THE DAVID W. TAYLOR MODEL BASIN

UNITED STATES NAVY

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OIL-FILM TURNTABLE BEARINGS FOR HEAVY GUN TURRETS

MODEL TESTS

BY R. T. McGOLDRICK

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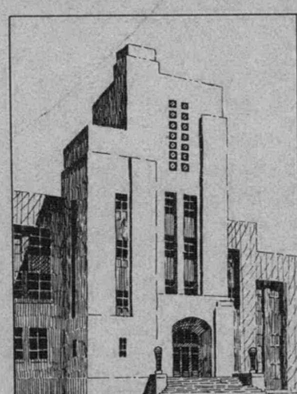

APRIL 1943

REPORT 514

~~RESTRICTED~~

NAVY DEPARTMENT

DAVID TAYLOR MODEL BASIN

WASHINGTON, D.C.

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THE DAVID TAYLOR MODEL BASIN

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This development has been in progress since early in 1938, having been started at the suggestion of Captain E.F. Eggert, (CC),USN,(Ret). Since Captain Eggert's retirement it has been under the direct supervision of Captain H.E. Saunders, USN, and Lieutenant Commander R.D. Conrad, USN. The tests on the first model were carried out by J. Vasta, formerly a member of the U.S. Experimental Model Basin staff. The experimental work since July 1938 and the preparation of this report have been carried out by R.T. McGoldrick. The digest is the work of Captain H.E. Saunders, USN.

DIGEST*

Since the invention of the pivot gun and the rotating gun turret, the conventional method of support for the rotating members has been a partial or complete ring of balls or rollers on a circular path.

In recent years there have been definite signs that unless both the design and the materials could be improved, the double-flanged tapered roller resting between two conical roller paths, now standard in the United States Navy, could no longer be considered an acceptable device for supporting a heavy major-caliber turret.

Many years ago Captain E.F. Eggert, (CC),USN, at that time Director of the U.S. Experimental Model Basin, suggested that major-caliber turrets in new capital ships be supported on circular bearings in which the metallic surfaces would be kept separated by an oil film maintained by a pressure supply of the lubricant. The use of a moderate oil pressure, he argued, would be sufficient to support loads considerably greater than any then in prospect on rings of the general dimensions of the roller paths then in use. The complete separation of the metallic surfaces would eliminate, once and for all, the problems involving concentrated loads of high intensity.

In the oil-flotation type of bearing the greater part of the area of one part of the bearing is recessed, and this recess, or lake as it is called, is surrounded by a relatively narrow land which fits closely to the other part of the bearing, as shown in Figure 1, reproduced in this digest. The oil is pumped into the recess in one of the bearing surfaces, and sufficient pressure is maintained to force the surfaces apart, and to keep the oil flowing continuously over the lands at the edges of the recess.

The development of the oil-film bearing for the support of rotating turrets has so far taken place in three stages. The first stage consisted of experiments with a "sandwich" arrangement in which a cast-iron block slid between two steel blocks fitted to receive oil under pressure and forced together in a hydraulic testing machine. The force required to draw the slider through between the steel blocks was measured by a Chatillon spring dynamometer while oil was being pumped into the two outer members. The general arrangement of the test apparatus is shown in Figure 2 on page 4, and full details of the test are given in Appendix 1, beginning on page 23.

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* This digest is a condensation of the text of the report, containing a description of all essential features and giving the principal results. It is prepared and included for the benefit of those who cannot spare the time to read the whole report.

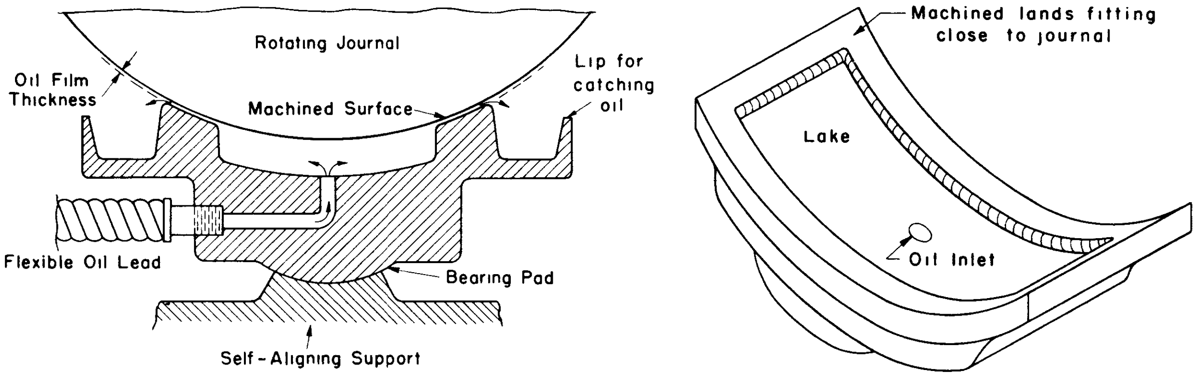

Figure 1 - Schematic Diagram of Oil-Film Bearing

Oil is pumped into the recess or lake in the bearing pad until the total upward pressure exerted by the oil just exceeds the downward load on the journal. The journal then lifts sufficiently to permit an oil film to be established over the entire periphery of the boundary lands and the journal is then free to rotate without metallic contact at any point.

A certain small amount of oil leaks out all the time; it is caught in the lip around the bearing pad and returned to the bearing by the pressure pump. The lip is not shown in the view at the right.

The journal can rotate with equal facility in either direction.

This device had no provision for equalizing the load and for maintaining a definite thickness of oil film over the whole bearing area. Consequently it was found unworkable because the bearing blocks would tilt and make firm metal-to-metal contact on one side while all the oil escaped on the other side.

The second stage consisted of experiments with a model more nearly of the form of a turret, in which there was a cylindrical foundation carrying a lower bearing ring, and a conical member carrying an upper ring. The lower ring was fitted with six recesses or lakes, into which oil was pumped. Around these lakes were lands with a complicated system of oil grooves, as shown in Figure 5, page 6. The general arrangement is given in Figure 3, and some of the details are shown in Figure 4, page 6.

The initial behavior of this second model, the tests of which are described in detail in Appendix 2, beginning on page 27, was unsatisfactory as there was found to be still no provision for equalizing the load. A slight tilting of the cone to one side would cause most of the oil to escape on the other side, with a resulting drop in pressure and a seizing of the bearing.

