TECHNICAL MEMORANDUMS

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

No. 1057

Piston rings - Press. distrib.
Press. distrib. - Piston rings
author

RESEARCHES ON THE PISTON RING

By Keikiti Ebihara

Scientific Papers of the Institute of Physical and Chemical Research
Vol. 10, No. 182, February 25, 1929

Washington
February 1944

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL MEMORANDUM NO. 1057

RESEARCHES ON THE PISTON RING*

By Keikiti Ebihara

I. INTRODUCTION

In internal combustion engines, steam engines, air compressors, and so forth, the piston ring plays an important role. Especially, the recent development of Diesel engines which require a high compression pressure for their working, makes, nowadays, the packing action of the piston ring far more important than ever.

Though a number of papers have been published in regard to researches on the problem of the piston ring, none has yet dealt with an exact measurement of pressure exerted on the cylinder wall at any given point of the ring. The only paper that can be traced on this subject so far is Mr. Nakagawa's report (reference 1) on the determination of the relative distribution of pressure on the cylinder wall, but the measuring method adopted therein appears to need further consideration.

No exact idea has yet been obtained as to how the obturation of gas between the piston and cylinder, the frictional resistance of the piston, and the wear of the cylinder wall are affected by the intensity and the distribution of the radial pressure of the piston ring. Consequently, the author has endeavored, by employing an apparatus of his own invention, to get an exact determination of the pressure distribution of the piston ring. By means of a newly devised ring tester, to which piezo-electricity of quartz was applied, the distribution of the radial pressure of many sample rings on the market was accurately determined.

Since many famous piston rings show very irregular pressure distribution, the author investigated and achieved a manufacturing process of the piston ring which will exert uniform pressure on the cylinder wall.

*The 31st Rep. of Ōkōchi Res. Lab.; Scien. Papers of the Inst. of Phys. and Chem. Res. (Tokyo), vol. 10, no. 182, Feb. 25, 1929, pp. 107-185.

Temperature effects on the configuration and on the mean spring power have also been studied.

Further, the tests were performed to ascertain how the gas tightness of the piston ring may be affected by the number or spring power.

The researches as to the frictional resistance between the piston ring and the cylinder wall were carried out out, too.

The procedure of study, and experiments conducted by the author, on this subject will be fully described in the following paragraphs.

II. MEASURING DEVICES

1. Mechanical Device

Mechanical methods were first resorted to and corresponding devices introduced, as shown in figures 1 and 2. In figure 1, a ring 2 was compressed by applying external forces on its peripheral surface radially by means of bell cranks 3, the diameter of the ring being exactly equal to the inner diameter of the cylinder. The pressure can be calculated from the readings of the weights hanging at the ends of bell cranks.

In the same year when the author contrived the apparatus above-mentioned, a piston ring tester similar to that of the author was published in a German patent (reference 2); this is very interesting and at the same time an accidental coincidence. The results of experiments, however, showed that it is almost impossible to determine exactly the pressure distribution of the piston ring on the cylinder wall by means of such a device. For, to determine the pressure distribution of the piston ring, the equilibrium should be attained between the weights hanging on the bell cranks and the spring power of the piston ring, and at the same time, the inner faces of steel balls 1 attached to the lower ends of all the bell cranks must be exactly equal to the inner diameter of the cylinder wall. If the weights are adjusted so as to make the top of bell cranks agree with the cylinder diameter, the equilibrium will be disturbed and all bell cranks will be displaced slightly. Since the same thing will take place in all the other bell cranks when adjusted, the adjustment of all the bell cranks can hardly be finished up in practice. Moreover, the frictional resistance between the ring surface and the ends of bell cranks, caused by the sliding of the ring during the operation, cannot be neglected. In figure 2, a circular hole is made in a cast iron circular disk 1, its inner diameter being equal to that of the cylinder. After putting the ring 2 to be examined into the bore of the disk, an external force is applied by a segmental block 3. Any slight deflection of the ring is observed by the rotation of a small mirror 4. In case when the ring exactly coincides with the inner surface of the disk, the pressure of a spring 5 is equal to the normal pressure exerted by the arc length 3' 3" of the ring. By continuing the same method in succession by rotating the ring, the pressure distribution on the cylinder wall can be determined. The result, however, is not fully satisfactory owing to the inaccuracy of the measurement due to the enormous friction between the ring and the cylinder wall. From these experiments it can be seen that the piston ring tester should be so designed as to fulfill the following requirements in order to get the correct figures.

(a) The pressure measurement of the rings must be done under the condition entirely free from friction between the ring surface and the cylinder wall; hence, sliding the piston ring along the peripheral surface of the cylinder wall while measuring the pressure of the ring must be avoided.

(b) The inner surface of the testing apparatus must be so accurately machined as to form an exact right cylindrical surface of a given diameter. If there exists any irregularity of the surface, however slight, the pressure distribution will be affected accordingly, so it is disadvantageous to use many pressure gages or bell cranks as shown in figure 1.

(c) Even when only one pressure gage is used as shown in figure 2, it does not give any satisfactory result, if the pressure gage is flexible and the pressure intensity is estimated under a large quantity of strain.

2. Piezo-Electric Device

Considering that the mechanical methods do not very well satisfy above conditions, the author has attempted for the study of this subject to utilize piezo-electricity.

The principle of the measuring device is diagrammatically shown in figure 3 in which T is the ring tester, and E an electrometer charged by batteries B for measuring the statical electricity produced in the tester T having a bore of exactly the same diameter as the engine cylinder in which the ring under test is set to work. A segment 3 is cut away from the wall of the tester and is supported by two disks 4 of piezo-electric crystals, so that the pressure on the segment exerted by the ring 2 is transmitted to the crystals and causes piezo-electricity in them. The amount of this electric charge, which is proportional to the change of applied pressure, is measured by the electrometer E. If the segmental surface exactly coincides with the surface of the bore as shown in the figure and the ring is turned through an angle φ successively after each measurement, the pressure distribution of the ring on the cylinder wall may be accurately determined. The calibration of the charge of the piezo-electricity is done simply by applying a known pressure on the said segment, for instance, by hanging a weight at one end of a bell crank and pushing the segment at its other end.

3. A Piston-Ring Tester of Type I (Chuck Type)

Figure 4 represents an apparatus of type I for determining the pressure distribution of the piston ring; 1 is a thick steel or cast iron plate, having its inner bore well machined and finished by a grinder to the diameter identical with the inner diameter of the cylinder in which the ring is to be placed for use; 2 is a piston ring to be tested; 3 is a piece taken from the cut 3' 3" in the steel plate 1; 4 are two quartz crystals, each being 3.5 centimeters in diameter and 1.2 centimeters in thickness, between which a thin copper plate is inserted so as to lead the electric charge produced on their surfaces; 5 are thin steel disks and 3, 4, and 5 are tightly held by four bolts 7 to the steel plate 1, so that the crystals are initially compressed under a certain pressure. After the quartz crystals have been fixed rigidly to the steel plate 1, the inner walls both of the bore and of the segmental block 3 are turned and finished at the same time by the grinder. The inner surface of the segmental block 3, therefore, coincides so accurately with the surface of the cylinder wall that it can be regarded as one part of the cylinder wall, there being no discontinuity along the inner surface of the segmental block and the cylinder wall. Now, if there is inserted a standard ring, which was specially finished to obtain a very good contact with the cylinder wall of true circular form, and it is illuminated by a small electric lamp from its under surface, no discontinuity will be observed along the inner surface. One example of the measures of the bore in all directions is $D = 127.08$ millimeters ± 0.008 millimeter, irregularity of diameters being very small. Thus the radial pressure on the segment 3 exerted by the arc element of the ring 3' 3'' is completely borne by the crystals. Eight are hemp or copper strips, all of which are used to chuck the ring lightly toward the radial direction to keep the ring from dislocation.* The phenomenon of dislocation of the ring will be fully explained later. One of the strips 8 is used to set the ring free from the block 3 so as to unload the crystals, and the change of pressure on the crystals is measured by the amount of the electric charge.

4. A Piston-Ring Tester of Type II (Stroke Type)

The piston-ring tester of type I mentioned above must be so improved as to fulfill practical demands. Figure 5 shows the improved ring tester of the so-called stroke type; 1 is a cast iron cylinder in which the ring is inserted; 2 is the piston ring to be tested; 3' is a rectangular hole made in one part of the cylinder wall 1 and the two edges of the hole subtend an angle φ at the axis of the cylinder, φ being 15° to 30° according to the diameters of the ring. A thin copper plate is inserted between the quartz plates 4 to be connected with the above terminal 6 which is perfectly insulated by ambroid; 7 is a cast iron box in which the quartz plates are kept and sealed so as to be protected from electromagnetic disturbances. A pipe connects the box 7 with a desiccator, by means of which the air in the box is kept perfectly dry, thus preventing any leakage of piezo-electricity. Quartz crystals in the box are put under a certain initial pressure by means of a screw 5, and such a specially constructed piezo-electric pressure gage is attached rigidly to the

*The word "dislocation" is used to mean the state of a piston ring being fitted into a cylinder wall and subjected to the disturbance upon the radial pressure distribution caused by the friction between the ring surface and the cylinder wall.

cylinder wall 1. After the piezo-electric pressure gage is attached to the cylinder, the inner wall is finished by the grinder as explained in the description of the ring tester of type I. A piston 10 can be reciprocated up and down by a handle 13 and the piston ring to be tested is put in it in the same state as if it were inserted in the actual piston groove. When the piston ring comes just upon the segment 3 the ring pressure is transmitted to the piezo-electric crystals. At the top end of cylinder 1 there is a closed ring 11 in which the piston ring may be caught and rotated through any desired angle; 12 is a bell crank used to calibrate the piezo-electric pressure gage, and the electrometer, by known weights.

Figure 6 shows the photograph of the ring tester of type II of a large diameter, in which the piston 10 goes up and down by screw drive, the other parts being nearly the same as mentioned above.

Thus, the author was able to avoid the various troubles usually accompanying the piezo-electric pressure gage by improvements in its construction, and also the disturbing effect on the radial pressure distribution due to the tangential friction between the ring surface and the cylinder wall.

5. A Piston-Ring Tester of Type III (Variable Diameter Type)

Since the above-mentioned ring testers have each a definite diameter, the piezo-electric pressure gage being fixed into the cylinder wall, there can be examined by the respective ring tester the piston ring of the same diameter only and no other rings of different diameters. So, the author designed a ring tester in which the cylinders of various sizes can be put and the piston rings of the corresponding diameters tested. Figure 7 shows the piston-ring tester of variable diameter type. A cylinder 1 of desired diameter is put concentric with a shallow cylinder, the upper part of the mother body 1', in which a piston ring to be tested is introduced. On the arm of the mother body 1' a cylindrical guide 5' is fixed, its central axis being directed perpendicular to the central axis of the mother body. The piezo-electric pressure gage can travel along the cylindrical guide 5' and is able to be fixed at any desired position by means of a fine adjusting screw. The curvature of the inner surface of the segmental block 3, the top of the piezo-electric pressure gage, coincides with the arc of the piston ring of the largest diameter to be examined. In the case of smaller rings, only one part of the arc coincides with the inner surface of the cylinder wall. As many piston rings, however, are sufficiently thick as to be inflexible, the errors of the pressure intensity due to a partial coincidence of the inner surface of the segmental block with the cylinder wall will be so small as to be negligible.

III. DETERMINATION OF THE PRESSURE DISTRIBUTION

The results of measurements concerning the pressure distribution of many kinds of piston rings carried out by the piezo-electric piston-ring testers are to be summed up as follows:

1. Specifications of Sample Rings

The specifications of piston rings examined are given in table I, and the ring testers used, in table II.

TABLE I.- SPECIFICATION OF SAMPLE RINGS NOS. 1 to 13

No. of rings	Diam. of cylinder D (cm)	Thickness (mean) t (cm)	Breadth b (cm)	Form of gap ends	Remarks	Manufacturer
1	12.70	0.406	0.793	Stepped	Inner side is left as cast	P.R. Co. (U.S.A.)
2	12.70	.417	.793	--do---	--do----------	Do.
3	12.70	.407	.793	--do---	--do----------	Do.
4	12.70	.440	.600	--do---	--do----------	M. & H. Co. (U.S.A.)
5	12.70	(.220 / .503)	.798	Diagonal (45°)	Eccentric type	I.P.C.R. (for trial)
5'	8.85	(.260 / .561)	.283	--do---	--do----------	Air-brake Co.
6	12.70	.460	.798	--do---	Simplest type	I.P.C.R. (for trial)
7	12.70	.392	.793	Stepped	Inner side is left as cast	Do.
8	12.70	.432	.397	Straight	--do----------	Do.
9	12.70	.389	.397	--do----	--do----------	Do.
10	12.70	------	.794	--do----	--do----------	Do.
11	8.89	.315	.635	Diagonal (45°)	Inner side is hammered	Brico (England)
12	15.00	.376	.340	Stepped	Inner side is left as cast	I.P.C.R. (for trial)
13	58.90	1.57	1.90	Diagonal (45°)		Do.

(Specimens are all virgin rings of cast iron except No. 5'.)

TABLE II.- KINDS OF RING TESTERS

Ring tester	Type	Diam. of cylinder D (cm)	φ (deg)
No. 1	I (old)	12.70	28
No. 2	I (old)	8.89	20
No. 3	II (new)	58.90	15
No. 4	II (new)	15.00	20
No. 5	II (new)	8.85	20

2. Method of Observations

General arrangements of a ring tester, an electrometer, and batteries are shown in figure 8; T is the piston-ring tester, E is Shimizu's sensitive electrometer, and B is a group of storage batteries. The silvered quartz fiber of the electrometer is charged with 100 to 150 volts, one of the poles being grounded and the other connected to the piezo-electric pressure gage of the ring tester. To avoid electro-magnetic induction, the apparatus and lead wires are shielded in iron boxes and pipes L. To determine the pressure distribution, the ring is oiled on its outer surface and inserted in the ring tester. Dislocation of the ring owing to the friction between the cylinder wall is adjusted by pulling lightly the hemp strips as in the case of the ring tester of type I, or by reciprocating the piston up and down as in the case of type II. Next, the zero-point of the needle of the electrometer is fixed, the residual charge of the quartz being grounded; then the ring is separated or slid away from the segmental block 3, and the deflection of the needle read. It does not matter how the pressure is taken off the quartz crystals, whether suddenly or slowly, the results being always independent of the manner of treatment. If the sensitivity of the electrometer is known, which can be calibrated by loading or unloading known weights on or off the quartz crystals by means of a bell crank, the amount of the total force F exerted by the ring over the fixed arc length can be obtained. Examples of calibration curves are shown in figure 9, they being nearly straight. It has been confirmed by a series of careful examinations that such a simple treatment as using hemp strip or as reciprocating the piston is quite satisfactory for destroying the dislocation of the ring and for fitting it freely to the inner surface of the cylinder.

Let R and b be the inner radius of the cylinder and the breadth of the ring, respectively; then the following relation can be obtained, assuming the intensity of radial pressure p_θ on the arc $R\theta$ to be constant,

$$F = 2bR \int_0^{\varphi/2} p_\theta \cos\theta \, d\theta = 2bRp \sin\frac{\varphi}{2}, \quad \text{or} \quad p = \frac{F}{2bR \sin\frac{\varphi}{2}} \qquad (1)$$

Then the intensity of pressure along the circumference of the ring is to be calculated from the observed value of F.

3. The Pressure Distribution in Many Sample Rings

Nos. 1 to 3 in table I, are all virgin rings to be used in Diesel or petrol engines, manufactured by the Piston Ring Co., U.S.A. As shown in figures 10 to 12 and tables III to V, all of them show three maximum pressures: namely, one at the gap end and the other two at approximately 120° from the gap end. It will also be noted that there are three minimums nearly opposite to the maximums, the pressure distributions being pretty irregular. Tests of many other sample pieces manufactured by the same company showed a similar configuration of the pressure distribution. The manufacturing process of these rings is not known, but it may be noticeable that the inner side of the ring is left as cast and the structure of material is very fine, so that the elasticity and the mean spring power of these rings are very high.

No. 4 is a ring manufactured by M. & H. Piston Ring Co., U.S.A. Its sectional view is as shown in figure 13', a steel band being put in the groove to exert side pressure on the piston groove. The pressure distribution of this ring, as shown in figure 13 and table VI, is very irregular, and at the gap ends the pressure intensity is especially large.

A ring No. 5 is an eccentric ring, of which the thickness is 2.20 millimeters at the gap ends and 5.03 millimeters at the opposite side. Now, according to the theory of elasticity (reference 3), the thickness at the gap ends of an eccentric ring must vanish in order to exert uniform pressure on the cylinder wall, but such a ring will be impossible to be used. The pressure distribution of the ring No. 5 is very far from being uniform as shown in figure 14 and table VII. Of course, if the thickness and gap length of the ring were made less, the distribution of the pressure would be more improved; but, at any rate, it would not become uniform. As a practical example, such an eccentric ring as No. 5 had been used in the air brake of the locomotive and there was observed a considerable wear on the two sides of the ring near its gap ends. So, the maker tried to change his manufacturing process by turning off those side parts of the ring much more, but the fitting of the ring to the cylinder wall was not improved. Hence, the author measured its pressure distribution of the ring thus treated by his ring tester, the result of which is shown in figure 15. The pressure exists only at the gap ends and at the opposite side, the lateral parts not being in contact with the wall. In short, the treating of an eccentric ring in such a manner is not good in practice.

Moreover, there is another defect with an eccentric ring: permanent set or breakdown would often occur when it is put into the piston groove owing to a large thickness of the opposite side of the gap and the resulting inflexibility. For a practical example, the bronze eccentric rings, as No. 5', are used in the air brake of the locomotive at present, and permanent set or breakdown often occurs, when they are put into the piston. The conclusion is inevitable that the eccentric type of the piston ring should be improved in form and not in the quality of material used.

No. 6 is a ring manufactured by the simplest method; a number of rings are cut off from a cast iron pipe of a proper cylindrical cross section, the diameter of the pipe being somewhat larger than the inner diameter of the cylinder, and a proper gap is made for each ring to obtain the spring action. The ring thus made has a true circular form in its free state; then, if it is put into the cylinder, its gap ends will become discontinuous and point sharply. The pressure distribution for this ring is the worst, as seen in figure 16 and table VIII; an enormously intense pressure is presented on the gap ends and the pressure on the opposite side is irregularly distributed. On both sides of the gap ends there exist considerable clearances which will allow free passage of the high pressure gas or steam in running state. Of course, if the gap length and the thickness of the ring is made smaller, the irregularity of the pressure distribution will be reduced. Such an irregular pressure distribution is not only bad for the packing of a high pressure gas, but also for the abrasion of the cylinder wall. Something about the local abrasion of the cylinder wall will be remarked later on in this paper. Nowadays, the ring of this sort is still used in marine and land engines, only because its manufacturing process is the simplest and its cost is the cheapest; notwithstanding, it must be an urgent problem to improve it.

No. 7 is a ring manufactured by a new process in the I.P.C.R. laboratory. The outer surface was turned and finished by the grinder with the gap ends kept clamped tangentially by a small pin. If a ring is finished in its outer surface by the new process, it will exert nearly uniform pressure on the cylinder wall. Figure 17 and table IX show the pressure distribution of the newly manufactured ring. Distinct improvements will be observed there, when compared with ordinary ones described before.

The rings Nos. 8 and 9 are reduced in their breadths just to half of those made by the Piston Ring Co. The pressure distributions of these rings, shown in figure 18 and tables X and XI, are similar to those in the rings Nos. 1 to 3. Now, by combining such rings so as to mutually cancel the irregular parts of the pressure distribution, there is made the so-called compound ring as No. 10; and the pressure distribution in such a compound ring is very much equalized, as shown in figure 18 and table XII. It is worth while to remark that the resultant pressure of the compound ring measured actually, coincides exactly with the sum of the pressure intensities in the two rings. The compound ring has excellent properties, but it costs so high that it is only used for a special purpose.

The ring No. 11 is a Brico hammered ring made in England, and its pressure distribution, shown in figure 19 and table XIII, is very similar to that of the ring No. 4. In several other Brico hammered rings of different sizes, the pressure distribution is also irregular and the intense pressure acts at the gap ends. Any other kind of hammered rings examined has been found not to exert uniform pressure on the cylinder wall; especially, for those having the stepped gap ends the pressure distribution is very irregular.

No. 12 is a ring for the airplane engine manufactured by the author's process similar to the ring No. 7 but not finished by the grinder, its mean spring power being very weak. Therefore, its pressure distribution, as shown in figure 20 and table XIV, is not uniform as that of the ring No. 7. No. 13 is a ring for a locomotive engine, having a very large diameter. Its pressure distribution is pretty irregular, as shown in figure 21 and table XV; the intense pressure acts near the gap ends and the opposite sides, both of the other sides suffering no pressure.

Next, the rings Nos. 1 and 7 were put into the cylinder of the diameter of 127.0 millimeters and annealed in the electric furnace for 10 hours at 300° C. The pressure distribution of these annealed rings was measured and is shown in figures 22, 23, and tables XXVI (XXVI'), (XXVIII), respectively. It is worth while to remark, that when the ring is annealed in the compressed state in the cylinder, its mean spring power becomes weak, but the configuration of the pressure distribution remains nearly unaltered similar to that of the virgin ring. Even if the annealing temperature becomes higher, say 500° C, the similarity is still maintained, and further, it may be said that the irregularity of the pressure distribution of the ring would not be altered greatly by annealing it in the running state of engines.

From the experiments previously described, it becomes clear that there is surprisingly a wide range in pressure irregularity, which is of course an undesirable condition for the efficiency and maintenance of the engine. Especially, the pressure distribution in the ring of an eccentric type or in the ring as No. 6 made by the simplest process, is the worst; besides, the hammered ring seems not so good as expected. Only the ring No. 7 manufactured by the new process exerts nearly uniform pressure on the cylinder wall. The mean spring power of many sample rings is 0.3 to 1.0 kilograms per square centimeter according to the kind of engines.

IV. DISCUSSION OF THE RESULTS OF MEASUREMENTS

1. Calculation of the Bending Moment

As previously described, the pressure distribution of the piston ring on the cylinder wall was determined by the ring tester, in which piezo-electricity of quartz was applied. The author intended to examine whether or not the results of measurements were correct.

In the case when the piston ring is put into the cylinder or the ring tester, the bending moment at any section of the ring is to be calculated by the following formulas:

$$M_\theta = \left\{EI\left(\delta \frac{1}{\rho}\right)\right\}_\theta = \left\{\frac{Ebt^3}{12}\left(\delta \frac{1}{\rho}\right)\right\}_\theta \qquad (2)$$

$$M_\theta = bR^2 \int_0^\theta p \sin\,(\theta - \varphi)\,d\varphi \qquad (3)$$

where M is the bending moment, E Young's modulus, $\delta \frac{1}{\rho}$ the variation of curvature of the ring, t the radial thickness, b the breadth, R the radius of the cylinder, p the pressure intensity as measured by the ring tester, and θ the angular distance measured from the gap ends.

The bending moment at any section of the piston ring within the cylinder is to be independently calculated either by the variation of curvature or by the pressure distribution diagram, using the formula (2) or (3).

Supposing p to be constant in the formula (3), there is obtained

$$M_\theta = pbR^2\,(1 - \cos\theta) \qquad (4)$$

In this case, the piston ring exerts perfectly uniform radial pressure on the cylinder wall, which is an ideal case.

To examine the experimental results, the variations of bending moment calculated by the formulas (2) and (3) with those for the ideal case were compared. The ratio M_θ/M_π was taken in the comparison, in order to simplify the value and also to eliminate unknown elastic constant E, where M_π represents the bending moment of the ring section opposite the gap.

2. Bending Moment Obtained from the Variation of Curvature

For the calculation of the bending moment of ring by means of the formula (2), the curvatures of the piston ring both in its free and compressed states must be measured. But, if the peripheral surface of the ring entirely fits well to the cylinder wall when it is compressed in the cylinder, the curvature of the ring is constant everywhere. Thus, the bending moment will be obtained by measuring the curvature of the ring only in its free state.

The measuring apparatus of the curvature of the ring at free state is as shown in figure 24; a piston ring 2 is put within a groove of a ring holder 1 and is fixed by clamps 3. A mirror holder 4, which stands on three equidistant knife-edges of equal heights, is mounted on the ring surface.

The central knife-edge is magnetized by an electromagnet 5 to prevent a slip when the mirror holder is tilted through an angle α; 6 is a small mirror and the amount of tilt of the mirror holder is measured by a method of the optical lever. In the actual case the distances between the knife-edges of the mirror holder are 6.004 and 6.002 millimeters, respectively, and the width of the holder was 10 millimeters. If the angle α is determined, the curvature $1/\rho$ of the corresponding part is to be evaluated by the formula (5)

$$\frac{1}{\rho} = \frac{\sin \frac{a}{2}}{\frac{l}{2}} \tag{5}$$

If the curvature of the ring at its free state is thus determined along its entire surface, the variation of the curvature at any part and the corresponding bending moment can be calculated.

As the rings No. 7" and 1" become truly circular at their outer surfaces when they are put in the cylinder, the initial curvatures of these rings was measured, and the bending moment along the entire circumference was calculated when they were bent to the circle of the cylinder. In table XVI, the third and fourth columns show the values M_θ/M_π of these rings, respectively, and in figures 25a and 25b these are shown graphically with marks (o). On the other hand, consider the case of the ideal piston ring which would exert uniform pressure p on the cylinder wall. The bending moment at any section is calculated by the formula (4) and the value of $\left\{\frac{M_\theta}{M_\pi}\right\}_i$ is shown in the second column in table XVI, and the chain lines in 25 a, b, and c show the variation of $\left\{\frac{M_\theta}{M_\pi}\right\}_i$.

Comparing the ideal case with that of the ring No. 7", which exerts nearly uniform pressure on the cylinder wall as shown in figure 23, it is seen that the value of M_θ/M_π almost coincides in both cases as in figure 25a. Precisely speaking, both results coincide better within an interval $\theta = 0 \approx \pi$, while they deviate slightly within the interval $\theta = \pi \sim 2\pi$. Referring to figure 23, the pressure distribution diagram in the interval $\theta = 0 \sim \pi$, the pressure is nearly constant, but there exist some irregularities in the interval $\theta = \pi \sim 2\pi$, thus presenting the similar characteristics as the former.

In case of the ring No. 1", the value M_θ/M_π does not coincide with that of the ideal ring, much irregularity being observed in figure 25b; then it is seen that the curve form of this ring at the free state differs from the ideal one remarkably.

3. Bending Moment Obtained from the Pressure Distribution Diagram

Next, examine the variation of bending moment of the ring obtained from the pressure distribution diagram by using the formula (3). To calculate the value $\int_0^\theta p \sin(\theta-\varphi)d\varphi$, the method of graphical integration was resorted to - that is, putting $\theta = \theta_o$, say, $\theta_o = 20^\circ$, 40° . . . π, and plotting the value of $p \sin(\theta-\varphi)$ against the base of φ, there is obtained the curve shown in figure 25'. When the area of this curve between the limits by the planimeter is measured, the value $\int_0^\theta p \sin(\theta-\varphi)d\varphi$ can be obtained. Hence, M_o of the ring at any angle θ can be found by the formula (3) and consequently M_θ/M_π is known. The bending moment for the ring No. 1" from the pressure distribution diagram shown in figure 22 was calculated by the graphical method, and the calculated values M_θ/M_π are given in the fifth column in table XVI, and plotted in figure 25c with marks x. Comparing this bending moment curve with that of the ideal ring, it is seen that the deviations of those two curves are nearly similar as in the case shown in figure 25b. By the piezo-electric measurement, the pressure intensity is taken constant in a comparatively wide range of the contact arc of the ring, say, $\varphi = 28^\circ$ in the case of the ring No. 1"; so the deviation of the bending moment curve calculated thereby from the ideal curve is somewhat smaller as shown in figure 25c.

By these calculations of the bending moment from the variation of curvature or the pressure distribution diagram and the resemblance of two curves in figure 25b and c, it would be understood that the pressure distribution measured by the newly devised ring tester gives a correct figure.

4. Verification of the Measured Pressure Distribution

Being Free from the Frictional Force

Next, when a piston ring is put in a cylinder, it must be in an equilibrium under radial pressures and frictional forces between the ring surface and the cylinder wall. Accordingly, the radial pressure distribution of the ring on the cylinder wall may, more or less, be disturbed by the frictional forces and the ring may lean to one side. The phenomenon of dislocation of the ring when its surface is not finished by the grinder and roughly turned has often been observed. In the mechanical tester shown in figure 1 or 2, the ring must slide along the circumference of the cylinder wall in order to be compressed to a fixed diameter. There appear on bell cranks or pressure gages such great frictional forces besides the radial pressure of the ring, that an accurate measurement of the radial pressure becomes very difficult. The best method to avoid the dislocation of the ring is to reciprocate it in direction of the cylinder axis, supplying a rich lubricant between the ring surface and the cylinder wall. Or, as described in the ring tester of type I, nearly the same result will be obtained by chucking lightly hemp strips wound on several positions of the ring.

To examine whether the radial pressure distributions of the rings Nos. 1", 7", and 12 satisfy the conditions of equilibrium, a system of rectangular coordinates is taken in the pressure diagram, the center of which coincides with the origin of the coordinates, and calculate x and y components of radial pressure. Let p_x and p_y be x and y components, respectively, which can be easily estimated from the pressure diagram by the graphical method and then a resultant force $p_r = \sqrt{p_x^2 + p_y^2}$ can be obtained. Values of p_x, p_y, and p_r are shown in table XVII; they are all very small and the resultant force p_r is not much more than 3 percent compared with the total radial pressure distributed on the whole peripheral surface of the ring. Therefore, it may be said, that the disturbing effect of the frictional force on the equilibrium of the ring is very small, and the radial pressures distributed are nearly in equilibrium.

5. Some Notes on I.P.C.R. Method of Measuring the Pressure Distribution

In the case where the pressure dibtribution of the ring is very irregular, the values obtained for the pressure intensity at the part having sharp pressure gradient, are somewhat inaccurate; because the pressure intensity was taken by assuming the radial pressure p_θ on the segmental arc $R\omega$ to be constant. Also, in the case of such rings as Nos. 5 and 6, the values corresponding to the peak pressure at the gap ends or other parts are unreliable on account of the contact area being unknown. To get rid of these errors the angle ω should be selected as small as possible, but the measurement will become thereby difficult.

