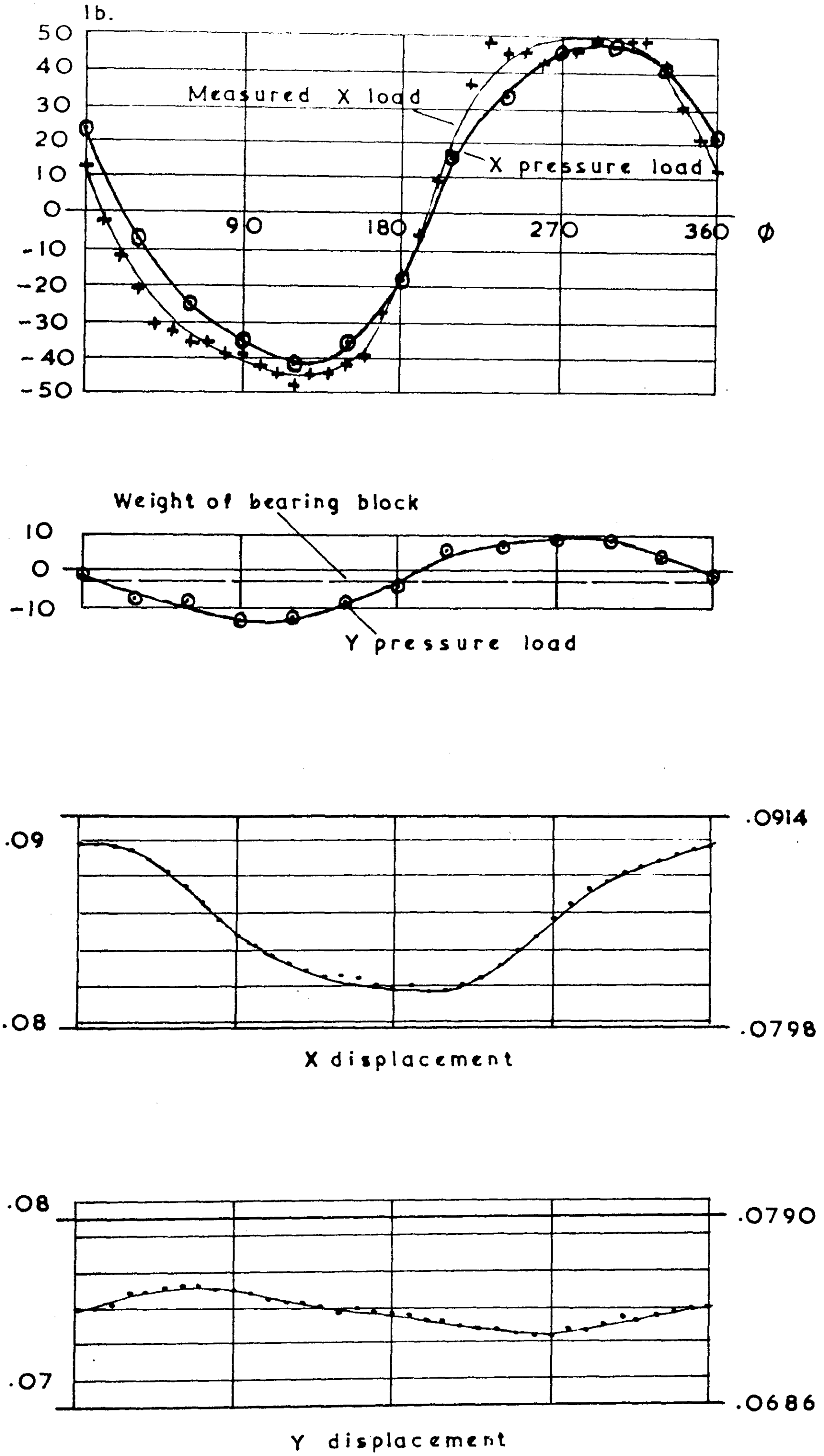
The oil pressure supply was successful in maintaining an oil film, where previously the oil had been almost completely expelled from the bearing in a short time./



OIL SUPPLY PRESSURE

Measured with no load and shaft stationary

FIG. 46



FLUCTUATING LOAD — SHAFT ROTATING 1:1 WITH PRESSURE FEED

FIG. 47

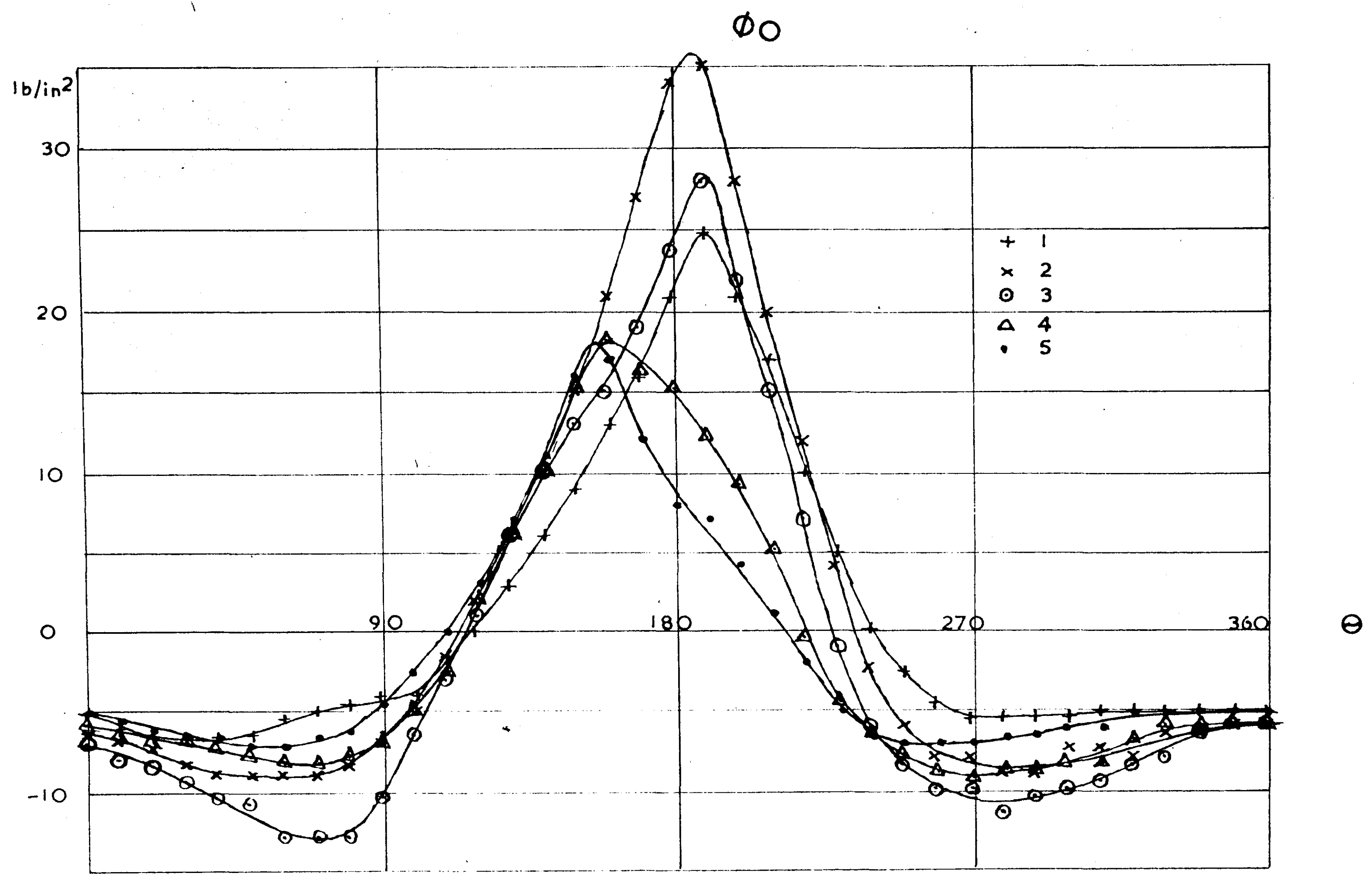
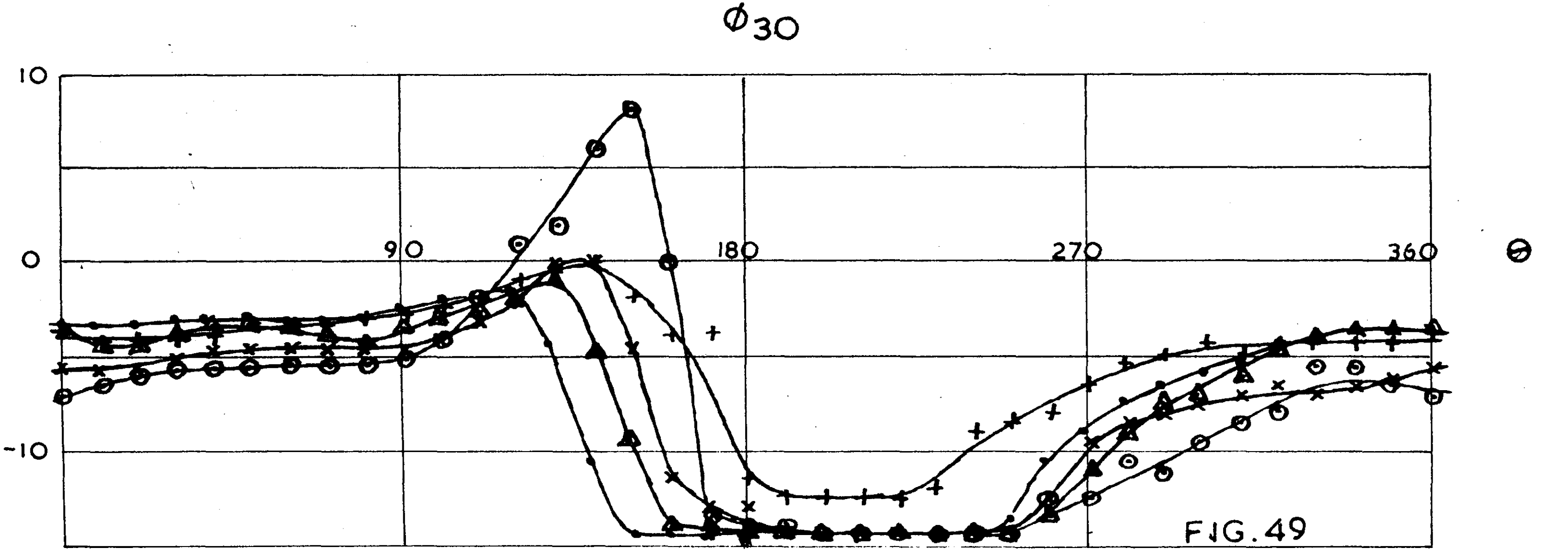
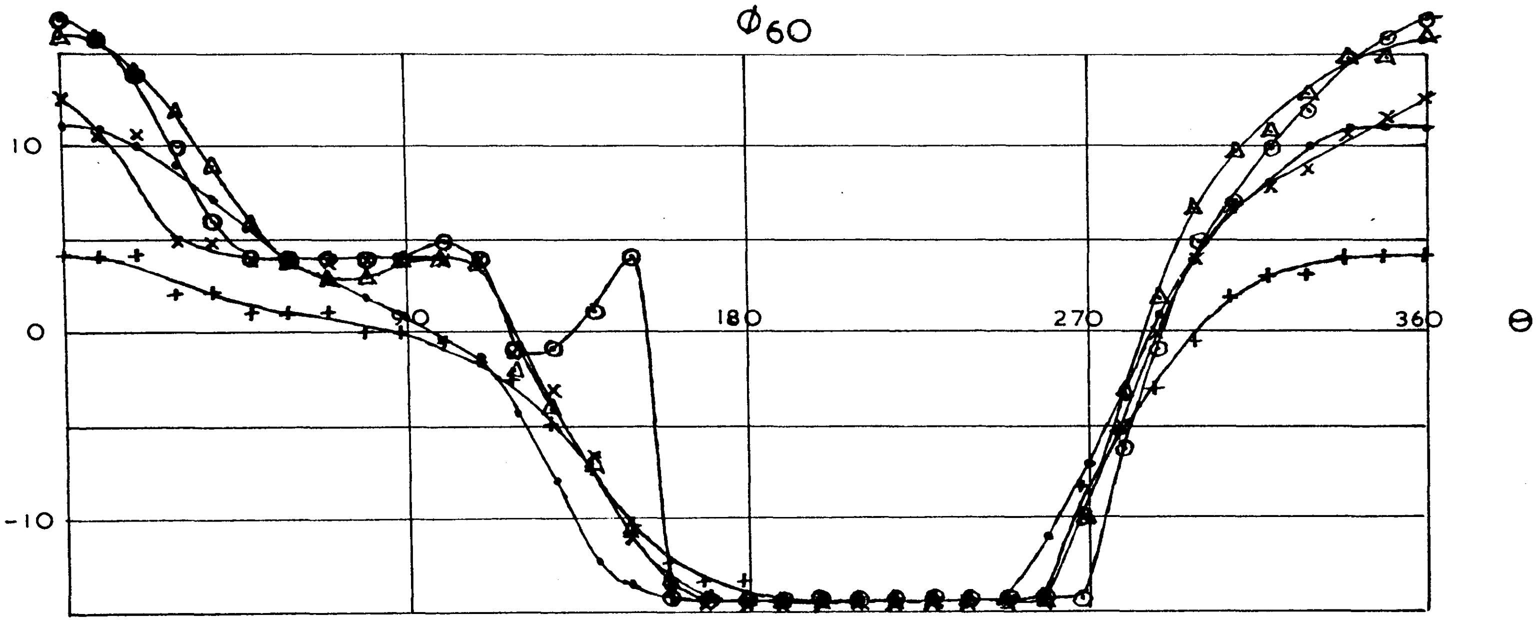