The problem was solved by restricting the oil supply for each lake to the absolute minimum required for maintaining a proper oil film over the lands around that lake under the maximum load. A plug with a small regulating orifice was fitted in the oil inlet to each lake; this orifice was small enough to permit the passage of adequate oil for the film around that lake,

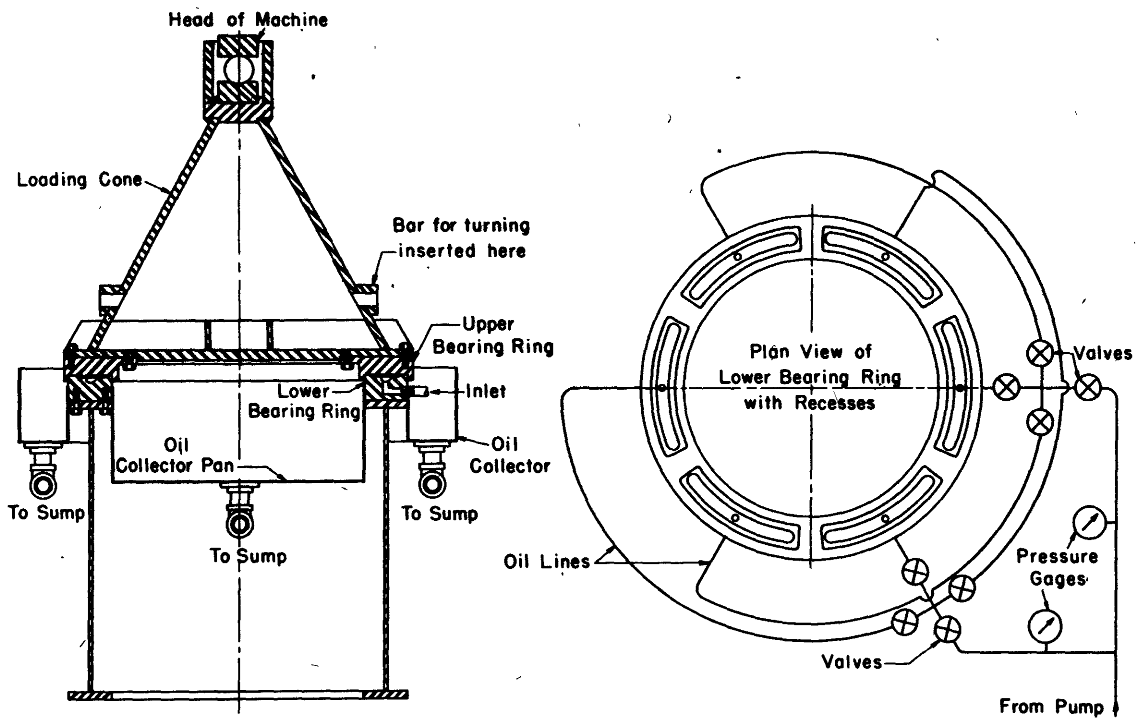

Figure 3 - Sectional Elevation and Oil Supply Arrangement of Second Preliminary Model

The lower fixed bearing ring is supported by a cylindrical shell of 1/4-inch steel plate, which simulates a turret foundation. It has 6 recesses or lakes in its upper surface.

The upper rotating bearing ring has a lower bearing surface which is perfectly flat, and it carries projections which fit over the edges of the lower bearing ring. It is loaded through a steel cone with a hardened circular plate at the top. Load is applied from the testing machine through a 3-inch steel ball which permits rotation of the upper bearing member when a downward force is exerted upon it.

but small enough so that if the upper part of the bearing lifted over that lake, not enough oil was lost to reduce the pressure on the other lakes to the point where each could no longer carry its own load.

The second model, after this change, would lift with full load when oil pressure was applied to the lakes, and it would rotate easily without the slightest sign of tilting or seizing. There remained, however, the problem of eliminating the friction in the loading ball, simulating the rolling and listing of a ship, and testing the model under a transverse load such as that due to gun recoil.

The problem was reformed by drawing up a full set of specifications and by preparing a preliminary mechanical design for a ship installation of this kind. These specifications are given in full on page 11 and the preliminary design is shown in Figure 6 on page 9 of the report.

The third model was much more representative of the full-scale bearing than the first two, and it was capable of demonstrating many more of the

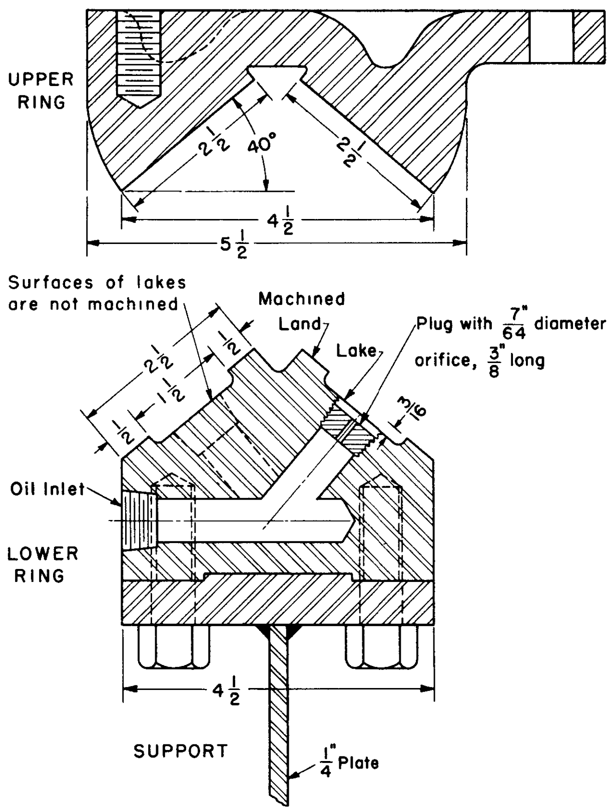

Figure 7 - Vertical Transverse Sections through Upper and Lower Bearing Rings of Third Model

For this model both the upper and lower rings were of cast steel. If scoring or dragging were to develop, it was considered that it would show up clearly with this combination of metals.

ship requirements. To take the horizontal recoil load, which is over twice as great as the vertical gravity load, and to take the horizontal component of the gravity load when the turret is rolling or is tilted, the lower bearing was made up of two surfaces forming a V-shaped section, as shown in Figures 7 and 8.

To demonstrate that the training of a turret mounted on a bearing of this type could be accomplished under the transverse loads due to rolling and listing, the weight of the entire full-scale turret was simulated by metal weights. The whole turret model, with the oil-film bearing, was mounted on a framework which could be tilted statically up to 30 degrees and rolled dynamically up to 10 degrees on either side. The stool or foundation carrying the lower bearing ring was mounted on a frame supported in trunnion bearings, as shown in Figure 10.

As for the third model, it was found that starting the pump and admitting oil to the lakes would lift the load and establish the oil film without difficulty. Despite the high pressure used, up to and exceeding 400 pounds per square inch, and the relatively narrow lands, 1/2 inch wide, the oil did not squirt from the clearance spaces, but simply ran down the vertical outer surface of the lower bearing ring as it would from an overflowing oil container.

With the axis of the turret model vertical and an oil pressure in the header not exceeding 300 pounds per square inch, the model could be rotated easily. The coefficient of friction was of the order of 0.000001. When the turret was listed to about 10 degrees it was necessary to increase the header pressure to about 600 pounds per square inch to rotate it. The explanation for this difficulty will be found in the diagram, Figure 13, on page 16.