As regards the mean spring power of the ring, the usual practice is to measure it by compressing the ring with a flexible band wound about its peripheral surface until the gap ends are closed. If the tension W of the band is known, the mean spring power p_{mno} of the ring can be calculated by the formula,

$$p_{mno} = W/bR \tag{6}$$

The mean spring power p_{mno} of the rings determined by such a method is shown on the fourth column in table XVIII, and the values p_{mn} measured by the piston ring tester explained before are on the third column. Comparing the values of p_{mn} with p_{mno}, it is seen that the differences are very small in all cases, being smaller as the pressure distribution becomes more uniform. When the ordinary piston ring is compressed by uniform radial pressures, its outer surface deviates much from a true circle, being elongated in the direction of the axis through the gap ends. It may be said that the gap ends of these rings may be more easily closed under the compression by a flexible band. On the other hand, the rings which exert nearly uniform pressure on the cylinder wall take approximately a true circular form under uniform pressure; so the difference between the values p_{mn} and p_{mno} is smaller in this case.

V. MANUFACTURING PROCESS OF THE PISTON RING EXERTING UNIFORM PRESSURE

Tests of many sample pieces of piston rings show that there is surprisingly a wide range of pressure irregularity which is disadvantageous not only for the packing action but also for the abrasion of the cylinder wall. If there may be found a narrow clearance between the ring surface and the cylinder wall, gas or steam in high pressure and temperature will freely escape through it; and moreover, the lubricating oil will be pumped up into the combustion chamber. In consequence, the incomplete combustion of the lubricating oil happens in the combustion chamber, on the piston surface, and so forth, resulting in a very undesirable feature for the engine practice.

Now it seems that the manufacturing processes of the piston ring in usual practice are imperfect and cannot make the piston ring of uniform pressure expediently. The simplest and cheapest method is described before in case of the ring No. 6, but its pressure distribution, as shown in figure 16, is the worst.

Next, there is another process in which a large number of rings, made by the same simplest method, are put into the cylinder of a moderate diameter and tightened from both sides of these rings; then, the cylinder is removed and they are finished to their outer diameter. This process is comparatively easy and used very widely, but in this method, as the ring is finished in the state excessively compressed in the lateral direction through the gap ends, the intense pressure is produced at the gap ends of rings and at the opposite sides.

The so-called hammered process is also used widely which gives the spring power to the ring by cold workings, say hammering, rolling, or scratching. In these methods, the curve form of the finished ring changes variously according to the degree and method of cold workings, to the kind of material used, to the dimension of rings, and to the form of gap ends. It may, therefore, be very difficult to make piston rings which will exert uniform pressure each on the cylinder wall at all times.

Concerning an eccentric ring, a rule about its thickness variation to fulfill the condition, that the ring may exert uniform pressure on the cylinder wall, is explained in textbooks. Recently the thickness variation of an eccentric ring was calculated more accurately by K. Aichi (reference 4); but after all, the difference of thickness calculated by his formula from that found by the foregoing simple calculation is very small. The eccentric ring designed theoretically to exert uniform pressure on the cylinder wall, will not be employed in practice on account of the variation of thickness being wide in ranges. And, the eccentric rings used at present, as described above, have much irregularity in their pressure distribution.

R. Bennet (reference 5) calculated theoretically the form of the ring in its free state, which exerts uniform pressure on the cylinder wall. The rings are turned out from a cast iron cylinder under the guide of a templet, which has a form similar to the ring of uniform pressure in its free state. These rings were used for the superheated steam locomotives of Swedish railways, in which the rings of the ordinary hammered type were also used. After running 80,000 kilometers, both the rings and the cylinders were carefully examined and the Bennet rings showed exceedingly good results. This manufacturing process of the piston ring is too complicated and does not, admit, it seems, of mass production. Moreover, the uniformity and elasticity of ring material are at first assumed, and then its curve form, thickness, and gap length are found by theoretical calculations, so that the finished ring must satisfy these conditions in order to give uniform pressure on the cylinder wall. But, in practice, these conditions initially expected will not be easily fulfilled in each piston ring, the permanent set of materials or other disturbing effects during the turning process being not small.

A. Inokuty (reference 6) studied theoretically the manufacturing process of the piston ring of uniform pressure and deduced an important principle which runs as follows.

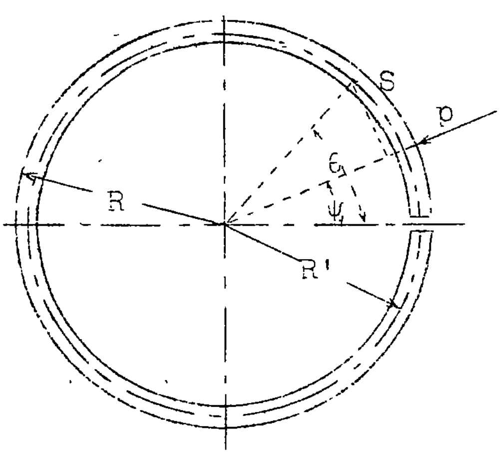


Figure A

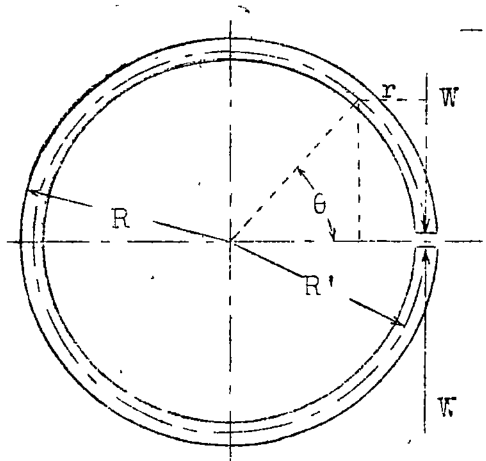


Figure B

In figure A, a ring is compressed by the radial pressure p equally distributed on its peripheral surface. In figure B, a ring is compressed by tangential forces W acting at the gap ends.

The bending moment at S,

in figure A, $$M_\theta = pbRR' \int_0^\theta \sin(\theta-\psi)d\psi = pbRR'(1-\cos\theta) \qquad (a)$$

and in figure B, $$M_\theta = Wr = WR'\ (1-\cos\theta) \qquad (b)$$

Let $W = pbR$, the formulas (a) and (b) will exactly coincide; therefore if a piston ring be compressed by any tangential force W acting at the gap ends, the result will be the same as if it were compressed by the radial pressure $p = \frac{W}{bR}$, equally distributed on its peripheral surface.

Inokuty devised a practical method of making a piston ring of uniform thickness exerting uniform pressure, but his manufacturing process may be considered somewhat complicated.

However, this principle is interesting and so important that the author applied it to the manufacturing process of the piston ring of uniform pressure as described hereafter.

Many other processes have been proposed concerning the manufacture of piston rings of uniform pressure. In one of them, a long,hollow cylinder having a necessary gap is compressed by a flexible band or rope of sufficient strength, and after finishing its diameter, an individual ring is to be cut off. In another, a cylindrical blank is clamped tightly at its gap ends by rivets or bolts and rings are turned thereof. These processes, however, are also imperfect and to be much improved.

Now, what shall be done to make a piston ring of uniform pressure? It will be an ideal process to finish a piston ring in the state compressed by radial pressures equally distributed on the peripheral surface, but it can hardly be realized in practice. If, however, a ring is tightened only by the tangential forces acting at the gap ends, the bending moment at any point of the ring will be the same as when it is compressed by uniformly distributed radial pressures. This principle was proposed in reference 6, and remarks on such an idea are also to be found in foreign papers; notwithstanding, no further investigations for finding an appropriate method applying this principle have yet been made.

Here, the author tried to find the simplest and best method to apply this ingenious principle for the manufacture of a piston ring of uniform pressure. Several processes for machining and a large number of trials have been experimented. In one of them, he clamped the gap ends of a piston ring by means of a small pin, and fixed the ring between two flat plates as shown in figure 26. Then, the ring was turned and finished on its outer surface equal to the cylinder diameter. In the actual working of this process, the ring was at first roughly turned and then released from the side pressure by loosening the clamping. The object of doing this was that the ring may take a free curve form with its gap ends clamped, care being taken to keep the center of the ring from getting out of position when the ring is released. After this operation the ring was again moderately clamped and received its final finish. The piston rings manufactured by such a new process* exert nearly uniform pressure on the cylinder wall, as shown in the ring No. 7 (fig. 17). Moreover, it was also ascertained by measurements, that the curve form of this ring in its free state coincides with that of an ideal ring which exerts uniform pressure against the cylinder wall. In this manufacturing process the ring is finished in nearly the same state as compressed under uniform radial pressures; therefore, it is not necessary to make the gap length and thickness of the ring agree accurately with dimensions previously determined. As the variation of thickness is very small and negligible in comparison with the diameter of the ring, the uniformity of the pressure distribution is not much disturbed even if the outside diameter only is finished. So, if the inside of the piston ring is to remain as cast, it can be done comparatively easily without much disturbing the uniformity of the pressure distribution. This insures many advantages: the simplification of the working process, the economy of material, and improvements of qualities. Another advantage of the new manufacturing process is that neither the requirements concerning the property of material and the dimension of ring nor any complicated calculations are necessary for the provision of a cylinder blank, from which piston rings are to be manufactured, as in the case of Bennet's or other manufacturing processes. It will be noted that this working process is applicable in cases where the diameter of the ring may be either large or small, and also where the material may be of cast iron, bronze, brass, and so forth. But in this process, a precise caution must be taken in the last finish of the outer surface, that the side pressure of two flat plates, between which the ring is inserted, must be so weak that it will not disturb the free curve form of the ring, the gap ends being clamped by means of a pin joint. The disturbing effect coming from the boundary force during the finishing operation on the uniformity of the pressure distribution of ring is not negligible, especially when the spring power of the ring is weak, or when a large number of such rings piled on a core drum is clamped together and turned to the desired diameter. In the case of the ring No. 12 where its spring power is weak and finished by a bite not by a grinder, the ring is pressed more heavily than in the case of a grinder finish, so the disturbance arising from the boundary force is more noticeable than that of the ring No. 7; accordingly, the

*Japanese patent, 70144; French Patent, No. 617,919; British patent, No. 278,048; U.S. Patent, No. 1,666,343.

pressure distribution of the former is somewhat worse when compared with that of the latter.

VI. HANDY METHOD FOR TESTING ANY PISTON RING

As described in the foregoing paragraph, a piston ring which exerts nearly uniform pressure on the cylinder wall can easily be made if it is turned to a true circle by clamping its gap ends by means of a pin joint. Then, if a piston ring of uniform pressure is clamped at its gap ends by a pin joint, it is easy to understand that the ring will take a true circular form at its peripheral surface. A ring which does not take a true circular form when clamped at its gap ends by a pin joint will not exert uniform pressure on the cylinder wall, and the greater the irregularity the worse will be the pressure distribution. Hence, it will be convenient, though not rigorous, to compare the quality of the piston ring by measuring how much its periphery deviates from a true circular form when clamped at its gap ends. Figure 27 shows the measuring arrangement of a ring for this purpose; 1 is a turning table which is able to turn accurately around a conical center, its periphery being graduated to 1°. A piston ring 2 to be examined is rested on a circular plate 3 and clamped lightly by clamping jaws 4. The circular plate 3 is adjustable with regard to the turning table 1, thus enabling the position of a ring easily adjusted on the plate 1. In the actual operation, a piston ring to be examined is at first clamped at its gap ends and fixed gently on the plate 3, and then by turning the table 1 the position of the ring is adjusted so that the readings of the dial indicator 5 show $r_1 = r_3$ and $r_2 = r_4$.

Specifications of sample rings are shown in table XIX. Most of them are made in famous manufacturing companies in Japan, Europe, and America. A pin hole at the gap ends is drilled in each ring while being compressed in the bore of a diameter accurately equal to that of the engine cylinder as shown in figure 28. Variations δr of the radius vector of each sample ring were precisely measured by the apparatus in figure 27 and shown graphically in figures 29 to 48 on convenient scales. Comparing the variation of radius vector with the irregularity of the pressure distribution of the rings in figures 29 to 32, it is found that in the case of the ring No. 14 where the pressure distribution is nearly uniform, the value of δr is very small and its periphery nearly forms a true circle. In the case of the ring No. 15, the value of δr becomes very large with the increase of the irregularity of the pressure distribution, and the worst pressure distribution accompanies the largest value of δr as in the ring No. 17. Any ring in which the pressure distribution is irregular is never found to have a constant radius vector when clamped at its gap ends.

Another example of a smaller diameter is shown in figures 33 to 35, and a similar relation between the variation of radius vectors and the irregularity of the pressure distribution is found as in case of the larger diameter, the smaller being the value δr the more uniform the pressure distribution. In comparing rings of different diameters and thicknesses rigorously, it is not, of course, sufficient to consider only the value of δr. The functional relation among the radius vector, the thickness, and the pressure distribution, is very complex; and it will be almost impossible to obtain the relation from measurements of these data. Nevertheless, it is convenient in practice to examine roughly whether or not the quality of a ring is good by measuring its deviation from a true circle when the gap ends are clamped as previously mentioned.

No. 21 is one of the rings made by the new process in the I.P.C.R., consisting of about 50 pieces of rings piled together with specially designed attachments; the variation of radius vector in figure 36 is very small, the ring forming nearly a true circle. For No. 22, though it was finished by the same process as No. 21, the result in figure 37 is not so good as No. 21, on account of its breadth being narrow and mean spring power very weak. Of the ring made by the new process in the I.P.C.R., the author measured simply the diameter in all directions, and examined the deviation from a true circle, the diameter ranging between 25.4 and 530 millimeters. The results are all good, showing them to be approximately true circles.

A ring No. 23 made by The Piston Ring Co. shows a pretty good result as shown in figure 38. No. 24 is a ring of an automobile engine; its radius vector diagram (fig. 39) shows small irregularities but gives a pretty good result. On the contrary, a ring No. 25 is very irregular and unsymmetrical (fig. 40); and will naturally give an irregular pressure distribution. No. 26 is a ring specially constructed (fig. 48) in which a steel expander having an oval form is put in a triangular groove of the outside bronze ring to give it the spring action. In spite of the complicated construction and consequent high cost, the result of measurement shows it to be the worst, (fig. 41) among the sample pieces, except the ring No. 17. The ring ought to be improved more in the combination of its steel expander and outside ring.

All the rings Nos. 27 to 32 are the so-called hammered rings, being hammered, rolled, or scratched inside to give the spring action. No. 27 is a ring to be used for an aero-engine and shows a good quality, having very small irregularity of radius vectors as seen in figure 42. No. 28 is a ring to be used for a solid-injection high-speed Diesel engine of the latest type, for which the value of δr as shown in figure 43 is not small. Nos. 30 and 31 are for aero-engines, and Nos. 29 and 32 are for automobile engines, respectively. They all show very much irregularity in the radius vectors as shown in figures 44 to 47.

On the whole, it may be said that hammered rings may have comparatively good qualities, when their gap ends are of the diagonal or the straight forms. Because, in the case where the spring action is given by cold working heaviest at the opposite side of the gap ends, decreasing gradually toward the neighborhood of gap ends where no cold working is given, the curvature near the gap ends will not be altered from its initial value. When the gap ends are of the stepped form, especially in case of a ring having a long step, there is a pretty great difference of curvatures between the ring surface at the free state and inside of the cylinder; so it is very difficult to fit all the ring surface to the cylinder wall. It seems that the manufacturing process of piston ring by means of cold working should be much more carefully studied. It is ascertained from the results stated above, that all the piston rings made by ordinary manufacturing processes do not possess the curve form in their free state which will give uniform pressure on the cylinder wall; only, the rings Nos. 24 and 27 are better than others.

If the ring made by the new manufacturing process in this Institute is compared with those made by ordinary processes in Europe and America, it can readily be seen how much improved the new manufacturing process is. It may be worth while to remark that this manufacturing process is applicable for all cases where the dimension of piston ring is either large or small, and also where the material is cast iron, bronze, brass, and so forth. Moreover, this process requires no gross equipment to begin with.

It may not be out of place to remark in this connection, that the author in course of inventing this new manufacturing process felt much indebtedness to the late Prof. A. Inokuty for his distinguished essay (reference 6).

VII. EFFECT OF ANNEALING ON THE PRESSURE DISTRIBUTION

It is not yet clearly known how the pressure distribution of the piston ring is affected by the rise of temperature during the running of an internal combustion engine under load, though many experimental researches (reference 7) have been reported as to the temperature of the piston and piston rings in question. Under a normal state of running, it seems that the temperature of the piston ring is rather low, the maximum estimated value not exceeding 300° C. Since the heat effect may not be small when the ring is heated to temperature higher than 500° C under an abnormal state of running, the temperature effect was taken into consideration by the author. To obtain the data of the ring pressure on the cylinder wall in the hot state is, of course, very difficult; so the measurement was made on annealed rings in the cold state. The heat treatment of sample rings is shown in table XX. For annealing purposes a cast iron hollow cylinder was prepared, its inner diameter being equal to that of the engine cylinder, and the rings to be annealed are placed inside it with graphite powder and then heated to a required temperature in an electric furnace for several hours. Figures 49 to 51 and tables XXI to XXVIII show the effect of annealing on the pressure distribution, and figure 52 shows the same on the mean spring power. It is interesting to observe that the configuration of the pressure distribution is not much altered by annealing below 400° C, both the irregularity and the uniformity of the pressure distribution being not unchanged. The mean pressure, however, decreases gradually with the rise of the annealing temperature, rather rapidly beyond 300° C, dropping down to half the pressure of the virgin ring at about 500° C. The effect of annealing on the mean pressure is comparatively large for the first few hours and becomes weak afterwards. Further experiments were carried out on the heat effect, and the duration of annealing, on the mean spring power for other sample rings. Table XXIX shows the specification and heat treatment of these sample rings. In these cases, the mean spring power was determined by winding a thin steel band around the ring surface using the formula (6), the annealing process being the same as explained before. Figure 53 shows the decrease of the mean spring power of the piston ring with the rise of the annealing temperature, the full line representing the case of ordinary piston rings and the dotted line that of the hammered rings. The effect of annealing on the mean spring power is nearly the same as in the case shown in figure 52, though it seems more effective for the hammered rings.

The time effect on the annealing is shown in figure 54, from which it is clearly understood that it is, as previously stated, noticeable only for the first few hours, and after repeated heating and cooling for a long time below 300° C, it becomes very small. It may be said from these experimental results, that the heat effect will not be dangerous, if the piston ring is not heated beyond 300° C in the running condition.

Next, examine the pressure distribution of piston rings after the actual service for a long time. Figures 55 to 57 and tables XXX to XXXII show the pressure distribution of the rings made by the The Piston Ring Co. and used for half a year in a 6 horsepower four-cycle Diesel engine of vertical type (bore 127 mm and stroke 191 mm). It would seem that the irregularity of the pressure was much equalized and the maximum pressure at the gap ends is considerably reduced, slight depressions being formed at these ends. The ratio of maximum and minimum pressures becomes smaller in the old ring and in the case of ring No. 44, which was used at the top of the piston. It seems that the mean pressure is not reduced more than 10 percent of its original value.

VIII. RELATION OF THE GAS TIGHTNESS OF PISTON RINGS TO THEIR NUMBER AND SPRING POWER

The pressure distribution of the piston ring on the cylinder wall having been ascertained, an examination is now made of the relation between the gas tightness and the pressure of the ring on the wall. For this experiment, two kinds of piston rings were used - one made abroad and the other in this Institute - and the engine used was a 3-horsepower four-cycle Diesel engine having 89 millimeter bore and 140 millimeter stroke.

a_1 virgin rings, same kind as those of No. 23 in table XIX explained before

a_2 same kind as a_1 but old one having been used for a month; its outer surface self-ground and well-fitted to the cylinder wall

b_1 and b_2 virgin rings specially made in this Institute to produce uniform distribution of pressures but with their outer surfaces rough

The amount of mean spring power was measured by compressing the ring with a thin steel band until the gap becomes just the same as when the ring was put into the cylinder. To reduce the amount of mean spring power, the rings were annealed in a hollow cylinder - as explained before. The engine was driven by an electric motor and the maximum pressure at the end of compression stroke was measured by means of Dobbie McInnes spring indicator. The maximum pressure thus measured is plotted in figure 58 and shown in table XXXIII with regard to the number of revolutions. By the figure it is seen that this maximum pressure decreases with the number of revolutions. It will be noted that no distinct rise of pressure is recognized even if the number of rings be increased up to 3, 4, and 7.

In order to compare the merits of rings for gas tightness more accurately, slow revolutions of the engine are preferable; and so the number of revolutions was kept to about 130 to 135 per minute, which was attained by hand driving. The inner surface of the cylinder and the piston rings were carefully washed with gasoline to remove the lubricant used, and only one ring was placed at the top of the piston groove.

The maximum pressures thus measured are shown in table XXXIV and in figure 59. In the case of an old ring a_2 the highest pressure was obtained; but comparing the virgin rings b_1 and b_2 with a_1, the compression pressure was higher in the former cases even if the outer surface of these rings was not satisfactory, owing to the vibration of the grinder during their finishing. The rings a_1 and a_2 were made by The Piston Ring Co.; the pressure distribution was somewhat irregular but all peripheral surface of the ring was well in contact, no partial touch or clearance between the ring and the cylinder wall being observed. When the rings of a worse pressure distribution than ring No. 6 was used, the packing action was found to decrease considerably; one such example was fully stated in another paper (reference 8). In short, the perfect fitting of the ring surface to the cylinder wall is necessary for the gas tightness. It will also be noted that the packing action of the ring is not much altered by decreasing the mean spring power of the ring (fig. 59), especially in the case of a ring of uniform pressure.

A perfect contact of the two surfaces is sufficient in practice to prevent the leakage of gases under any pressure. For example, a hollow cylinder having its inner surface well ground to fit a solid piston of good finish will not permit any passage of the gas which is compressed by the piston, provided the cylinder wall is not expanded by the internal pressure. It is, therefore, an absurd idea to keep a higher pressure between the two contact surfaces in order to get a better obturation performed. In fact, the pressure existing between the outer surface of the piston ring and cylinder wall plays no principal role in impeding the leakage of gases, but a perfect contact of the two surfaces is a necessary condition to be attained by all means for the obturation purpose, and the said pressure is preferred to be low enough to bring the outer surface of the ring just in contact with the cylinder wall, even in case where the cylinder is expanded under the gas pressure or as the effect of high temperature.

Many engine makers, on the contrary, are using a ring which has a very strong spring power, exerting an enormous pressure on several parts of the cylinder wall; so the ring is strained by this spring action and obliged to make a slight contact with the other part of the wall, where no touch may be found when the spring power of the ring is weak. Again, a strong spring power may be convenient for so-called rubbing operation, the ring surface being ground rapidly. This may be the only merit of the ring with a strong spring action, and its bad workmanship is concealed by the spring action. It is now clear that the most important thing is the uniform pressure distribution of a light pressure of packing ring on the cylinder wall with a perfect contact, for the sake of minimizing the frictional resistance of the piston and the wear of cylinder wall in a long run of the engine. When a ring of strong spring action with an irregular pressure distribution is used, not only the mechanical efficiency of the engine is diminished by undue frictional resistance, but also the eccentric wear of the cylinder soon becomes conspicuous enough to need the reboring or grinding of the cylinder.

The previous considerations refer to internal combustion engines, but the same reasoning holds good for air or gas compressors, steam engines, and any other engines using piston-packing rings. In the case of low pressure engines, however, the defects of ordinary rings are not so noticeable and, consequently, the question of obturation draws less attention.

Some examples of the local abrasion of piston rings are as follows. In figure 60, a ring was used for only 3 weeks in a marine steam engine of the vertical type, having an initial uniform thickness of 31.8 millimeters. As this ring was manufactured by hand hammering, its initial curve form would have been, perhaps, very irregular; after only 3 weeks' service it wore so much locally that its maximum thickness amounted to about ten times the minimum. Another example is a hammered ring used for several months for an air compressor of Ingersoll-Rand Co. A considerable local abrasion is observed, the maximum and the minimum thicknesses being 3.6 millimeters and 0.4 millimeter, respectively. In these examples, it was not clearly ascertained if these local abrasions were due only to the irregularity of the pressure distribution of the ring on the cylinder wall, but it seemed quite probable; because, in a ring having a good quality of uniform pressure, there appears no such enormous local abrasions. Many other examples of local abrasions are seen in the case of piston rings of locomotive engines on account of the many rings employed therein being manufactured by the simplest method as No. 6. An engineer told the author, that when such a ring made by the simplest method in the cylinder of a steam hammer of 254-millimeter diameter was used, the cyllinder bore became rapidly oval during only two weeks' service; and the difference of the maximum and the minimum diameters amounted to about 2 millimeters. In short, it is surely unprofitable in various respects that the ring has an irregular pressure distribution on the cylinder wall.

Next, the relation between the number of rings and the packing action of the ring by the same method was examined, and the results are shown in figure 61 and table XXXV. When the ring is removed - that is, the piston works in the cylinder with no ring - the maximum pressure is quite low; when only one ring is put in, the abrupt increase of the pressure occurs; but by increasing the number of rings, any higher compression pressure is not obtained in the case of no lubricant, this being a very singular and interesting phenomenon. Similar results are seen in the case of using sufficient lubricant, but a tendency is observed that the increase of the number of rings accompanies a slight increase of the compression pressure. The piston rings used in these experiments are not fresh, their outer surface being self-ground and well-fitted to the cylinder wall. From these results, it may surely be said that when a piston ring has good quality for the packing action, its tightness for a high pressure gas is attained well enough by the first one or two rings, the other rings serving only for a spare. Nearly the same conclusion was given by A. Turner (reference 9) in his experimental reports.

The effect of lubricant for the packing action of rings is very great, and in the case of a rich lubricant the compression pressure is about 20 percent higher than that of no lubricant. The circulation of lubricant is very important, not only for the gas tightness but also for minimizing the frictional resistance. On the other hand, an excessive lubricant in practical operation will introduce many undesirable effects for the driving of engines; so, careful attentions should be paid to this matter. In an internal combustion engine, an excessive lubricant often causes its incomplete combustion. An exact fitting, however, of the piston ring to a cylinder and its stepped gap ends is very effective in preventing the lubricant from being pumped up into the combustion chamber. It will be more interesting if the action of piston rings on gas tightness be carefully studied in the working conditions of internal combustion engine, but the experiment will be very difficult.

IX. FRICTIONAL RESISTANCE OF THE PISTON RING

The frictional resistance of the piston rings of engines is said to be pretty large (reference 10), and it will be interesting to examine how much the frictional resistance of rings would be in comparison with the other frictional and mechanical resistances. The 3-horsepower Diesel engine with the a_2 rings was driven by an electric motor, the cylinder cover being removed to prevent compression. The driving belt was suddenly taken off the engine when its revolution became steady, and the reduction of the rotation was recorded in relation to the time of running. If the relation between the retardation of engine speed and the time elapsed is known, a comparison can be made of the frictional resistance due to the piston ring with the total resistance of the engine driving - except pumping and valve-gearing losses. Now, let w be the revolution of the engine at the time t from the beginning of the retardation; then there is obtained the relation between w and time t as seen in tables XXXVI to XXXVIII. In the case where the piston friction is small - namely, few piston rings - the functional relation between time and speed is expressed by the formula

$$w = w_0 e^{-k't} \tag{7}$$

where w_0 is the initial value of w, and k' is a constant proportional to the coefficient of retardation. This formula does not hold good for the case when the friction becomes large according to the increase of the number of rings. But an empirical formula,

$$w^{\frac{1}{2}} = w_0^{\frac{1}{2}} - kt \tag{8}$$

holds better for all the cases, so that the relation between $w^{\frac{1}{2}}$ and t is expressed exactly by a linear relation as shown in figure 62. Since the piston friction is very small, it may be considered proportional to the mean piston speed in the observed interval, the formula (7) holding good. However, if the number of piston rings increases, small clearances between the piston rings and their grooves would affect the retardation of the reciprocating motion of the piston and the formula (7) would become more and more inapplicable. The experimental data in tables XXXVI to XXXVIII taken on the retardation tests of the engine, when plotted, give a series of straight lines (fig. 62). These straight lines show that the speed w and time elapsed t are related as given by the formula (8). So, compare the frictional resistance of the piston rings with the total frictional resistance by estimating the values k in the formula (8); the results are shown in table XXXIX. From these experiments, it is found that the frictional resistance of rings is estimated to be so large as 60 percent of the total resistance, excepting the pumping and valve-gearing resistances. If the number of rings is reduced to two or three, the total frictional resistance will be diminished by about 25 percent; if the mean pressure of rings which is 0.80 to 0.90 kilograms per square centimeter in present practice (which in the author's opinion is intensely high) be reduced to about one half, there will be a considerable gain of mechanical efficiency of the engine.