FIG. 48



ϕ_{30}
10
0
-10
90
180
270
360
θ

FIG. 49



ϕ_{60}
10
0
-10
90
180
270
360
θ

FIG. 50

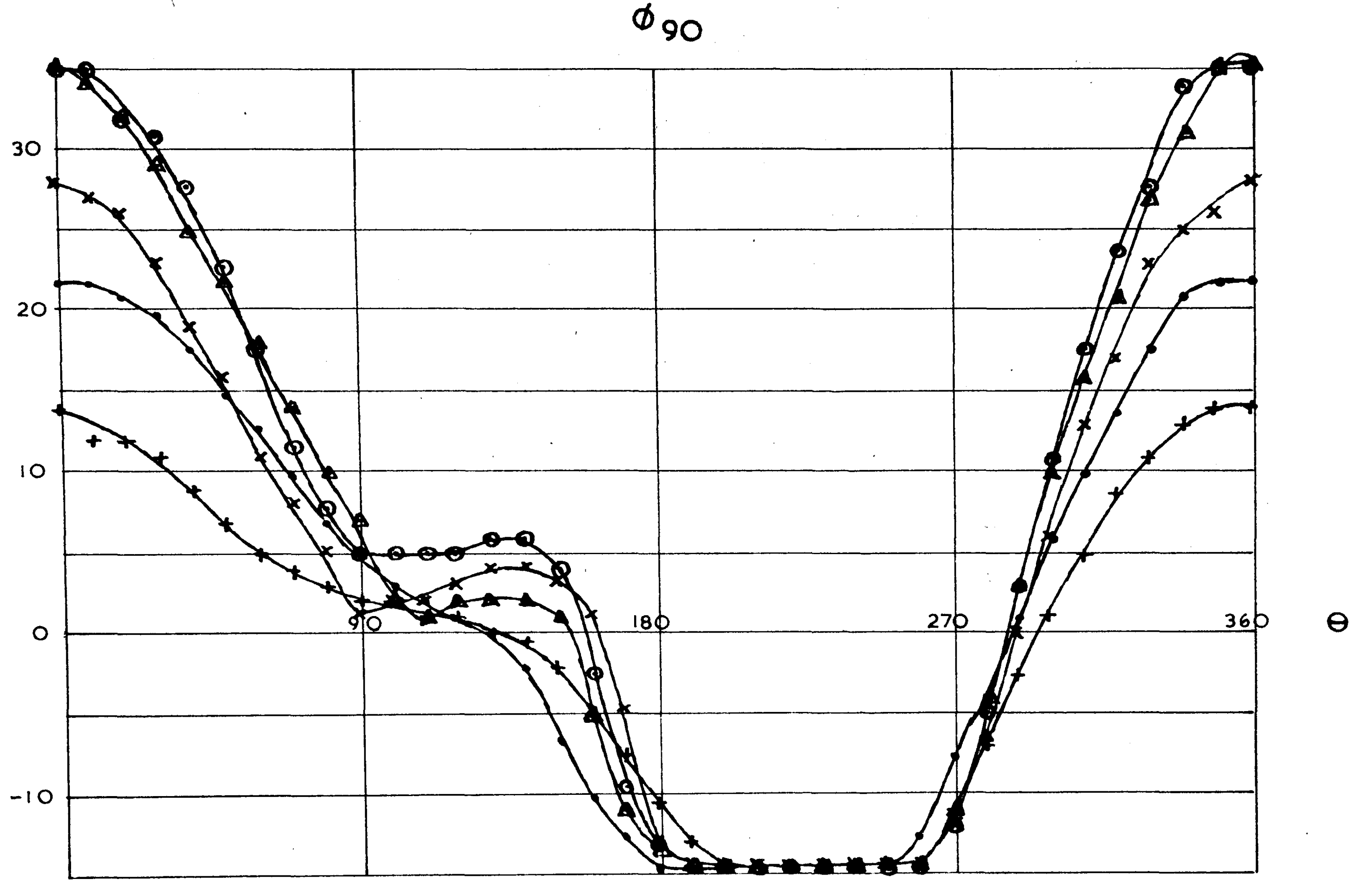


FIG. 51

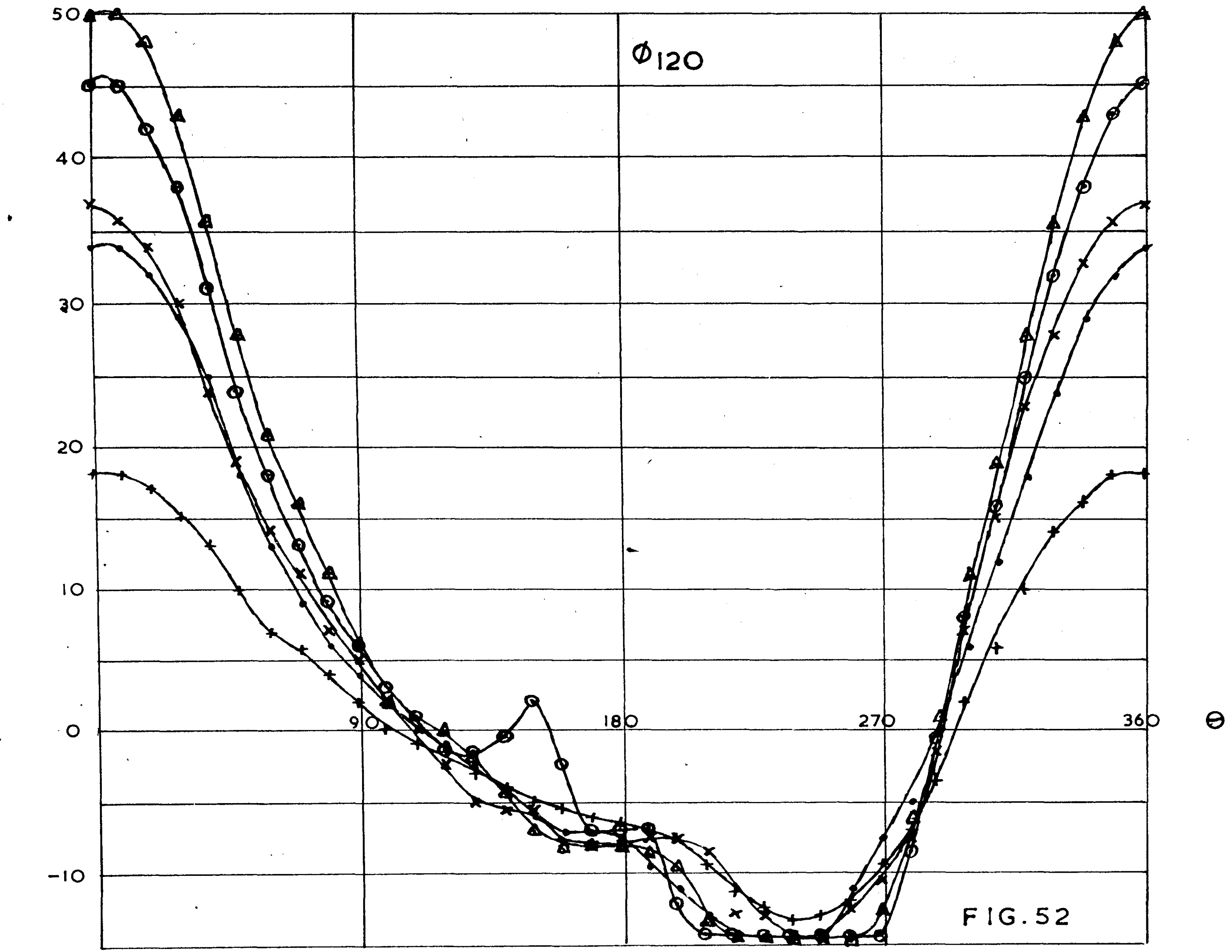


FIG. 52

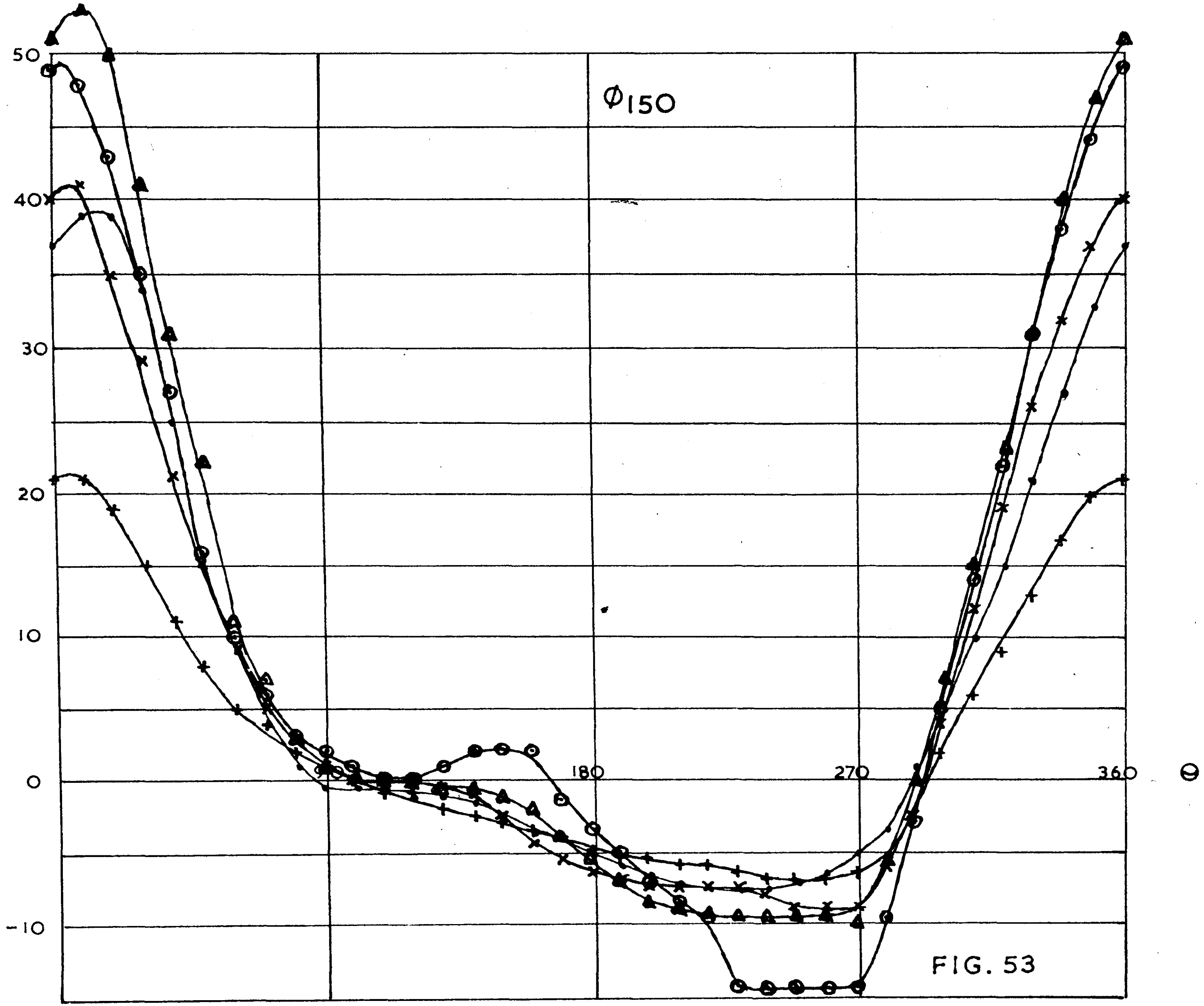


FIG. 53

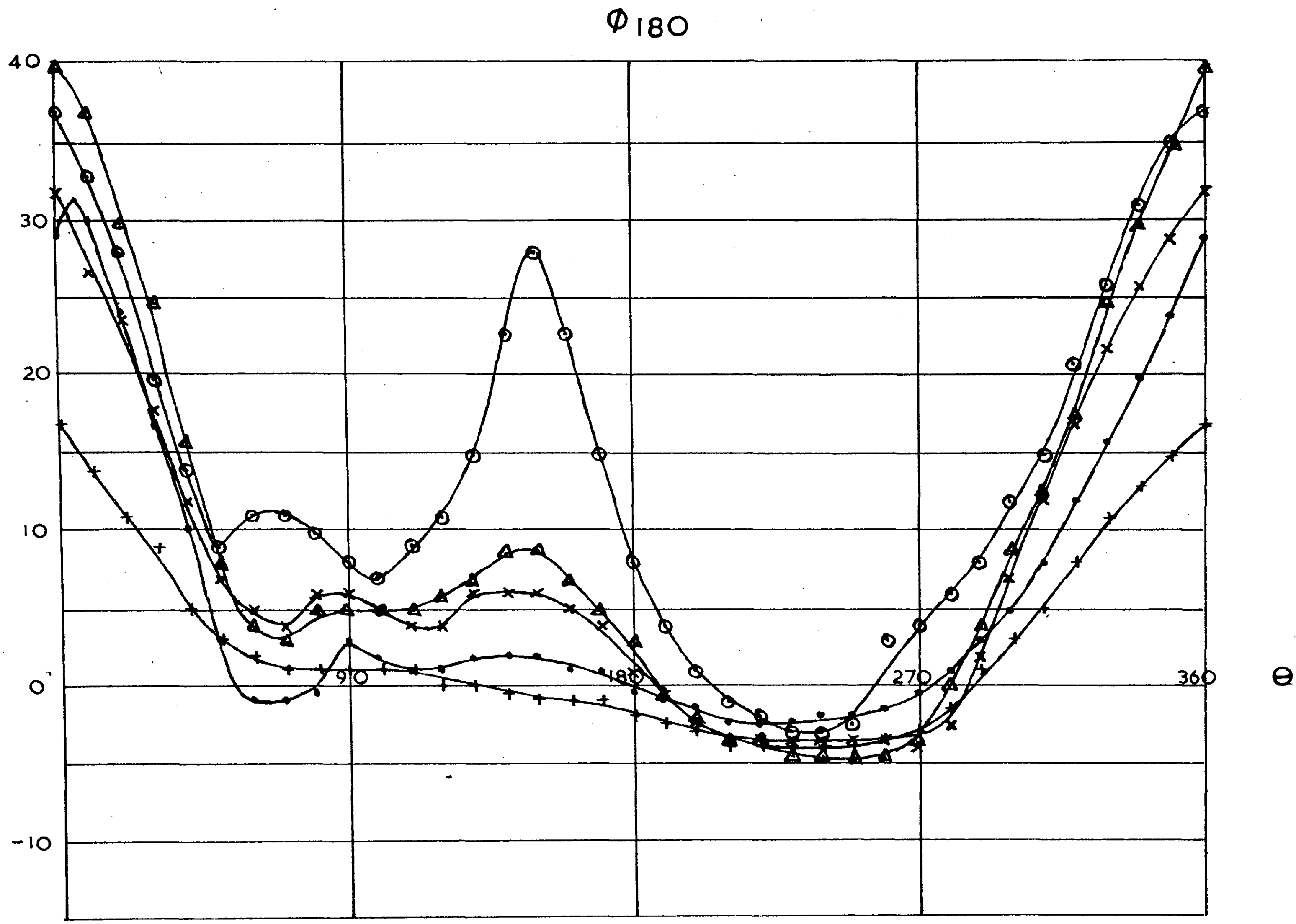


FIG. 54

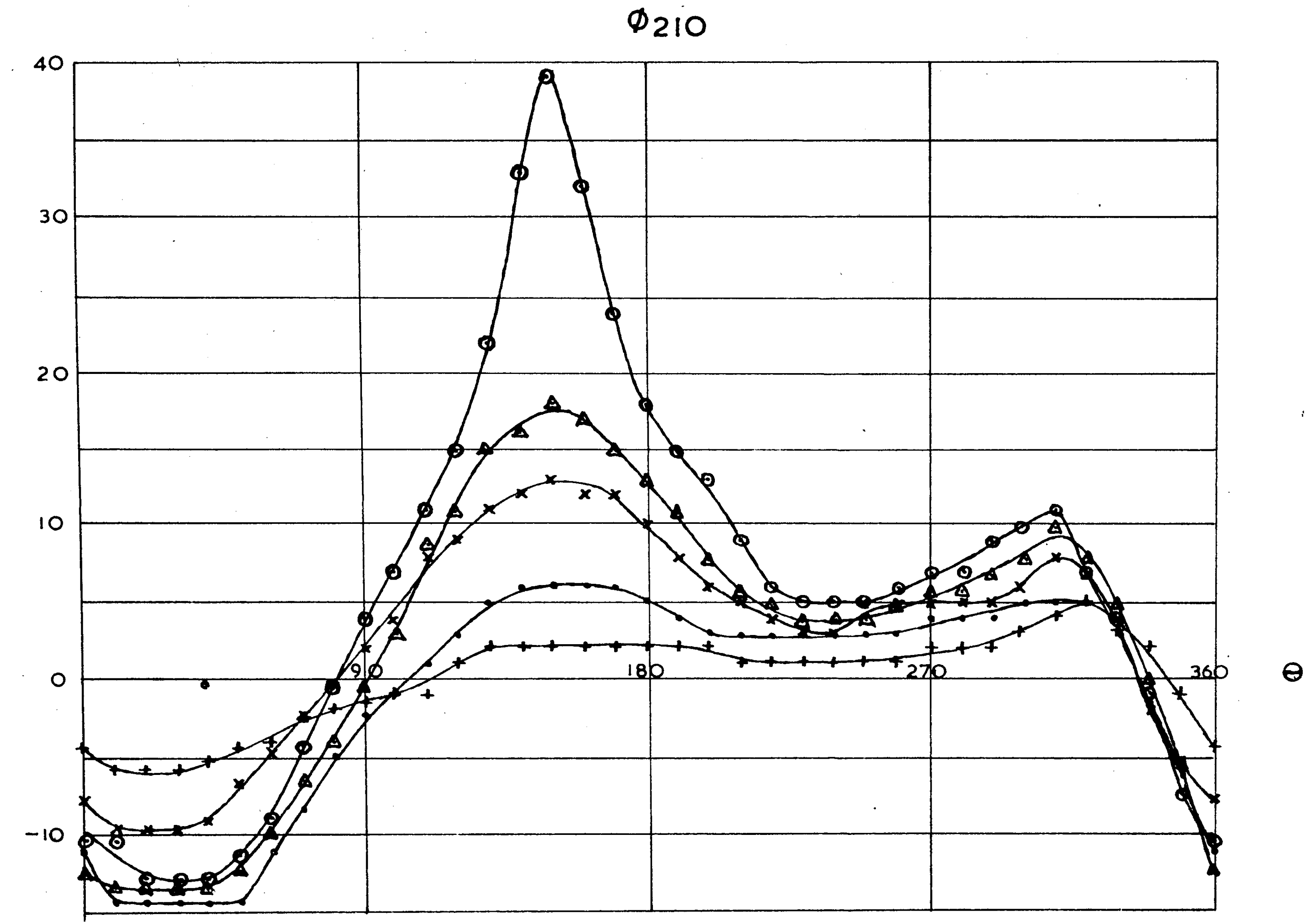


FIG. 55

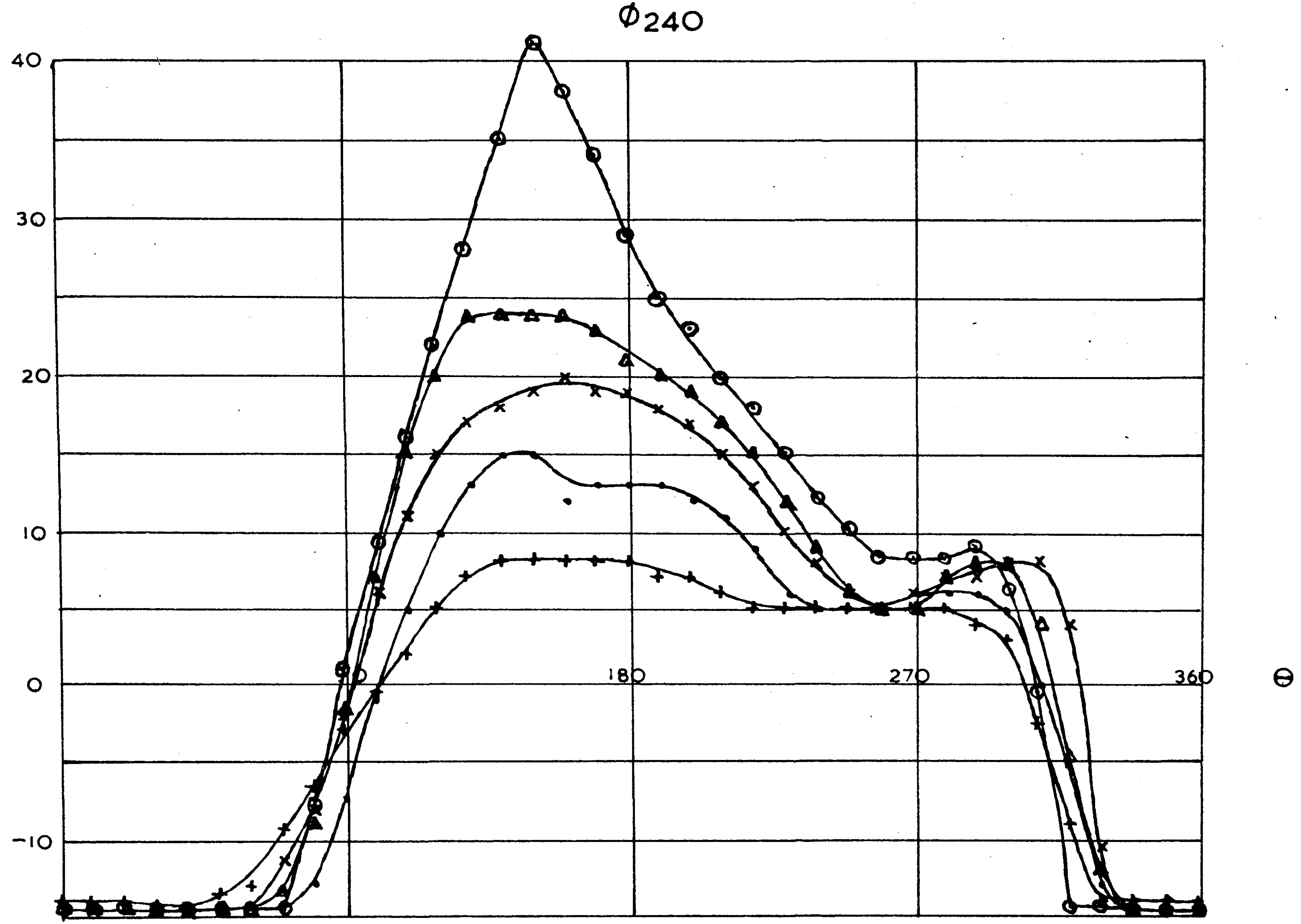


FIG. 56

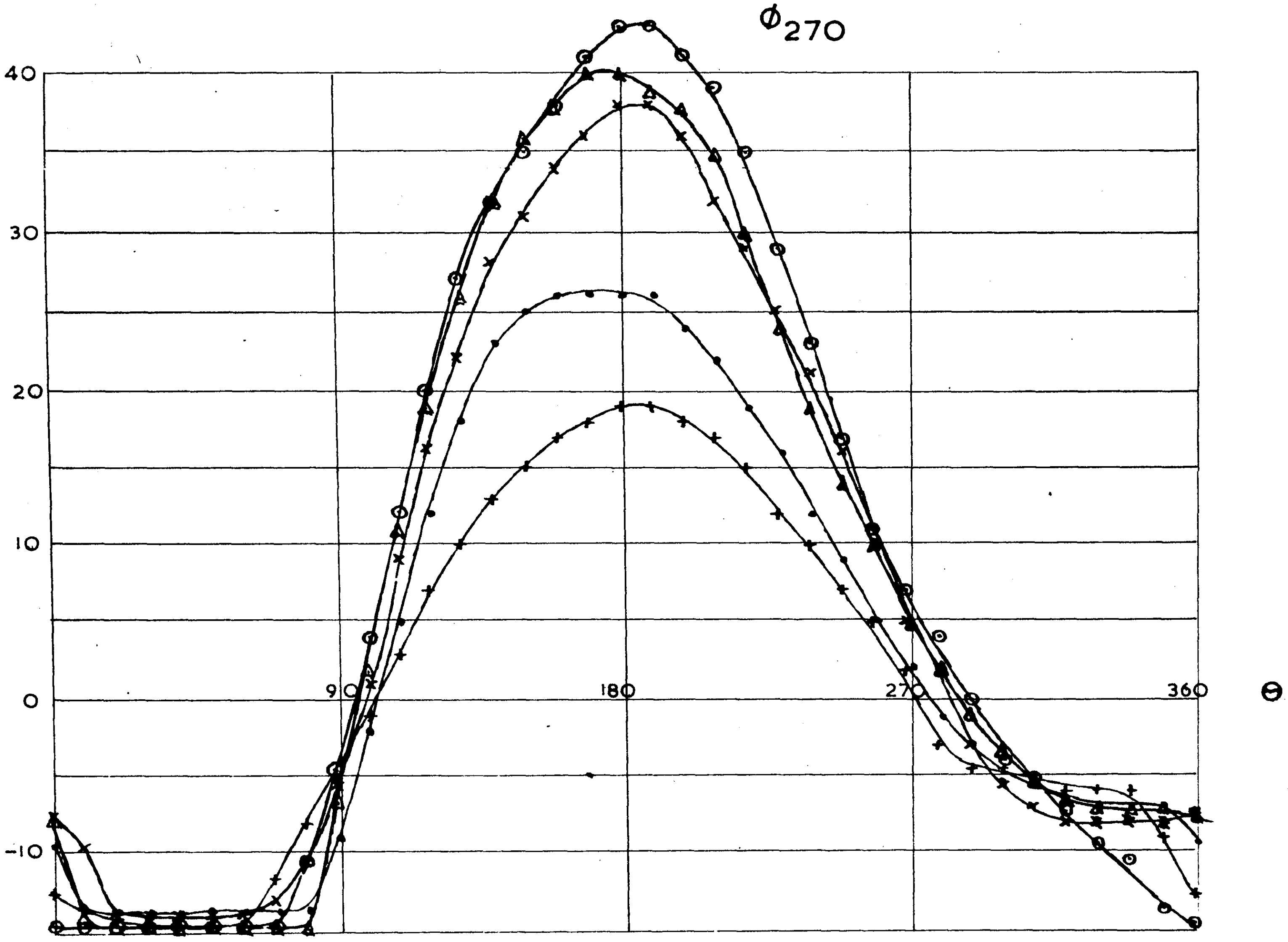


FIG. 57

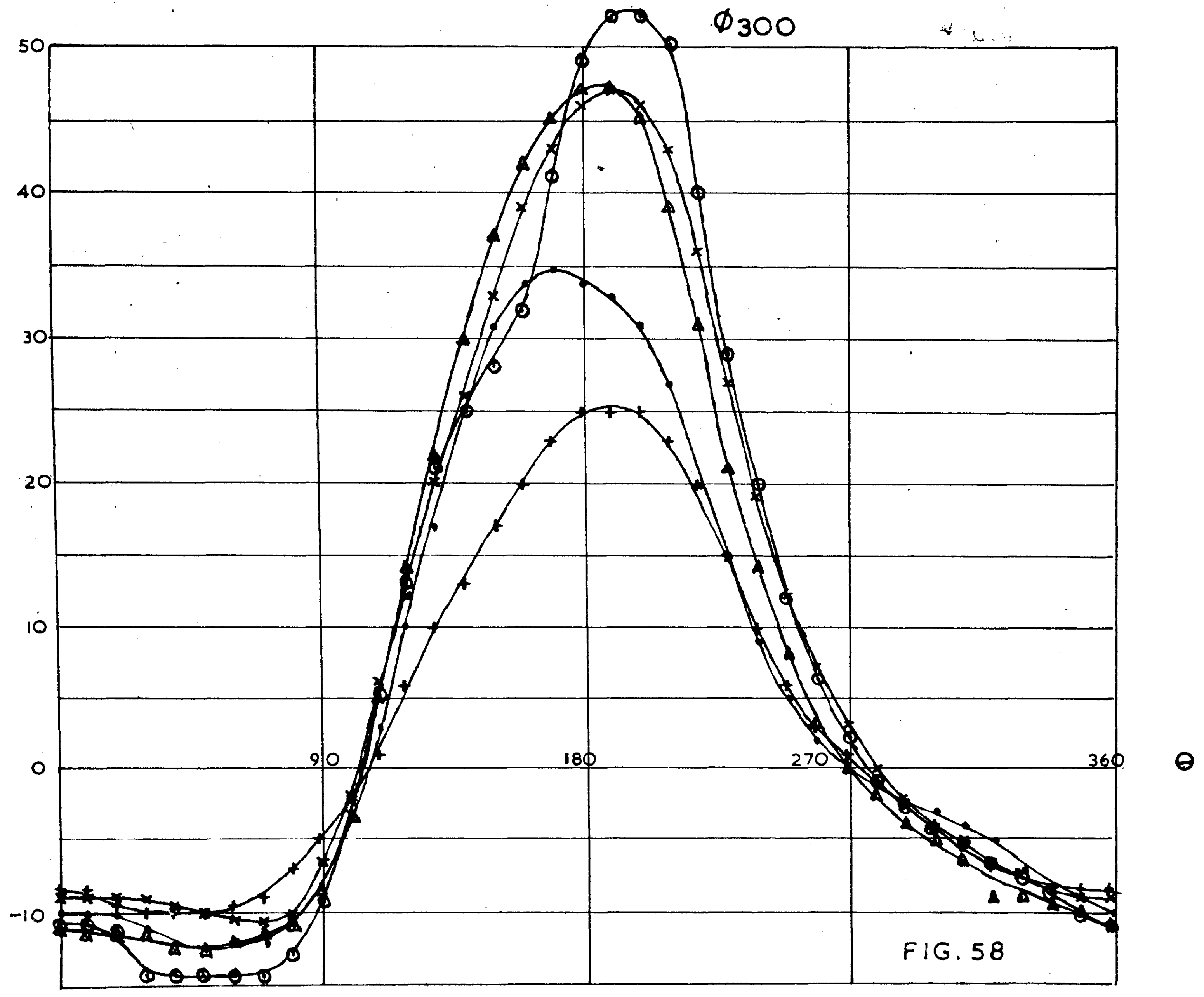


FIG. 58

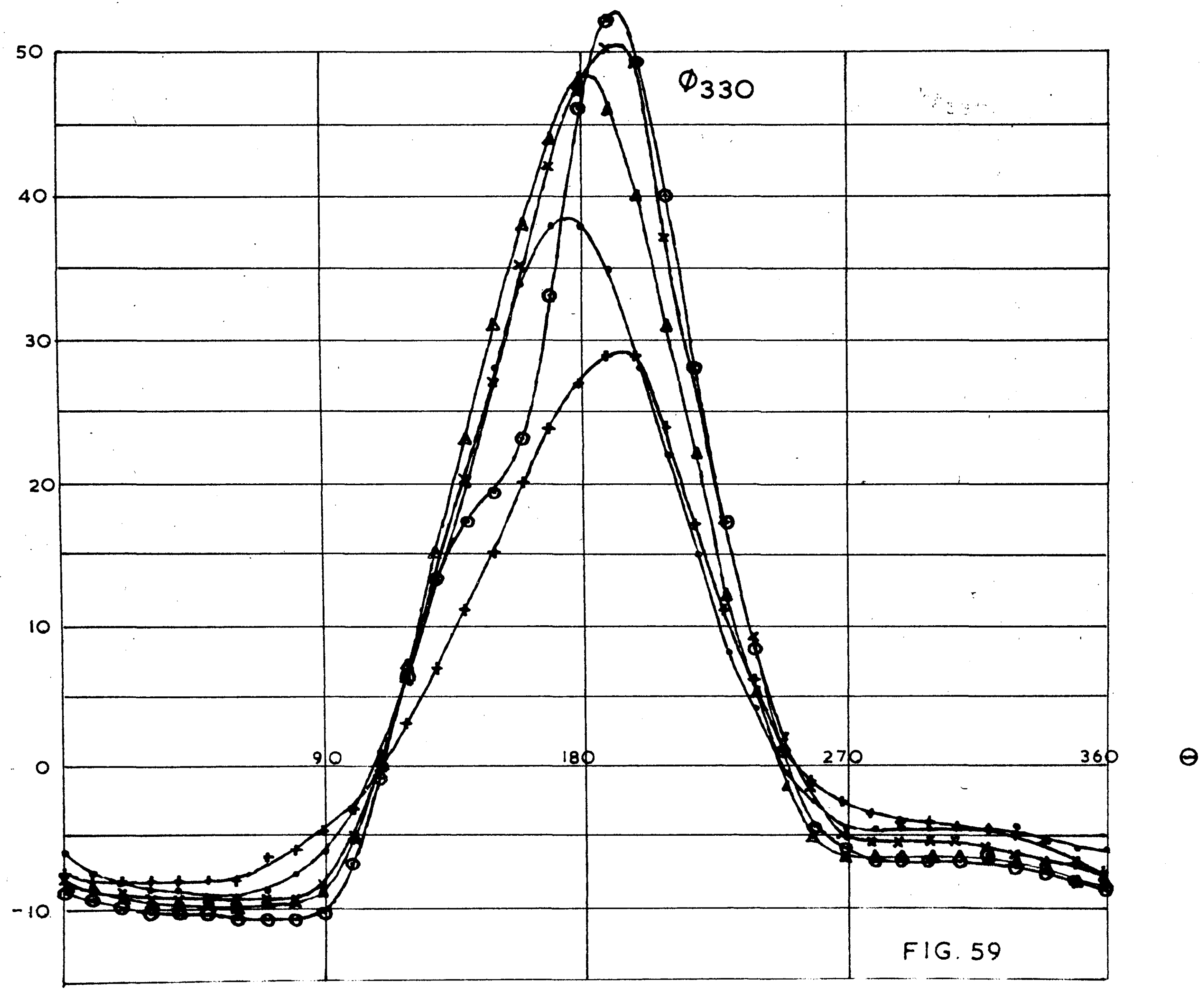


FIG. 59

time.

4.8 ROTATING LOAD - SHAFT ROTATING 1:1 WITH PRESSURE FEED

4.81 Test

This test was made immediately after that recorded in §4.7 above, with all settings undisturbed except that the Y-spring was replaced. The