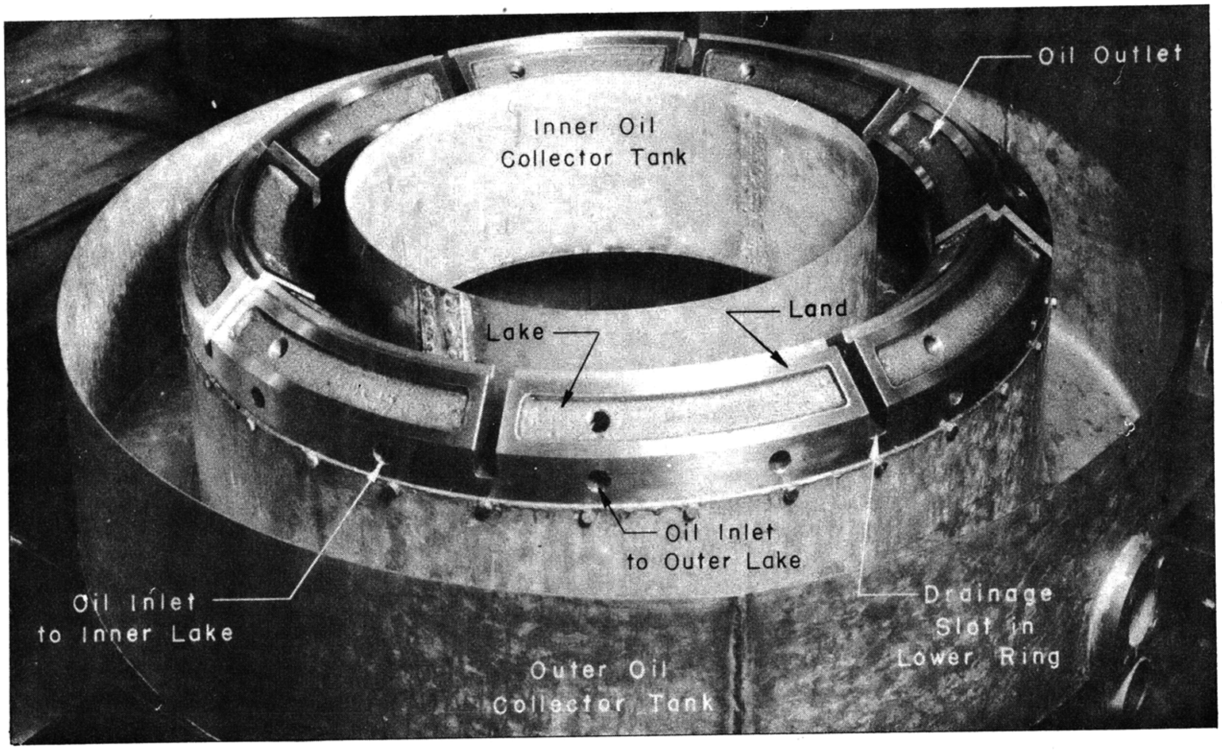

Figure 8 - Lower Bearing Ring of Third Model

The lower ring of the model is cast in one piece with transverse slots representing the spaces between the full-scale segments. There are only 16 lakes instead of 64. Note that the depressed surfaces of the lakes are not machined.

This 10-degree list is probably close to the maximum at which major-caliber guns on a capital ship could be kept firing.

It was found possible to train this model by hand for an oscillation or roll of 10 degrees to each side at a frequency of 7 cycles per minute, corresponding to a rolling period of about 8.6 seconds. This gave approximately the same angular acceleration as would be developed by a 30-degree roll at a 15-second period on a battleship. The average coefficient of friction for this condition was about 0.003, which is normal for an oil-lubricated plain bearing of orthodox design.

The thickness of the oil films for all model tests was about 0.003 to 0.008 inch. It is expected that the oil film on the full-scale installation would be of the same order of thickness.

The development described in this report has shown that an oil-film bearing at least has possibilities as a support for turrets of the heaviest type. Naturally such a radical change from present design practice would require much further experimentation than has been carried out so far.

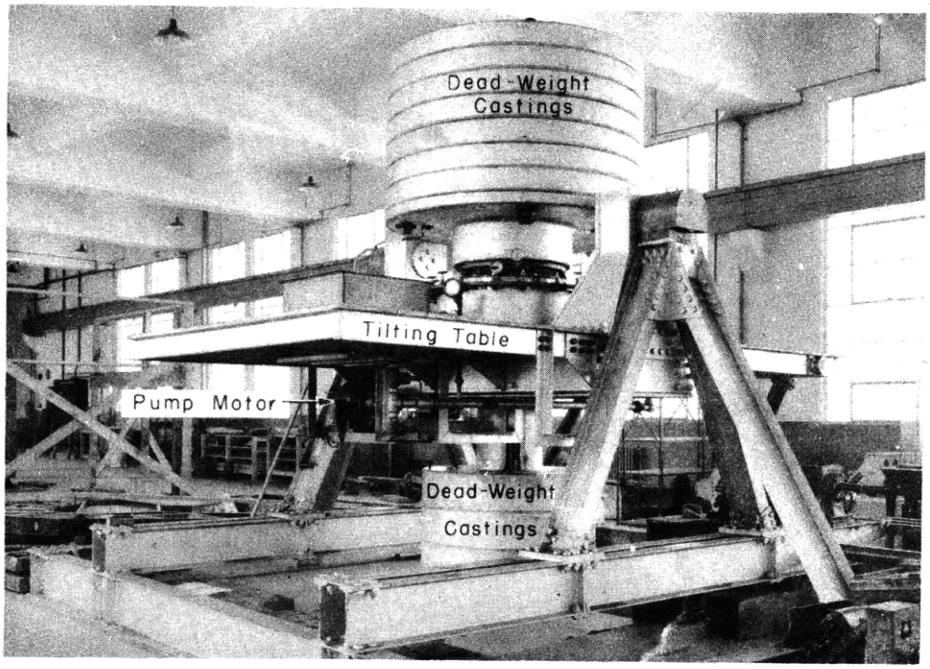

Figure 10 - Third Model Assembled Ready for Test

80,000 pounds in the form of castings bolted together is carried by the bearing to represent the dead weight of the turret. The weights are disposed vertically to place the center of gravity of the assembly in the proper position relative to the bearing.

The chief questions yet to be answered are:

1. whether machining such a bearing after installation on the ship is feasible
2. whether a bearing of this type can withstand the most severe conditions of transverse loading on a combatant vessel
3. whether the fire hazard can be overcome, and
4. whether the lubrication system and the bearing surfaces can be adequately protected against damage in battle.

A rather extensive discussion of these phases of the problem will be found on pages 17 to 20 of the report.

Because of the lack of trained personnel and the low priority assigned, this work has been at a standstill for a year or more past, but the report has been prepared to describe the progress to date, and to bring forth comment and suggestions which it is hoped will aid in a final solution to the problem.

It is planned to continue the work as opportunity affords, and in the not distant future to design an installation which can be tested on a medium-caliber mount on an active naval vessel.