X. NOTE ON THE DESIGN OF PISTON RING

The piston rings are generally strained under an excessively large stress when they are put into the piston groove or under the working state; accordingly, there often happens a permanent set or a breakdown in actual operation. If the maximum bending stress in their working state is calculated by the formulas for curved beams, it becomes of surprisingly high values, say, 2000 kilograms per square centimeter or more. Dr. Bach (reference 11) states that in the case of cast iron the ultimate strength against bending may be taken as high as 1.7 times that of its ultimate for the case of the tension or compression. Therefore, the piston ring in use under such a high stress will not immediately break down; but it cannot be treated as an elastic material at its extreme outer or inner layer. The hysteresis loops in the load-strain diagrams of the following experiments may be regarded as supplying evidence of what has just been mentioned. By connecting the threads to the gap ends of the ring, it was expanded to the diameter until it could be put into the groove cut in piston body, and then compressed until the gap ends were closed. The relation between the gap length and the forces acting in the tangential direction at the gap ends are plotted in figures 63 and 64. The load strain diagrams drawn for every ring give pretty large hysteresis loops, and many other sample rings show nearly the same results. It may be said that the maximum bending moment is greater in the case when the ring is expanded to slip over the piston body than when it is compressed, but the difference between the two is small so far as the experiments show. Though the rings used in the experiment had fine structure and good quality, they described such hysteresis loops; accordingly, in the case of piston rings of ordinary cast iron very large hysteresis loops may well be expected to be drawn. It is undesirable for a piston ring to yield under the permanent set, when it is put into the corresponding piston groove or in its working state. If the curvature of the piston ring be deformed permanently during the operation of putting in or taking out of the groove, the elaborate task performed in the manufacture of the ring to insure uniform pressure would become entirely useless. Thus, the maximum stress must be taken as small as possible in the design of piston rings, so as not to produce a permanent set. Since it is very complex, as described later (sec. XI), to estimate the true value of maximum fiber stress in each piston ring under its working state, it is convenient to determine the dimensions of piston rings by calculating an apparent stress, though not the true stress. For the sample rings Nos. 3 to 33, the apparent maximum stress under the state of being put within the cylinder, was calculated by the formula for curved beams (reference 12)

$$\sigma_{max} = \frac{W}{f} + \frac{M}{fr} + \frac{M}{I} \frac{t/2}{1 + \frac{t/2}{r}}$$

if $2r = D$, there is obtained for the tension side,

$$\sigma_{max\ t} = \frac{3p_{mn}\ D^2}{t^2} \frac{1}{1 + \frac{t}{D}}$$

where $\sigma_{max\ t}$ is the maximum tensile stress; p_{mn}, D, and t already have been defined. Calculated values of $\sigma_{max\ t}$ are shown in table XL, from which it is seen that all the rings except Nos. 12 and 13 are strained under excessive stresses beyond the limit of applicability of the formulas; these formulas are supposed to hold within the limit of elasticity of the material. Though the values of the maximum stresses thus calculated are not true but only apparent, these values should be taken into account in the design of piston rings, as famous ring makers in Europe and America do.

Next, the author measured the dimensions of various rings employed in practice and obtained relations between the thickness t, breadth b, gap length l, and the diameter D. From the results of measurement shown in figure 65, it is found that the values of t, b, and l are all proportional to the diameter within the range of experiments, though b and l are somewhat irregular. As to the value of b, it was taken for a very wide range according to the kind of engines. In high speed services, especially in aero-engines, very narrow breadth is adopted as shown with a dotted line in figure 65. Even in the case of larger diameters the breadth does not increase so much. For example, a piston ring used in a locomotive engine of 530-millimeter diameter has breadth 20 millimeters and another valve ring of 210-millimeter diameter has breadth 6 millimeters, and so forth. As previously stated, the number of rings play no principal role for the packing action, only resulting to increase the frictional resistance of piston; hence, no large breadth of the ring will be necessary. Moreover, in the case where the cylinder expands or wears conically owing to a temperature gradient, and so forth, along the wall, the high pressure gas would leak easily through the wedge-shape space between the ring surface and the cylinder wall. It is easily understood that the narrower is the breadth of the ring the better contact of the all-ring surface with the cylinder wall, and the less frictional resistance will be experienced. Therefore, the breadth of the ring should be as narrow as its strength permits. In view of this, it is thought that the piston rings used in aero-engines require much improvement.

The mean spring powers of the piston rings explained before, it seems, are excessively strong; but from experiments, it may be said that the exact fitting of the all-peripheral surface of the ring to the cylinder wall is only necessary for the gas tightness, an excessively intense pressure being not only useless but injurious. Hence, the spring power of the piston ring should be made comparatively weak. For this purpose, the question arises whether its thickness t is to be chosen smaller or the gap length l taken shorter. By the theory of elasticity it is known that $p_{mn} \propto l \times t^3$ (reference 6) and $\sigma_{max} \propto \frac{p_{mn}}{t^2}$ approximately, hence; $\sigma_{max} \propto l \times t$ (reference 14). If gap length l is taken smaller, it will not be convenient to put the ring into the piston groove as in the case of an eccentric ring No. 5 or 5', the permanent set being easily established. On the contrary, when t is smaller, even slightly, the diminution of the mean spring power is considerable and the ring will be more flexible, σ_{max} becoming correspondingly smaller. In short, in the design of piston rings it is convenient and sure to determine their dimensions according to the previously stated conception, taking into consideration the dimensions of piston rings of famous makers.

XI. CALCULATION OF TRUE STRESS ON THE SKIN OF PISTON RING

The maximum bending moment of the piston ring under the working condition is very large, and the maximum stress estimated by the ordinary formulas for curved beams appear to be so excessive that the formulas could not be used for the stress calculation of cast-iron rings. In order to calculate the true value of maximum stress, the author tried to introduce here the "Potenz Gesetz" $\epsilon = a\sigma^m$ according to Bach, where σ and ϵ are stress and strain, respectively, and a, m are certain constants.

In figure 66,

	initial	final
Radius of curvature at the central surface	ρ_o	ρ
Length of arc element.	ds_o	ds
Angle subtended by the arc element at the center of curvature	$d\varphi$	$d\varphi + \Delta d\varphi$
Half the thickness of the ring	h	
Breadth of the ring	b	
Distance of the neutral surface from the central surface	e	
Strain of arc element distant by η from the neutral surface;		
tensile strain.	ϵ_t	
compressive strain.	ϵ_c	

Referring to figure 66,

$$ds_o = (\rho_o - e + \eta)d\varphi, \quad ds = (\rho + \eta)(d\varphi + \Delta d\varphi)$$

$$\epsilon = \frac{ds - ds_o}{ds_o} = \frac{(\rho + \eta)(d\varphi + \Delta d\varphi) - (\rho_o - e + \eta)}{(\rho_o - e + \eta)d\varphi}$$

$$= \frac{(\rho - \rho_o + e)d\varphi + (\rho + \eta)\Delta d\varphi}{(\rho_o - e + \eta)d\varphi} \qquad (10)$$

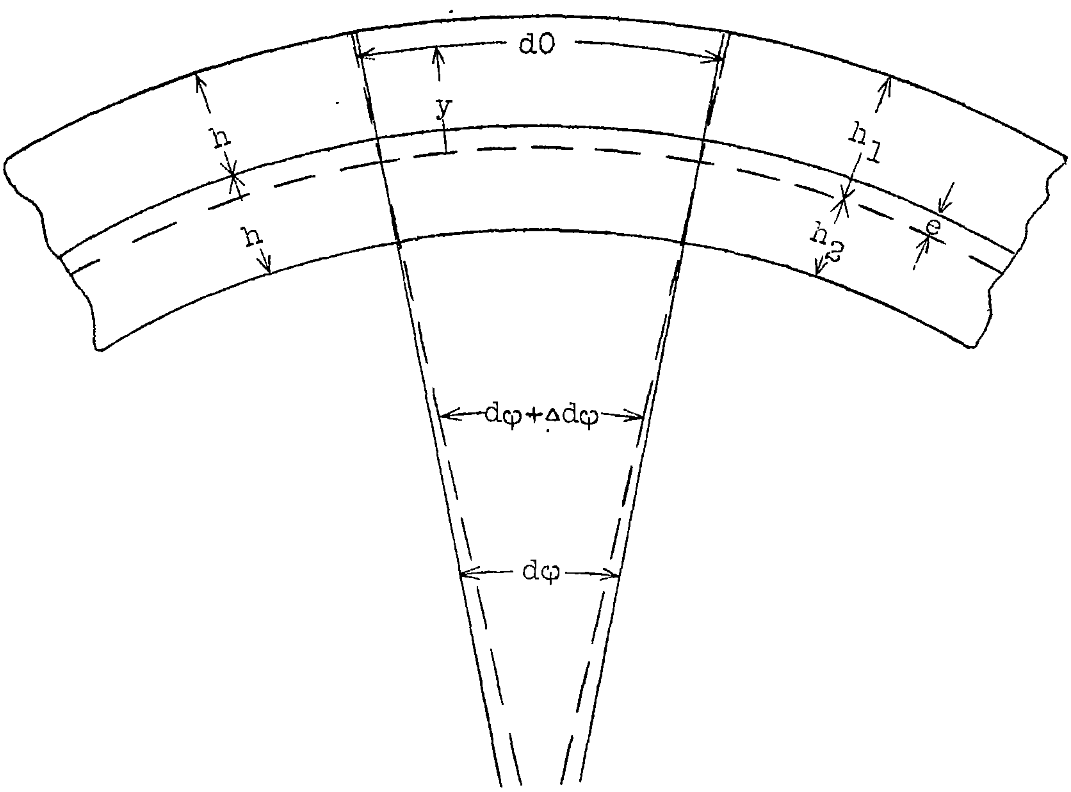


Figure 66.

$$\because (\rho_o - e)d\varphi = \rho(d\varphi + \Delta d\varphi)$$

$$\therefore \quad \frac{d\varphi + \Delta d\varphi}{d\varphi} = \frac{\rho_o - e}{\rho}$$

$$\therefore \quad \frac{\Delta d\varphi}{d\varphi} = \frac{\rho_o - \rho - e}{\rho} \tag{11}$$

By combining formulas (10) and (11),

$$\epsilon_t = \frac{\rho - \rho_o + e}{\rho_o - e + \eta} + \frac{(\rho + \eta)(\rho_o - \rho - e)}{(\rho_o - e + \eta)\rho}$$

$$= \frac{(\rho_o - \rho - e)\eta}{(\rho_o - e + \eta)\rho} \tag{12}$$

In the same way,

$$\epsilon_c = \frac{(\rho_o - \rho - e)\eta}{(\rho_o - e - \eta)\rho} \tag{13}$$

Introducing the relations $\epsilon_t = a_1\sigma_t^{m_1}$ and $\epsilon_c = a_2\sigma_c^{m_2}$, there is obtained the stress for tension side,

$$\sigma_t = \left\{\frac{(\rho_o - \rho - e)\eta}{a_1\rho(\rho_o - e + \eta)}\right\}^{\frac{1}{m_1}}$$

or

$$\sigma_1(\text{max, stress}) = \left\{\frac{h_1(\rho_o - \rho - e)}{a_1\rho(\rho_o - e + h_1)}\right\}^{\frac{1}{m_1}} \tag{14}$$

and for the compression side,

$$\sigma_c = \left\{\frac{(\rho_o - \rho - e)\eta}{a_2\rho(\rho_o - e - \eta)}\right\}^{\frac{1}{m_2}}$$

or

$$\sigma_2(\text{max. stress}) = \left\{\frac{h_2(\rho_0 - \rho - e)}{a_2\rho(\rho_0 - e - h_2)}\right\}^{\frac{1}{m_2}} \tag{15}$$

From the equilibrium of the stresses and bending moment, the following equations will be obtained,

$$\int_0^{h_1} \sigma_t df - \int_0^{h_2} \sigma_c df = 0 \tag{16}$$

and

$$\int_0^{h_1} \sigma_t \eta df + \int_0^{h_2} \sigma_c \eta df = M \tag{17}$$

Substituting the values of σ_t and σ_c, there is obtained

$$b\int_0^{h_1} \left\{\frac{(\rho_0 - \rho - e)\eta}{a_1\rho(\rho_0 - e + \eta)}\right\}^{\frac{1}{m_1}} d\eta - b\int_0^{h_2} \left\{\frac{(\rho_0 - \rho - e)\eta}{a_2\rho(\rho_0 - e - \eta)}\right\}^{\frac{1}{m_2}} d\eta = 0 \tag{18}$$

and

$$b\int_0^{h_1} \left\{\frac{(\rho_0 - \rho - e)\eta}{a_1\rho(\rho_0 - e + \eta)}\right\}^{\frac{1}{m_1}} \eta d\eta + b\int_0^{h_2} \left\{\frac{(\rho_0 - \rho - e)\eta}{a_2\rho(\rho_0 - e - \eta)}\right\}^{\frac{1}{m_2}} \eta d\eta = M \tag{19}$$

From the relation given in the formulas (14) and (15),

$$\left\{\frac{\rho_0 - \rho - e}{a_1\rho}\right\}^{n_1} = \sigma_1\left\{\frac{\rho_0 - e + h_1}{h_1}\right\}^{n_1} \tag{20}$$

and

$$\left\{\frac{\rho_0 - \rho - e}{a_2\rho}\right\}^{n_2} = \sigma_2\left\{\frac{\rho_0 - e - h_2}{h_2}\right\}^{n_2} \tag{21}$$

where $n_1 = \frac{1}{m_1}$ and $n_2 = \frac{1}{m_2}$

By substituting these values in the formulas (18) and (19), ρ will be eliminated,

$$\sigma_1 \left\{\frac{\rho_0 - e + h_1}{h_1}\right\}^{n_1} \int_0^{h_1} \frac{\eta^{n_1}}{(\rho_0 - e + \eta)^{n_1}}\, d\eta$$

$$-\sigma_2 \left\{\frac{\rho_0 - e - h_2}{h_2}\right\}^{n_2} \int_0^{h_2} \frac{\eta^{n_2}}{(\rho_0 - e - \eta)^{n_2}}\, d\eta = 0 \qquad (22)$$

and

$$b\sigma_1 \left\{\frac{\rho_0 - e + h_1}{h_1}\right\}^{n_1} \int_0^{h_1} \frac{\eta^{n_1+1}}{(\rho_0 - e + \eta)^{n_1}}\, d\eta$$

$$+ b\sigma_2 \left\{\frac{\rho_0 - e - h_2}{h_2}\right\}^{n_2} \int_0^{h_2} \frac{\eta^{n_2+1}}{(\rho_0 - e - \eta)^{n_2}}\, d\eta = M \qquad (23)$$

The integration of the values $\int^{h_1} \frac{\eta^{n_1}}{(\rho_0 - e + \eta)^{n_1}}\, d\eta$, and so forth, is made as follows:

$$Is_1 = \int_0^{h_1} \frac{\eta^{n_1}}{(\rho_0 - e + \eta)^{n_1}}\, d\eta = \int_0^{h_1} \eta^{n_1} (\rho_0 - e + \eta)^{-n_1}\, d\eta$$

$$\frac{1}{(\rho_0 - e)^{n_1}} \int_0^{h_1} \eta^{n_1} \left(1 + \frac{\eta}{\rho_0 - e}\right)^{-n_1} d\eta = \frac{1}{(\rho_0 - e)^{n_1}} \int_0^{h_1} \eta^{n_1} \left\{1 - \frac{n_1}{\rho_0 - e}\, \eta \right.$$

$$\left. + \frac{n_1(n_1 + 1)}{2(\rho_0 - e)^2}\, \eta^2 - \frac{n_1(n_1 + 1)(n_1 + 2)}{6(\rho_0 - e)^3}\, \eta^3 + \ldots \right\} d\eta$$

$$= \frac{1}{(\rho_0 - e)^{n_1}} \left\{ \frac{h_1^{\,n_1+1}}{n_1 + 1} - \frac{n_1}{\rho_0 - e} \times \frac{h_1^{\,n_1+2}}{n_1 + 2} + \frac{n_1(n_1 + 1)}{2(\rho_0 - e)^2} \times \frac{h_1^{\,n_1+3}}{n_1 + 3} \right.$$

$$\left. - \frac{n_1(n_1 + 1)(n_1 + 2)}{6(\rho_0 - e)^3} \times \frac{h_1^{\,n_1+4}}{n_1 + 4} + \ldots \right\}$$

In the same way,

$$Is_2 = \int_0^{h_2} \frac{\eta^{n_2}}{(\rho_o - e - \eta)^{n_2}} d\eta = \frac{1}{(\rho_o - e)^{n_2}} \left\{ \frac{h_2^{n_2+1}}{n_2 + 1} + \frac{n_2}{(\rho_o - e)} \times \frac{h_2^{n_2+2}}{n_2 + 2} \right.$$

$$\left. + \frac{n_2(n_2 + 1)}{2(\rho_o - e)^2} \times \frac{h_2^{n_2+3}}{n_2 + 3} + \frac{n_2(n_2 + 1)(n_2 + 2)}{6(\rho_o - e)^3} \times \frac{h_2^{n_2+4}}{n_2 + 4} + \ldots \right\}$$

$$Im_1 = \int_0^{h_1} \frac{\eta^{n_1+1}}{(\rho_o - e + \eta)^{n_1}} d\eta = \frac{1}{(\rho_o - e)^{n_1}} \left\{ \frac{h_1^{n_1+2}}{n_1 + 2} - \frac{n_1}{(\rho_o - e)} \times \frac{h_1^{n_1+3}}{n_1 + 3} \right.$$

$$\left. + \frac{n_1(n_1 + 1)}{2(\rho_o - e)^2} \times \frac{h_1^{n_1+4}}{n_1 + 4} - \frac{n_1(n_1 + 1)(n_1 + 2)}{6(\rho_o - e)^3} \times \frac{h_1^{n_1+5}}{n_1 + 5} + \ldots \right\}$$

$$Im_2 = \int_0^{h_2} \frac{\eta^{n_2+1}}{(\rho_o - e - \eta)^{n_2}} d\eta = \frac{1}{(\rho_o - e)^{n_1}} \left\{ \frac{h_2^{n_2+2}}{n_2 + 2} + \frac{n_2}{(\rho_o - e)} \times \frac{h_1^{n_2+3}}{n_2 + 3} \right.$$

$$\left. + \frac{n_2(n_2 + 1)}{2(\rho_o - e)^2} \times \frac{h_2^{n_2+4}}{n_2 + 4} + \frac{n_2(n_2 + 1)(n_2 + 2)}{6(\rho_o - e)^3} \times \frac{h_2^{n_2+5}}{n_2 + 5} + \ldots \right\}$$

Substituting the values Is_1, Is_2, Im_1, and Im_2 in the formulas (22) and (24), there is obtained

$$\sigma_1 \left\{ \frac{\rho_o - e + h_1}{h_1} \right\}^{n_1} Is_1 - \sigma_2 \left\{ \frac{\rho_o - e - h_2}{h_2} \right\}^{n_2} Is_2 = 0$$

and

$$b\sigma_1 \left\{ \frac{\rho_o - e + h_1}{h_1} \right\}^{n_1} Im_1 + b\sigma_2 \left\{ \frac{\rho_o - e - h_2}{h_2} \right\}^{n_2} Im_2 = M$$

$$\therefore \quad \sigma_2 = \sigma_1^{\frac{n_2}{n_1}} \left\{ \frac{a_1 h_2 (\rho_0 - e + h_1)}{a_2 h_1 (\rho_0 - e - h_2)} \right\}^{n_2} \tag{24}$$

$$\therefore \quad \sigma_1 \left\{ \frac{\rho_0 - e + h_1}{h_1} \right\}^{n_1} I s_1 - \sigma_1^{\frac{n_2}{n_1}} \left\{ \frac{a_1 (\rho_0 - e + h_1)}{a_2 h_1} \right\}^{n_2} I s_2 = 0 \tag{25}$$

$$b\sigma_1 \left\{ \frac{\rho_0 - e + h_1}{h_1} \right\}^{n_1} I m_1 + b\sigma_1^{\frac{n_2}{n_1}} \left\{ \frac{a_1 (\rho_0 - e + h_1)}{a_2 h_1} \right\}^{n_2} I m_2 = M \tag{26}$$

From the formulas (24), (25), and (26) the following results are obtained

$$\sigma_1 = \left(\frac{a_2}{a_1} \right)^{\frac{n_1 n_2}{n_2 - n_1}} \left(\frac{h_1}{\rho_0 + h} \right)^{n_1} \left(\frac{I s_1}{I s_2} \right)^{\frac{n_1}{n_2 - n_1}} \tag{27}$$

and

$$M = b \left(\frac{a_2}{a_1} \right)^{\frac{n_1 n_2}{n_2 - n_1}} \left\{ \left(\frac{I s_1}{I s_2} \right)^{\frac{n_2}{n_2 - n_1}} I m_1 + \left(\frac{I s_1}{I s_2} \right)^{\frac{n_2}{n_2 - n_1}} I m_2 \right\} \tag{28}$$

The relation between the bending moment M and the value of e in the formula (26) can be obtained graphically as to the piston ring of given dimensions, by calculating the bending moment with respect to each assumed value of e; then from this graphical relation the value of e corresponding to a given bending moment can easily be determined. Putting the determined value of e in the formulas (27) and (24), maximum bending stresses σ_1 and σ_2 can be obtained.

For the verification of the result obtained, if ρ_0 is infinite in the formula (27) it will just coincide with the formula obtained by Bach (reference 12, p. 277) in the case of a straight beam.

For a numerical example, the case where the ring No. 1" is put into the cylinder is taken, in which the bending moment at the side opposite the gap ends is 47.7 kilogram-centimeters calculated from the pressure distribution diagram in figure 23, the radius of curvature ρ_0 in the unstrained state 6.514 centimeters, thickness of the ring and the cross-sectional area 0.420 centimeter and 0.333 square centimeter, respectively.

Assuming

$$a_1 = \frac{1}{24956000000}, \quad a_2 = \frac{1}{20000000}, \quad m_1 = 2.623, \quad \text{and} \quad m_2 = 1.491$$

as given by Bach, the result of calculation by the formulas (27) and (28) gives $\sigma_1 = 1200$ kilograms per square centimeter, approximately. (In the above calculation the effect of the normal compressive force on the cross section is neglected as it is very small.)

This value of σ_1 still may be too large for working conditions, but is very small compared with the value of σ_1, 1997 kilograms per square centimeter, calculated by the ordinary formula for curved beams. If a_1, a_2, m_1, and m_2 were known for the ring material, the value of maximum stress would be estimated more accurately.

XII. RÉSUMÉ

~~From~~ what has been described in the preceding pages, the author believes that the following facts concerning the characters of piston rings have been brought into light by him.

1. Mechanical devices were at first designed for measuring the pressure intensity of the piston ring upon the cylinder wall, but they only gave unsatisfactory results.

2. Next, a specially designed apparatus was introduced to measure the pressure intensity of the ring to which piezo-electricity was applied, and the absolute value of the pressure intensity distributed on the cylinder wall was measured accurately by this testing apparatus.

3. This new testing apparatus was so improved as to be easily used in practice - that is, the piezo-electric crystals were closed in a cast-iron box connected with the desiccator shielded from the electro-magnetic induction and moisture, a piston ring to be examined being able to be moved up and down within the cylindrical barrel in the same state as in the actual piston groove.

4. The pressure distribution in many sample rings on the cylinder wall is found to be very irregular, especially so with the rings manufactured by simple and common methods.

5. The mean radial pressure of the mean spring power of sample rings is less than 1.0 kilogram per square centimeter.

6. A new simple manufacturing process of piston rings which exert nearly uniform pressure on the cylinder wall has been discovered, and there is now established a practical possibility of obtaining a piston ring of uniform pressure.

7. A handy method for examining the quality of piston rings by means of a dial gage tester was devised by which piston rings can easily be compared.

8. By annealing the piston rings below 600° C under the working state, it was ascertained that the configuration of the pressure distribution is not much altered, while the mean pressure is considerably reduced, say, 50 percent at 500° C, and if annealing temperature is below 300° C, both the configuration and the mean pressure are not much different from their initial state.

9. The pressure distribution of a piston ring becomes better after being used in an engine for a long period of time, even though it was irregular in the virgin ring.

10. The effects of the pressure distribution, the mean spring power and the number of rings on the packing action were examined, and it may surely be said that only an exact fitting of all peripheral surface of the ring to the cylinder wall is the necessary and sufficient condition for the gastightness, the strong contact pressure or a large number of rings being not so effective, as far as the experiments are concerned.

11. The frictional resistance of the piston ring in the engine cylinder is fairly high - being about 60 percent of the total frictional resistance except pumping and valve-gearing resistances. This large amount of piston friction is diminished considerably by reducing the number of rings and also their mean spring power.

12. Stress calculation shows that the maximum skin stress of the piston ring under working state is very large, and the maximum stresses found out by the ordinary formulas for a curved beam which hold within the limit of elasticity are only apparent values.

13. Mathematical investigation for the calculation of the true maximum stress for a curved beam of nonelastic substance, the relation between stress σ and strain ϵ being expressed by $\epsilon = a\sigma^m$, was carried out and the formula applicable to the estimation of the most probable maximum stress of the cast-iron ring was found by the author.

In conclusion, the author wishes to express his cordial thanks to Dr. M. Ōkōchi, for his suggestions and constant guidance while the present investigations were being carried out.

REFERENCES

1. Nakagawa, K.: Proc. Soc. Nav. Arch., vol. 14, 1914.

2. Roux, E.: Einrichtung zum Messen des Dichtungsdruckes federnder Dichtungsringe und der Verteilung dieses Druckes längs des Umfanges. Patent 417714 Kl. 42 k Gr. 24.

3. Prescott, John: Applied Elasticity. Longmans, Green and Co. (London), 1924, p. 299.

   Devillers, Rene: Automobile and Aero-Engines. Spon & Chamberlain, New York, N. Y., 1919, pp. 157-159.

4. Aichi, K.: Jour. Soc. Mech. Eng. (Tokyo), vol. 25, 1922, no. 74, pp. 51-59.

5. Bennet, R.: VDI, vol. 68, 1924, pp. 253-254.

6. Inokuty, A.: Jour. Soc. Mech. Eng. (Tokyo), vol. 24, 1921, pp. 24-30.

7. Riehm, W.: VDI, vol. 65, 1921, pp. 923-925.

Asakawa, G., Ishizuka, S., and Hashimoto, U.: Jour. Soc. Mech. Eng., vol. 24, 1921, pp. 7-16 and 21-44.

Suhara, T. and Satō, N.: Jour. Soc. of Mech. Eng., vol. 25, 1922, pp. 35-70.

Sulzer: VDI., vol. 70, 1926, pp. 429-436.

Gibson, A. H.: Inst. of Mech. Eng., 1926, pp. 221-249.

8. Ōkōchi, M., and Ebihara, K.: Sci. Pap. I.P.C.R., vol. 6, 1927, p. 76.

Ōkōchi, M., and Ebihara, K.: Jour. Soc. of Mech. Eng. (Tokyo), vol. 30, 1927, p. 217.

9. Turner, A.: Engineering, vol. CXXV, 1928, p. 462.

Ōkōchi, M., and Ebihara, K.: Jour. Soc. of Mech. Eng. (Tokyo), vol. 30, 1927, pp. 222-223.

10. Judge, Arthur W.; Automobile and Aircraft Engines. Sir Isaac Pitman & Sons (London), p. 363.

Judge, Arthur W.: Auto. Eng., 1925, pp. 48-50.

Ricardo, Harry R.: The Internal Combustion Engine, I (London), 1922, pp. 58-59.

11. Bach: Elastizität und Festigkeit, 9th ed., p. 292.

12. Bach-Baumann: Elastizität und Festigkeit, 9th ed., p. 509.

13. Ono, Kansei: Zairyo-Ryokugaku (Strength of Material, in Japanese), p. 228.

14. Hütte: vol. II, 25th ed., p. 72.

NOTATIONS IN THE FOLLOWING TABLES.

θ.....Angular distance measured from the gap ends of the piston ring in degrees.

d.....Deflection of needle of the electrometer in cm. in the scale of microscope.

F.....Total force exerted on the piezo-electric pressure gauge by the piston ring in kg.

p.....Intensity of radial pressure in kg./cm^2.

γ.....Ratio of the maximum and minimum values of p.

TABLES

	III (Fig. 10)			IV (Fig. 11)			V (Fig. 12)		
$\theta°$	d(cm.)	F(kg.)	p(kg./cm^2.)	d(cm.)	F(kg.)	p(kg./cm^2.)	d(cm.)	F(kg.)	p(kg./cm^2.)
0	4·27	4·48	1·84	4·21	4·56	1·87	6·50	6·63	2·71
20	3·15	3·27	1·34	1·70	1·78	0·73	2·53	2·60	1·07
40	1·47	1·48	0·61	1·43	1·49	0·61	0·78	0·77	0·32
60	0·58	0·58	0·24	2·01	2·13	0·87	0·65	0·64	0·26
80	0·30	0·30	0·12	1·50	1·57	0·64	1·50	1·50	0 62
100	3·06	3·18	1·30	1·40	1·46	0·60	2·50	2·56	1·05
120	4·40	4·62	1·89	4 70	5·10	2·09	6·20	6·32	2·59
140	2·33	2·38	0·98	3·20	3·44	1·41	2·61	2·68	1·10
160	1·54	1·56	0·64	0·71	0·73	0·30	0·71	0·70	0·29
180	2·74	2·82	1·15	0·48	0·49	0·20	0·22	0·21	0·09
200	2·83	2·93	1·20	1·35	1·40	0·57	0·68	0·68	0·28
220	2·58	2·66	1·09	3·32	3·57	1·46	3·16	3·25	1·33
240	3·09	3·20	1·31	4·07	4·39	1·78	6·15	6 27	2·57
260	3·00	3·12	1·28	2·57	2·74	1·12	1·90	1·92	0·79
280	0·90	0·90	0·37	2·23	2 37	0·97	0·53	0·52	0·21
300	0·59	0·59	0·24	1·12	1·18	0·48	0·88	0·87	0·36
320	1·52	1·54	0·63	2·10	2·22	0·87	1·50	1·50	0·62
340	2·60	2·67	1·09	2·52	2·68	1·10	3·00	3·08	1·26
			$p_{mn}=0·964$			$p_{mn}=0·980$			$p_{mn}=0·974$
	Ring No. 1, D=12·70 cm.			Ring No. 2, D=12·70 cm.			Ring No. 3, D=12·70 cm.		