oil supply pressure is therefore as shown in Fig.46. X&Y loads and displacements were recorded, and also loads with shaft removed, to indicate the amount of friction. These results are presented in Fig.60.

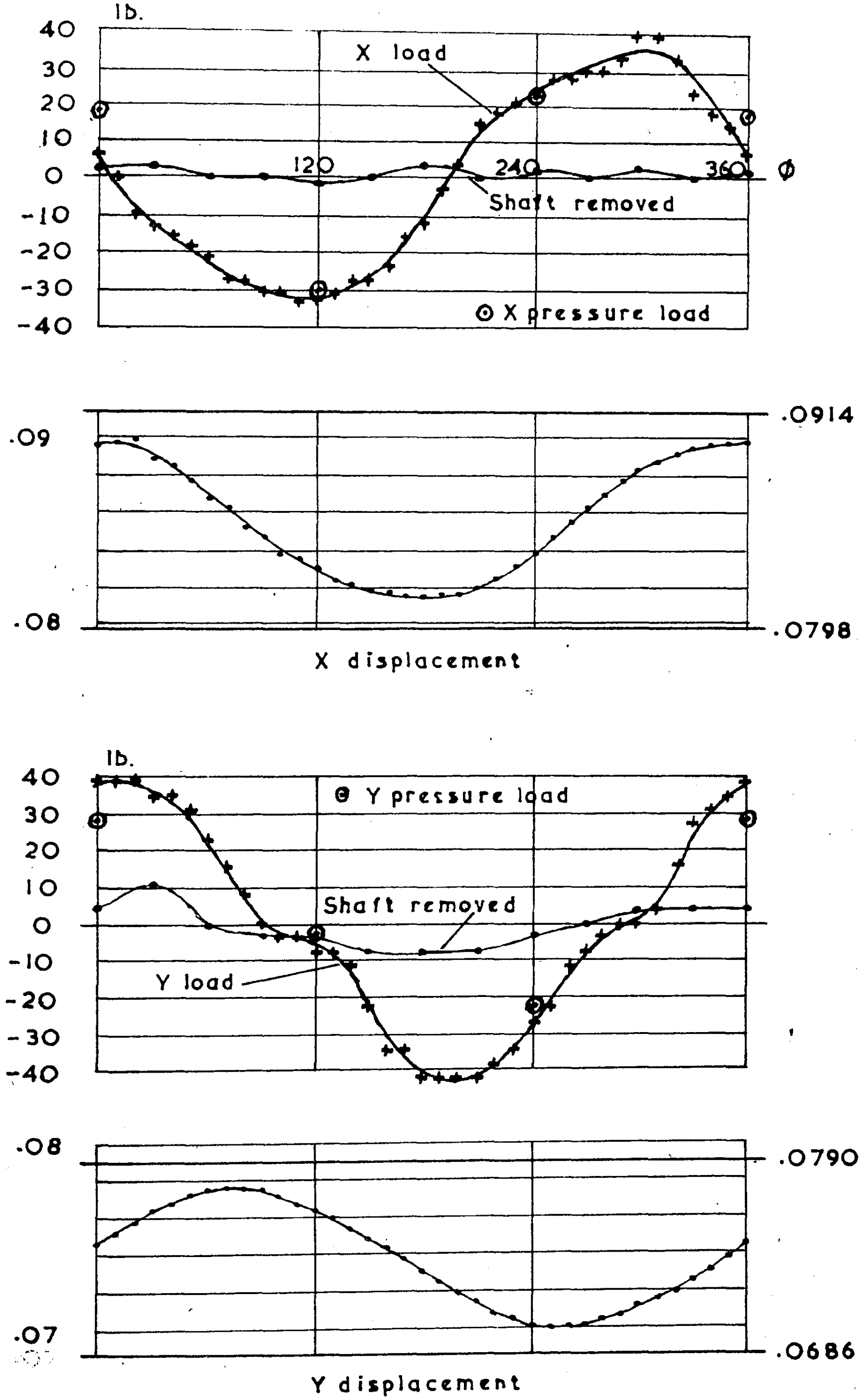
The pressure distribution was examined at three points in the load cycle, $\emptyset$ = 0,120 & 240, see Figs.61,62&63, and again the pressures were integrated horizontally and vertically for comparison with the measured applied load, see Fig.60.

The peak value of film pressure dropped by approximately 3 lb. per in^2. in the first few minutes running, but thereafter remained constant. A steady stream of bubbles was again observed, and on stopping the machine, a number of big bubbles came from the bearing. The film pressures were again very stable.