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ABSTRACT

To meet the need for a type of rotatable support for heavy gun turrets of combatant vessels superior to the present double-flanged rollers and roller paths, the practicability of using an oil-film bearing for this service has been investigated.

Three models have been built and tested, to about 1/8 scale, in which the bearing surfaces are separated by a film of oil delivered to the bearing under pressure. When definite separation of the metallic surfaces can be assured, the coefficient of friction is of the order of 1×10^{-6}, and the quantity of oil required is moderate.

The third bearing model was loaded by dead weights simulating the actual turret, and was arranged for rolling and listing to simulate ship operation. By the use of an oil pressure of 600 pounds per square inch it was possible to rotate the turret model assembly with a list of 10 degrees or with a roll of 10 degrees to either side.

While a full solution of the problem has not been achieved, the development shows promise, and a program of future research is briefly outlined.

INTRODUCTION

Since the invention of the pivot gun and the rotating gun turret, the conventional method of support for the rotating members has been a partial or complete ring of balls or rollers on a circular path (1) (2).*

In the course of development of the armored steel fighting ship, however, design of the roller assembly and the tracks has barely kept pace with the increase in weight of the rotating turret and the increased recoil loads. The universal use of director firing systems has brought with it the necessity for maintaining the plane of rotation within accurate limits throughout the arc of train. Indeed, in recent years there have been definite signs that unless both the design and the materials could be improved, the double-flanged tapered roller resting between two conical roller paths, now standard in the United States Navy, could no longer be considered an acceptable device for supporting a heavy major-caliber turret.

In any roller design the crushing loads are naturally very great owing to the theoretically linear contact between the roller and the track. The chief cause of high loading, however, is not the dead weight of the turret but the horizontal recoil load due to gunfire, which must be taken up by

* Numbers in parentheses indicate references on page 20 of this report.

the roller flanges. The resulting tilt of the rollers puts heavy crushing loads on the outer edge of the lower track at the rear and on the inner edges of the lower track at the front.

Many years ago Captain E.F. Eggert, (CC),USN, at that time Director of the U.S. Experimental Model Basin, suggested that major-caliber turrets in new capital ships be supported on circular bearings in which the metallic surfaces would be kept separated by an oil film maintained by a pressure supply of the lubricant. The use of a moderate oil pressure, he argued, would be sufficient to support loads considerably greater than any then in prospect on rings of the general dimensions of the roller paths then in use. The complete separation of the metallic surfaces would eliminate, once and for all, the problems involving concentrated loads of high intensity.

The proposal was so radical and so untried that for a long time it did not progress beyond the idea stage. However, difficulties with the roller path designs of the post-war U.S. battleships, and the adoption of pressure oil-film bearings for supporting certain heavy parts of the 200-inch telescope mounting on Mount Palomar (3) renewed interest in this type of bearing as a turret support. Early in 1938 Captain Eggert initiated a series of experiments looking to its use for this purpose.

The type of bearing under consideration has been variously described as a pressure-type plain bearing, or as an oil-film or oil-flotation bearing. The essential feature of a bearing of this kind is that complete physical separation of the metal surfaces is maintained, with an oil film in between, whenever it is desired to have the bearing turn. This condition is sought after in any type of plain bearing but is not usually attained, at least not for the whole area of the bearing, and for all speeds.

In the oil-flotation type the greater part of the area of one part of the bearing is recessed, and this recess, or lake as it is called, is surrounded by a relatively narrow land which fits closely to the other part of the bearing, as shown in Figure 1. The oil is pumped into the recess in one of the bearing surfaces, and sufficient pressure is maintained to force the surfaces apart, and to keep the oil flowing continuously over the lands at the edges of the recess. Obviously the distinction between this bearing and a plain bearing with oil grooves and forced circulation is not very sharp. In the new type of bearing the oil film is maintained more or less uniformly over the whole area by oil pressure and oil supply, whereas in an orthodox plain bearing, the supply of oil to the space between the journal and the bearing and the pressure to maintain the oil film is derived from the motion of the journal in the bearing.

Two of the chief distinguishing features of the oil-film type of bearing under discussion are 1, that in its present stage of development it

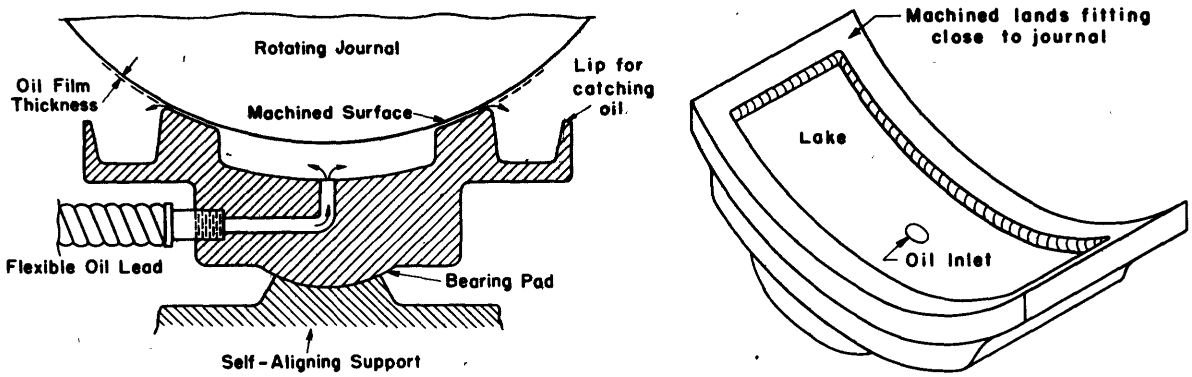


Figure 1 - Schematic Diagram of Oil-Film Bearing

Oil is pumped into the recess or lake in the bearing pad until the total upward pressure exerted by the oil just exceeds the downward load on the journal. The journal then lifts sufficiently to permit an oil film to be established over the entire periphery of the boundary lands and the journal is then free to rotate without metallic contact at any point.

A certain small amount of oil leaks out all the time; it is caught in the lip around the bearing pad and returned to the bearing by the pressure pump. The lip is not shown in the view at the right.

The journal can rotate with equal facility in either direction.

is essentially a low-speed bearing, and 2, that no special bearing materials are required, such as babbitt, bronzes, or brasses. Since no metal-to-metal contact exists when there is relative motion, any two materials that can stand the internal stress can be used. As will be pointed out later, however, it is preferable to use materials of good wearing qualities, that will not gall in case the oil pressure is lost and the surfaces come into metallic contact while they are still moving relative to one another.

It is obvious that in any form of plain bearing no such concentration of load exists as is to be found on a roller path, and that consequently better wearing qualities are to be expected of plain bearings under shock load. Furthermore, no locking device is needed for a turret carried by an oil-film bearing when the oil supply is shut off. Finally, it is to be expected that the rotating friction will be even less than that encountered in turning roller-bearing structures.