TABLES

θ°	VI (Fig. 13) d(cm.)	F(kg.)	(pkg./cm².)	VII (Fig. 14) d(cm.)	F(kg.)	p(kg./cm².)	VIII (Fig. 16) d(cm.)	F(kg.)	p(kg./cm².)
0	4·98	5·92	3·00	1·32	3·84	1·57	5·82	13·2	5·40
20	0·02	0·02	0·00	0	0	0	0	0	0
40	0·15	0·15	0·08	0	0	0	0	0	0
60	1·25	1·39	0·71	0·42	1·24	0·51	0	0	0
80	2·02	2·30	1·17	3·14	9·23	3·78	0	0	0
100	1·18	1·13	0·57	0·36	1·06	0·48	0·08	0·26	0·11
120	1·73	1·96	1·00	0	0	0	1·77	4·08	1·67
140	1·86	2·12	1·08	0	0	0	0·29	0·56	0·28
160	0·87	0·94	0·48	0 53	1·58	0·65	0·45	0·94	0·89
180	1·22	1·35	0·68	0·90	2·66	1·09	3·19	7·21	2·95
200	3 29	3·98	2·02	1·35	3·95	1·62	1·93	4·46	1·83
220	1·35	1·50	0·76	1 20	3 47	1·42	0·59	1·28	0·52
240	1·70	1·92	0·98	0·44	1·28	0·52	0·80	1·80	0·74
260	1·77	2·01	1·02	0 84	2·48	1·02	0·93	2·10	0·86
280	1·17	1·30	0·66	2·41	7·08	2·90	0·05	0·18	0·07
300	1·57	1·76	0·89	0·46	1·36	0·56	0	0	0
320	0·80	0·94	0·48	0	0	0	0	0	0
340	0	0	0	0	0	0	0	0	0
			p_{mn} = 0·864			p_{mn} = 0·892			p_{mn} = 0·820
	Ring No. 4, D = 12·70 cm.			Ring No. 5, D = 12·70 cm.			Ring No. 6, D = 12·70 cm.		

θ°	IX (Fig. 17) d(cm.)	F(kg.)	p(kg./cm².)	X (Fig. 18) d(cm.)	F(kg.)	½p(kg./cm².)	XI (Fig. 18) d(cm.)	F(kg.)	½p(kg./cm².)
0	1·64	1·67	0·74	2·20	2·07	0·85	2·39	2·67	1·09
20	2·03	2·10	0·86	1·00	0·90	0·37	1·30	1·40	0·57
40	2·19	2·27	0·93	0·85	0·77	0·31	0·65	0·67	0·27
60	2·19	2·27	0·93	1·12	1·02	0·42	0·95	0·99	0·41
80	2·22	2·31	0·95	2·05	1·92	0·79	1·51	1·62	0·66
100	2·12	2·20	0·90	1·64	1·52	0·62	1·70	1 85	0·76
120	2·35	2·46	1·01	1·60	1·49	0·61	2·20	2·44	1·00
140	1·78	1·82	0·75	0·91	0·83	0·34	1·17	1·24	0·51
160	1·84	1·88	0·77	0·31	0·28	0·12	0·05	0·05	0·02
180	2·47	2·58	1·06	0·99	0·89	0·36	0·05	0·05	0·02
200	2·52	2·65	1·08	2·43	2·29	0·94	0·65	0·67	0·27
220	2·91	3·08	1·26	2·19	2·06	0·85	2·47	2·76	1·13
240	1·94	1·98	0·81	1·85	1·24	0·51	2·05	2·26	0·93
260	1·83	1·87	0·77	1·09	0·99	0·40	1·18	1·26	0·52
280	2·11	2·20	0·90	0·78	0·70	0·29	0·78	0·81	0·33
300	2·33	2·44	1·00	0·65	0·59	0·24	0·66	0·68	0·28
320	2·50	2·63	1·08	1·30	1·20	0·49	0·89	0·92	0·38
340	1·77	1·80	0·74	1·65	1·53	0·63	1·53	1·66	0·68
			p_{mn} = 0·922			½p_{mn} = 0·507			½p_{mn} = 0·502
	Ring No. 7, D = 12·70 cm.			Ring No. 8, D = 12·70 cm.			Ring No. 9, D = 12·70 cm.		

TABLES

	XII (Fig. 18)			XIII (Fig. 19)			XIV (Fig. 20)		
θ°	d(cm.)	F(kg.)	p(kg./cm².)	d(cm.)	F(kg.)	p(kg./cm².)	d(cm.)	F(kg.)	p(kg./cm².)
0	2·28	2·54	1·04	4·23	2·17	2·11	1·10	0·306	0·35
20	2·27	2·53	1·03	0·40	0·25	0·20	1·15	0·320	0·36
40	2·90	3·32	1·36	1·22	0·63	0·61	0·70	0·194	0·22
60	2·27	2·53	1·03	2·12	1·09	1·06	0·98	0·272	0·31
80	2·12	2·34	0·96	1·33	0·68	0·66	1·80	0·500	0·57
100	2·17	2·40	0·98	1·58	0·81	0·79	1·60	0·445	0·50
120	2·55	2·88	1·18	2·65	1·36	1·32	1·00	0·278	0·31
140	2·55	2·88	1·18	2·33	1·19	1·16	0·66	0·183	0·21
160	2·65	3·00	1·23	1·69	0·87	0·84	0·75	0·208	0·24
180	2·33	2·60	1·06	1·00	0·51	0·50	2·00	0·555	0·64
200	2·13	2·36	0·97	0·65	0·33	0·32	0·70	0·194	0·22
220	2·03	2·24	0·92	0·95	0·49	0·47	1·08	0·300	0·34
240	2·00	2·20	0·90	1·03	0·53	0·51	0·80	0·222	0·25
260	3·28	3·80	1·55	3·00	1·54	1·50	0·80	0·222	0·25
280	2·63	2·97	1·22	2·85	1·46	1·42	1·40	0·389	0·44
300	1·89	2·06	0·84	2·55	1·31	1·27	1·35	0·375	0·42
320	2·05	2·26	0·93	0·54	0·28	0·27	1·90	0·528	0·59
340	1·03	2·12	0·87	0·16	0·08	0·08	0·90	0·250	0·28
			p_{mn}=1·05			p_{mn}=0·840			p_{mn}=0·360

Ring No. 10, D=12·70 cm. Ring No. 11, D=8·89 cm. Ring No. 12, D=15·0 cm.

TABLE XV. (Fig. 21)

θ°	d (cm.)	F (kg.)	p (kg./cm².)
0	2·70	11·2	0·77
15	3·16	13·1	0·90
30	2·04	8·45	0·58
45	0·56	2·32	0·16
60	0·05	0·21	0·01
75	0	0	0
90	0	0	0
105	0	0	0
120	0·30	1·25	0·09
135	1·28	5·31	0·36
150	2·05	8·50	0·58
165	2·50	10·4	0·71
180	3·40	14·1	0·97
195	3·12	13·0	0·89
210	1·25	5·8	0·36
225	0·15	0·62	0·04
240	0	0	0
255	0	0	0
270	0	0	0
285	0	0	0
300	0·40	1·66	0·11
315	1·50	6·22	0·43
330	2·40	9·95	0·68
345	2·70	11·2	0·77
			p_{mn}=0·351

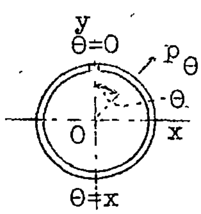
Ring No. 13 D=58·9 cm.

TABLE XVI.- COMPARISON OF BENDING-MOMENT DISTRIBUTIONS

θ	$\left\{\frac{M_\theta}{M_\pi}\right\}_i = \frac{1 - \cos\theta}{1 - \cos\pi}$	$\frac{M_\theta}{M_\pi} = \frac{\left\{t^3\left(\frac{1}{R} - \frac{1}{\rho}\right)\right\}_\theta}{\left\{t^3\left(\frac{1}{R} - \frac{1}{\rho}\right)\right\}_\pi}$		$\frac{M_\theta}{M_\pi}$
	Ideal case	No. 7" (fig. 25a)	No. 1" (fig. 25b)	No. 1" (fig. 25c)
0	0	----------	----------	0
10	.0076	0.01	0.01	----------
20	.030	.04	.07	.06
40	.117	.12	.30	.20
60	.250	.28	.38	.37
80	.413	.42	.52	.51
100	.588	.59	.52	.64
120	.750	.75	.56	.69
140	.883	.82	.73	.86
160	.970	.90	.83	.92
180	1.00	1.00	1.00	1.00
200	.970	1.02	.93	1.00
220	.883	.82	.96	.87
240	.750	.81	.79	.74
260	.588	.58	.71	.60
280	.413	.49	.56	.48
300	.250	.34	.49	.34
320	.117	.16	.31	.19
340	.030	.08	.12	.05
350	.0076	.02	.01	----------
360	0	----------	----------	0

TABLE XVII

Number of rings	p_x (kg)	p_y (kg)	$p_r = \sqrt{p_x^2 + p_y^2}$ (kg)	$\frac{p_r}{P}$ (percent)
1"	-0.054	-0.809	0.811	3.0
7"	+.108	-.378	.393	1.6
12	+.056	+.089	.104	1.8



$$p_x = bR\int_0^{2\pi} p_\theta \sin\theta \, d\theta \qquad P = bR\int_0^{2\pi} p_\theta \, d\theta$$

$$p_y = bR\int_0^{2\pi} p_\theta \cos\theta \, d\theta \qquad \doteq 2\pi \, Rbp_{mn}$$

TABLE XVIII

No. of rings	D (cm)	p_{mn}	p_{mno}	$\frac{p_{mno}}{p_{mn}}$
1"	12.7	0.853	0.786	0.92
7"	12.7	.756	.721	.95
12	15.00	.360	.335	.94

p_{mn} mean pressure intensity or mean spring power measured by the ring tester

p_{mno} mean spring power measured by the ordinary method in which the ring is compressed by winding a thin band about the ring surface

TABLE XX

No. of rings	Heat treatment
1	Ring same as No. 1 in table I.
1'	Ring No. 1 annealed at 300° C for 2 hours.
1"	Ring No. 1' annealed at 300° C for 8 hours.
2	Ring same as No. 2 in table I.
2'	Ring No. 2 annealed at 200° C for 3 hours.
2"	Ring No. 2' annealed at 300° C for 2 hours.
2‴	Ring No. 2" annealed at 400° C for 2 hours.
2""	Ring No. 2‴ annealed at 550° C for 2 hours.
7	Ring same as No. 7 in table I.
7'	Ring No. 7 annealed at 300° C for 2 hours.
7"	Ring No. 7' annealed at 300° C for 8 hours.

TABLE XXIX.- SPECIFICATION OF SAMPLE RINGS NOS. 34 to 43

No. of rings	D (cm)	b (cm)	t (cm)	p_{mno} (kg/cm^2)	Remarks	Manufacturer
34	8.89	0.477	0.296	0.57		(I.P.C.R.)
35	8.89	.474	.336	1.04	Left as cast	(P.R. Co.)
36	8.89	.634	.320	.87	Hammered	(Brico)
37	8.89	.574	.352	1.02	Hammered	----------
38	8.89	.474	.335	1.04	Left as cast	(P.R. Co.)
39	8.89	.474	.352	1.02	Hammered	----------
40	8.89	.474	.316	.89	Left as cast	(P.R. Co.)
41	8.89	.634	.322	.88	Hammered	(Brico)
42	8.89	.477	.293	.56		(I.P.C.R.)
43	8.89	.474	.256	.32		----------

(Specimens are all virgin rings of cast iron.)

TABLE XIX.- SPECIFICATION OF SAMPLE RINGS NOS. 14 to 32

No. of rings	Diam. of cylinder D (cm)	Thickness (mean) t (cm)	Breadth b (cm)	Gap length l (cm)	Form of gap-ends	Remarks	Manufac-turer
14	12.70	0.392	0.793	1.65	Stepped	Same as No. 7	I.P.C.R. (Japan)
15	12.70	.406	.793	1.74	---do---	Same as No. 1	P.R. Co. (U.S.A.)
16	12.70	.435	.793	1.28	---do---	Hammered	A.H.P.R. Co. (U.S.A.)
17	12.70	.460	.798	1.46	Diagonal (45°)	Same as No. 6	
18	8.89	.320	.473	.82	Stepped		I.P.C.R. (Japan)
19	8.89	.351	.473	.80	---do---	Hammered	A.H.P.R. Co. (U.S.A.)
20	8.89	.295	.473	1.10	Diagonal (45°)		McQ.-Nor. Co. (U.S.A.)
21	8.50	.335	.497	.70	Stepped		I.P.C.R. (Japan)
22	12.0	.407	.197	.85	Diagonal (30°)		Do.
23	8.89	.336	.473	.98	Stepped		P.R. Co. (U.S.A.)
24	8.57	{.230 .361	.468	.53	---do---	Eccentric type	Buick (U.S.A.)
25	9.50	{.271 .349	.623	.86	Diagonal (45°)		Ford (U.S.A.)
26	10.15		.628	1.50	---do---	Specially constructed	Reus Co. (U.S.A.)
27	12.0	.410	.198	1.10	---do---	Hammered	Lorraine (France)
28	12.0	.365	.596	1.45	Stepped	---do---	Benz (German)
29	12.70	.437	.794	1.25	Diagonal (45°)	---do---	Brico (England)
30	14.0	.417	.497	1.96	---do---	---do---	Daimler (German)
31	13.8	.345	.697	1.77	Stepped	---do---	N.A.G. (German)
32	9.03	.345	.625	1.30	Diagonal (30°)	Scratched	Woseley (England)

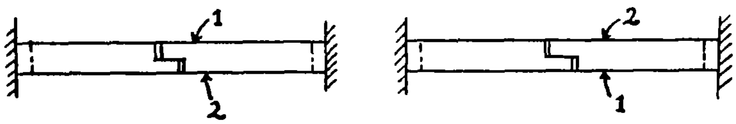
(Specimens are all virgin rings; except the I.P.C.R. rings, several pieces as obtained in the Japan market may not be representative of all others,

Tables

	XXI (Fig. 49)			XXII (Fig. 49)			XXIII (Fig. 49)		
θ°	d(cm.)	F(kg.)	p(kg./cm².)	d(cm.)	F(kg.)	p(kg./cm².)	d(cm.)	F(kg.)	p(kg./cm².)
0	3·40	3·59	1·47	3·28	3·44	1·41	2·68	2·75	1·13
20	1·70	1·73	0·71	1·88	1·94	0·80	1·83	1·84	0·76
40	1·31	1·33	0·55	1·23	1·26	0·52	1·33	1·32	0·54
60	1·86	1·90	0·78	1·70	1·75	0·72	1·19	1·18	0·48
80	1·78	1·81	0·74	1·70	1·75	0·72	1·24	1·22	0·50
100	1·51	1·54	0·63	1·72	1·76	0·72	1·53	1·53	0·63
120	4·49	4·76	1·95	4·10	4·32	1·77	3·71	3·82	1·57
140	3·17	3·30	1·35	2·90	3·04	1·25	2·93	3·00	1·23
160	0·41	0·41	0·17	0·40	0·40	0·16	0·40	0·40	0·16
180	0·38	0·38	0·16	0·33	0·33	0·14	0·34	0·34	0·14
200	0·75	0·75	0·31	0·90	0·91	0·37	0·53	0·53	0·22
220	3·44	3·62	1·48	3·30	3·46	1·42	2·87	2·94	1·21
240	3·54	3·74	1·53	3·43	3·60	1·48	2·85	2·93	1·20
260	2·65	2·76	1·13	2·40	2·49	1·02	2·01	2·04	0·84
280	2·10	2·16	0·89	1·45	1·48	0·61	1·36	1·36	0·56
300	1·27	1·28	0·52	1·20	1·22	0·50	1·23	1·22	0·50
320	1·70	1·73	0·71	1·70	1·75	0·72	1·80	1·80	0·73
340	2·14	2·20	0·90	2·25	2·36	0·97	2·20	2·23	0·92
			$p_{mn}=0·888$			$p_{mn}=0·847$			$p_{mn}=0·740$
	Ring No. 2′			Ring No. 2″			Ring No. 2‴		

	XXIV (Fig. 49)			XXV (Fig. 50)			XXVI (Fig. 22)		
θ°	d(cm.)	F(kg.)	p(kg./cm².)	d(cm.)	F(kg.)	p(kg./cm².)	d(cm.)	F(kg.)	p(kg./cm².)
0	1·30	1·30	0·53	4·00	4·20	1·72	3·68	3·61	1·48
20	0·80	0·80	0·33	3·00	3·12	1·28	3·20	3·10	1·27
40	0·73	0·73	0·30	1·18	1·19	0·49	2·05	1·94	0·80
60	0·60	0·60	0·25	0·30	0·30	0·12	0·54	0·51	0·21
80	0·70	0·70	0·29	0·40	0·40	0·16	0·47	0·42	0·17
100	0·75	0·75	0·31	2·98	3·10	1·27	1·70	1·60	0·66
120	1·68	1·68	0·69	3·70	3·87	1·59	4·51	4·52	1·85
140	1·29	1·29	0·53	2·24	2·28	0·89	2·50	2·38	0·98
160	0·80	0·80	0·33	1·96	2·00	0·82	2·14	2·02	0·83
180	0·33	0·33	0·14	1·51	1·53	0·63	2·15	2·04	0·84
200	0·30	0·30	0·12	2·58	2·66	1·09	2·70	2·58	1·06
220	1·33	1·33	0·55	2·68	2·77	1·13	2·40	2·29	0·94
240	1·30	1·30	0·53	2·80	2·90	1·19	2·70	2·58	1·06
260	0·42	0·42	0·17	2·63	2·71	1·11	3·15	3·05	1·25
280	0·72	0·72	0·29	1·00	1·00	0·41	1·06	0·98	0·40
300	0·60	0·62	0·25	0·85	0·85	0·35	0·83	0·75	0·31
320	1·10	1·10	0·45	1·23	1·23	0·50	1·42	1·30	0·53
340	0·80	0·80	0·33	1·99	2·02	0·83	2·15	2·04	0·84
			$p_{mn}=0·355$			$p_{mn}=0·865$			$p_{mn}=0·857$
	Ring No. 2⁗			Ring No. 1′			Ring No. 1″		

Values shown in Tables 26 and 26′ are the results of the measurements of the ring No. 1″ in the following states *a* and *b*;



a *b* (Reversed)

TABLES

	XXVI′ (Fig. 22)			XXVII (Fig. 51)			XXVIII (Fig. 23 & 51)		
θ°	*d*(cm.)	*F*(kg.)	*p*(kg./cm².)	*d*(cm.)	*F*(kg.)	*p*(kg./cm².)	*d*(cm.)	*F*(kg.)	*p*(kg./cm².)
0	3·50	3·54	1·45	1·30	1·36	0·61	1·40	1·22	0·54
20	2·97	2·93	1·20	1·75	1·86	0·76	2·01	1·78	0·73
40	2·18	2·14	0·88	1·86	1·98	0·81	2·15	1·91	0·78
60	0·35	0·35	0·14	1·84	1·96	0·80	2·28	2·03	0·83
80	0·35	0·35	0·14	2·01	2·16	0·89	2·24	1·99	0·82
100	2·18	2·14	0·88	1·95	1·95	0·80	2·30	2·05	0·84
120	4·00	4·10	1·68	1·68	1·78	0·73	2·00	1·78	0·73
140	2·20	2·16	0·89	1·44	1·51	0·62	1·80	1·59	0·65
160	2·05	2·01	0·82	1·66	1·76	0·72	1·75	1·55	0·64
180	2·04	2·00	0·82	1·83	1·95	0·80	2·28	2·03	0·83
200	2·43	2·42	0·99	2·04	2·19	0·90	2·63	2·36	0·97
220	2·23	2·19	0·90	2·16	2·34	0·96	2·58	2·32	0·95
240	2·64	2·61	1·07	1·45	1·53	0·63	1·96	1·74	0·71
260	2·85	2·83	1·16	1·43	1·50	0·61	1·64	1·44	0·59
280	1·08	1·05	0·43	1·65	1·75	0·72	1·91	1·67	0·69
300	0·88	0·86	0·35	2·20	2·37	0·97	2·45	2·18	0·90
320	1·32	1·29	0·53	2·20	2·25	0·88	2·55	2·28	0·94
340	2·30	2·24	0·93	1·51	1·60	0·66	1·48	1·29	0·53
			$p_{mn}=0{\cdot}848$			$p_{mn}=0{\cdot}773$			$p_{mn}=0{\cdot}756$

Ring No. 1″ Ring No. 7′ Ring No. 7″

TABLES

	XXX (Fig. 55)			XXXI (Fig. 56)			XXXII (Fig. 57)		
$\theta°$	d(cm.)	F(kg.)	p(kg./cm².)	d(cm.)	F(kg.)	p(kg./cm².)	d(cm.)	F(kg.)	p(kg./cm².)
0	1·30	1·31	0·54	2·70	2·76	1·13	2·65	2·51	1·03
20	3·00	3·17	1·30	4·01	4·18	1·71	2·97	2·83	1·16
40	2·15	2·22	0·91	2·10	2·12	0·87	2·18	2·12	0·87
60	1·10	1·10	0·45	1·40	1·39	0·57	1·80	1·67	0·68
80	1·31	1·32	0·54	1·50	1·51	0·62	2·14	1·99	0·81
100	2·30	2·38	0·98	3·48	3·61	1·48	3·00	2·86	1·17
120	3·75	4·00	1·64	4·40	4·50	1·84	3·63	3·49	1·43
140	3·68	3·92	1·61	2·43	2·46	1·01	3·20	3·05	1·25
160	2·33	2·42	0·99	0·94	0·92	0·38	3·04	2·90	1·19
180	1·65	1·68	0·67	0·75	0·74	0·30	2·49	2·34	0·96
200	1·53	1·57	0·64	1·95	1·97	0·81	1·80	1·67	0·68
220	2·35	2·43	0·99	4·75	4·86	1·99	2·38	2·24	0·92
240	3·27	3·46	1·42	4·10	4·20	1·72	2·61	2·56	1·05
260	2·00	2·06	0·84	1·48	1·47	0·60	2·57	2·43	1·00
280	1·30	1·31	0·54	0·90	0·89	0·36	2·48	2·33	0·96
300	1·10	1·10	0·45	1·30	1·29	0·53	1·60	1·47	0·60
320	3·10	3·29	1·35	3·00	3·09	1·27	2·43	2·28	0·93
340	1·97	2·02	0·83	3·50	3·60	1·47	3·78	3·64	1·49
			$p_{mn}=0·927$			$p_{mn}=1·03$			$p_{mn}=1·01$

Ring No. 44, D=12·70 cm. Ring No. 45, D=12·70 cm. Ring No. 46, D=12·70 cm.

TABLE XXXIII (fig. 58)

RELATION BETWEEN THE MAXIMUM PRESSURE AT THE END OF THE COMPRESSION STROKE AND THE NUMBER OF REVOLUTIONS OF ENGINE

No. of rings 3		No. of rings 4		No. of rings 7		Remark
rpm	Maximum pressure (kg/cm^2)	rpm	Maximum pressure (kg/cm^2)	rpm	Maximum pressure (kg/cm^2)	
140	30.2	140	29.9	141	30.2	Lubricant poor in all cases
190	32.8	182	32.4	203	32.8	
236	33.4	247	33.4	247	34.3	
267	34.6	261	34.3	292	34.5	
308	35.4	300	34.5	324	35.5	
333	35.6	324	35.6	364	36.2	
353	36.1	365	36.2	390	37.1	
400	36.9	393	36.9	429	37.6	
414	37.4	414	37.3	455	38.2	
428	37.7	444	37.7	490	38.8	
462	38.0	462	38.1	521	39.0	
500	38.3	490	38.4			
521	38.6	510	38.7			
		522	39.0			

TABLE XXXIV (fig. 59)

RELATION BETWEEN THE MAXIMUM PRESSURE AT THE END OF THE COMPRESSION STROKE AND THE INTENSITY OF PRESSURE OF THE RINGS

Kind of rings	p_{mn}* (kg/cm^2)	Maximum pressure (kg/cm^2)	rpm	Remark
a_1	0.948	25.9	130	No lubricant in all cases
	.912	26.0	130	
	.604	25.2	132	
	.257	25.4	134	
a_2	0.840	26.9	131	
	.768	26.7	132	
	.527	26.9	132	
b_1	0.619	25.8	132	
	.562	25.9	132	
	.413	25.8	132	
b_2	0.675	25.9	132	
	.604	25.9	132	
	.461	25.8	133	
	.210	25.8	135	

*The mean spring power p_{mn} of the rings is diminished by annealing within the cylinder.

TABLE XXXV (fig. 61)

RELATION BETWEEN THE MAXIMUM PRESSURE AT THE END OF THE COMPRESSION STROKE AND THE NUMBER OF RINGS

No. of rings	Maximum pressure (kg/cm^2)	rpm	Condition of oil	Remarks
0	8.9	134	none	Kind of rings is a_2 p_{mn} = 0.8 to 0.9 kg/cm^2
1	27.2	131	none	
2	27.2	130	none	
4	27.3	126	none	
6	27.2	120	none	
0	10.2	137	poor	
1	28.8	134	poor	
2	29.7	132	poor	
4	29.9	131	poor	
1	31.6	133	rich	
2	33.2	133	rich	
4	33.4	132	rich	

TABLE XXXVI

RETARDATION OF THE ANGULAR VELOCITY OF THE ENGINE AFTER THE DRIVING BELT HAS BEEN REMOVED

Time (sec)	ω (rad/sec) oil poor				ω (rad/sec) no oil	
	Ring 3	Ring 3	Ring 7	Ring 7	Ring 2	Ring 7
0	57.1	57.1	57.1	57.1	57.1	57.1
1	57.1	57.1	57.1	57.1	57.1	57.1
2	52.4	53.7	54.6	53.1	54.6	54.6
3	48.4	49.4	50.4	49.0	51.3	50.3
4	44.8	46.5	45.5	45.5	48.4	46.5
5	41.8	44.8	41.8	43.3	45.5	42.4
6	39.2	41.8	37.0	39.2	43.3	38.0
7	35.9	40.0	34.9	35.3	40.6	34.4
8	34.9	37.0	32.2	33.1	38.1	32.2
9	32.2	35.5	29 9	29.5	36.4	28.6
10	29.9	33.6	27.3	27.3	33.1	25.1
11	27.7	29.9	24.2	24.6	30.6	22.4
12	25.6	29.2	20.9	21.6	28.6	19.6
13	24.2	27.3	18.5	19.3	26.7	17.9
14	21.3	25.1	16.5	17.2	25.1	----
15	19.6	23.2	----	----	21.6	----
k	k_3 = 0.194		k_7 = 0.262		k_2 = 0.199	k_7 = 0.281

TABLE XXXVII

RETARDATION OF THE ANGULAR VELOCITY OF THE ENGINE AFTER

THE DRIVING BELT HAS BEEN REMOVED

Time (sec)	ω (rad/sec) oil rich			
	Ring 2	Ring 2	Ring 7	Ring 7
0	57.1	57.1	57.1	57.1
1	57.1	57.1	57.1	57.1
2	54.6	54.6	54.6	55.7
3	52.4	52.4	52.4	52.4
4	50.4	51.3	49.0	50.4
5	48.4	48.4	46.5	48.4
6	46.5	47.4	43.5	44.8
7	44.1	44.8	41.8	41.8
8	42.4	43.3	39.2	39.2
9	40.6	40.6	37.0	37.0
10	39.2	39.2	34.9	34.9
11	37.0	38.0	32.2	32.2
12	34.0	34.9	30.6	30.6
13	33.1	33.1	28.2	28.6
14	31.4	31.4	26.2	26.2
15	----	----	24.6	24.2
	$k_2 = 0.150$		$k_7 = 0.190$	

TABLE XXXIX

Value of k

Oil	No. of rings			Piston only (k_p)	Bearing only (k_b)	$\frac{k_7 - k_p}{k_7}$ (percent)	$\frac{k_7 - k_{3,2}}{k_7}$ (percent)
	7 (k_7)	3 (k_3)	2 (k_2)				
Rich	0.192	--------	0.150	--------	0.050	=	21.8
Poor	.262	0.194	--------	0.0974	-------	62.8	26.0
None	.281	=	.199	.0998	-------	64.4	29.2

TABLE XXXVIII

RETARDATION OF THE ANGULAR VELOCITY OF THE ENGINE AFTER

THE DRIVING BELT HAS BEEN REMOVED

Time (sec)	ω (rad/sec)		
	Bearing only	Without ring	
		Oil poor	No oil
0	57.1	57.1	57.1
2	57.1	57.1	57.1
4	54.6	52.4	54.6
6	53.1	49.0	50.6
8	52.4	46.5	48.6
10	50.4	44.8	44.8
12	48.4	41.8	41.8
14	47.4	39.2	39.6
16	46.5	37.0	38.0
18	44.8	34.9	35.5
20	43.3	32.2	32.2
22	42.4	30.6	29.7
24	40.6	27.3	28.0
26	39.2	25.3	25.3
28	39.2	23.7	24.2
30	37.0		
32	35.5		
34	34.9		
36	34.0		
38	33.1		
40	31.4		
	$k_b = 0.050$	$k_p = 0.0974$	$k_p = 0.0998$

TABLE XL

MAXIMUM TENSILE STRESS (APPARENT) OF THE PISTON RINGS UNDER WORKING CONDITIONS

No.	D (cm)	b (cm)	l (cm)	t_π (cm)	p_{mn} (kg/cm²)	$\sigma_{max\ t}$ (kg/cm²)
3	12.70	0.793	1.70	0.427	0.924	2372
4	12.70	.600	1.61	.443	.914	2178
7	12.70	.793	1.65	.427	.822	2110
11	8.89	.635	.86	.315	.850	1961
12	15.00	.340	1.80	.504	.360	925
13	58.90	1.900	3.25	1.795	.351	1100
19	8.89	.473	.80	.351	.974	1804
24	8.57	.468	.53	.361	.826	1340
25	9.50	.623	.86	.349	.810	1737
27	12.00	.198	1.10	.410	.598	1486
28	12.00	.596	1.45	.365	.520	1637
30	14.00	.497	1.96	.417	.565	1855
31	13.80	.697	1.77	.345	.317	1466
32	9.03	.625	1.30	.345	1.24	2454
33	14.60	.237	2.00	.507	.910	2188

t_π Thickness of the side just opposite the gap ends.

p_{mn} Mean spring power or mean pressure intensity measured by the ring tester in case of the rings Nos. 3 to 13 and by the ordinary method, the ring being compressed by a thin band, in case of the rings Nos. 19 to 32.