4.82 Pressure distribution

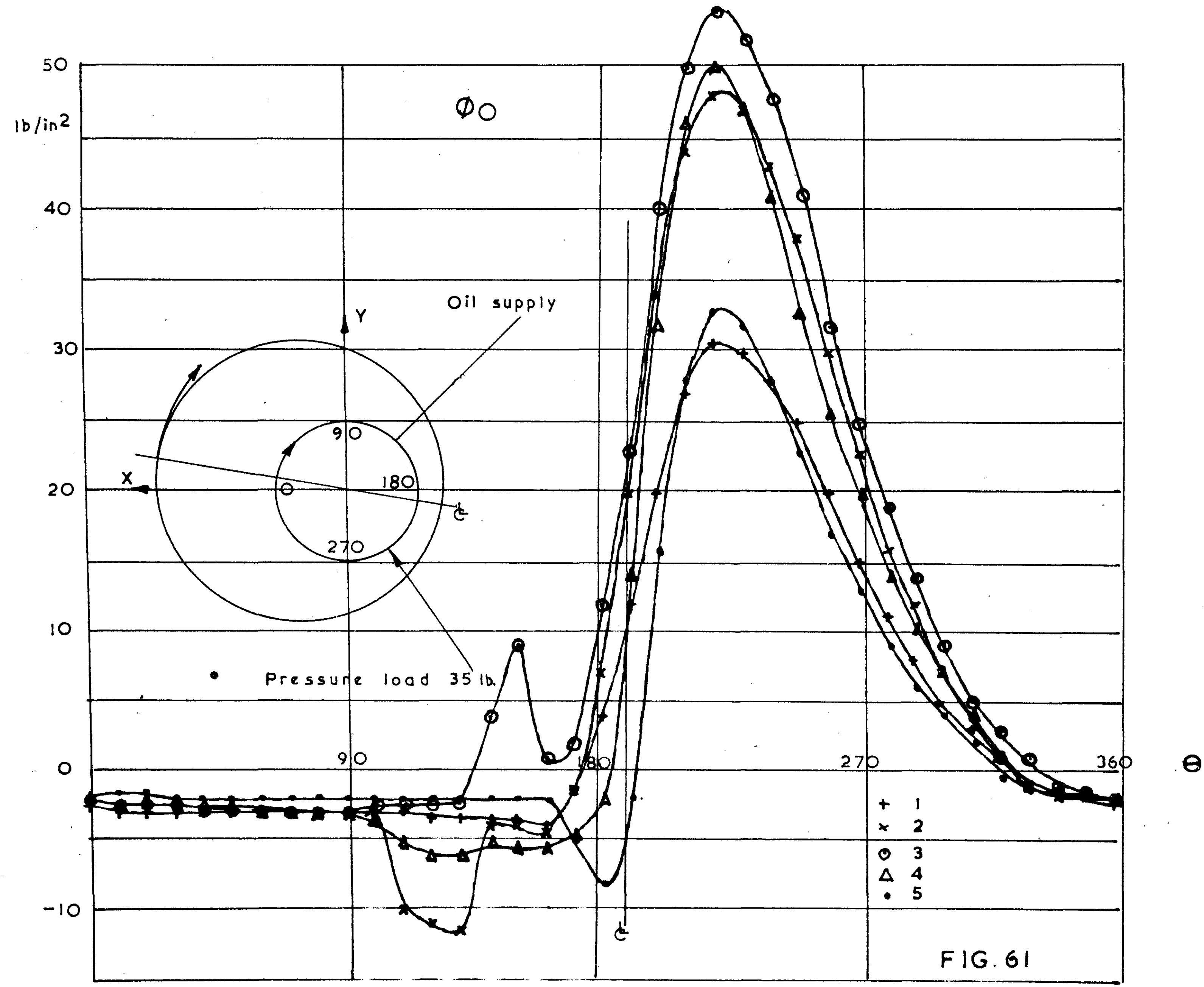
This test is of particular interest as it is similar to the normal case of a bearing under steady load, in which case the load is fixed relative to the bush, whereas in this test the load is fixed relative to the shaft. In order to give a clearer picture of the pressure distribution, Figs.61,62 & 63 have been redrawn in three-dimensional form, see Figs.64,65 & 66. It is seen that the axial pressure distribution is parabolic and that the pressure peak is followed by a sharp drop in pressure, approaching absolute zero over a narrow region. The remaining portion of the bearing, up to the point where the film commences, is at constant pressure, just slightly beneath atmospheric pressure.