GENERAL DEVELOPMENT

The development of the oil-film bearing for the support of rotating turrets has so far taken place in three stages. The first stage consisted of experiments with a "sandwich" arrangement in which a cast-iron block slid between two steel blocks fitted to receive oil under pressure and forced together in a hydraulic testing machine. The force required to draw the slider through between the steel blocks was measured by a Chatillon spring dynamometer while oil was pumped into the two outer members. The general arrangement

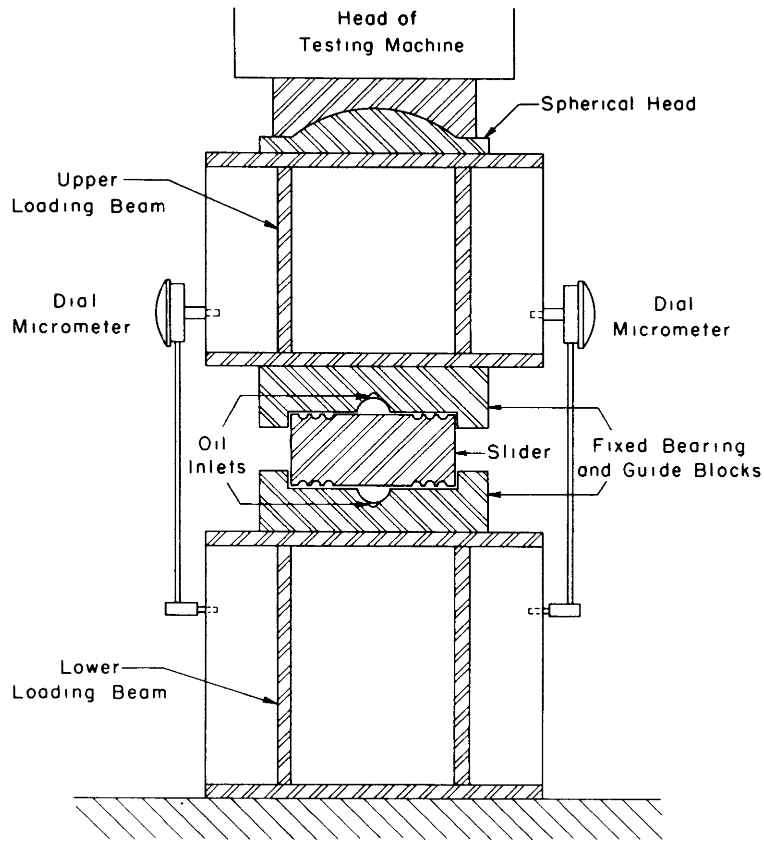


Figure 2 - Sectional View through Slider Type Bearing
used on First Preliminary Model

In this design, it is intended that the oil be delivered to the recesses near the inlets, thence that it be forced out through the narrow clearance spaces shown. The grooves are intended to restrict the quantity of oil passing through.

The separation between the upper and the lower bearing blocks is an indication of the double oil-film thickness. This is measured by four dial micrometers, reading to 0.0001 inch, at the four corners of the guide blocks.

The limit of longitudinal travel of the slider in this setup is 4 inches.

of the test apparatus is shown in Figure 2, and full details of the test are given in Appendix 1, beginning on page 23.

This device had no provision for equalizing the load and for maintaining a definite thickness of oil film over the whole bearing area. Consequently, it was found unworkable because the bearing blocks would tilt and make firm metal-to-metal contact on one side while all the oil escaped on the other side.

The second stage consisted of experiments with a model more nearly of the form of a turret, in which there was a cylindrical foundation carrying a lower bearing ring, and a conical member carrying an upper ring. The lower ring was fitted with six recesses or lakes, into which oil was pumped. Around

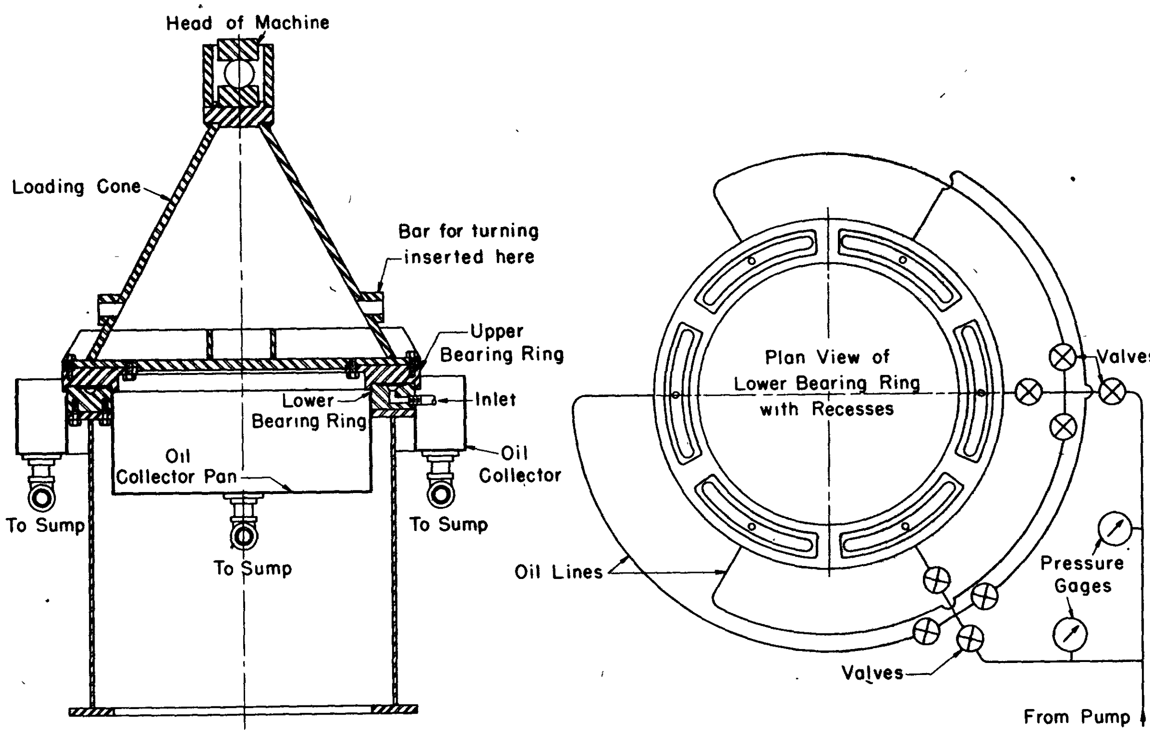


Figure 3 - Sectional Elevation and Oil Supply Arrangement of Second Preliminary Model

The lower fixed bearing ring is supported by a cylindrical shell of 1/4-inch steel plate, which simulates a turret foundation. It has 6 recesses or lakes in its upper surface.