$\sigma_{max\ t}$ Maximum tensile stress calculated by the formula of curved beam,

$$\sigma_{max\ t} = \frac{3p_{mn}\ D^2}{t^2} \cdot \frac{1}{1 + \frac{t}{D}}$$

TABLE XLI (fig. 65)

SPECIFICATION OF SAMPLE RINGS NOS. 47 to 79

No.	D (cm)	t (cm)	b (cm)	l (cm)	Gap form	Remarks	Manufacturer
47	5.08	0.218	0.474	0.50	Stepped	Left as cast	P.R. Co.
48	6.50	.260	.396	.90	Diagonal (45°)	Hammered	Fiat
49	6.98	.269	.474	.95	Stepped		U.S.A.
50	7.32	.286	.312	.84	---do---	Left as cast	Chance Co.
51	7.62	.280	.474	.74	---do---	---do-------	P.R. Co.
52	7.62	.275	.472	.96	---do---	---do-------	----------
53	7.62	.290	.200	.62	Diagonal (45°)	--- -	----------
54	7.62	.268	.466	.84	Stepped	Left as cast	M. & H. Co.
55	7.62	.278	.366	.75	---do---	Hammered	----------
56	7.62	.289	.316	.66	Straight		Ikegai
57	8.00	.302	.497	.80	Stepped		----------
58	8.25	.303	.315	.75	---do---		----------
59	8.25	.310	.480	1.10	---do---	Hammered	U.S.A.
60	8.50	.320	.497	.82	---do---		----------
61	8.89	.360	.474	.62	---do---	Left as cast	U.S.A.
62	9.00	.294	.396	1.14	Diagonal (45°)	Hammered	German
63	9.00	.289	.745	.90	---do---		Tobata
64	9.26	.319	.475	.71	Stepped	Hammered	----------
65	9.40	.306	.474	.90	---do---	---do-----	U.S.A.
66	10.00	.282	.742	.82	Diagonal (45°)		Tobata
67	10.00	.327	.648	1.02	Straight		Ikegai
68	10.50	.357	.397	1.43	Diagonal (45°)	Hammered	German
69	11.40	.393	.373	1.45	Straight		Rolls-Royce
70	11.50	.350	.497	1.02	Stepped		----------
71	11.50	.356	.745	1.47	Diagonal (45°)		----------
72	12.40	.447	.151	1.20	Straight		Siemens
73	12.70	.524	.948	1.35	---do---	Hammered	Ikegai
74	12.70	.431	.793	1.60	Stepped	Left as cast	P.R. Co.
75	14.50	.408	.374	1.50	Diagonal (45°)		Siddeley
76	15.20	.490	.950	1.80	Stepped	Left as cast	P.R. Co.
77	15.20	.450	.591	2.20	Diagonal	Hammered	Niigata
78	16.00	.604	.995	1.95	Stepped		----------

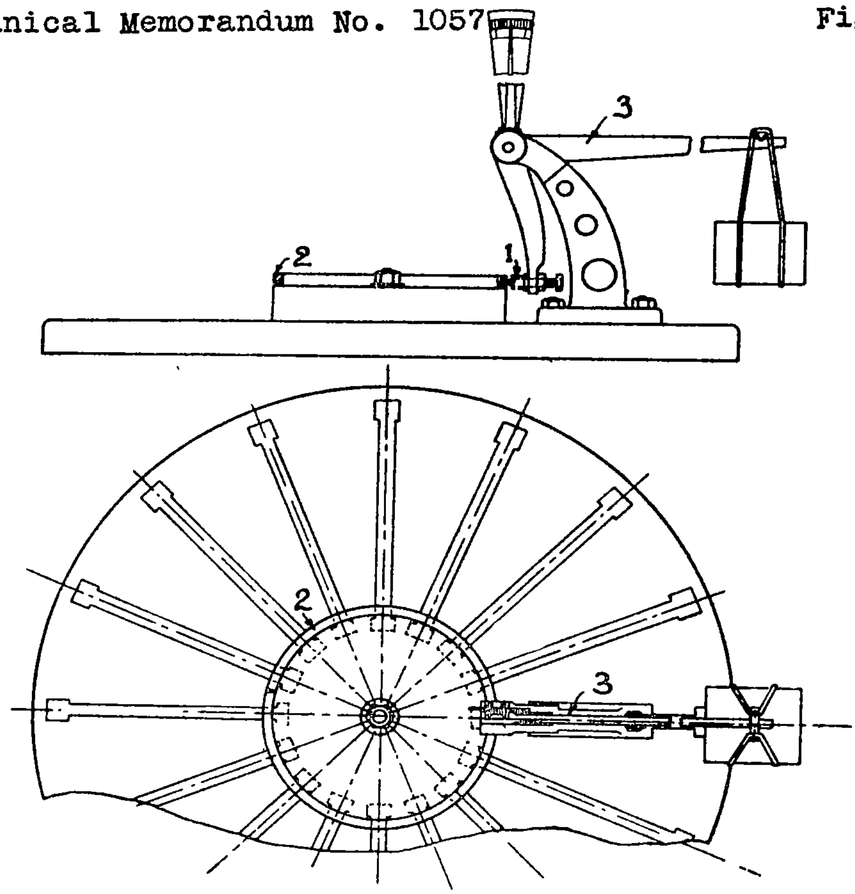


Fig. 1. (Mechanical device)

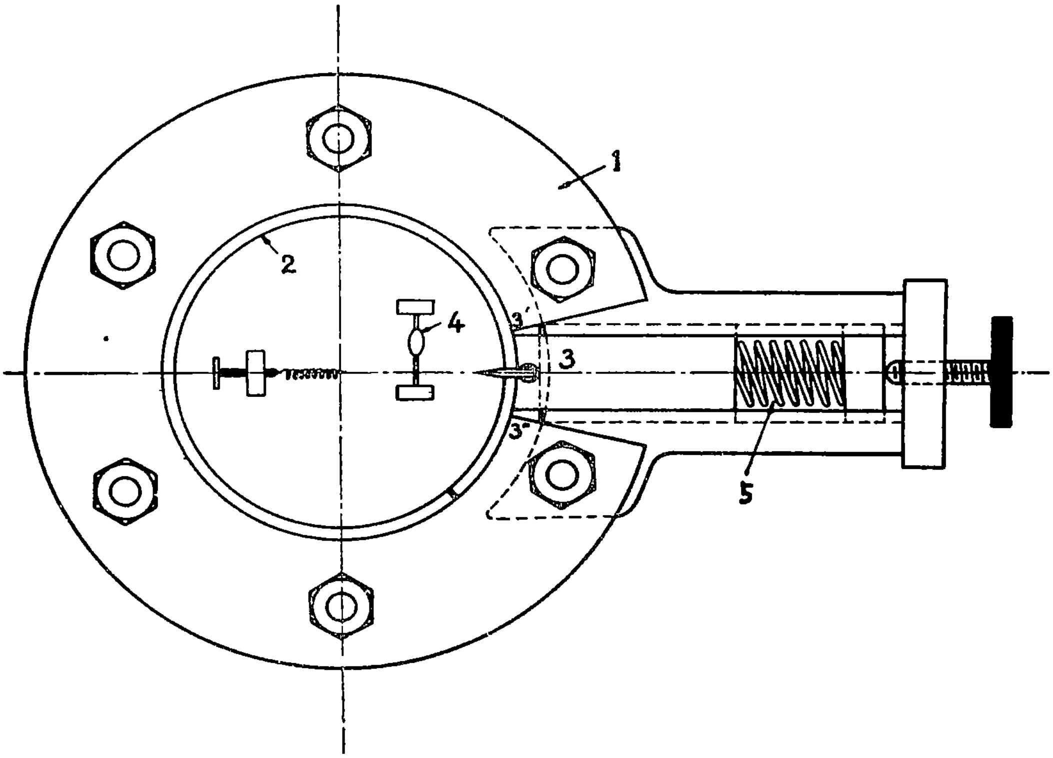


Fig. 2. (Mechanical device)

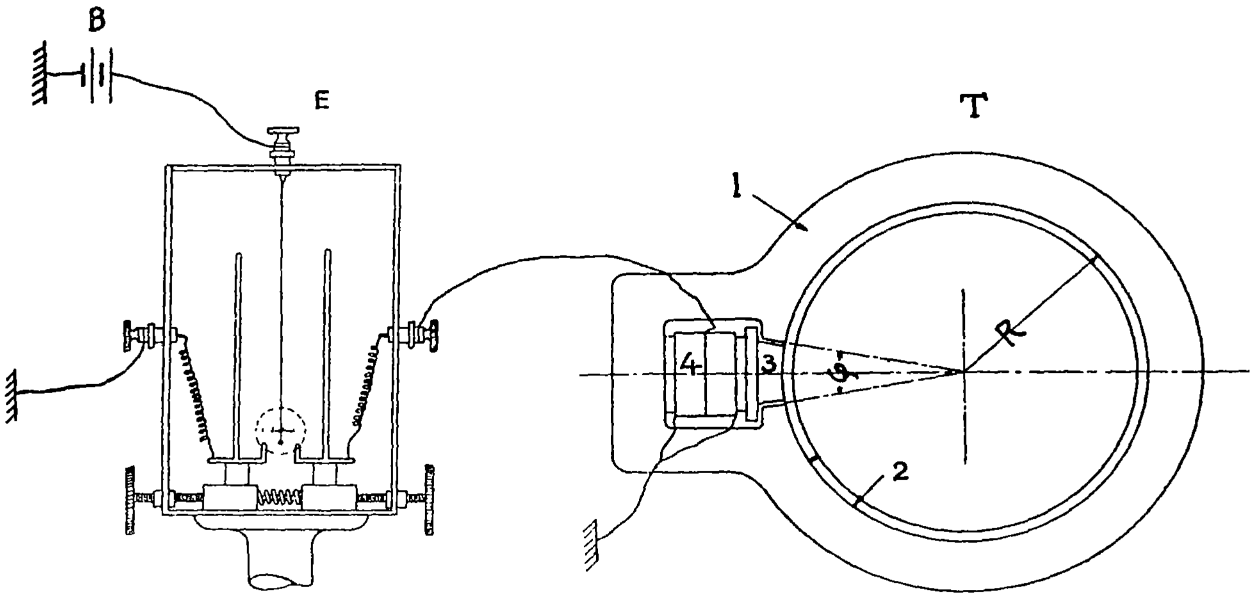


Fig. 3. (Piezo-electric device)

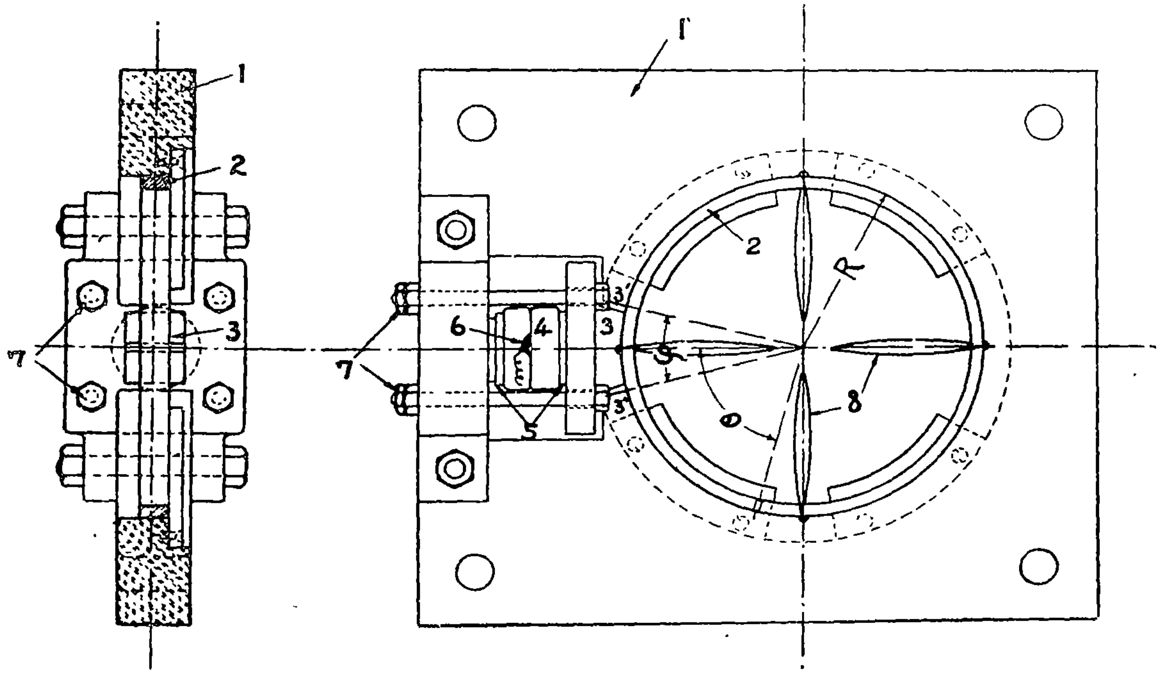


Fig. 4. (Ring Tester of Type I)

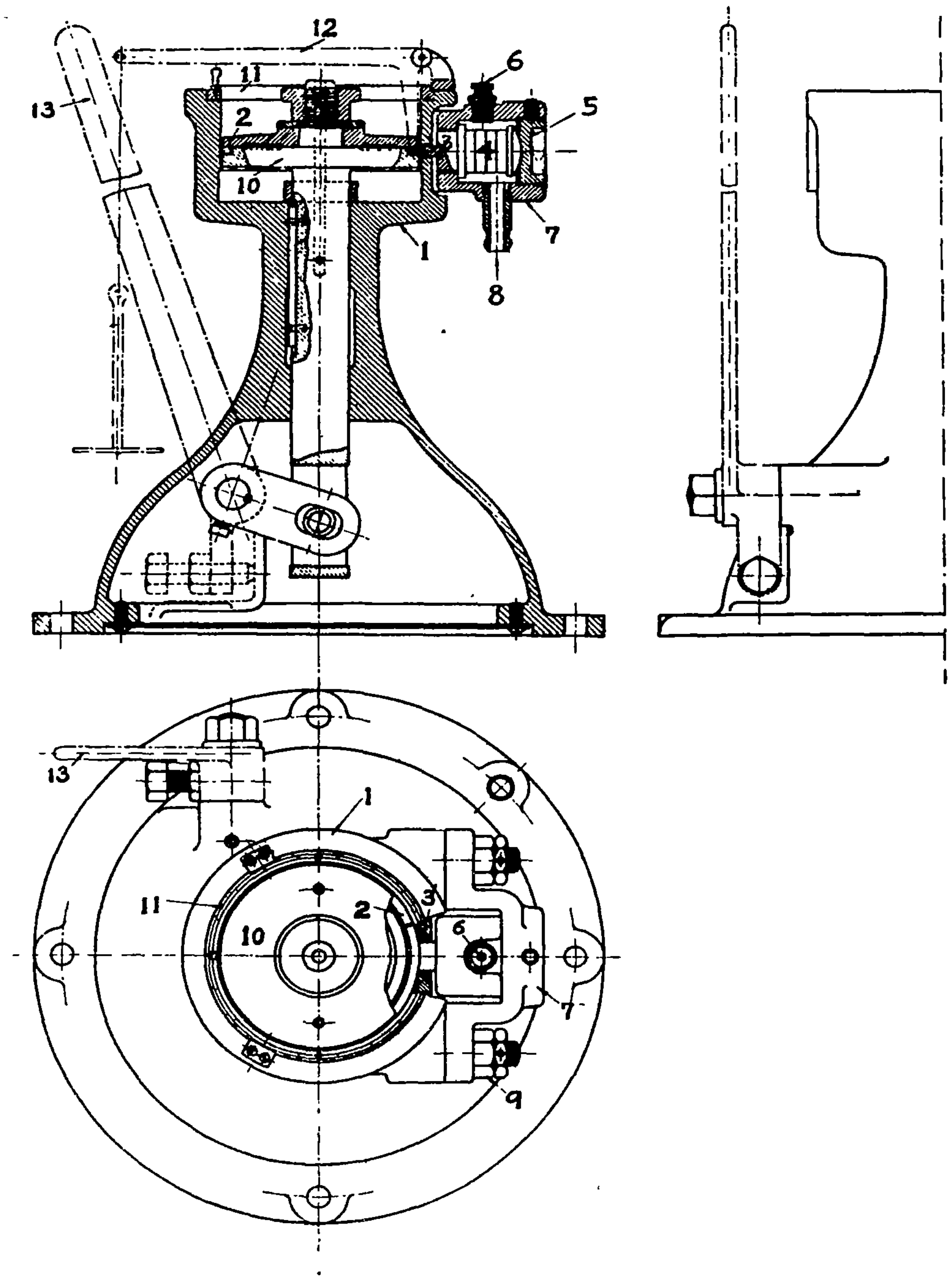


Fig. 5. (Ring Tester of Type II)

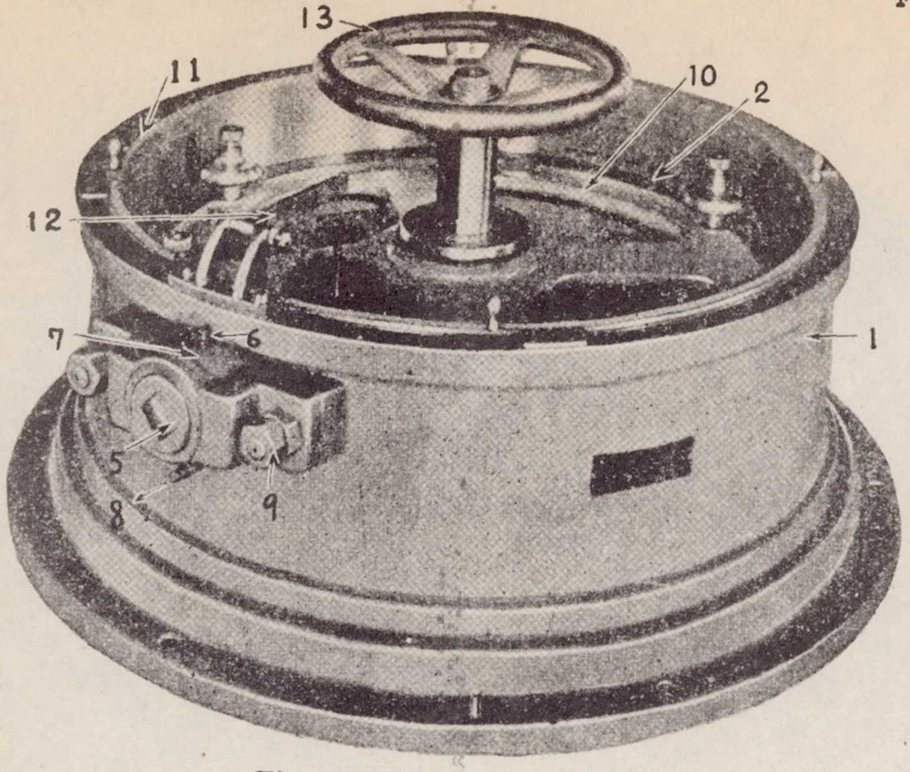


Fig. 6. (Dia. 250~600 mm.)
(Type II)

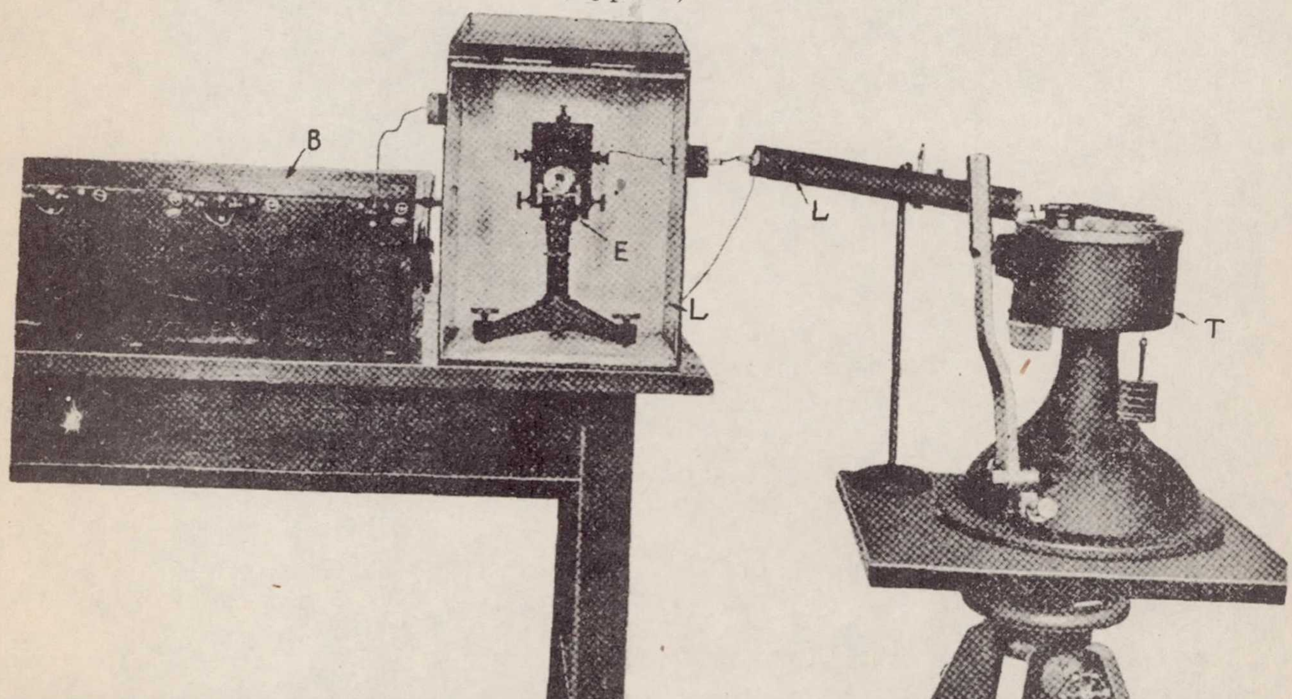


Fig. 8. (Dia. 70~250 mm.)

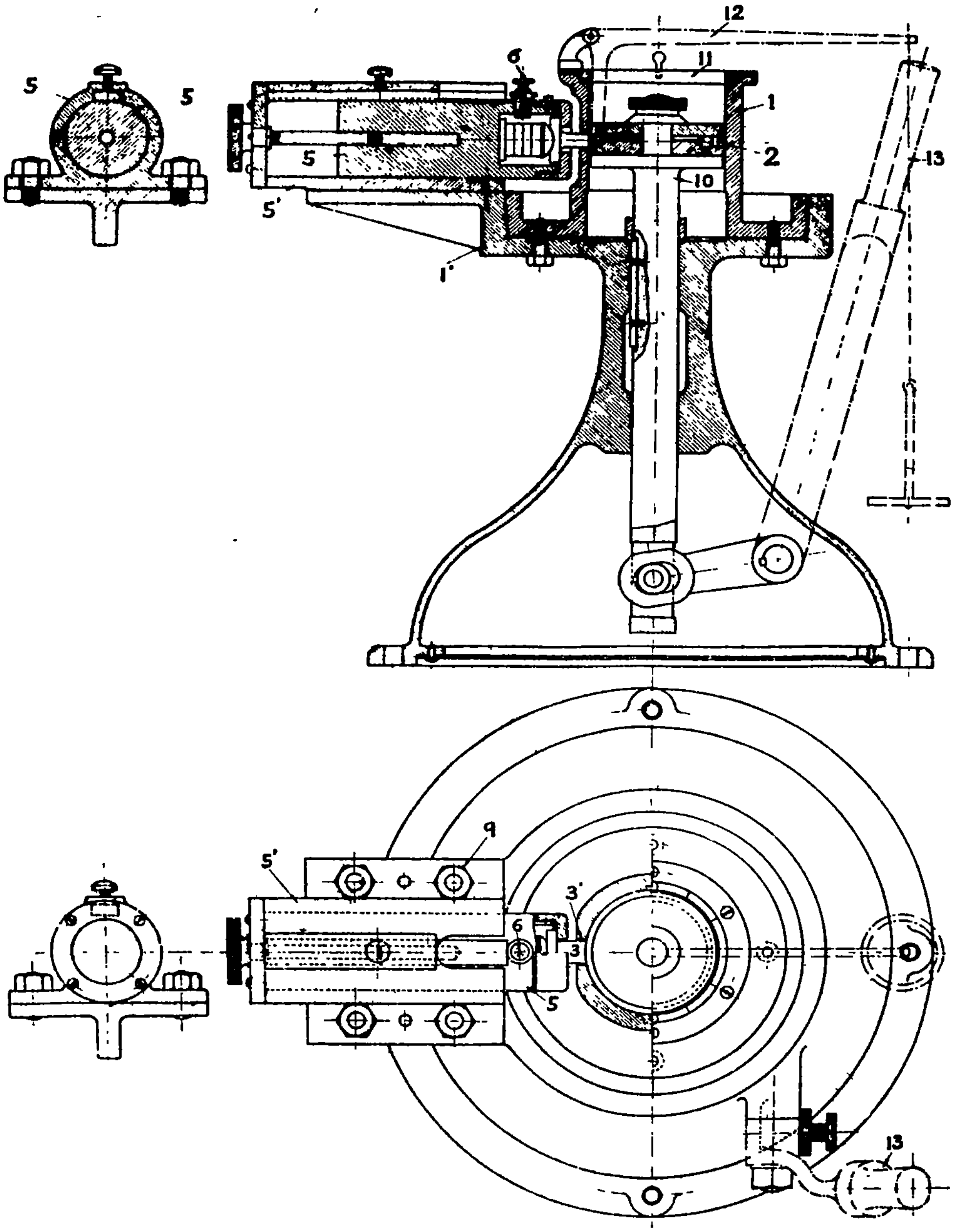


Fig. 7. (Ring Tester of Type III)

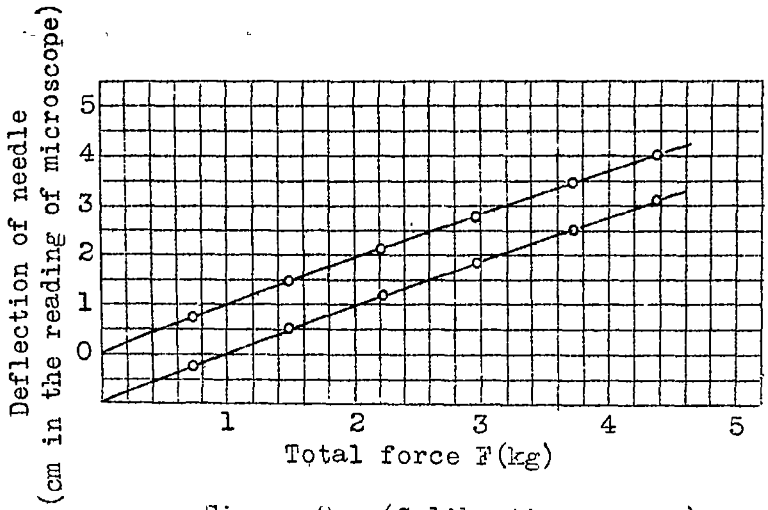


Figure 9.- (Calibration curves).

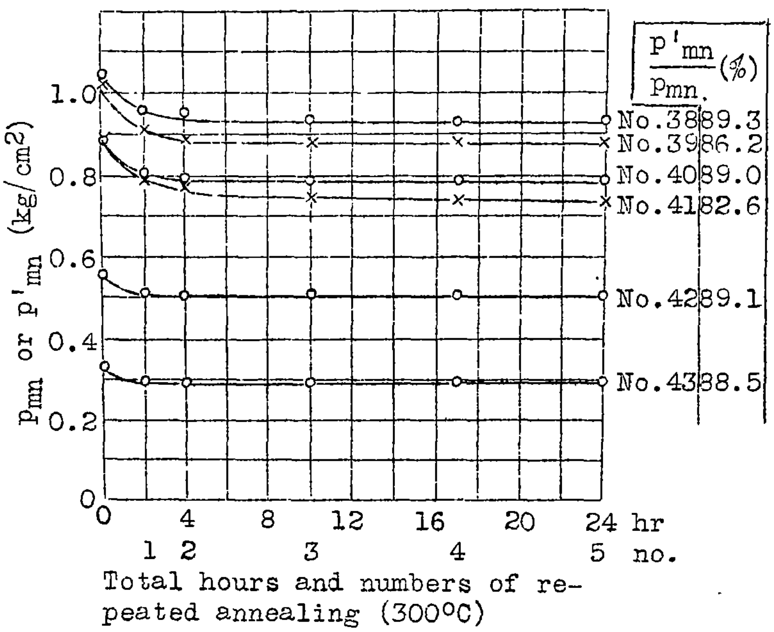


Figure 54.