Although the oil pressure supply can be detected in the pressure curves, it is seen that the effect is very local./



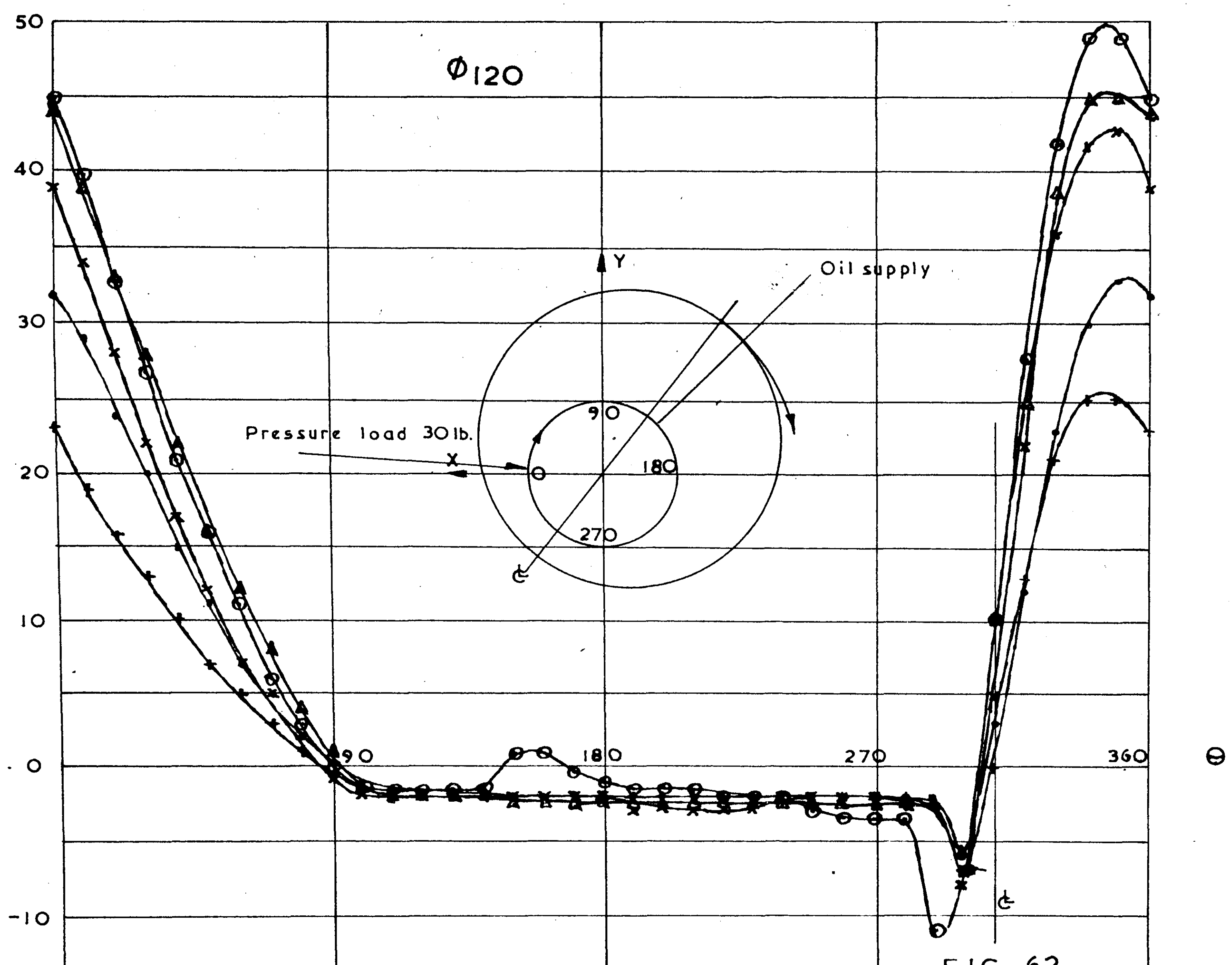
ROTATING LOAD –
SHAFT ROTATING 1:1
WITH PRESSURE FEED

FIG. 60



50
lb/in2
40
30
20
10
0
-10
Ø0
Oil supply
Y
X
90
180
270
0
Pressure load 35 lb.
90
180
270
360
θ
+ 1
× 2
⊙ 3
△ 4
• 5

FIG. 61



ϕ_{120}
50
40
30
20
10
0
-10
90
180
270
360
Θ
Y
X
Oil supply
Pressure load 30 lb.
90
180
270

FIG. 62

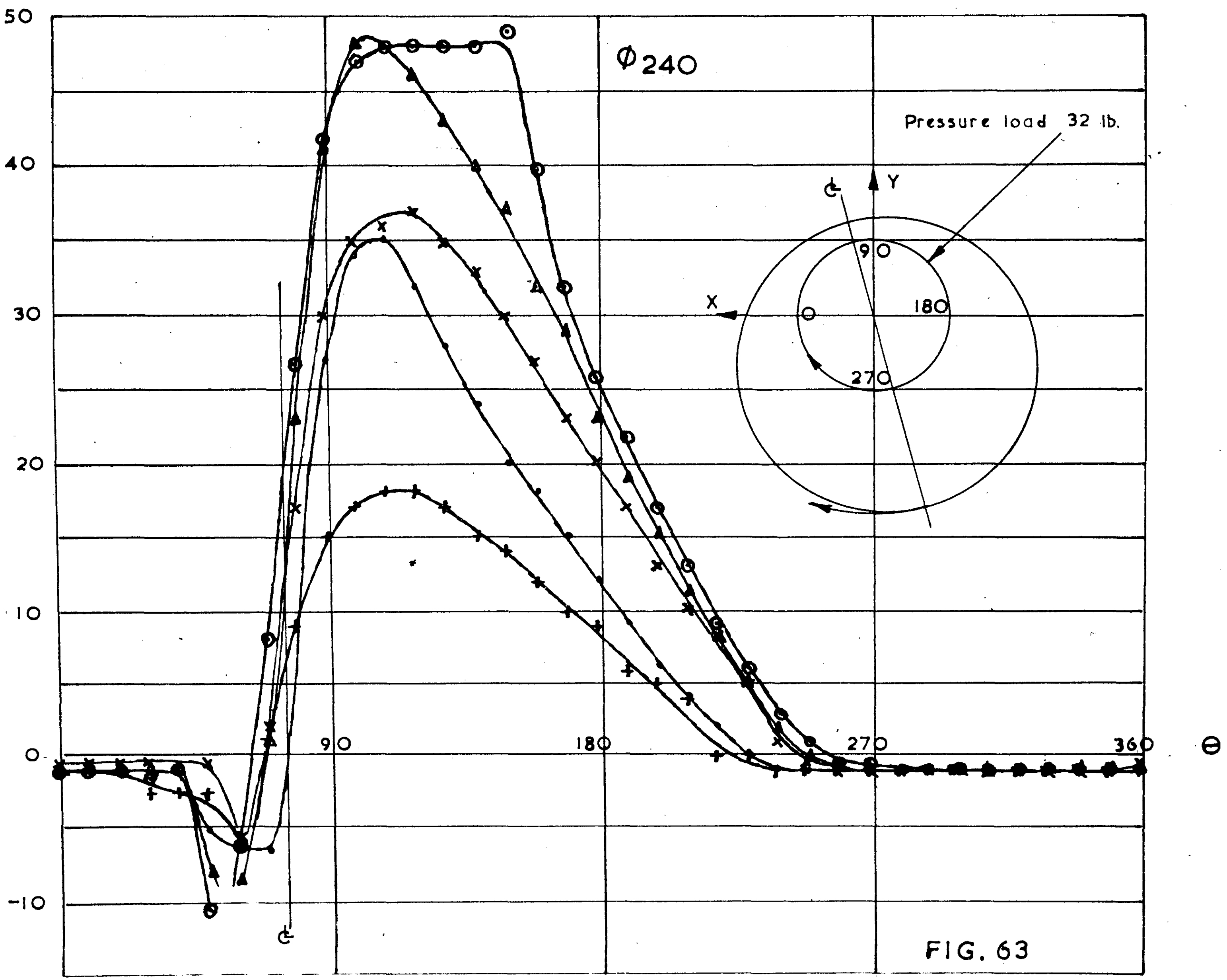
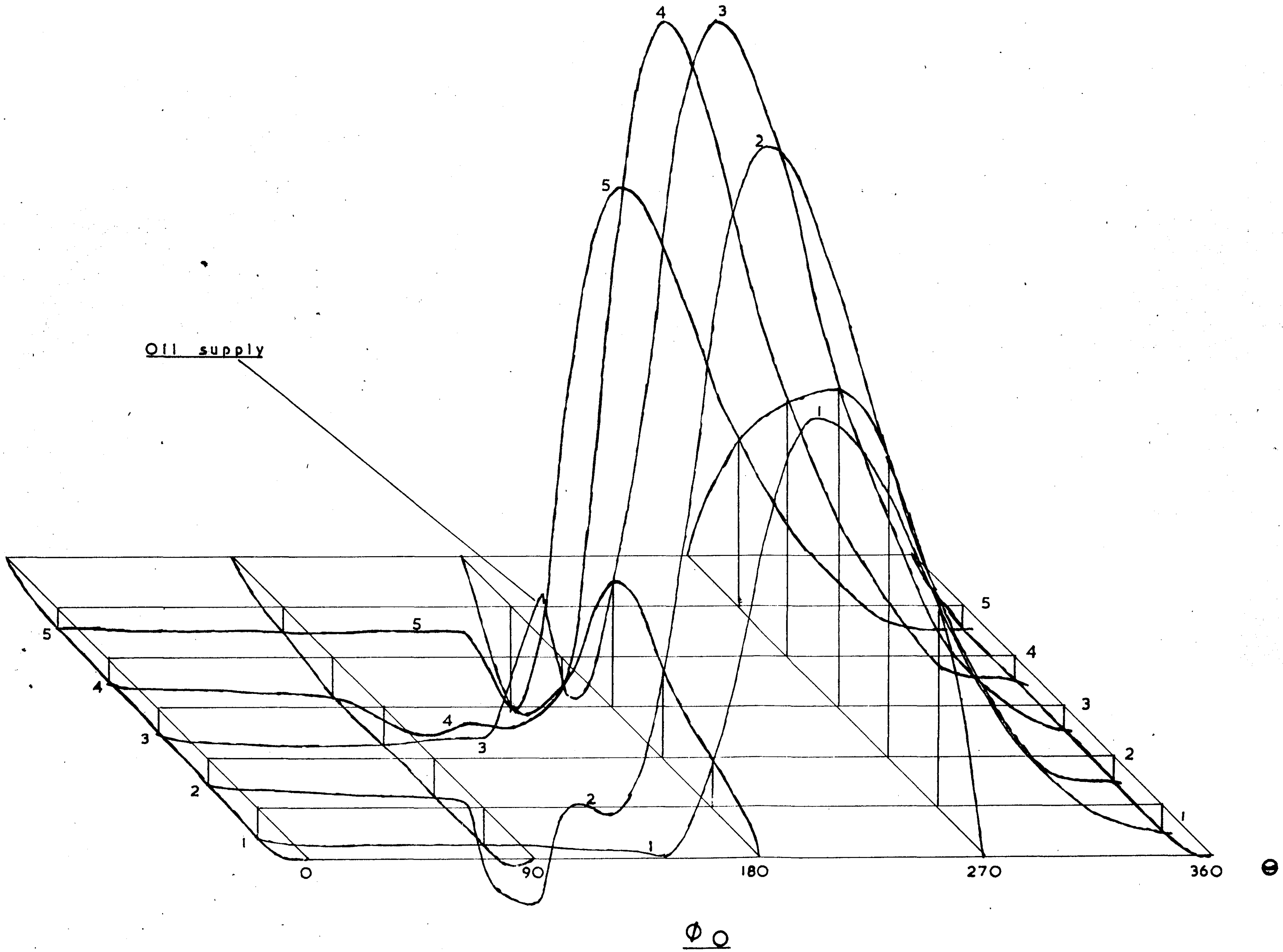