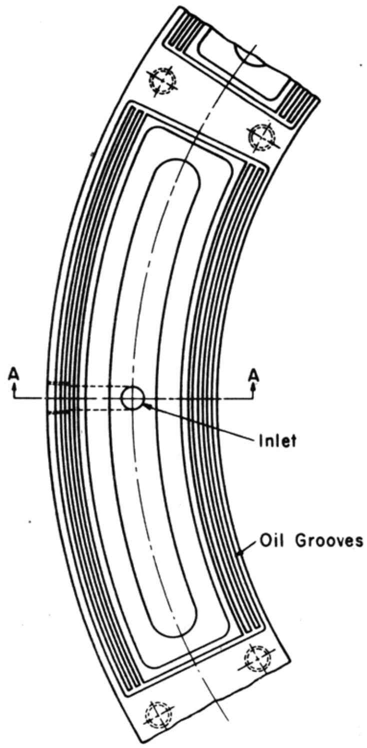
The upper rotating bearing ring has a lower bearing surface which is perfectly flat, and it carries projections which fit over the edges of the lower bearing ring. It is loaded through a steel cone with a hardened circular plate at the top. Load is applied from the testing machine through a 3-inch steel ball which permits rotation of the upper bearing member when a downward force is exerted upon it.

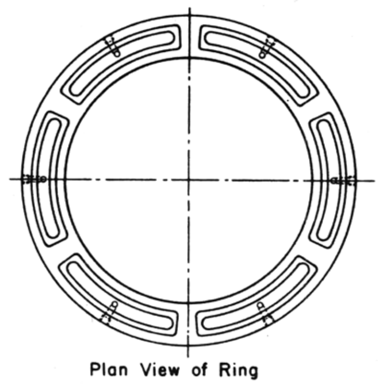
these lakes were lands with a complicated system of oil grooves, as shown in Figure 5, page 6. The general arrangement is given in Figure 3, and some of the details are shown in Figure 4, page 6.

This model was placed under a hydraulic testing machine and the load was applied through a steel ball at the top of the cone carrying the upper ring. This permitted rotation of the upper ring and an approximate measurement of the friction torque.

The initial behavior of this model, tests of which are described in detail in Appendix 2, was unsatisfactory as there was found to be still no provision for equalizing the load. A slight tilting of the cone would cause most of the oil to escape on one side with a resulting drop in pressure and a seizing of the bearing on the opposite side.

As it could not be expected that a turret would ever bear uniformly upon its support, certainly not with the ship rolling or listed, some solution had to be devised for maintaining an oil film of definite minimum





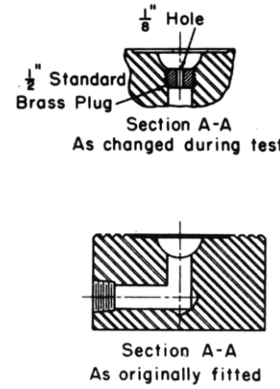


Figure 4 - Details of Lower Bearing Ring of Second Preliminary Model

A general view of this ring is shown in the photograph, Figure 5.

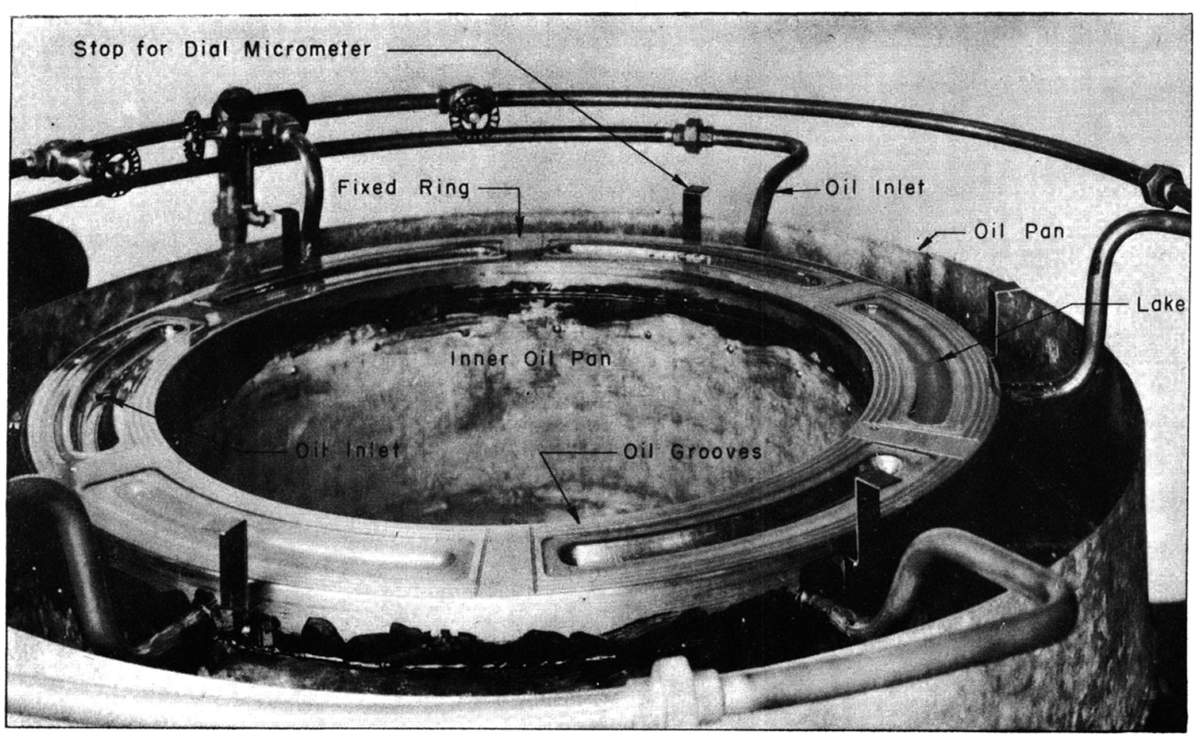


Figure 5 - Lower Bearing Ring of Second Preliminary Model

This view shows the recesses or lakes, the oil grooves surrounding them, and the high pressure valves and piping used to supply oil to the six segments of the bearing.

thickness under these conditions. The problem was solved by restricting the oil supply to each lake to that required for maintaining a suitable oil film over the lands around it under the maximum load. This could be accomplished by insuring a sufficient supply of oil at an adequate pressure to lift the upper part of the bearing and to keep an oil film under it.

The practical solution was to insert a plug with a small orifice in the oil inlet to each lake. This orifice was large enough to permit the passage of adequate oil for the film around that lake, but small enough so that if the upper part of the bearing lifted over that lake, not enough oil was lost to reduce the pressure in the other lakes to the point where each could not carry its own load. The modification is shown in the right hand portion of Figure 4.

With this modification the second preliminary model would lift with the full load when oil under pressure was applied to the lakes, and it would rotate easily without the slightest sign of tilting or seizing. There remained, however, the problem of eliminating the friction in the loading ball, of simulating the listing and rolling of a ship, and of testing the model under a transverse load such as that due to gun recoil.

Before the design of the third model was undertaken, the problem was re-studied and reformed. A set of detail requirements which the final full-scale design had to meet was prepared and a full-scale preliminary mechanical design was developed. The full-scale requirements follow.