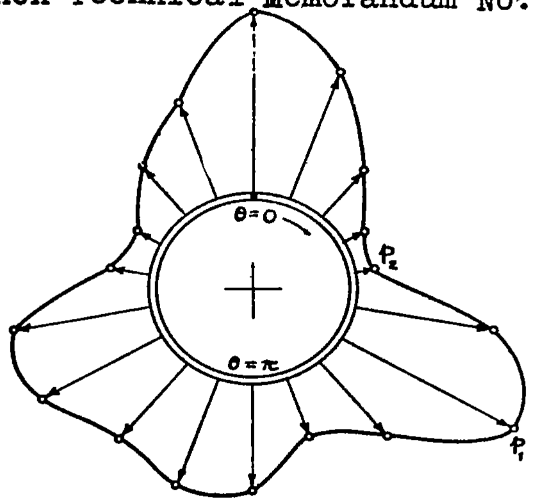


Fig. 10. (Table III)

p_1 = 1·89 kg./cm². (Ring No. 1)
p_2 = 0·12 ,,
p_{mn} = 0·964 ,,
r = 15·7

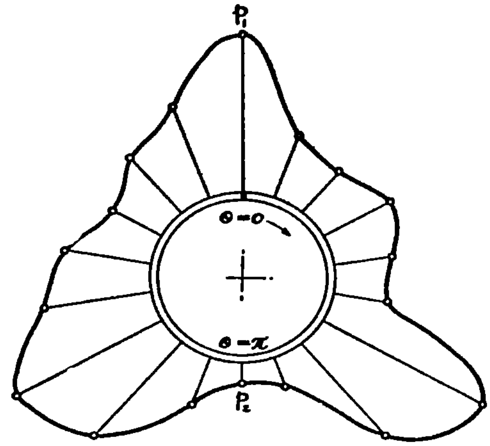


Fig. 11. (Table IV)

p_1 = 1·87 kg./cm². (Ring No. 2)
p_2 = 0·20 ,,
p_{mn} = 0·980 ,,
r = 10·5

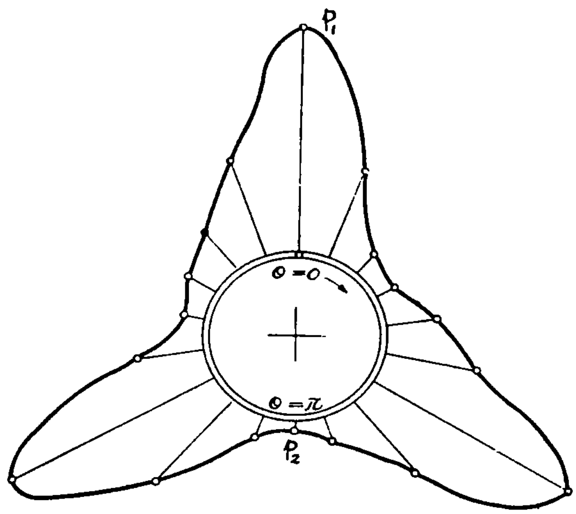


Fig. 12. (Table V)

p_1 = 2.71 kg./cm². (Ring No. 3)
p_2 = 0·09 ,,
p_{mn} = 0·974 ,,
r = 30·1

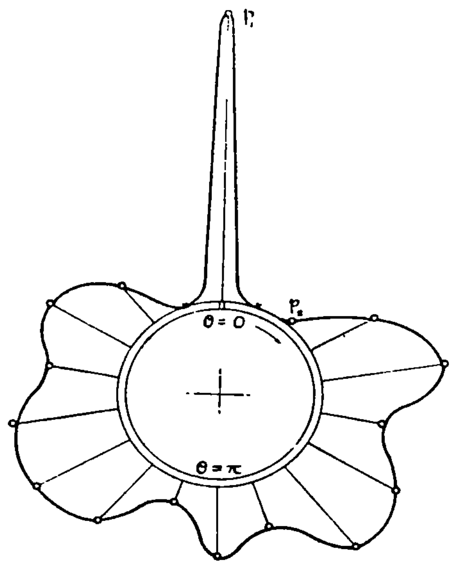


Fig. 13. (Table VI)

p_1 = 3·00 kg./cm². (Ring No. 4)
p_2 = 0·08 ,,
p_{mn} = 0·864 ,,

$$r = \frac{p_{max}}{p_{min}}$$

Fig. 13'

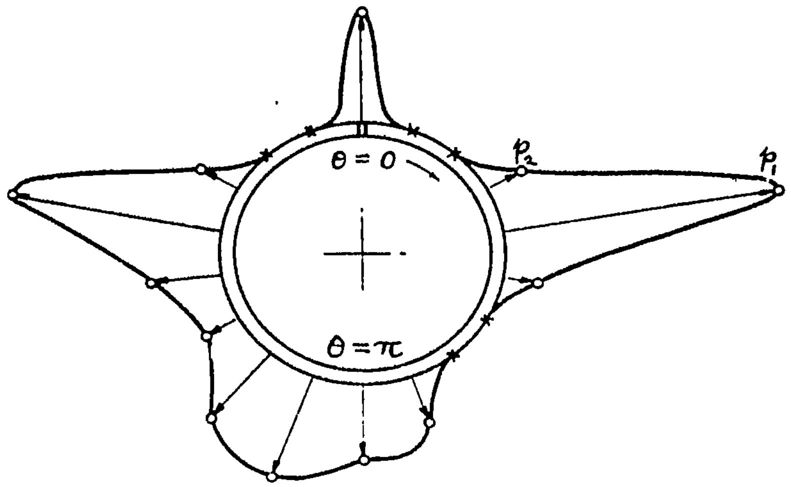


Fig. 14. (Table VII)

p_1 = 3.78 kg./cm².
p_2 = 0·51 ,, (Ring No. 5)
p_{mn} = 0.892 ,,

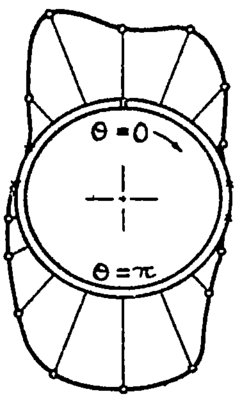


Fig. 15.
(Ring No. 5′)

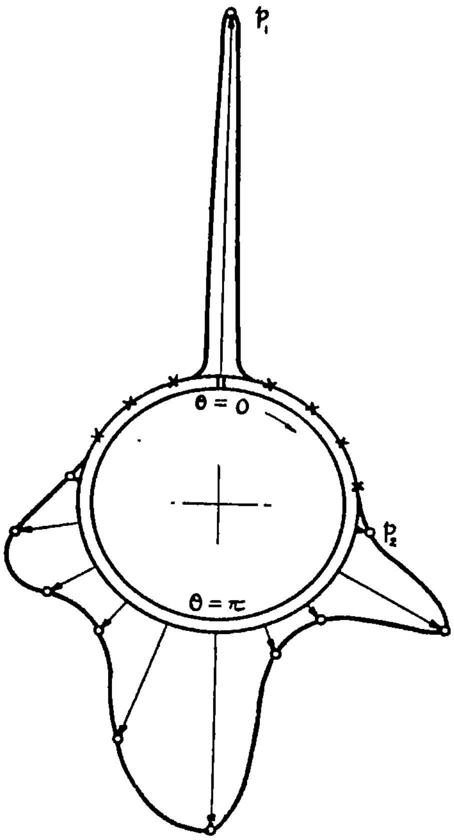


Fig. 16. (Table VIII)

p_1 = 5·40 kg./cm².
p_2 = 0·11 ,, (Ring No. 6)
p_{mn} = 0·820 ,,

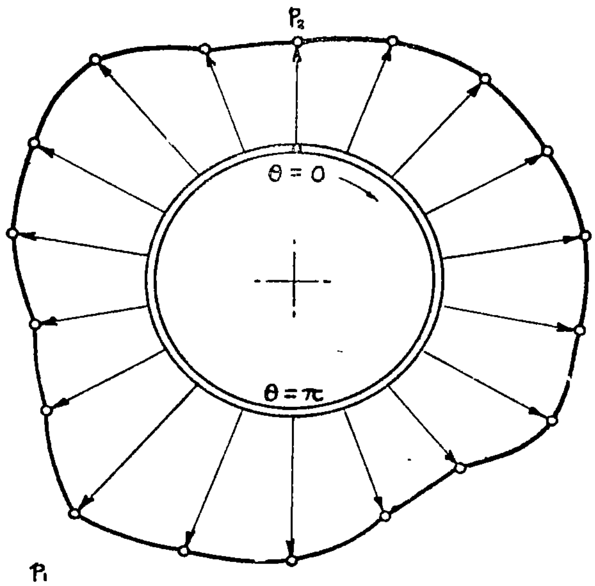
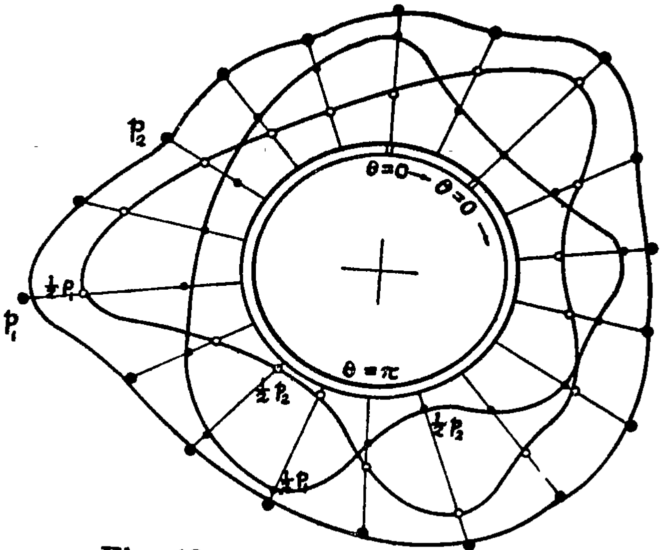


Fig. 17. (Table IX)

p_1 = 1·26 kg./cm².
p_2 = 0.74 ,, (Ring No. 7)
p_{mn} = 0·822 ,,
r = 1·7



$\frac{1}{2}p_1 = 0{\cdot}94$ kg./cm².
$\frac{1}{2}p_2 = 0{\cdot}12$,, (Ring No. 8)
$\frac{1}{2}p_{mn} = 0{\cdot}507$,,
$r = 7{\cdot}8$

$\frac{1}{2}p_1' = 1{\cdot}13$ kg./cm².
$\frac{1}{2}p_2' = 0{\cdot}02$,, (Ring No. 9)
$\frac{1}{2}p_{mn} = 0{\cdot}502$,,
$r = 56{\cdot}5$

Fig. 18. (Table X~XII)

$p_1 = 1{\cdot}55$ kg./cm².
$p_2 = 0{\cdot}84$,, (Ring No. 10)
$p_{mn} = 1{\cdot}05$,,
$r = 1{\cdot}9$

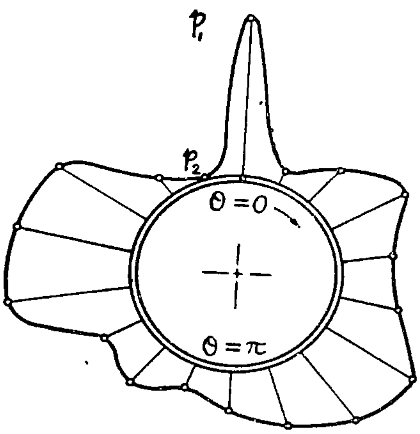


Fig. 19. (Table XIII)

$p_1 = 2{\cdot}11$ kg./cm².
$p_2 = 0{\cdot}08$,, (Ring No. 11)
$p_{mn} = 0{\cdot}850$,,

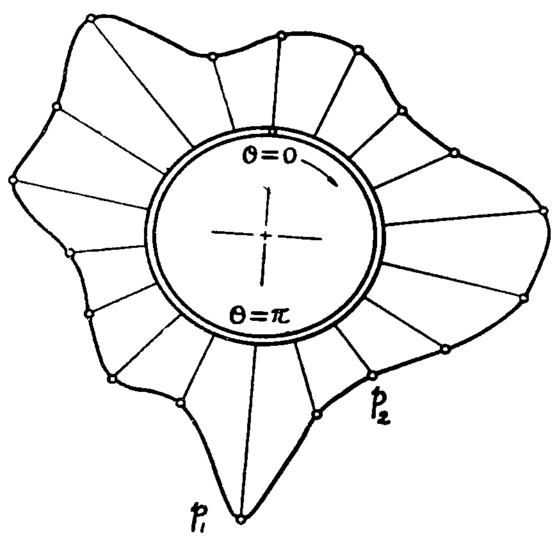


Fig. 20. (Table XIV)

$p_1 = 0.636$ kg./cm²
$p_2 = 0{\cdot}206$,, (Ring No. 12)
$p_{mn} = 0{\cdot}360$,,

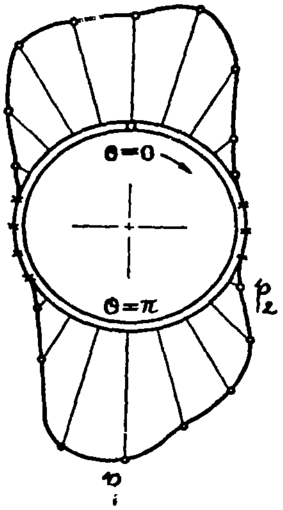


Fig. 21. (Table XV)

p_1 =0·965 kg./cm².
p_2 =0·086 ,, (Ring. No. 13)
p_{mn}=0·351 ,,

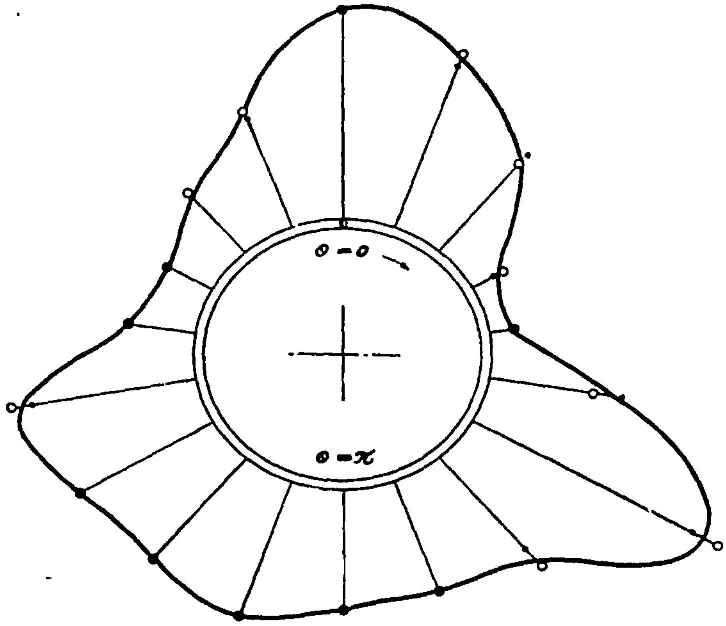


Fig. 22. (Table XXVI~XXVI′)

(○)... p_1 = 1·85 kg./cm². p_2 = 0·17 ,, p_{mn} = 0·857 ,, r =10.9 (Ring No. 1″) annealed at 300°C

(●)... p_1 = 1·68 kg./cm². p_2 = 0·14 ,, p_{mn} = 0·848 ,, r =12·9 (Ring No. 1″)

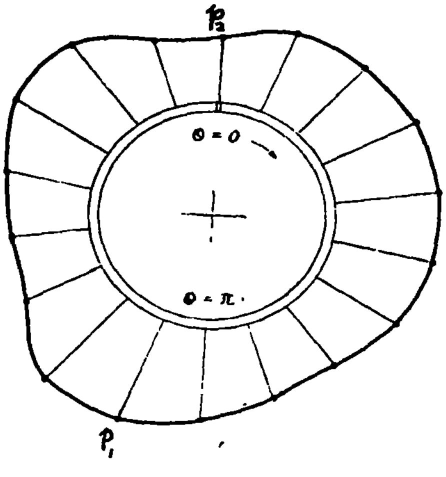


Fig. 23. (Table XXVIII)

p_1 =0·97 kg./cm².
p_2 =0·54 ,, (Ring No. 7″)
p_{mn}=0·756 ,, annealed at 300°C
r =1·8

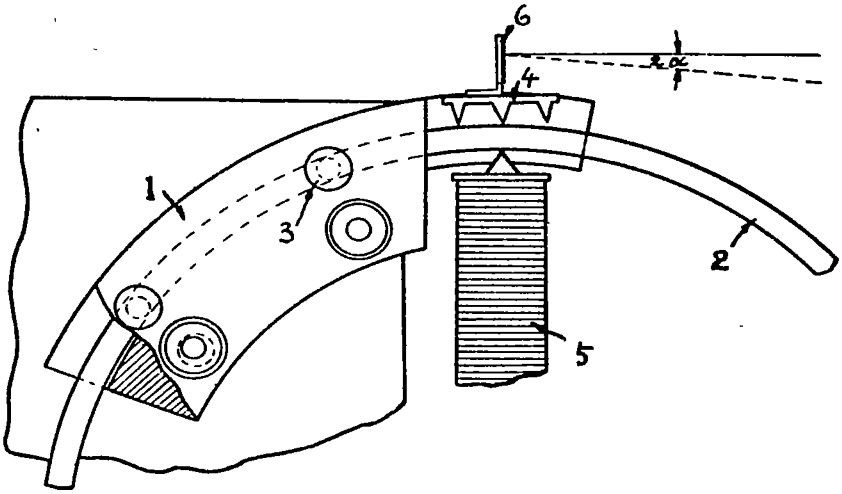

6
α
4
1
3
2
5

Fig. 24.

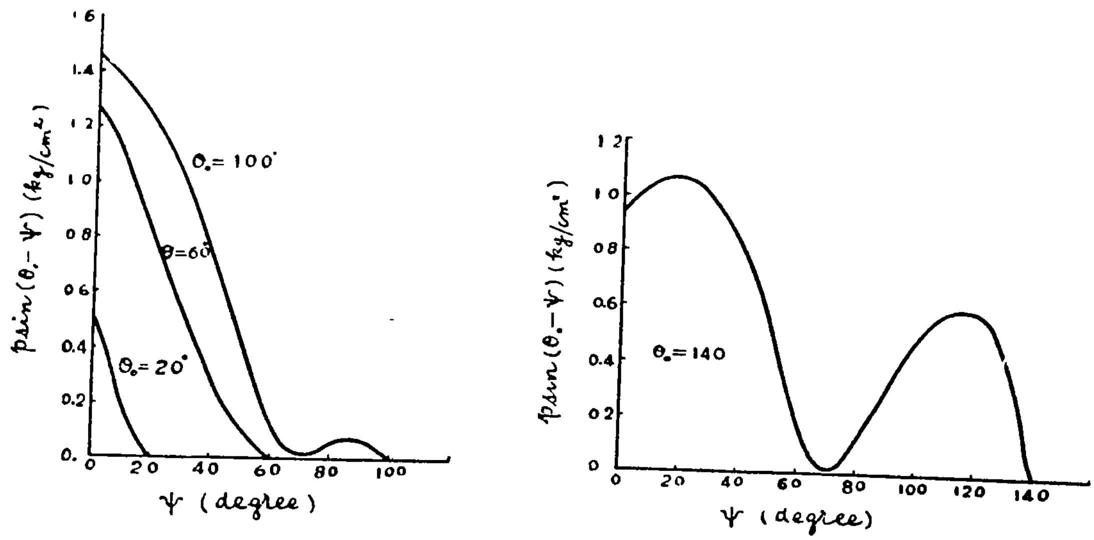

p sin (θ₀ − ψ) (kg/cm²)
θ₀ = 100°
θ = 60°
θ₀ = 20°
ψ (degree)
θ₀ = 140
ψ (degree)

Fig. 25'

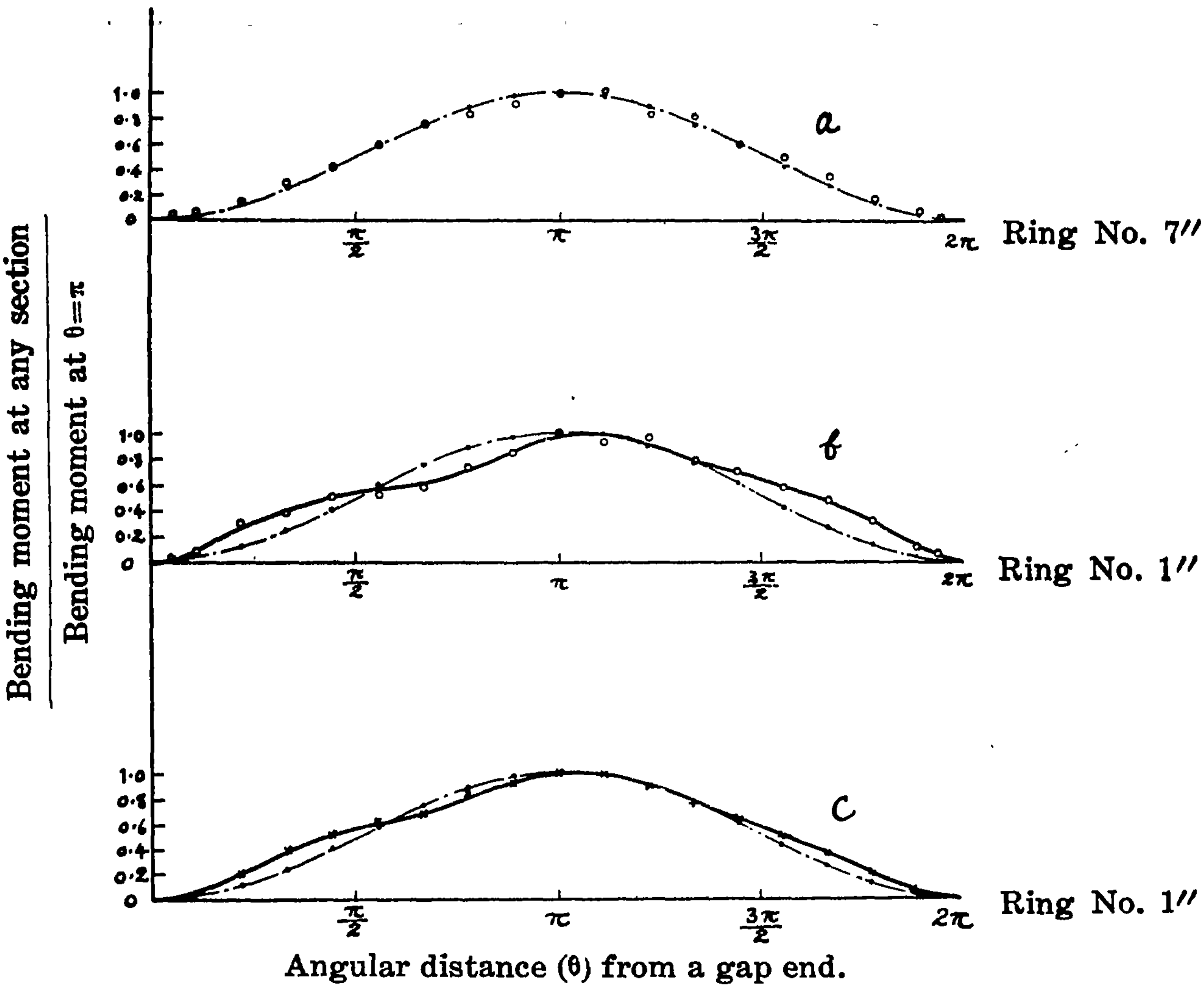


Fig. 25. (Table XVI)

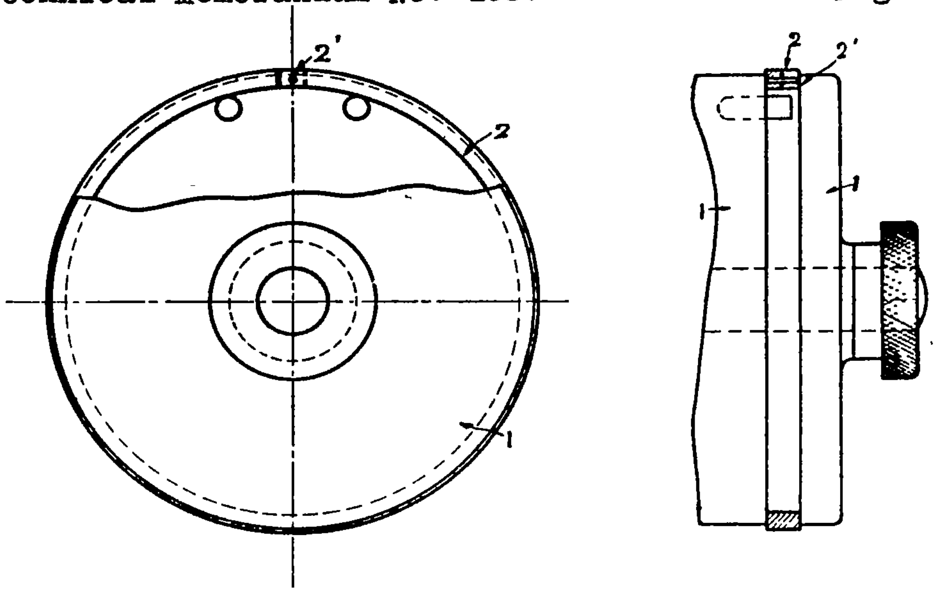


Fig. 26.

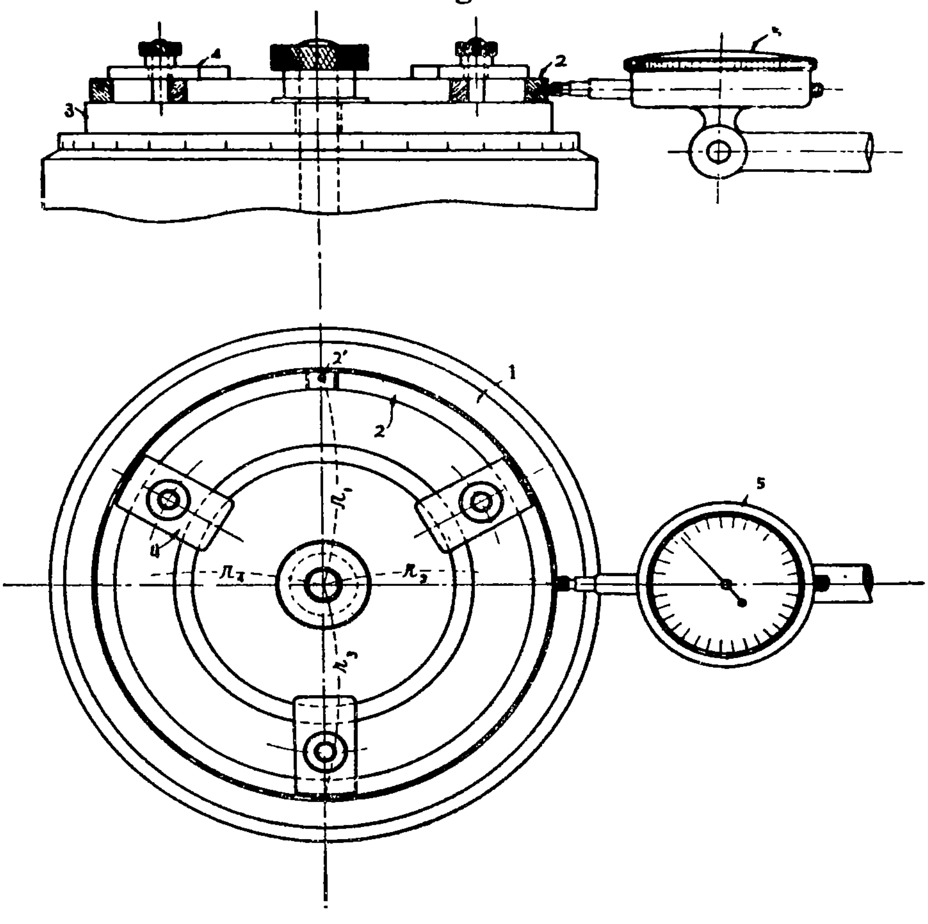


Fig. 27.

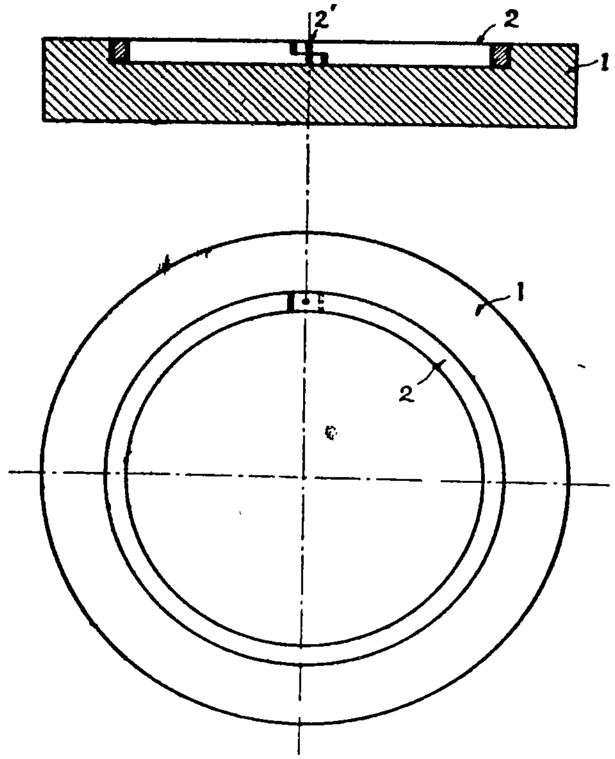


Fig. 28.

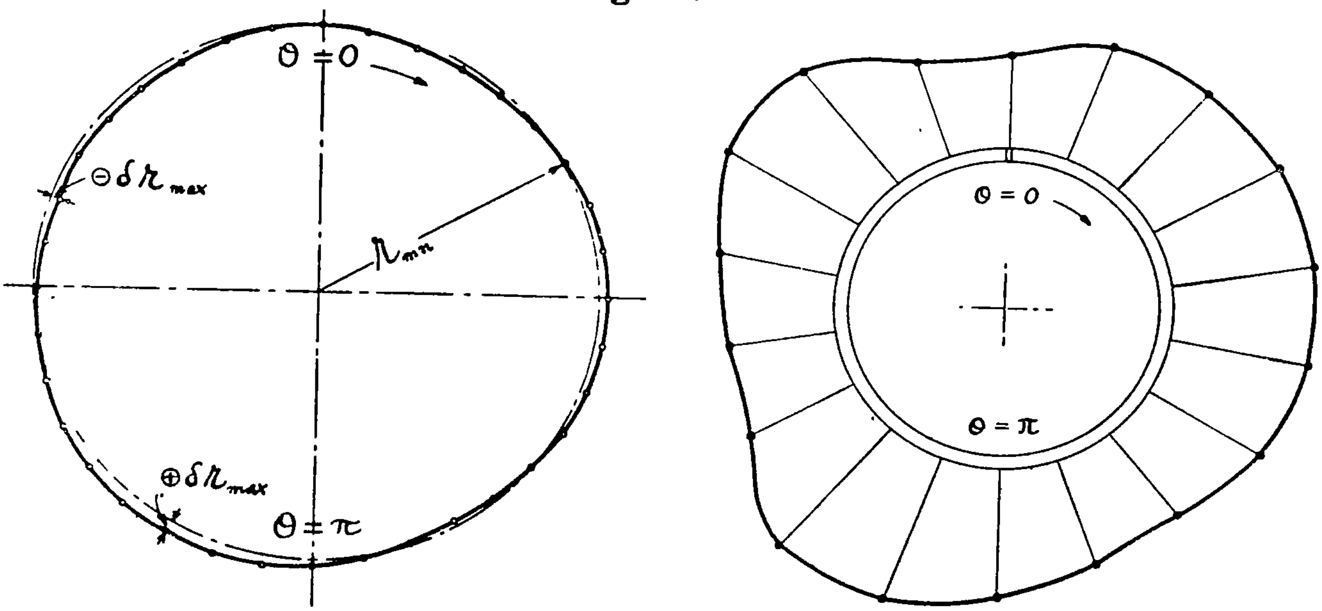


Fig. 29.