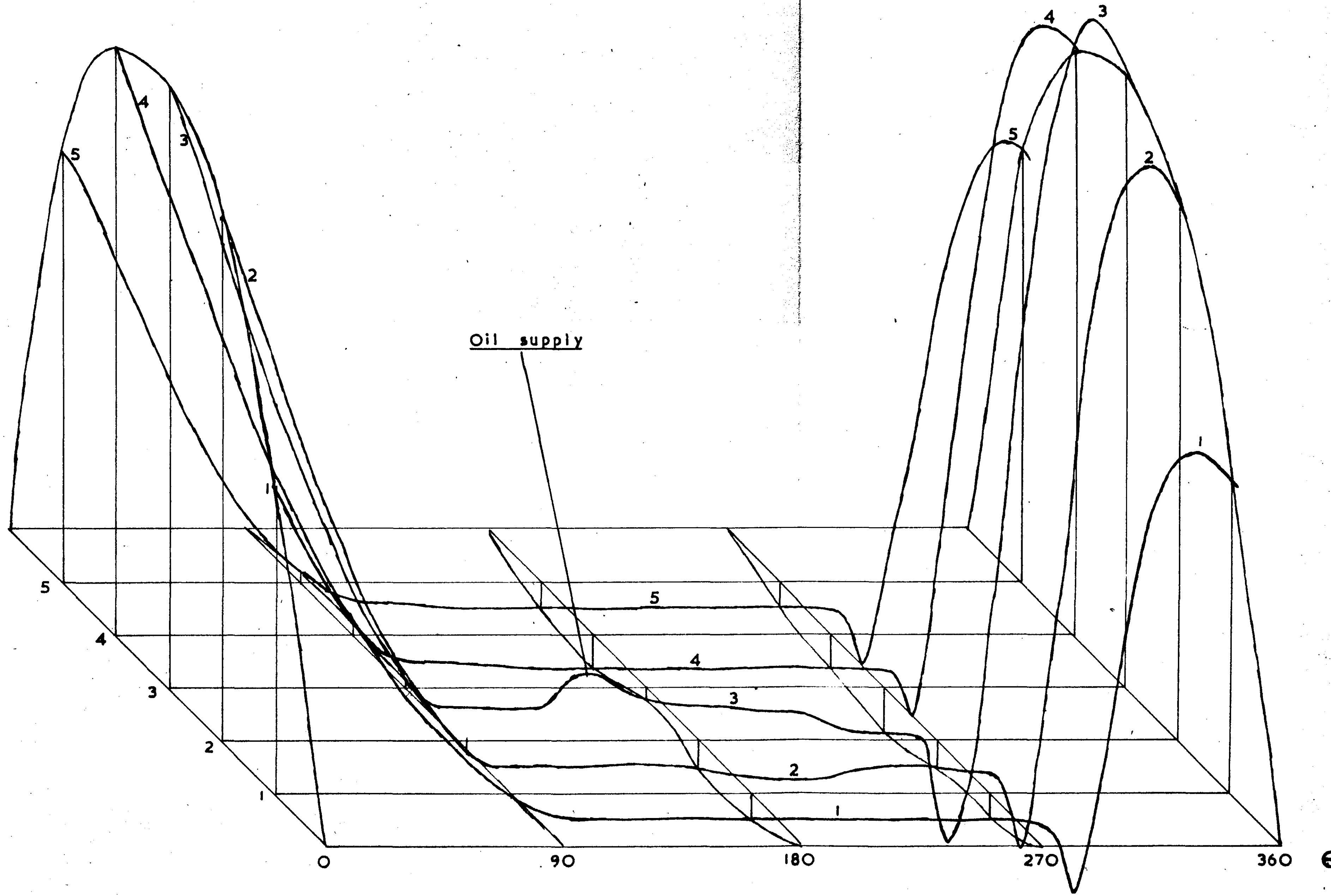
FIG. 63



ROTATING LOAD - SHAFT ROTATING 1:1

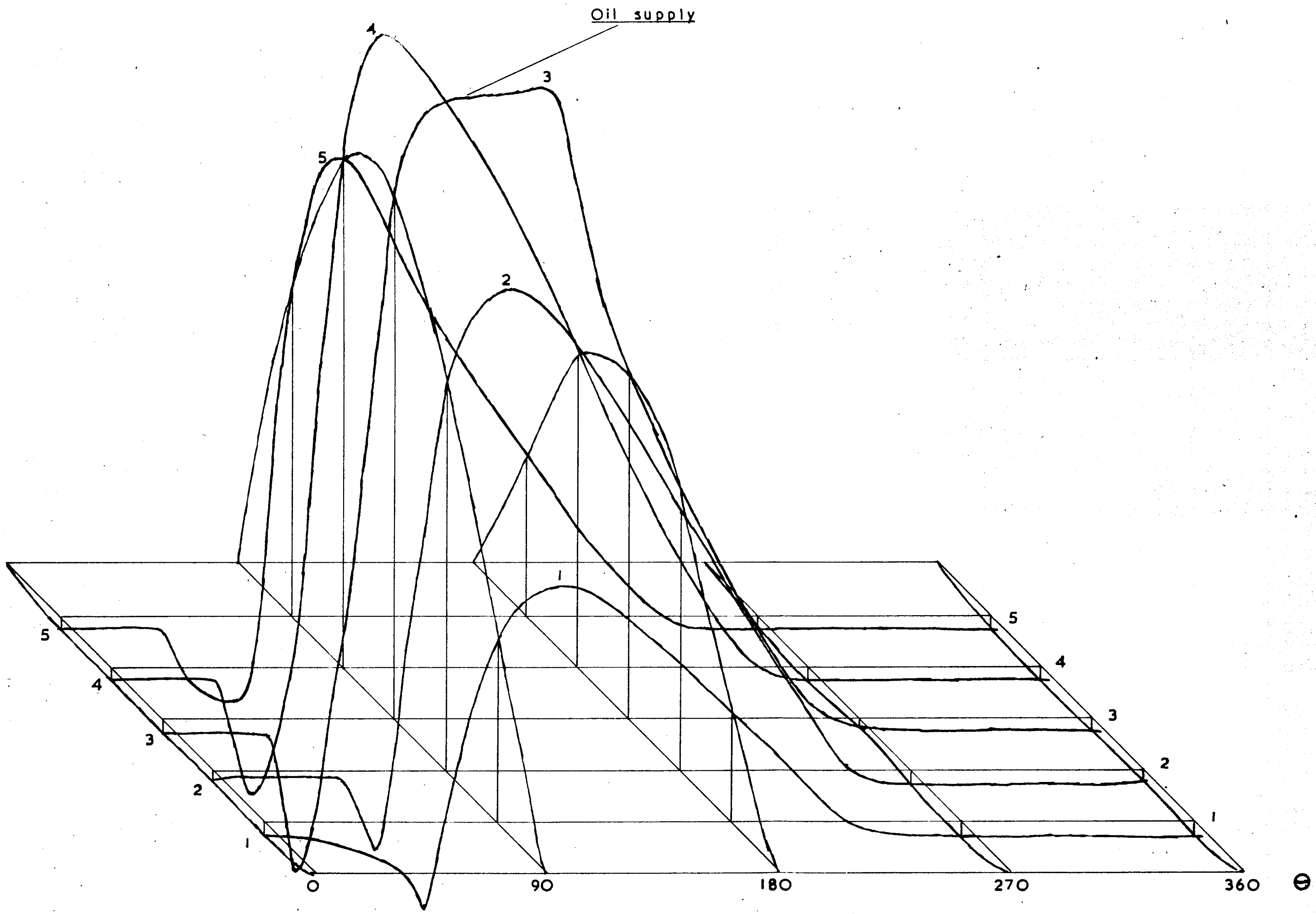
WITH PRESSURE FEED

FIG. 64



ϕ_{120}

FIG. 65



Φ_{240}

FIG. 66

local.

5. COMPARISON WITH THEORY

5.1 THEORETICAL PERFORMANCE

The existing theory of journal bearings under fluctuating load[3&6], contains a number of solutions for particular conditions. Two of these may be directly compared with experimental measurements now presented. Since the motion of the bearing, viscosity of oil, and clearance are accurately measured, the theoretical load may be evaluated for comparison with the experimental value obtained.

Firstly, when the shaft is stationary and the bearing oscillates along the line of centres, as in §4.1, the theoretical load per inch width, P, may be derived from*

$$P = \frac{12\pi\lambda V R^3}{r^3}\frac{1}{(1-\epsilon^2)^{3/2}} ,$$

where, R is radius of journal,

r is radial clearance,

ϵ is eccentricity,

V is velocity of bearing,

λ is viscosity./

* This equation is from Swift[6], also derived from Harrison[3] in Appendix 1.

viscosity.

The total X-travel of the bearing in this test was 0.0034 in., see Fig.20, giving an amplitude of 0.0017 in. about the mean position. The peak load will occur at the mean position, where the velocity of the bearing is greatest. Assuming sinusoidal motion, $V = 0.0017\,\omega$; at 530 R.P.M., $\omega = 55.5$ rad/sec., and so, $V = 0.0943$ in. per sec.

The viscosity of oil was measured as 290 cp., equivalent to 0.000042 Reyn.

Radius of journal is 0.49 in. and radial clearance is 0.0053 in.

Evaluating, P = 93 lb. giving a total load of <u>139 lb</u>. on a bearing 1.5 in. long.

The experimental value was approximately <u>40 lb</u>.

Secondly, when load and shaft are rotating at the same speed, as in §4.8, the theoretical load per inch width of bearing, W, may be obtained from the equation:

$$W = \frac{12\pi\lambda R^3}{r^2}\,\frac{\epsilon}{(2+\epsilon^2)(1-\epsilon^2)^{\frac{1}{2}}}\,\omega$$

where, R is radius of journal, r is radial clearance, ϵ is eccentricity, λ is viscosity, and ω is angular velocity of journal and load. This equation is again from Swift[6], and is also derived from Harrison[3] in Appendix 1./

1.

From Fig.60, the total travel of the bearing in both horizontal and vertical directions is 0.008 in., the diametral clearance being 0.0106 in., so that ϵ will be 0.008/0.0106 = 0.75.

As before, ω = 55.5 rad/sec.,

R = 0.49 in.,

r = 0.0053 in.,

λ = 2.9 poise = 0.000042 Reyn.

Evaluating, W = 162.0 lb., which for a bearing 1.5 in. long gives a total load of <u>243 lb</u>.

The experimental value was no more than <u>40 lb</u>.

5.2 REASONS FOR DISCREPANCY

Since a large clearance was employed in the experiments, reducing the effect of surface irregularities to a minimum, and practically eliminating temperature effects in the oil film, and since good agreement was obtained between measured applied load and integrated pressure load, the large difference between theoretical and experimental performance may be attributed to the following reasons:

(1) The bearing is not full of oil, as assumed by theory, but contains a large cavity, very evident from measurements of pressure distribution.

(2)/

(2) End leakage occurs in the experimental bearing, but is omitted in the theoretical treatment.

(3) It has been noted that small bubbles form in the oil film, so that the oil is no longer homogeneous and the effective viscosity will differ somewhat from the value assumed.

Of these, (1) is possibly by far the most important, followed by (2), while it is doubtful if the effect of (3) will compare with either (1) or (2).

5.3 CRITICAL CONDITION

While it was found that the load carrying capacity of the bearing was much less than that predicted by theory, it is note-worthy that the critical condition, indicated by theory, can be demonstrated experimentally.

This condition occurs when a bearing is subject to a rotating load while the journal rotates at twice the speed of the load, when theoretically, the pressure gradient along the oil film is zero. Under these conditions it was shown, refer to §4.5, that the oil film would carry no load, and that no pressure film was formed.

This indicates that the theory is fundamentally valid./

valid.

5.4 BOUNDARY CONDITIONS

The case of load and shaft rotating at the same speed is theoretically identical with a normal bearing under steady load, the one being an inversion of the other: under steady load, the load is fixed relative to the bearing, while with shaft and load rotating, the load is fixed relative to the shaft. With the present apparatus, the rotating load, produced by the combination of horizontal and vertical fluctuating loads, is not exactly of constant magnitude, but will have a small fluctuating component due to secondary effects in the mechanism, nevertheless, the results presented in §4.8 may be of value in the study of film boundaries. Till present, little knowledge of the low pressure region has been available, and yet it must be considered of first importance in the determination of film boundaries. Cameron, in his review of boundary conditions[2], states that the film commences at $p=0$, at the point of maximum clearance, and ends at the point where $p=dp/d\theta=0$, assuming a Newtonian fluid.

In the present experiments, it was found that the film commenced at a pressure slightly beneath atmospheric pressure and considerably beyond the point of maximum clearance, and ended beyond the point of minimum film thickness and at/

at a pressure approaching absolute zero. It cannot be said that the condition, $p=dp/d\theta=0$, was exactly satisfied.

6. DISCUSSION

6.1 CAVITATION

The most significant feature of this research is the general occurrence of cavitation: it has been previously suspected that cavitation would occur under extreme conditions of high speed, etc., but it is shown here as a persistent feature. There is abundant evidence to show that a cavity exists and that it has some considerable size despite the most optimistic conditions of these experiments.

Firstly, in the fluctuating load test with stationary shaft it was found that at two periods in the cycle the film pressure was everywhere negative throughout the bearing, which is a positive proof of discontinuity and the existence of a cavity, supported by an examination of the shape of the pressure distribution curves which show a dent on the loaded side of the bearing when the pressure begins to rise.

Later, when examining rotating loads, it was/

was found that the greater part of the bearing was at constant pressure just slightly beneath atmospheric pressure. When the machine was stopped, bubbles could be seen emerging from the bearing, and it was shown that the film pressure dropped as the quantity of air inside the bearing increased. The cavity was not eliminated by supplying oil under pressure even although the supply pressure was comparable with the maximum film pressure.

Lastly, the loads determined experimentally are much less than those calculated from theory, which assumes that the bearing is full of oil.

6.2 OIL FILM UNDER FLUCTUATING LOAD

An impression can now be formed of what occurs inside a bearing when it is subjected to a purely fluctuating load. The oil acts as a cushion and supports the load, but is displaced in the process; the displaced oil flows away in all directions, out of the ends of the bearing and round the journal to the other side; when the direction of the load is reversed, the oil is subject to high vacumm as the bearing surfaces separate, the film cavitating and air coming out of solution, while, at the same time, oil is drawn in from all directions, fresh oil from the ends of the bearing and old oil from the other side of the bearing;/

bearing; when the direction of the load is again reversed, it is this mixture of fresh oil and old cavitated oil which is cushioned between the bearing surfaces and which is partly expelled from the ends of the bearing where it can be seen to contain a large number of really minute bubbles.