REQUIREMENTS FOR FULL-SCALE TURRET BEARING

An oil-film bearing, to be acceptable for installation in a combatant naval vessel, must meet the following specifications:-

1. It shall support the dead weight of the turret or other rotating gun structure and shall at the same time withstand the recoil and shock loads encountered when the guns fire or when the turret is hit by a projectile which does not otherwise put it out of commission.

2. It shall support the turret and permit it to turn when the ship is rolling within the limits at which the guns can otherwise be fired.

3. It shall have a coefficient of friction no higher than that of the best roller bearing.

4. It shall be no more vulnerable to derangement by gunfire or shock or routine service operation than the best roller bearing.

5. It shall add no appreciable fire risk to that already existing in and around the turret.

6. The useful life of the bearing, without repairs to or replacement of any parts which cannot be handled by the forces afloat, shall be at least 20 years of continuous normal operation.

7. The overall weight of the installation, and the useful space occupied, shall be no greater than for the best roller bearing installation.

8. It shall be possible for the forces afloat to examine all bearing surfaces, when this is desired, and to make all necessary minor repairs.

9. The machining tolerances and requirements shall be no more difficult, time-consuming or expensive than those now laid down for roller type bearings.

10. The power requirements shall be moderate; they shall, if practicable, be taken from that now supplied to the training gear.

So much for the general ship requirements. The specific design requirements for the new type of bearing were no less exacting or extensive. They are listed here in some detail, because an understanding of them is necessary to appraise the value of the development work that has been accomplished.

1. Provision must be made for steady and for suddenly applied side loads as well as for down loads.

2. The lower bearing ring must be in sections or segments, to permit alignment and renewal.

3. To meet the vulnerability requirement, there must be at least two oil pumping systems so that it will be possible to rotate the turret if one oil pump is not functioning.

4. Oil must not be present in the sumps in large quantities, and it must not be permitted to overflow if the ship rolls or lists.

DEVELOPMENT AND TESTS OF THE THIRD MODEL

After these specifications and requirements were developed, the next step was to work up, in preliminary design form, an installation representing that in a battleship, so that the third model could be more representative of the full-scale bearing, and so that compliance with more of the ship requirements could be demonstrated.

The resulting arrangement, shown in Figure 6, was developed around a modern roller bearing design, retaining the turret foundation stool, pan

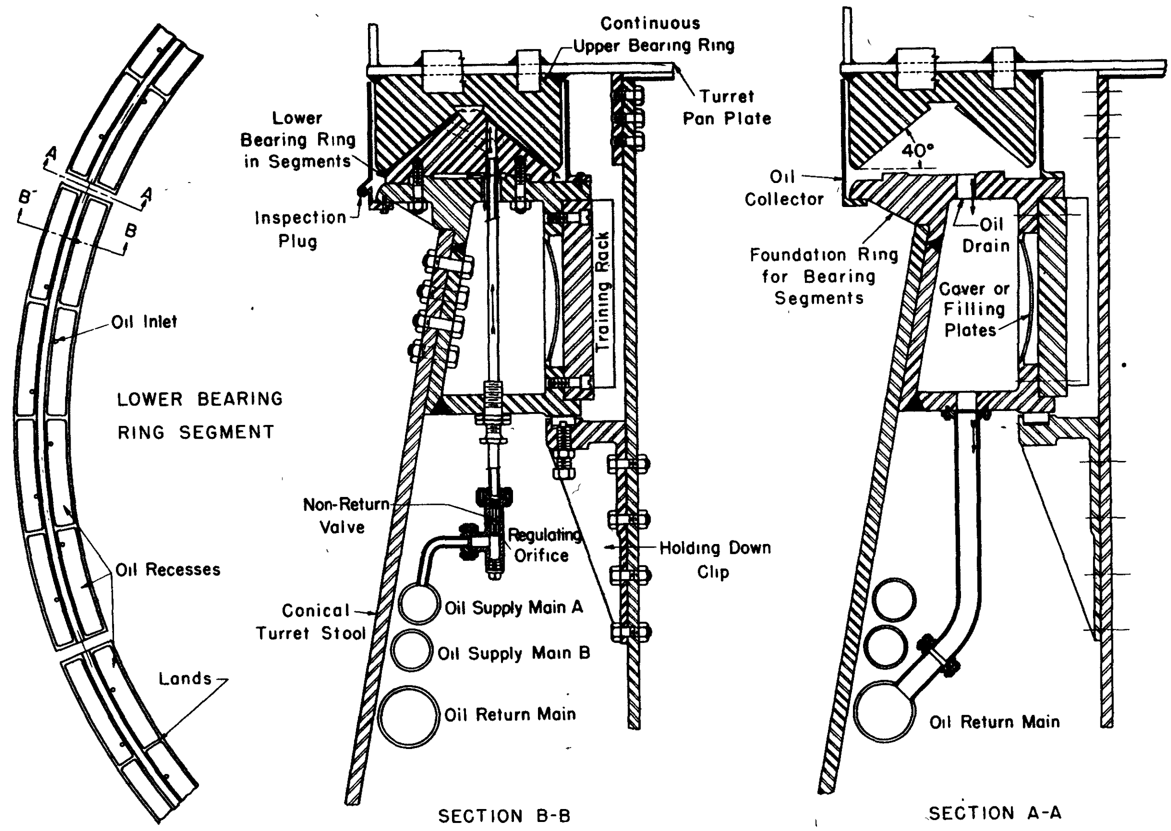


Figure 6 - Arrangement of Proposed Full-Scale Oil-Film Turret Bearing

At the left there is shown one of the proposed eight lower bearing ring segments, with portions of the adjacent segments, taken from TMB plan A-1387.

To take the recoil, rolling and listing loads, the lower bearing ring is shaped like a roof, with separate bearing surfaces and separate series of recesses on the two sides. Oil leaks off from the lands surrounding these recesses into a groove at the ridge and into collecting gutters at the sides, whence it flows back to the sump.

To permit adjustment of the orifices, they are placed in the pressure lines leading to the recesses, where they can be made accessible.

The vulnerability requirement is met by providing two oil pumps and two pressure mains. Alternate recesses on opposite sides of each lower bearing ring segment are supplied by each main, and oil in one such set of alternate recesses is sufficient to carry the turret.

plate, skirt plate, training rack and other principal features in their existing size and form, although in slightly different positions.*

A continuous circular member of box section, similar in shape to the present lower roller path, is bolted to the top of the conical foundation. This has a machined projection on its upper surface, over which and to which are bolted the eight segments of the lower bearing ring.

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* It is perhaps well to point out here that this ship arrangement represented what seemed best at the time it was devised, in 1939, and that it will, in the course of development on this project, be superseded by a better design.

To take the horizontal recoil load, which is over twice as great as the vertical gravity load, and to take the horizontal component of the gravity load when the turret is rolling or is tilted, the lower bearing is made up of two inclined surfaces forming a V-shaped section. As a compromise between a normal to the resultant of the two large forces and an angle which would give adequate bearing area in a reasonable space, the upper or bearing faces of the segments of the lower bearing lie at angles of 40 degrees with the horizontal.