No. 14. (I. P. C. R. or 'Riken')
$D = 12{\cdot}70$ cm.
$\delta r \begin{cases} +0{\cdot}021 \text{ mm.} \\ -0{\cdot}023 \text{ ,,} \end{cases}$

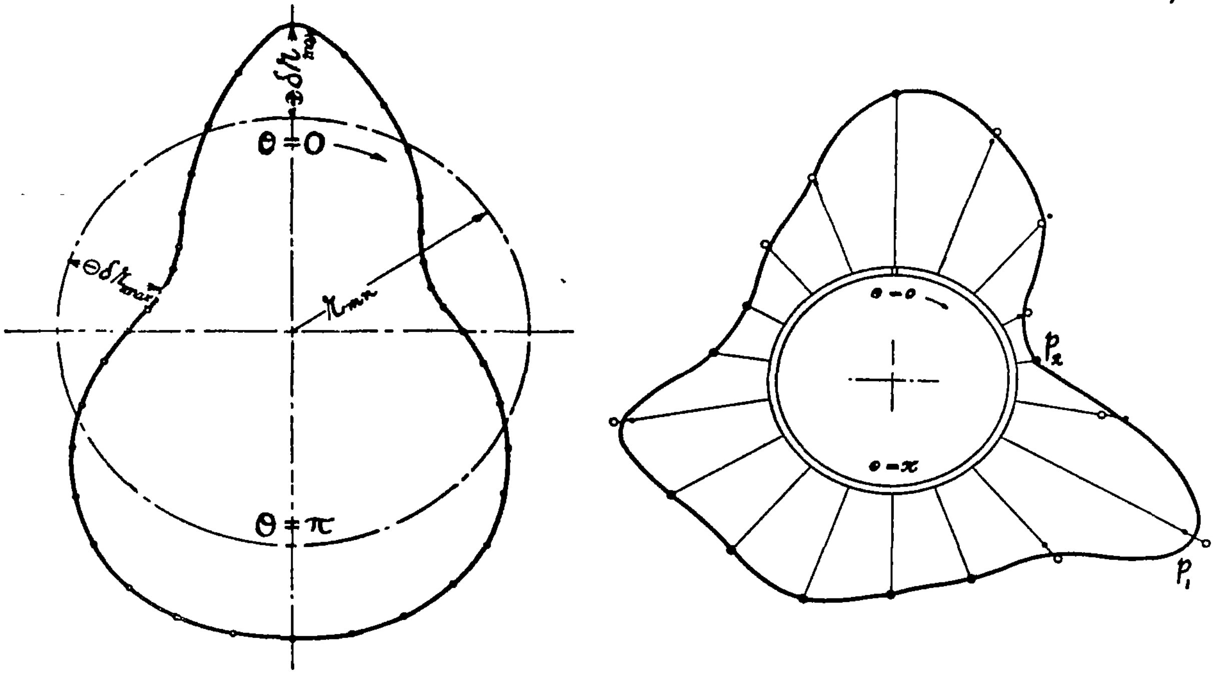


Fig. 30.

No. 15.
D=12·70 cm.
$\delta r \begin{cases} +0{\cdot}165 \text{ mm.} \\ -0{\cdot}171 \text{ ,,} \end{cases}$

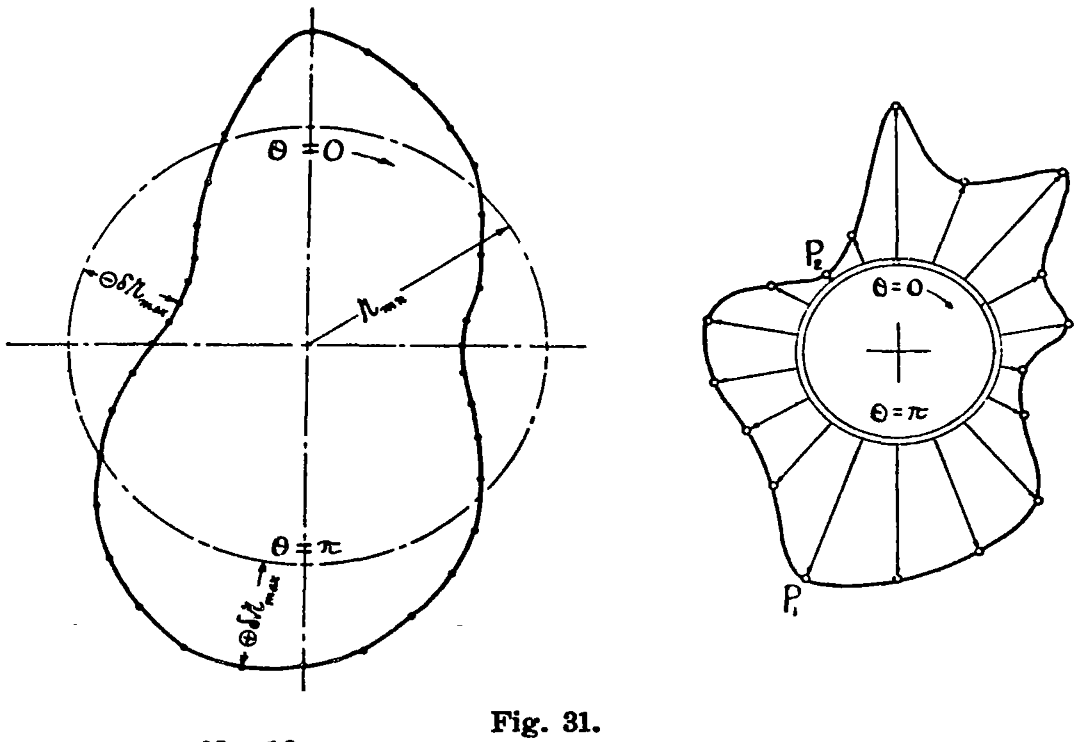


Fig. 31.

No. 16
D=12·70 cm.
$\delta r \begin{cases} +0{\cdot}195 \text{ mm.} \\ -0{\cdot}177 \text{ ,,} \end{cases}$

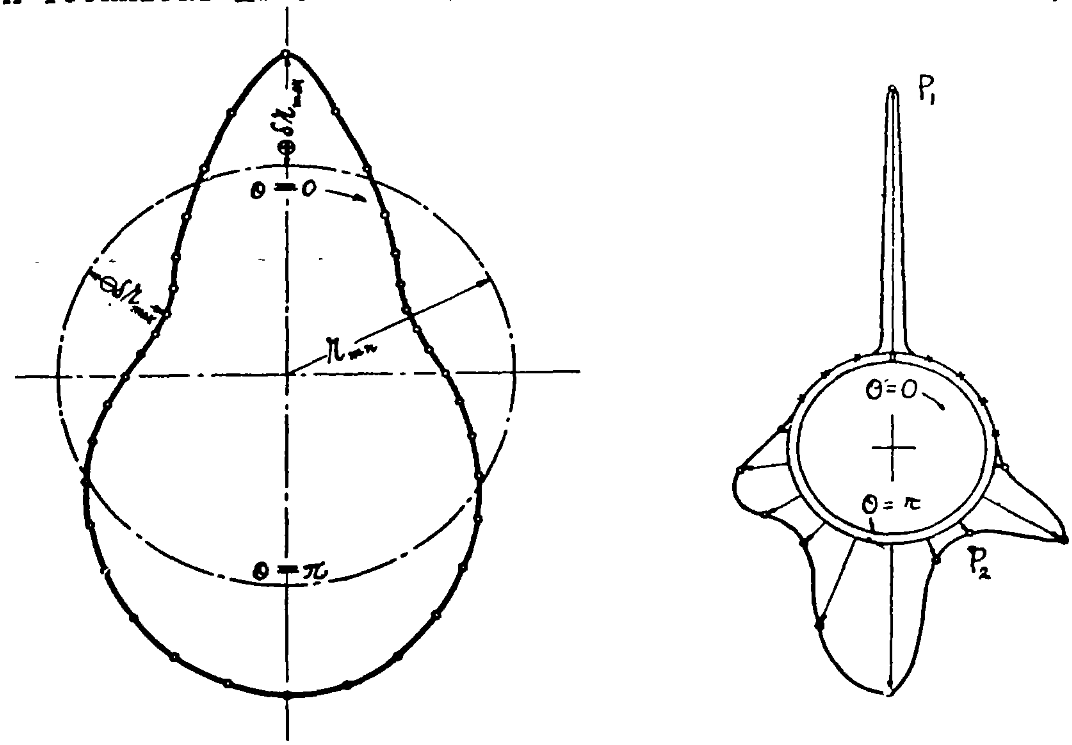


Fig. 32.

No. 17. (Made by the simplest method)
D=12·70 cm.
$\delta r \begin{cases} +0{\cdot}898 \text{ mm.} \\ -0{\cdot}672 \quad ,, \end{cases}$

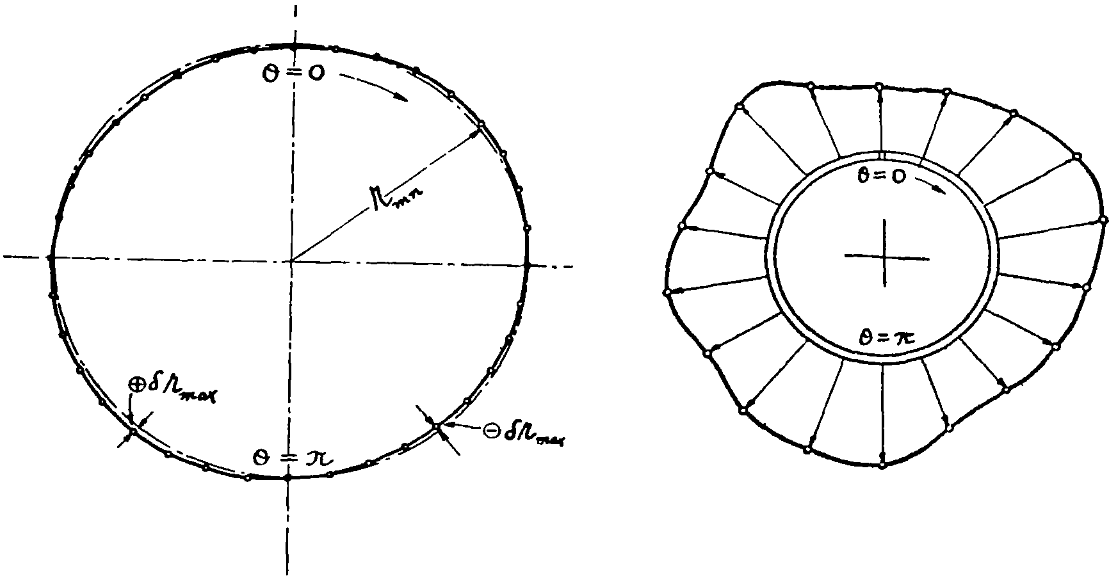


Fig. 33.

No. 18. (I. P. C. R. or 'Riken')
D=8·89 cm.
$\delta r \begin{cases} +0{\cdot}019 \text{ mm.} \\ -\underline{0{\cdot}020} \quad ,, \end{cases}$

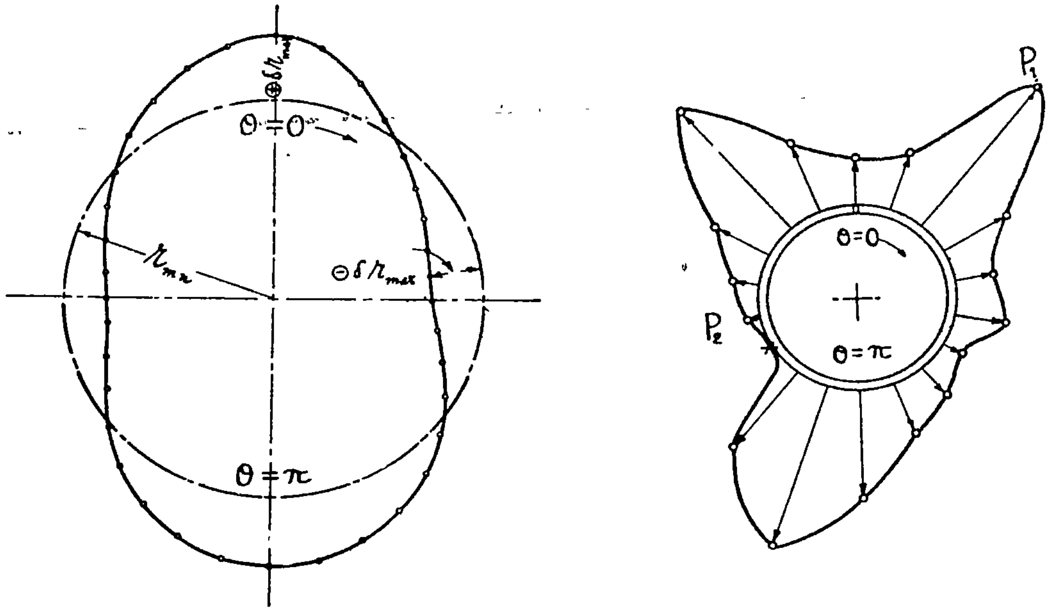


Fig. 34.

No. 19.
$D = 8{\cdot}89$ cm.
$\delta r \begin{cases} +0{\cdot}120 \text{ mm.} \\ -0{\cdot}112 \text{ ,,} \end{cases}$

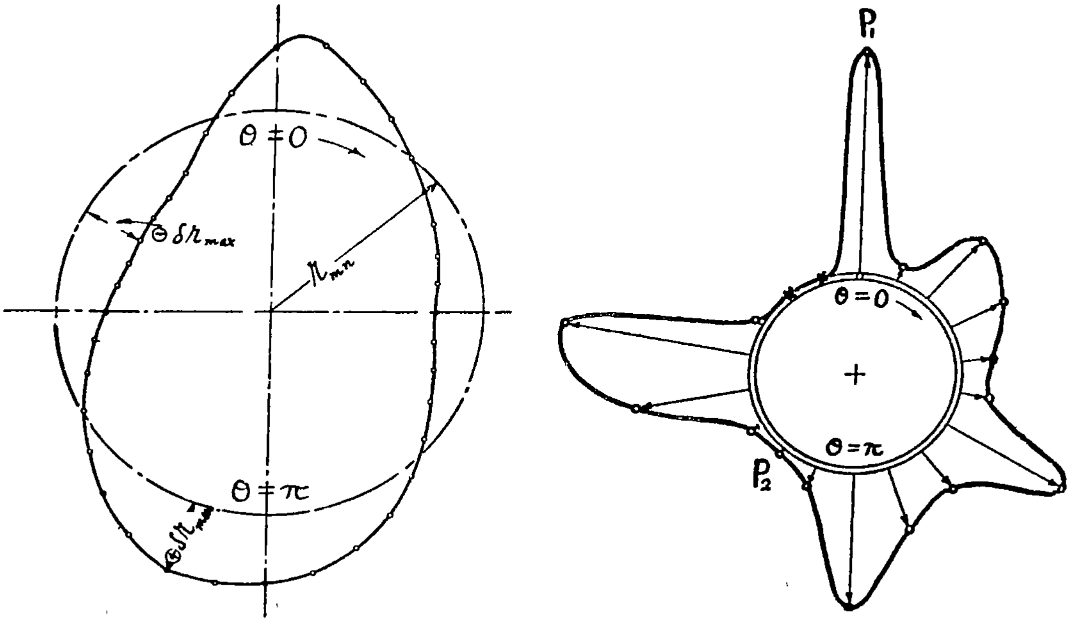


Fig. 35.

No. 20.
$D = 8{\cdot}89$ cm.
$\delta r \begin{cases} +0{\cdot}144 \text{ mm.} \\ -0{\cdot}113 \text{ ,,} \end{cases}$

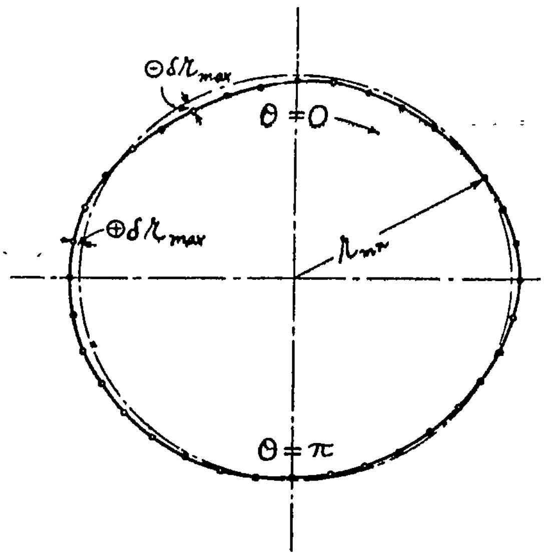


Fig. 36.

No. 21. (I. P. C. R. or 'Riken')
D=8·50 cm.
$\delta r \begin{cases} +0{\cdot}022 \text{ mm.} \\ -0{\cdot}025 \text{ ,,} \end{cases}$

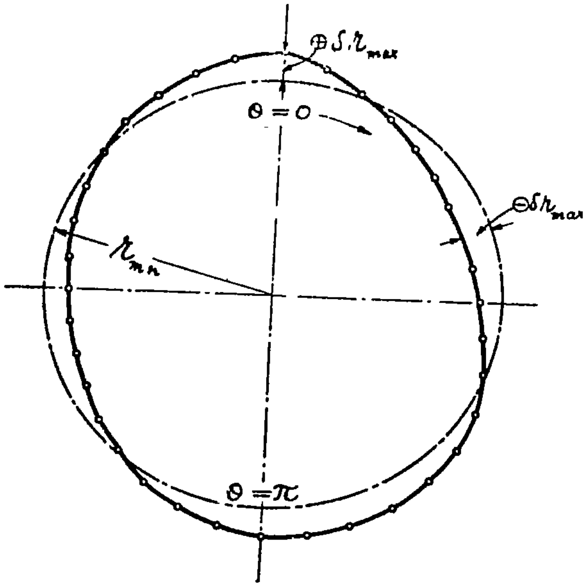


Fig. 37.

No. 22. (I. P. C. R. or 'Riken')
D=12·0 cm.
$\delta r \begin{cases} +0{\cdot}060 \text{ mm.} \\ -0{\cdot}055 \text{ ,,} \end{cases}$

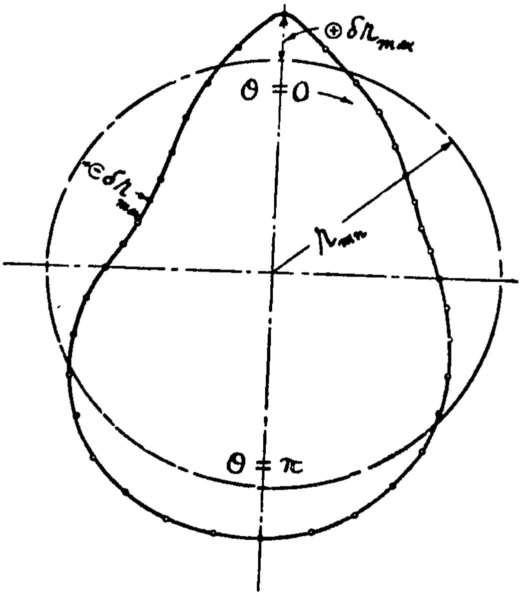


Fig. 38.

No. 23.
D=8.89 cm.
$\delta r \begin{cases} +0{\cdot}090 \text{ mm.} \\ -0{\cdot}129 \text{ ,,} \end{cases}$

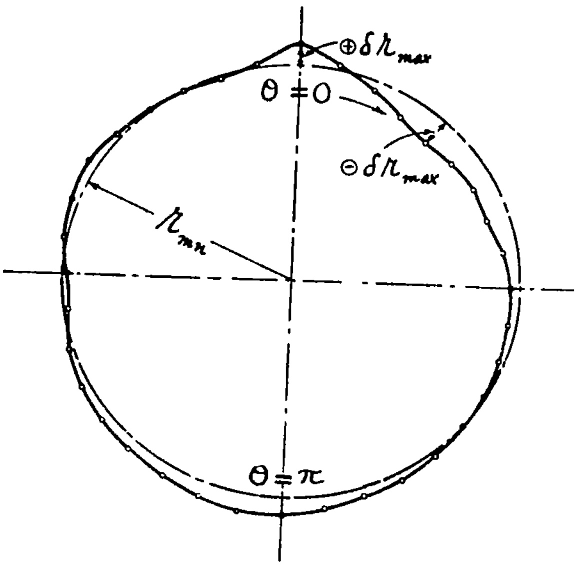


Fig. 39.

No. 24.
D=8·57 cm.
$\delta r \begin{cases} +0{\cdot}035 \text{ mm.} \\ -0{\cdot}054 \text{ ,,} \end{cases}$

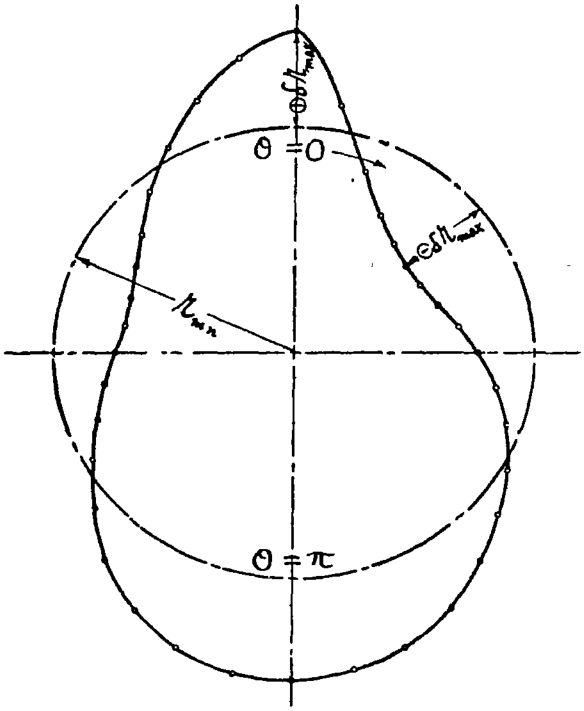


Fig. 40.

No. 25.
D=9·50 cm.
$\delta r \begin{cases} +0{\cdot}183 \text{ mm.} \\ -0{\cdot}169 \text{ ,,} \end{cases}$

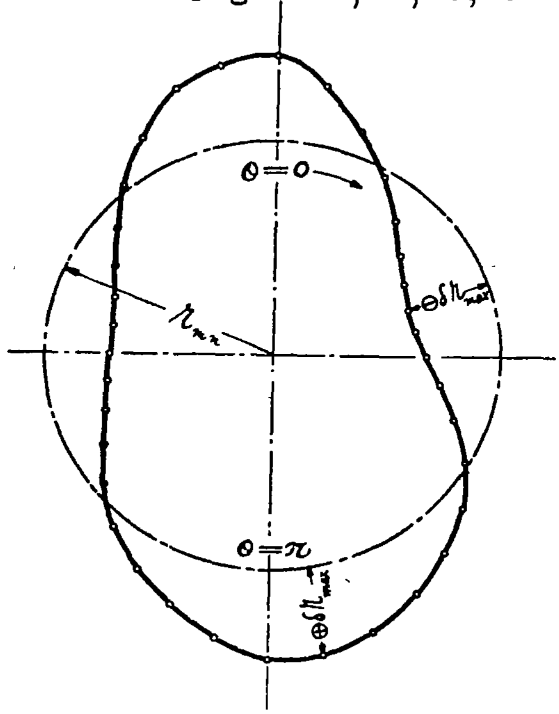


Fig. 41.

No. 26.
D=10·15 cm.
$\delta r \begin{cases} +0.305 \text{ mm.} \\ -0{\cdot}277 \text{ ,,} \end{cases}$

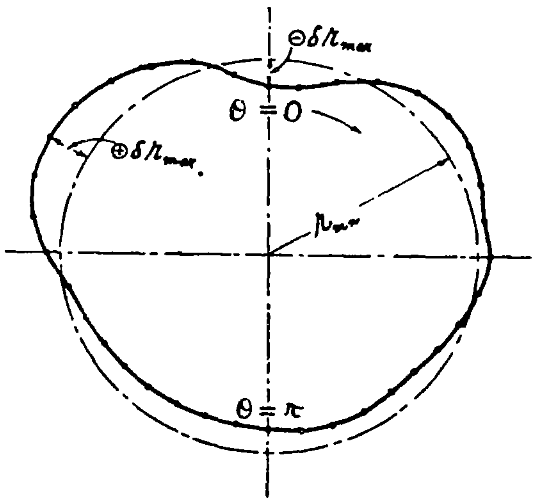


Fig. 42.

No. 27.
D=12·0 cm.
$\delta r \begin{cases} +0{\cdot}084 \text{ mm.} \\ -0{\cdot}052 \text{ ,,} \end{cases}$

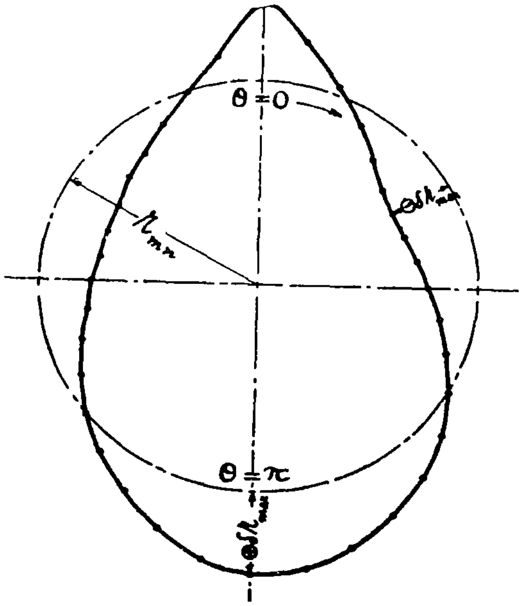


Fig. 43.

No. 28.
D=12·0 cm.
$\delta r \begin{cases} +0{\cdot}156 \text{ mm.} \\ -0{\cdot}118 \text{ ,,} \end{cases}$

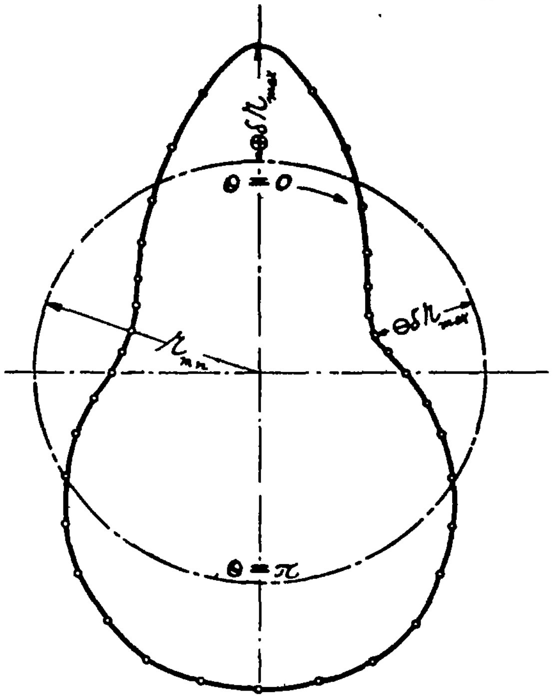


Fig. 44

No. 29.
D=12.70 cm.
$\delta r \begin{cases} +0{\cdot}242 \text{ mm.} \\ -0{\cdot}205 \text{ ,,} \end{cases}$

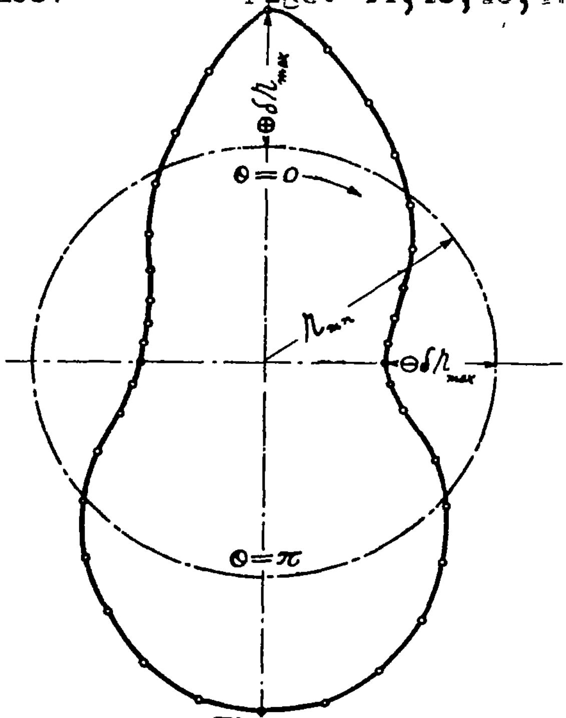


Fig. 45.

No. 30.
D=14·0 cm.
$\delta r \begin{cases} +0.235 \text{ mm.} \\ -0.185 \text{ ,,} \end{cases}$

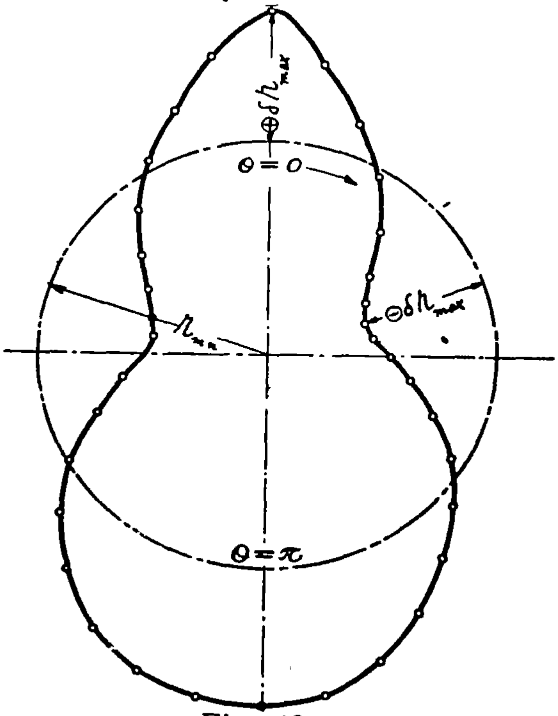


Fig. 46.