At this point, it is interesting to note some continuity of features under differing conditions. Comparing Figs.52,27 & 62, it is seen that the pressure distribution in the test with fluctuating load and rotating shaft shows similar features to the earlier test with stationary shaft, and also to the test with rotating load and rotating shaft, and may be considered an intermediate case.

It appears that with rotating load, and perhaps even with elliptical load, the oil film will travel round the bearing with the load, but it is also possible, as in the case of purely pulsating load, for the oil film to break up and reform continuously.

6.3 BOUNDARIES OF FILM

It was found that in all cases of rotating load, the pressure distribution round the journal showed a pressure peak followed by a sharp drop in pressure sometimes approaching absolute zero, the remainder of the circumference, usu-/

usually more than half of it, being at constant pressure slightly beneath atmospheric pressure. The film pressure was found to be considerable at the point of minimum film thickness. It was also shown that the bearing contained a large quantity of air.

It seems important to recognise that the bearing contains a large bubble of air (or more likely, a number of large bubbles situated side by side) which is trapped and is never expelled while the bearing is running. The bubble is situated in the large constant pressure region, and the fact that the bubble is slightly beneath atmospheric pressure indicates that it must lie in the central part of the bearing and does not extend to the edges of the bearing.

The oil film commences at that point in the convergent part of the bearing where the oil completely fills the clearance, the pressure being equal to that of the adjoining bubble, i.e. slightly beneath atmospheric pressure. In the divergent part of the film, beyond the point of minimum film thickness, the film pressure falls and goes beneath atmospheric pressure to pressures approaching absolute zero. At this point, the film cavitates and is met by the air bubble.

The extent of the oil film is determined by the quantity of oil in the bearing and the dispersion of air/

air throughout the oil. The film commences at the point where the bubble ends, and terminates at the point where the film pressure approaches absolute zero, the intervening space to the beginning of the film containing air and oil.

6.4 CONSTANT PRESSURE REGION

The following observations were made in an examination of oil under vacumm pressures: if the oil contains any air bubbles, these expand as the pressure is reduced; at a vacumm of about 28 in. Hg., air comes out of solution, the action becoming more vigorous as the pressure is further reduced; the bubbles appear to form mostly on the boundaries of the oil, but at very low pressures the oil froths and when the vacumm is broken, all the bubbles immediately collapse, leaving only tiny air bubbles throughout the oil. Accordingly, if very low pressure occurs at any point in the bearing, the oil will become perforated with tiny air bubbles which will circulate with the oil.

It seems fairly certain that the constant pressure region contains several long-shaped air bubbles with oil flowing through between them. In this way, the oil can pass freely round the bearing, and at the same time the air can remain in the low pressure region, which is the only region where it can exist. Small bubbles which flow with the oil/

oil coalesce with other bubbles and eventually join the big bubbles in the constant pressure region.

On the other hand, with fluctuating load, the large region of very low pressure and severe cavitation perforates the oil with minute bubbles; these can be observed in the oil expelled from the bearing. At the same time, in passing over the journal from one side to the other, some of these bubbles coalesce and form large air bubbles, perhaps again taking the long-shaped form described above, and remaining in a midway position which is constantly at a pressure slightly beneath atmospheric pressure. As noted during the experimental work, such air is periodically expelled and effects the oil film pressure.

6.5 LOAD CAPACITY

From investigations made, oil bath lubrication is not to be recommended when load carrying capacity is sought. Efficiency, however, is another consideration and it may be that, under prescribed conditions, there is an optimum quantity of oil which the bearing should contain.

It was found that, with oil supplied to the bearing under pressure, an oil film could be maintained in cases where previously, with only the oil bath, the film pressure had practically disappeared after a short period of running./

running. Cavities were still evident however, although the oil supply pressure was considerable and compared with the maximum film pressure.

It is thought that the supply pressure is unimportant, the essential being to displace the air filled cavity from the bearing. This would be best done by making conditions such that a cavity could not exist. It is suggested that this might be achieved by a central circumferential groove supplied with oil under pressure. If the bearing is submerged in oil, it might be equally effective, although not equally practical, to apply vacumm to the central groove.

A central groove divides the bearing into two parts, which may be considered as separate bearings with their inner ends at a higher pressure than their outer ends. Any air bubbles which form will travel towards the low pressure ends and be expelled from the bearings. With such a pressure gradient accross the bearings it would be impossible for a large bubble to exist inside them./

them.

7. CONCLUSIONS

Successful results were achieved with the apparatus and instruments developed in the course of this research, and good agreement was obtained between applied load and integrated film pressure.

The use of a lightly loaded bearing with large clearance proved to be a considerable advantage in this first experimental investigation of journal bearings under fluctuating load, and enabled attention to be concentrated upon the low pressure region and upon the phenomenon of cavitation, subjects about which little is known.

The conclusions to be drawn from the range of conditions investigated may be summarized as follows:

(1) Generally, a bearing contains a large region of sub-atmospheric pressure.

(2) Film pressures may approach absolute zero, notably under pulsating load.

(3) Generally, a bearing is not full of oil while running, but is cavitated.

(4)/

(4) The extent of the oil film is governed by the amount of oil in the bearing and the disposition of air in the oil film.

(5) When oil is supplied under pressure to a bearing, the load capacity is increased by virtue of the fact that the oil film is extended and maintained; cavitation and low pressure regions are not necessarily eliminated.

(6) With rotating load, the oil film travels round the bearing with the load, but with purely fluctuating load the oil film may break-up and reform continuously, particularly when the journal is stationary.

(7) In full accord with theory, the oil film carries no load and no pressure film is formed, when the bearing is subjected to a load rotating at half the speed of the journal./

journal.

8. RECOMMENDATIONS

The general design of the apparatus has not out-served its purpose, particularly in demonstration work and in searching out new fields. The use of large clearance should be exploited to the full on account of the invaluable simplification that it offers, although the amount of the clearance could be progressively reduced. Great care should be exercised to ensure precision throughout. Mechanical accuracy and elimination of friction are essential in loading systems of the type employed. Attention should be given to the durability of all components.

The use of simple electrical contacts in low voltage circuits is particularly to be recommended in instrumentation, on account of its extreme sensitivity and reliability. Professor Kingsbury[4] notes that the accuracy of such a contact is 0.00001 in.

It is thought that, while the displacement indicators can be read quite precisely to 0.0001 in. provided the contacts are clean and free of oil, greater accuracy/

accuracy could be obtained and the present uncertainty eliminated if it were arranged that the spring came up against a stop, against which it was preloaded by a small amount. The load indicators could be improved by employing accurate guides, micrometer screws, and more wear resistant materials for contacts.

The balanced pressure piston proved to be highly reliable in operation. The small time lag in making contact could be eliminated by a re-design embodying the principle of the Farnborough Indicator, where the element travels between two contacts. Readings would then be made only at the point of 'break', at which there is no time lag. A more direct means of supplying the back pressure oil should be sought, perhaps a small submerged pump would serve, as this would considerably speed the process of pressure plotting. It would be an advantage in the study of the low pressure region to have the oil bath and bearing pressurized to eliminate variations of atmospheric pressure.

The escapement and tripping device, employed for searching round the journal, has the advantage of speed in setting, but a disadvantage in that pressure readings cannot be taken at intermediate points; this could be overcome by employing a differential gear in the shaft-drive, in place of the escapement mechanism./

mechanism.

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APPENDIX 1

BASIS OF THEORY FROM HARRISON[3]

List of symbols

(In brackets is equivalent symbol used by Swift[6])

a	(R)	radius of journal,
η	(r)	radial clearance,
c	(ϵ)	eccentricity ratio,
μ	(λ)	viscosity,
V		velocity of journal centre,
U		surface speed of journal,
α		inclination of V to line of centres.

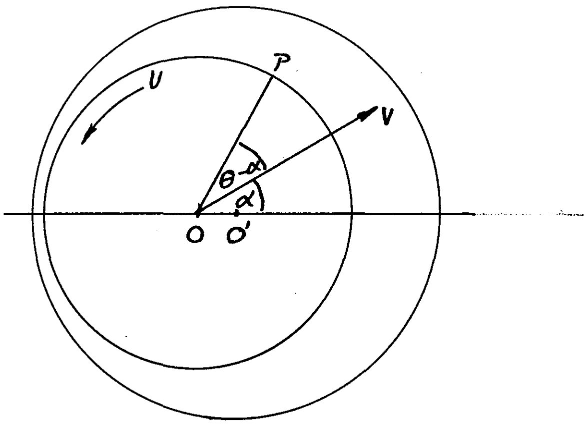


FIG.67

The coordinate x will be measured along the moving surface in the direction of motion, the coordinate y normal/

normal to this surface. The motion is steady and will be assumed to be two-dimensional.

If u, and v be the component velocities at any point in the liquid, p the pressure, the equations of motion are:

$$\frac{\partial p}{\partial x} = \mu\left[\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right] \quad . \quad . \quad (1)$$

$$\frac{\partial p}{\partial y} = \mu\left[\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right] \quad . \quad . \quad (2)$$

The equation of continuity is

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad . \quad . \quad . \quad (3)$$

The boundary conditions are

$$u = U,\ \bar{v} = V.\cos(\theta-\alpha),\ \text{when } y = 0,$$
$$u = 0,\ v = 0, \qquad \text{when } y = h, \quad . \quad (4)$$

where h is the variable distance between the surfaces and is a function of x.

Since the surfaces are nearly parallel v will be small compared with u, and the rate of variation of u in the direction of x will be very small compared with its rate of variation in the direction of y.

Accordingly equations (1) and (2) become/

become

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 u}{\partial y^2} \quad . \quad . \quad . \quad (5),$$

$$\frac{\partial p}{\partial y} = 0 \quad . \quad . \quad . \quad (6).$$

Integrating (5) we get

$$u = \frac{1}{2\mu}\frac{\partial p}{\partial x} y(y - h) + \frac{U(h - y)}{h},$$

and from (3) $\int_0^h \frac{\partial u}{\partial x}\, dy = -\left[v\right]_0^h = V.\cos(\theta - \alpha).$

Therefore,

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) = 6\mu\left[U\frac{\partial h}{\partial x} - 2V.\cos(\theta - \alpha)\right],$$

hence $h^3 \frac{\partial p}{\partial x} = 6\mu Uh - 12\mu Va.\sin(\theta - \alpha) + k,$

or, $\frac{1}{a}\frac{\partial p}{\partial \theta} = \frac{6\mu U\eta(1 + c.\cos\theta) - 12\mu Va.\sin(\theta - \alpha) + k}{\eta^3(1 + c.\cos\theta)^3}$

where k is a constant of integration.

Since p is a single-valued function, k may be found by integration,

$$k = -12\mu(1 - c^2)\left\{U\eta + 2V(a/c)\sin\alpha\right\}/(2+c^2) + 12\mu V(a/c)\sin\alpha .$$

And therefore,/

therefore,

$$p = \frac{6\mu a}{\eta^3(1 + c.\cos\theta)^2}\left[\left\{U\eta + 2V(a/c)\sin\alpha\right\}c.\sin\theta\ \frac{2 + c.\cos\theta}{2 + c^2} + V(a/c)\cos\alpha\right] + C,$$

where C is a constant of integration.

Due to the normal pressure the forces acting on the shaft per unit length are R in a downward direction perpendicular to OO', Fig.67, and S along O'O, where,

$$R = \frac{12\pi\mu a^2 c}{\eta^2(2+c^2)(1-c^2)^{\frac{1}{2}}}\left(U + \frac{2Va.\sin\alpha}{c\eta}\right),$$

$$S = \frac{12\pi\mu a^3 V.\cos\alpha}{\eta^3(1 - c^2)^{3/2}}.$$

These general relations can be employed to solve various particular cases:

Example 1.

Journal stationary but moving along the line of centres with velocity V.

In this case, $\alpha = 0$ and $U = 0$ consequently,

$$R = 0,$$

and $$S = \frac{12\pi\mu a^3 V}{\eta^3(1-c^2)^{3/2}}.$$

This is a force along the line of centres, opposing/

opposing motion, and is identical with the theory of Swift.

Example 2.

Journal rotating and at the same time moving round centre of bearing at the same speed.

In this case, $U = a\omega$, where ω is speed of rotation. Also, $V = c\eta\omega$ and $\alpha = 270^{\circ}$,

so that,

$$U + \frac{2Va.\sin\alpha}{c\,\eta} = a\omega + \frac{2c\eta\omega a(-1)}{c\,\eta}$$

$$= -a\,\omega.$$

Consequently,

$$R = \frac{-12\pi\mu a^3 c}{\eta^2(2+c^2)(1-c^2)^{\frac{1}{2}}}\,\omega\ ,$$

and $S = 0$.

This force is at right angles to the line of centres and acts upwards, again identical with the theory of Swift.

Example 3.

Journal rotating and centre of journal moving round the centre of the bearing at half speed.

In this case,

$$\frac{U}{a} = \frac{2V}{c\,\eta}\ , \quad \text{and } \alpha = 270^{\circ}\ ,$$

consequently, $R = 0$ and $S = 0$. This is the critical case when the oil film will support no load.

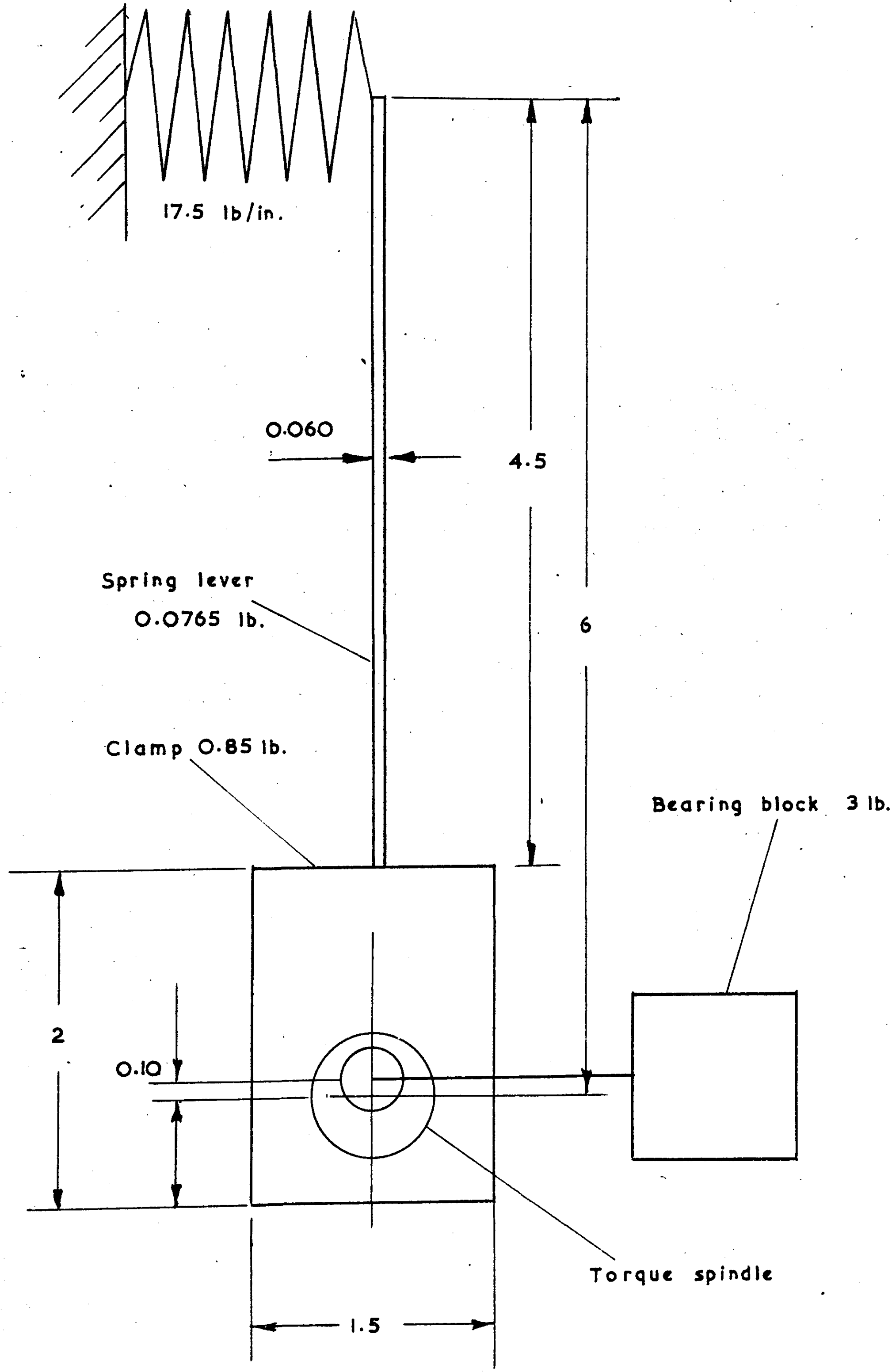
APPENDIX 2

SPRING LEVERS AND INERTIA EFFECTS

The spring levers were made from a machine hack-saw blade, ground to a thickness of 0.06 in. and width of 1 in. While the radius from centre of torque spindle to connecting rod is 6 in., the actual cantilever overhang is 4.5 in. Assuming a modulus value of 30×10^6 lb.per in^2., the rate of the spring is calculated as 17.75 lb.per in. When calibrated under static load this value was found to be 17.5 lb. per in. The coupling of load and attitude may therefore be calculated as, $17.5 \times (60)^2 = 63000$ lb. per in. i.e. 63 lb. per 1/1000 in. displacement.

The loading system can be reduced to a spring and mass system as shown in Fig.48. This arrangement makes a generous allowance for the inertia of the spring lever, and gives a correspondingly low estimate of the natural frequency. The relative inertias of the various components may be seen from the following table:

Part	Weight	K^2	Weight referred to 0.1 in. radius.
Lever	.0765 lb.	$3.75^2+2.25^2/3 = 15.8$	121 lb.
Clamp	.85	$1/3 + .75^2/3 = 0.66$	56
Block	3	$(0.1)^2 = .01$	3
			180 lb.



EQUIVALENT SPRING & MASS SYSTEM

FIG. 68

lb.

The spring stiffness referred to 0.1 in. radius is $17.5 \times 12 \times (60)^2 = 755000$ lb.per ft. The natural frequency of the system may therefore be calculated as follows:

$$n^2 = \frac{755000\ g}{180} = 0.135 \times 10^6,$$

$$n = 367 \text{ radians per second,}$$

and so, <u>f = 58.5 cycles per second</u>.

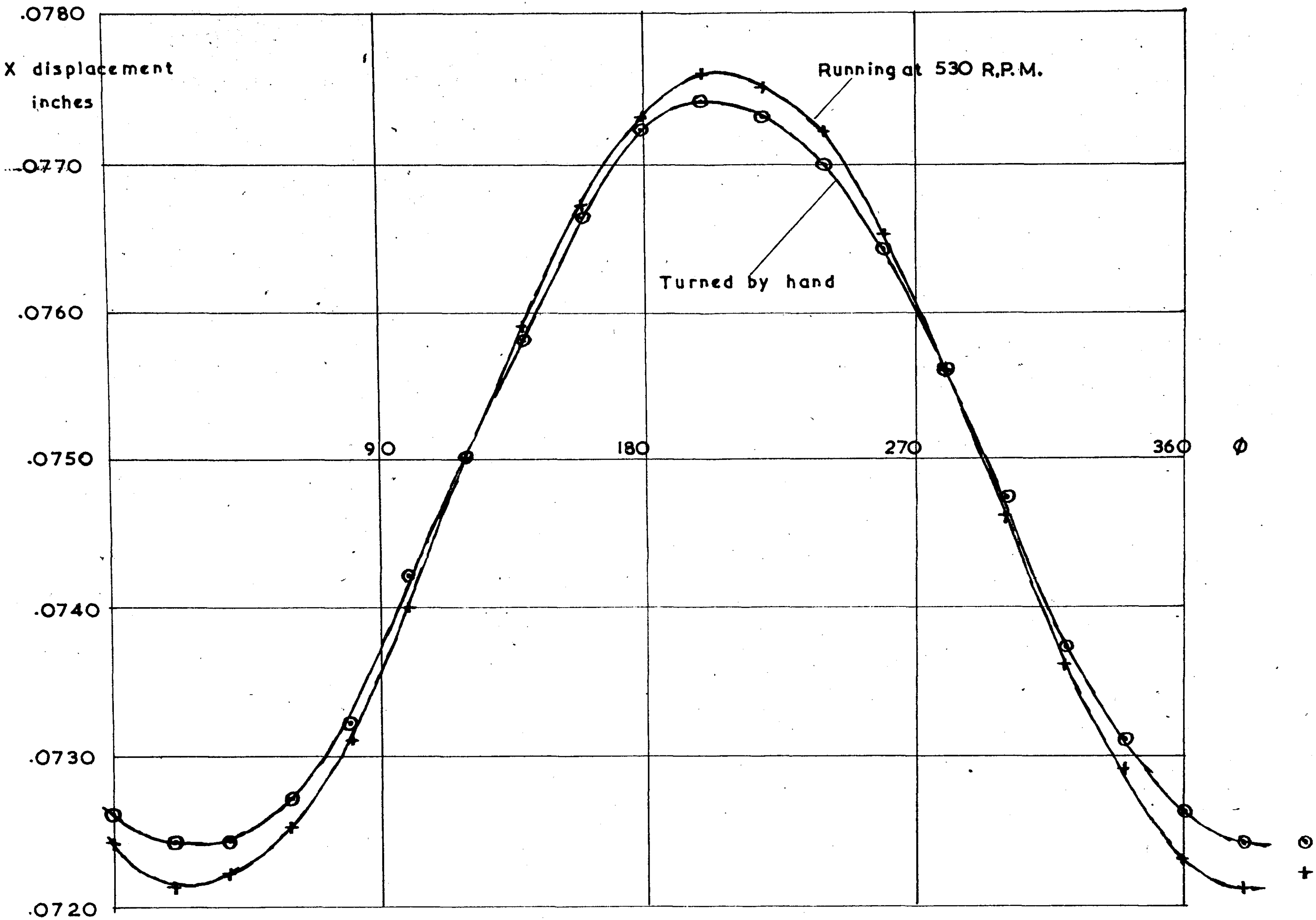
At the normal running speed of 530 R.P.M., or 8.8 c/s., the amplification factor is:

$$\frac{58.5^2}{58.5^2 - 8.8^2} = 1.024.$$

The maximum error due to inertia will therefore be,

$$(0.005 \text{ in.}) \times 0.024 = \underline{0.00012 \text{ in}}.$$

A test was made to determine the effect of inertia. The shaft was removed and X-displacement plotted against phase angle Ø, firstly with machine running at normal speed and secondly, turned over by hand. The two curves are shown together in Fig.69 and it will be seen that the difference is 0.00025 in., while the total amplitude of motion is 0.005 in. Although the difference between the curves is a small amount, it is considerably greater than that estimated above. The discrepancy is most likely due to back-lash in the mechanism.



EFFECT OF INERTIA etc.

FIG. 69

APPENDIX 3

DISPLACEMENT INDICATORS

It was assumed that contact between the point and the spring was made and broken when the spring was in a state of zero strain. This is not exactly true as the spring is excited to some extent by the movement of the bearing block to which it is clamped, also, it is possible that contact is broken a little sooner or a little later than the point of zero strain, and it is possible that when contact is being made the spring will still be vibrating from the last contact. The relative magnitudes of these errors is now estimated for the dimensions of spring actually employed.

The springs were formed from pieces of spring steel 3/16 in. wide and 0.017 in. thick, the length of the cantilever being $\frac{5}{8}$ in. The cross-sectional area of the spring will therefore be 0.00318 in^2. and the weight 0.0009 lb. per in. length. The inertia of the section, I is 0.077 in^4. The natural frequency of the spring from the formula,

$$n^2 = 6.23 \frac{2g(EI)}{w.L^4} ,$$

is 1430 cycles per sec. In comparison, the frequency of/

of movement of the bearing block is only 8.8 cycles per sec.

Since the springs are clamped to the bearing block they oscillate with it and are subject to the corresponding accelerations. The maximum amplitude of vibration of the bearing block is 0.005 in. and consequently at a speed of 530 R.P.M., the acceleration is 1.283 ft. per sec^2., assuming sinusoidal motion. From the formula,

$$\delta = \frac{WL^3}{8EI} ,$$

in which $W = \frac{5}{8} \times 0.0009 \times 1.283 = 0.000721$ lb., the deflexion at the tip of the spring is 9.52×10^{-6} in., an insignificant amount.

The problem may therefore be treated as though the point moves while the bearing and spring remain stationary, which is the reverse of the actual case.

Considering that the point moves in a sinusoidal manner, its displacement 'x' from the mean position may be written:

$$x = a.\sin\omega_p t.$$

Differentiating, $\dot{x} = a\omega_p \cos\omega_p t,$

$$\text{and, } \ddot{x} = -a\omega_p^2 \sin\omega_p t = -\omega_p^2 x. \qquad . \qquad (1)$$

Let the point move about I and let the zero strain position of the spring be II, see Fig.70, so that,

$$h + y = x, \qquad . \qquad . \qquad (2)$$

where y is deflexion of spring and h is distance between/

between I and II.

The upward acceleration of the point is ω_p^2x, and the upward acceleration of the spring, if free, is ω_s^2y. While in contact, the velocity and acceleration are equal, contact being broken when ω_p^2x becomes greater than ω_s^2y, i.e. at the point where,

$$\omega_p^2x = \omega_s^2y.$$

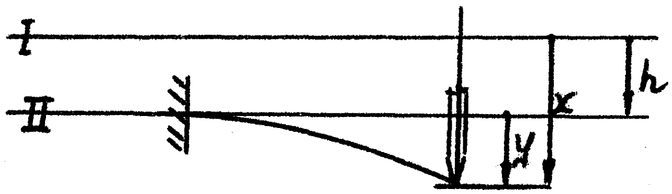


FIG.70

Substituting (2) and solving for y,

$$\omega_p^2(h + y) = \omega_s^2y$$

$$\text{or,} \quad \underline{y = h/(S^2 - 1)}, \quad . \quad . \quad (3)$$

where $\omega_s/\omega_p = S$.

This is the error on breaking contact and is of a very small order.

Substituting (3) in (2),

$$x = h + h/(S^2 - 1),$$

$$\text{or,} \quad x = hS^2/(S^2 - 1) = a.\sin\omega_p t. \quad . \quad (4)$$

Let 'v' be the common velocity at point of break, so that from (4),/

from (4),

$$v = \dot{x} = \dot{y} = a\omega_p\left[a^2(S^2-1)^2 - h^2S^4\right]^{\frac{1}{2}}/a(S^2-1) \quad . \qquad (5).$$

Motion of spring when free is $y = b.\sin\omega_s t$, so that, from (3),

$$\ddot{y} = -\omega_s{}^2h/(S^2-1) = -b\omega_s{}^2\sin\omega_s t,$$

therefore, $b = h/(S^2-1)\sin\omega_s t$, . . (6)

and from (5), $$\cos\omega_s t = \frac{a\omega_p\left[a^2(S^2-1)^2 - h^2S^4\right]^{\frac{1}{2}}}{a(S^2-1)b\omega_s}$$

$$= \frac{\left[a^2(S^2-1)^2 - h^2S^4\right]^{\frac{1}{2}}}{bS(S^2-1)} \quad . \quad . \qquad (7)$$

Hence, by substituting (7) in (6),

$$b^2 = \frac{a^2}{S^2} - \frac{h^2}{(S^2-1)} .$$

This implies that the greatest error, when contact is being made is when $h = 0$, the value being $\pm a/S$.

Evaluating, the maximum value of 'a' is 0.005 in., and S is 1430/8.8 = 163, so that the maximum error is 0.005/163 = <u>0.000031 in</u>., a negligible amount.