No. 31.
D=13·80 cm.
$\delta r \begin{cases} +0{\cdot}400 \text{ mm.} \\ -0{\cdot}348 \text{ ,,} \end{cases}$

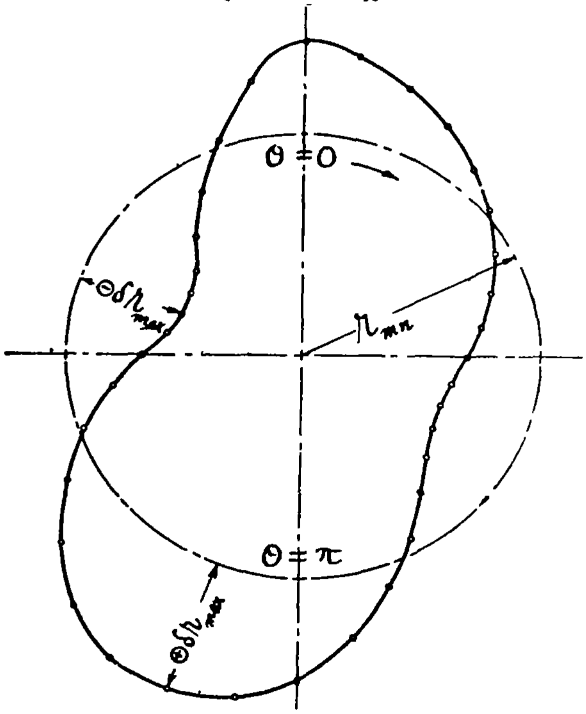


Fig. 47.

No. 32.
D=9·03 cm.
$\delta r \begin{cases} +0{\cdot}238 \text{ mm.} \\ -0{\cdot}185 \text{ ,,} \end{cases}$

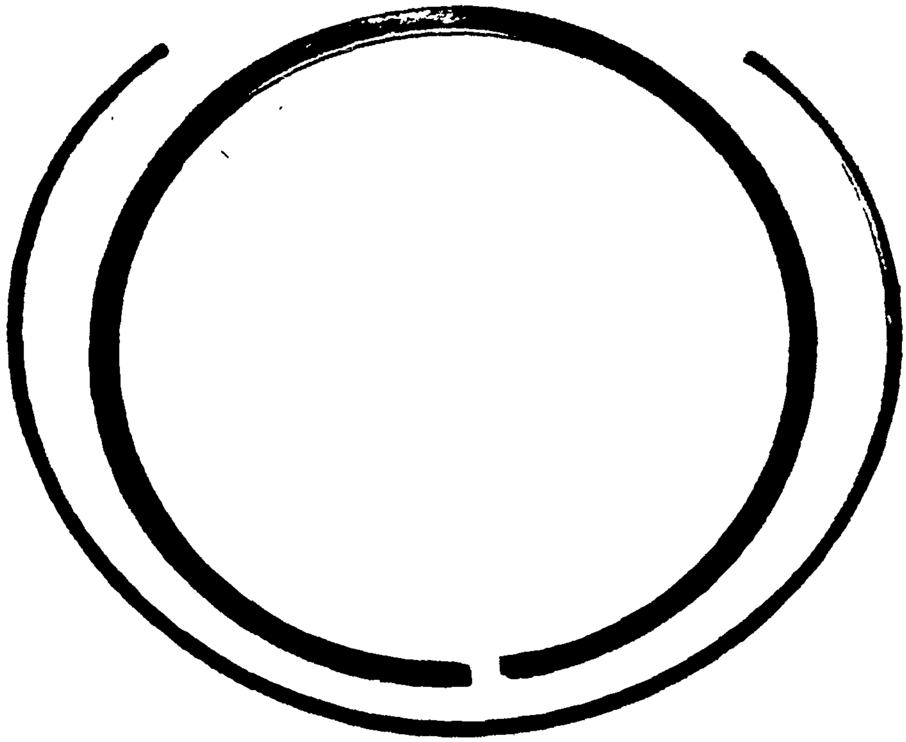


Fig. 48. (Ring No. 26)

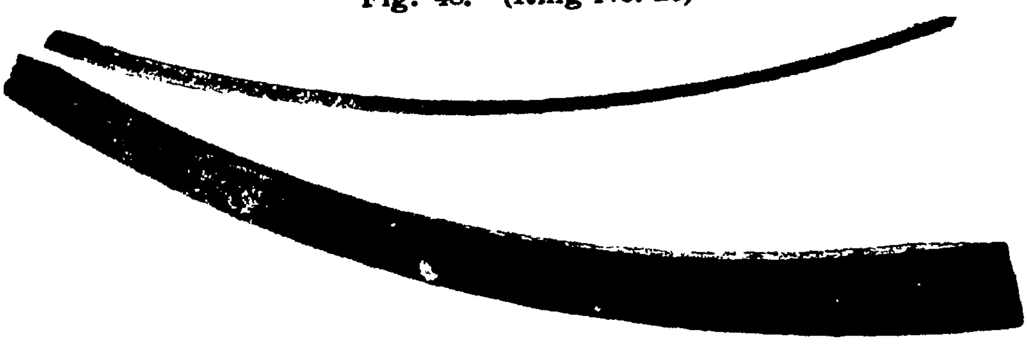


Fig. 60. Piston ring used in a marine steam engine of vertical type.

Dia. of cyl............660 mm.
Running interval ....about 3 weeks.
Kind of rings ....hammered ring.
Thickness :—

initial......uniform......31·8 mm.

final ..... { max. ....20·8 ,,
{ min. .... 2.5 ,,

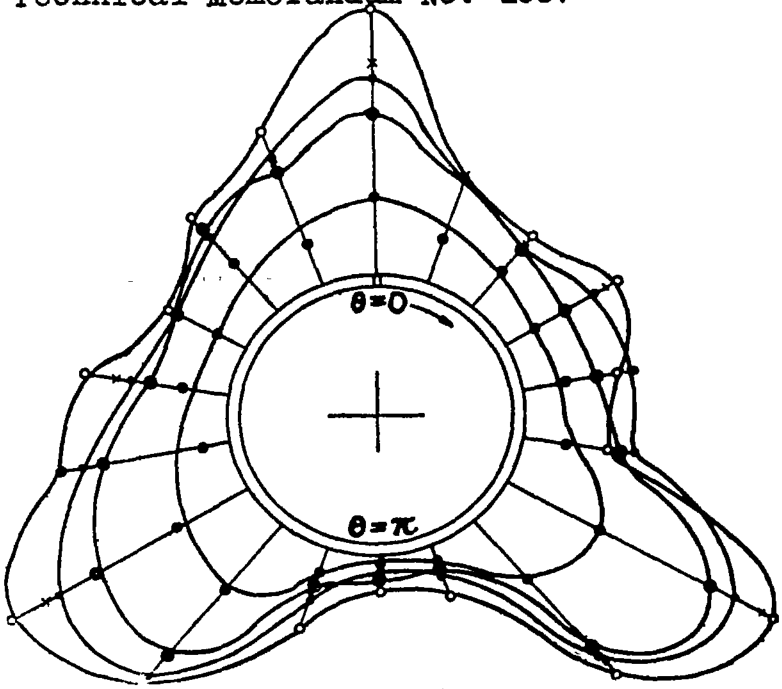


Fig. 49. (Table IV & XXI~XXIV)

		p_{mn}(kg./cm².)	r
○	No. 2	0·980	10·5
×	No. 2′	0·888	10·5
●	No. 2″	0·847	12·6
◎	No. 2‴	0·740	11·2
●	No. 2⁗	0·355	5·8

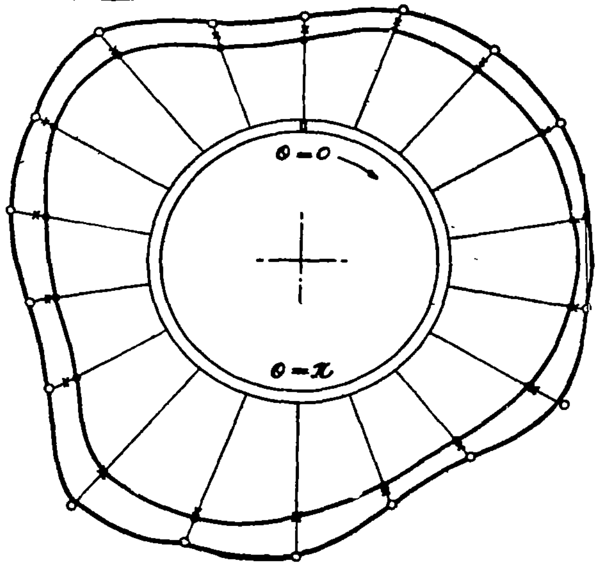


Fig. 51. (Table IX & XXVII~XXVII

		p_{mn}(kg./cm².)	r
○	No. 7	0·922	1·7
×	No. 7′	0·773	1·6
●	No. 7″	0·756	1·8

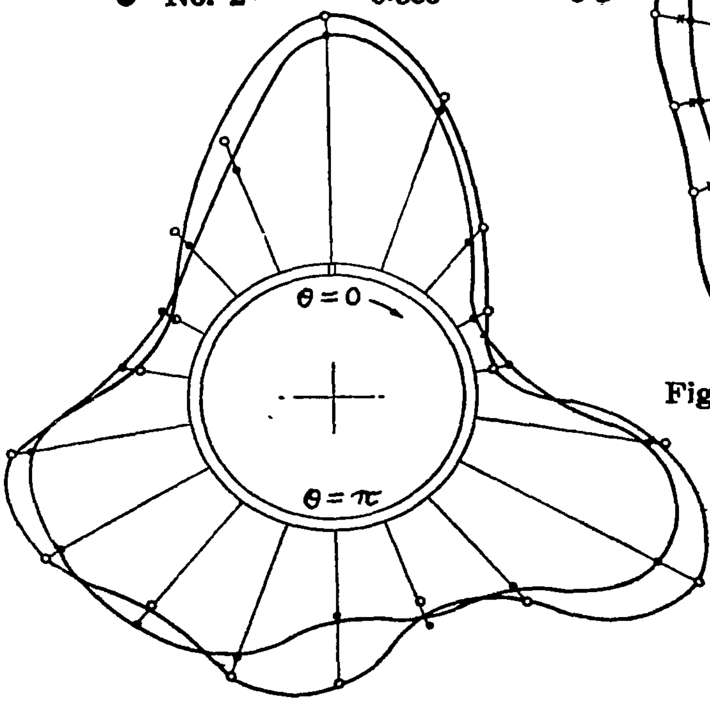


Fig. 50. (Table III & XXV)

		p_{mn}(kg./cm².)	r
○	No. 1	0·964	14·4
●	No. 1″	0·865	15·7

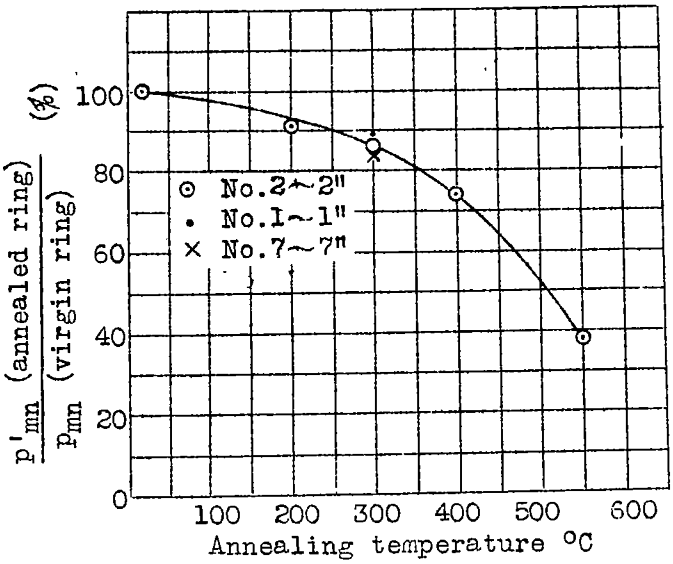


Figure 52.

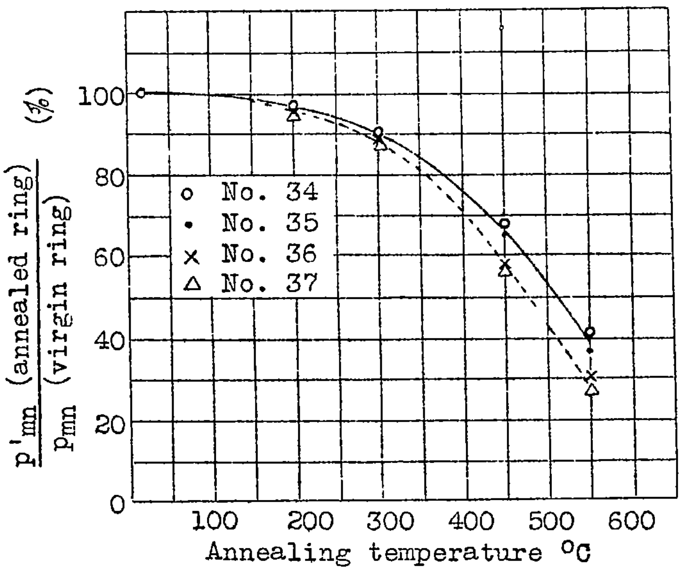


Figure 53.

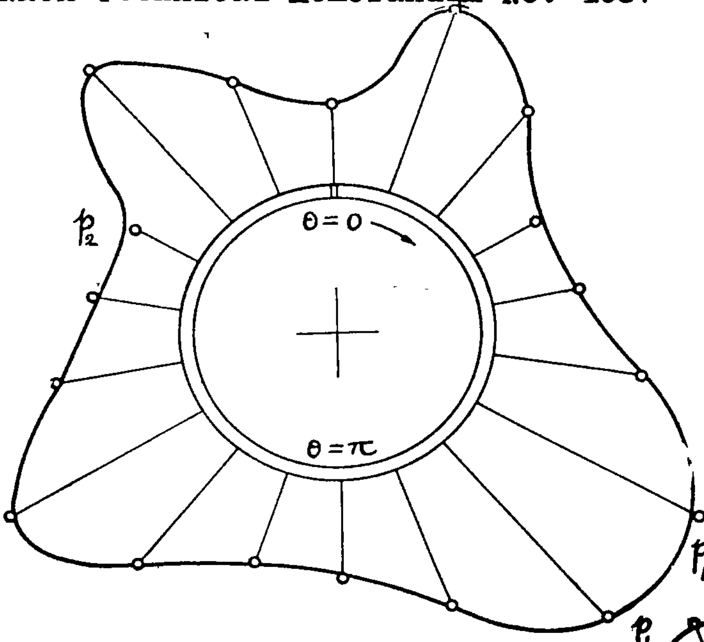


Fig. 55. (Table XXX)

$p_1 = 1{\cdot}61$ kg./cm².
$p_2 = 0{\cdot}45$,, (Ring No. 44.)
$p_{mn} = 0{\cdot}927$,,
$r = 3{\cdot}7$

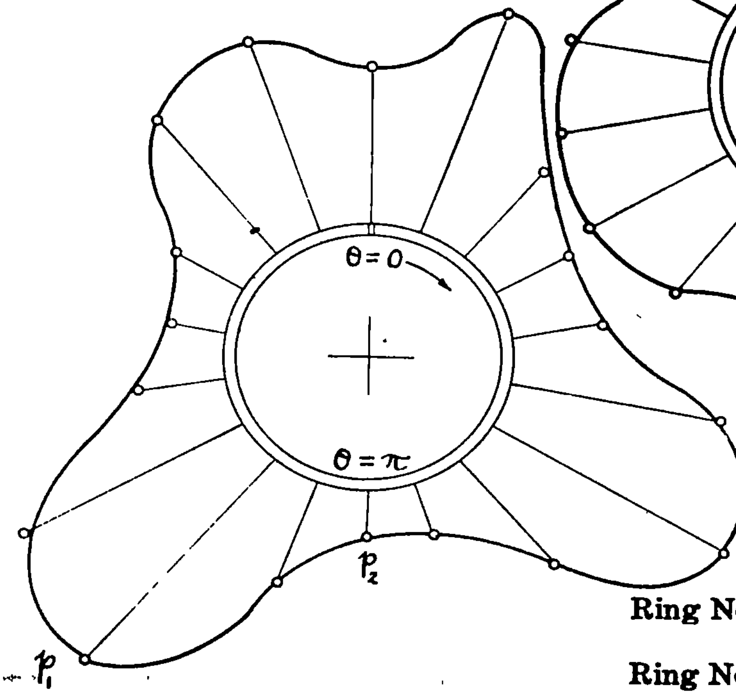


Fig. 56. (Table XXXI)

$p_1 = 1{\cdot}99$ kg./cm².
$p_2 = 0{\cdot}30$,, (Ring No. 45)
$p_{mn} = 1{\cdot}03$,,
$r = 6{\cdot}6$

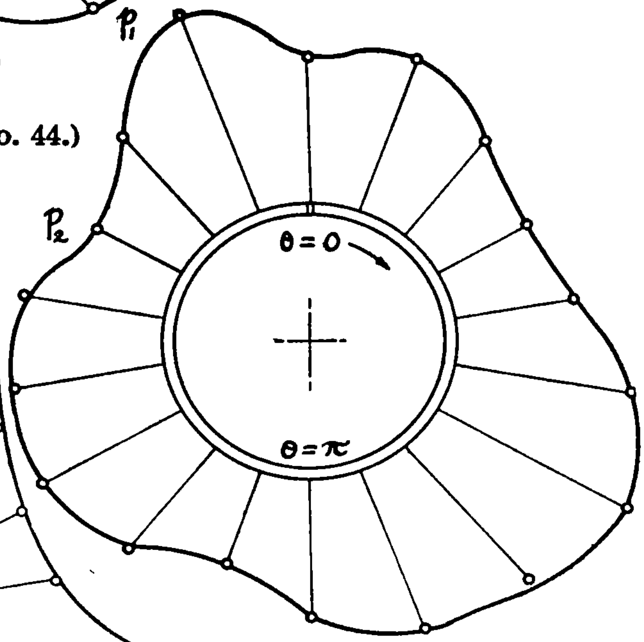


Fig. 57. (Table XXXII)

$p_1 = 1{\cdot}49$ kg./cm².
$p_2 = 0{\cdot}60$,, (Ring No. 46)
$p_{mn} = 1{\cdot}01$,,
$r = 2{\cdot}4$

Ring No. 44... After being used for half a year at the top of the piston of a Diesel engine.

Ring No. 45... After being used for half a year at the 3rd from the top of the piston of a Diesel engine.

Ring No. 46... After being used for half a year at the 5th from the top of the piston of a Diesel engine.

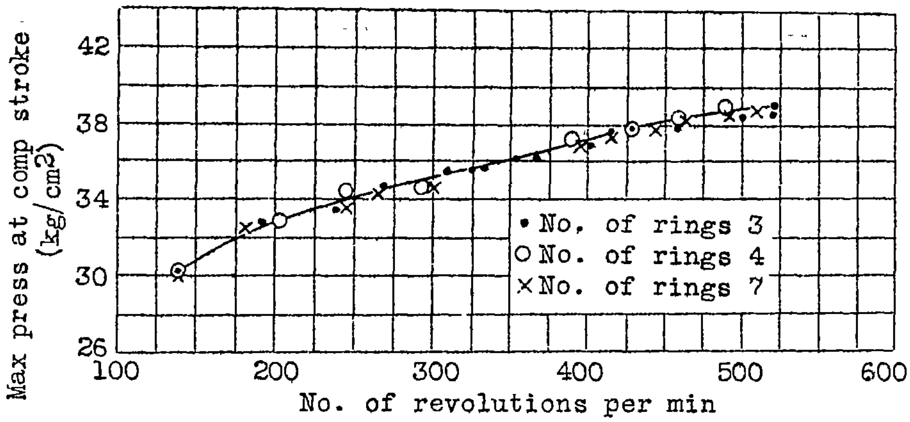


Figure 58.

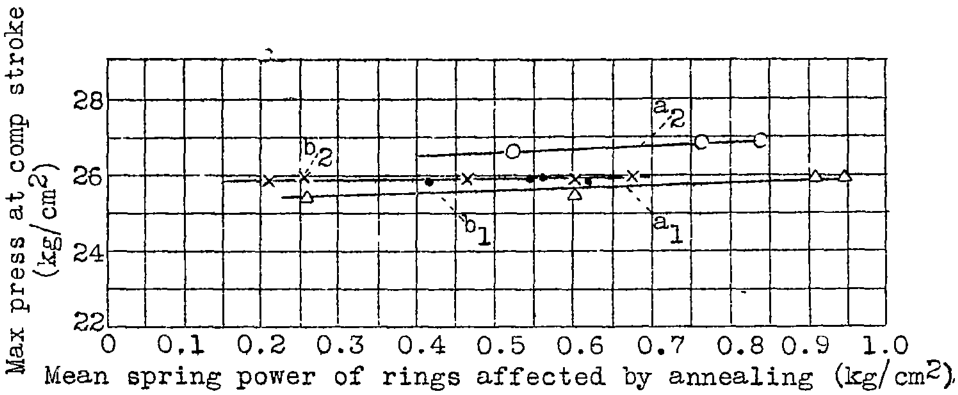


Figure 59.

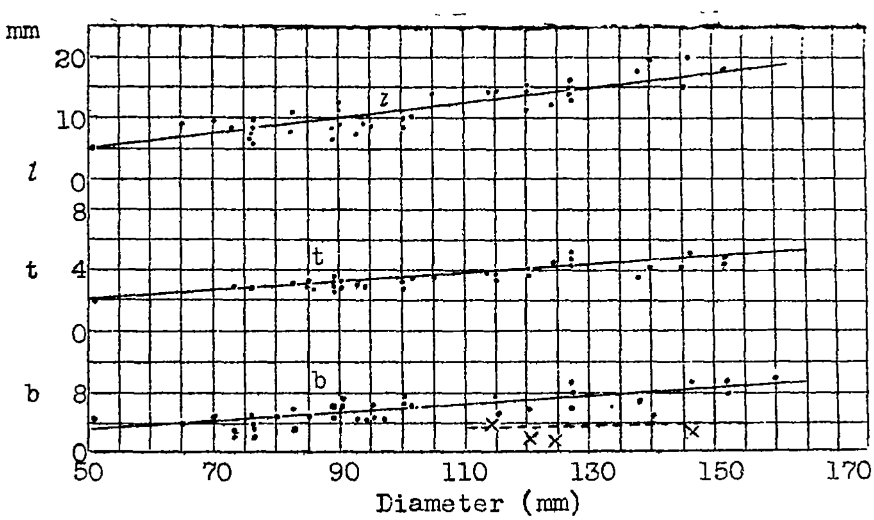


Figure 65.

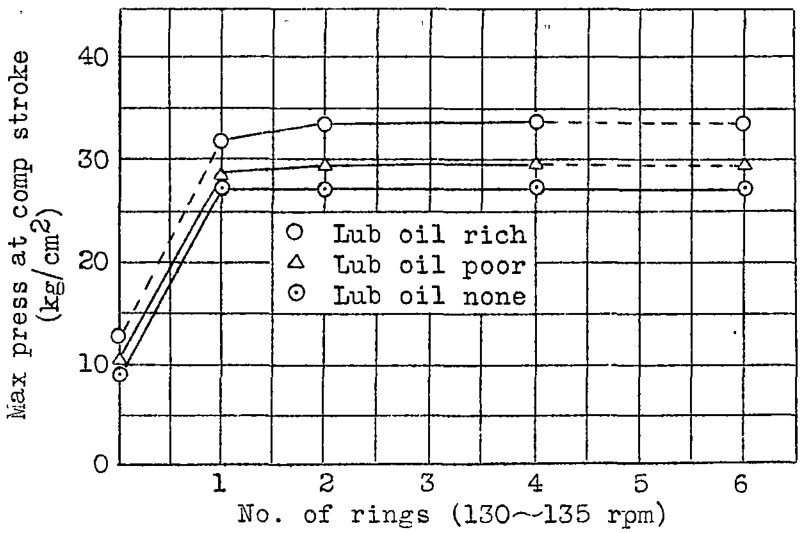


Figure 61.

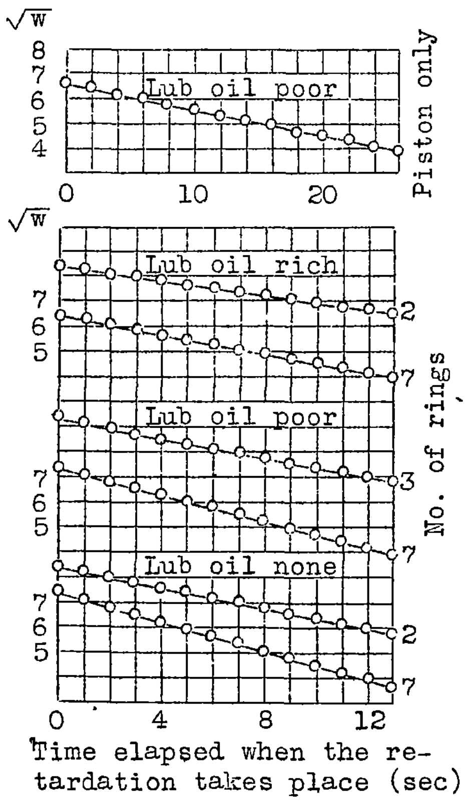
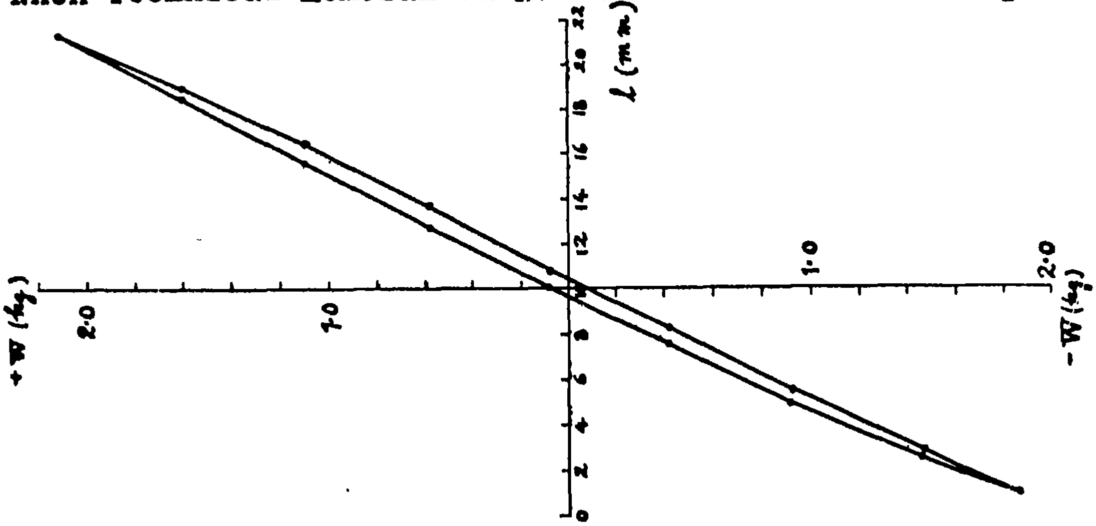


Figure 62.


+W (kg)
-W (kg)
l (mm)
2.0
1.0
1.0
2.0
0
2
4
6
8
12
14
16
18
20
22

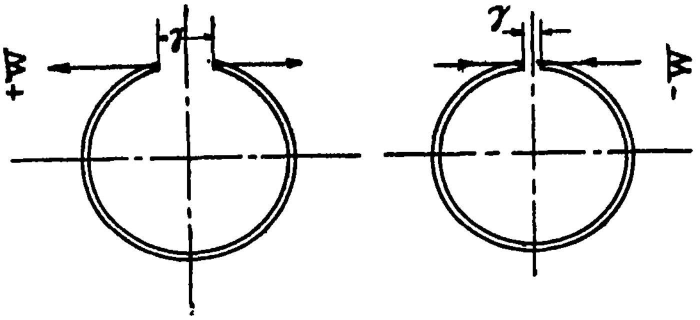

+W
-W
γ
γ

Fig. 64.

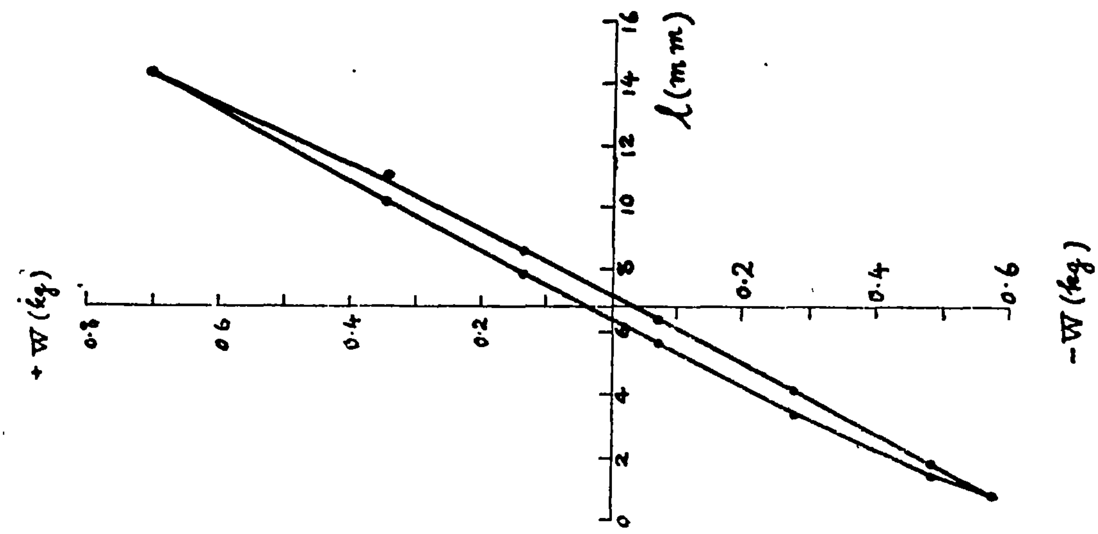

+W (kg)
-W (kg)
l (mm)
0.8
0.6
0.4
0.2
0.2
0.4
0.6
0
2
4
6
8
10
12
14
16

Fig. 63.