The upper ring is of one piece, keyed and welded to the under side of the turret pan plate. The conical bearing surfaces on the under side of this ring are machined perfectly smooth.

The lands on the lower bearing ring are also perfectly smooth, with no grooves such as were incorporated in the first two preliminary models. The lands are relatively narrow, so that by far the greater part of the bearing area is formed solely by the body of oil in the lakes. In fact, there is only enough width in the lands to restrict the leakage to a reasonable amount and to support the turret without excessive unit bearing loads, when the oil delivery is stopped.

To insure reliability, there are two independent high-pressure lines, supplying oil by individual pipes to 64 recesses, 32 connected to each supply main. There is an adjustable orifice and a non-return valve in an accessible position in each inlet line for restricting the flow of oil to each of the recesses in the bearing. The lines from each supply main go alternately to outer and inner recesses or lakes, so that if one supply system fails the distribution of support in the bearing will still be fairly uniform. Additional features of the full-scale design are given in Table 1.

DESCRIPTION OF THIRD MODEL

The next problem was to design a model which would possess as many as possible of the characteristics of the proposed full-scale installation, and which would permit testing the bearing under conditions approximating those to be found in service at sea and in action with the enemy.

The bearing itself is similar in design to that proposed for the full-scale prototype described, but is of considerably simpler construction. The scale ratio is approximately 1 to 12. The lower ring is a single steel casting with transverse slots to represent the gaps between the full-scale segments. The included angle between the upper faces is the same, namely 100 degrees. There are only 16 lakes, instead of 64, and the lands are 1/2 inch wide; those in the prototype are 1 inch wide. The regulating orifice plugs are threaded directly into the outlets in the lakes, which requires

TABLE 1

Detail Design Data

Proposed Oil-Film Turret Bearing for Full-Scale Ship Installation

All figures are for one turret. All data are for the original design of the third model, and some of these data were later modified.

Assumed dead-weight load of turret	1700 tons, acting vertically
Assumed maximum recoil load, including a dynamic factor of 2.0	3800 tons, acting horizontally
Width of lower bearing segments	18 inches overall
Projected width of each lake	6 inches (approximately)
Length of each lake	36 inches (approximately)
Projected area per lake	216 square inches (approximately)
Total number of lakes	64, 8 in each of 8 segments
Total vertically projected lake area	13,800 square inches
Estimated lake pressure required for lift with all lakes functioning*	275 pounds per square inch
Estimated pressure required at pump	370 pounds per square inch
Lake periphery	112 inches (approximately)
Estimated orifice diameter for supply to each lake	3/16 inch
Estimated oil flow, all lakes functioning	180 gallons per minute
Estimated power for pumping at 70 per cent efficiency	50 horsepower
Projected land area, total	4100 square inches (approximately)
Estimated maximum load on lands with oil delivery stopped	930 pounds per square inch (dead weight only) 16,000 pounds per square inch (dead weight plus recoil)
Bearing angle, 40 degrees with horizontal	

* This is calculated on the assumption that only the oil under pressure in the lakes actually carries load. The pressure in the oil film over a land varies from the full lake pressure on the inside to zero at the outside, so that some of the land area is effective in supporting the load.

disassembly of the model and separation of the bearing rings for modification or replacement. Sections through both upper and lower rings are shown in Figure 7 and the lower ring is illustrated in the photograph, Figure 8.

To demonstrate that the training of a turret mounted on an oil-film bearing could be accomplished under the transverse loads due to rolling of the ship and to gun recoil, the weight of the entire full-scale turret was simulated by metal weights. The whole turret model, with the bearing, was mounted on a framework which could be tilted statically up to 30 degrees and

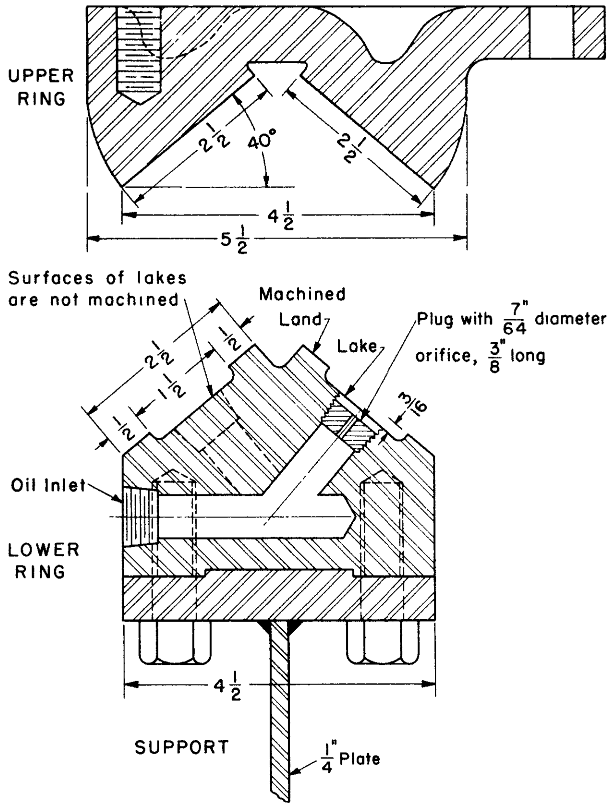

Figure 7 - Vertical Transverse Sections through Upper and Lower Bearing Rings of Third Model

For this model both the upper and lower rings were of cast steel. If scoring or dragging were to develop, it was considered that it would show up clearly with this combination of metals.

rolled dynamically up to 10 degrees on either side. The stool carrying the lower bearing ring was mounted on a tilting frame supported in trunnion bearings, as illustrated in the photographs, Figures 9 and 10. Castings having a total weight of 80,000 pounds were mounted on and suspended from the upper ring to simulate the dead weight of the rotating turret and to bring its center of gravity in the proper vertical position with relation to the bearing.

During a test, oil is supplied under pressure to the lakes by a variable-stroke (Waterbury) pump driven by an electric motor and reduction gear. As shown in the various photographs, Figures 9 to 12, the pump, motor, sump and oil piping are mounted on the tilting frame.

RESULTS OF TEST ON THIRD MODEL

The oil used for the tests was a Navy Contract SAE 50 lubricating oil, but it is possible that a considerably lighter oil might be found acceptable. The orifices in the plugs, where the oil entered the lakes, were 7/64-inch in diameter and 3/8-inch long, as shown in Figure 7.

As for the third preliminary model, it was found that starting the pump and admitting oil to the lakes would lift the load and establish the oil film without difficulty. Despite the high pressures used, up to and exceeding 400 pounds per square inch, and the relatively narrow lands, 1/2 inch wide, the oil did not squirt from the clearance spaces, but simply ran down the vertical outer surface of the lower bearing ring as it would from an overflowing oil container.

With the axis of the turret model vertical and an oil pressure in the header not exceeding 300 pounds per square inch, the model could be rotated easily. The coefficient of friction was found to be about 1×10^{-6}, the same as for the previous test with the circular flat bearing ring.

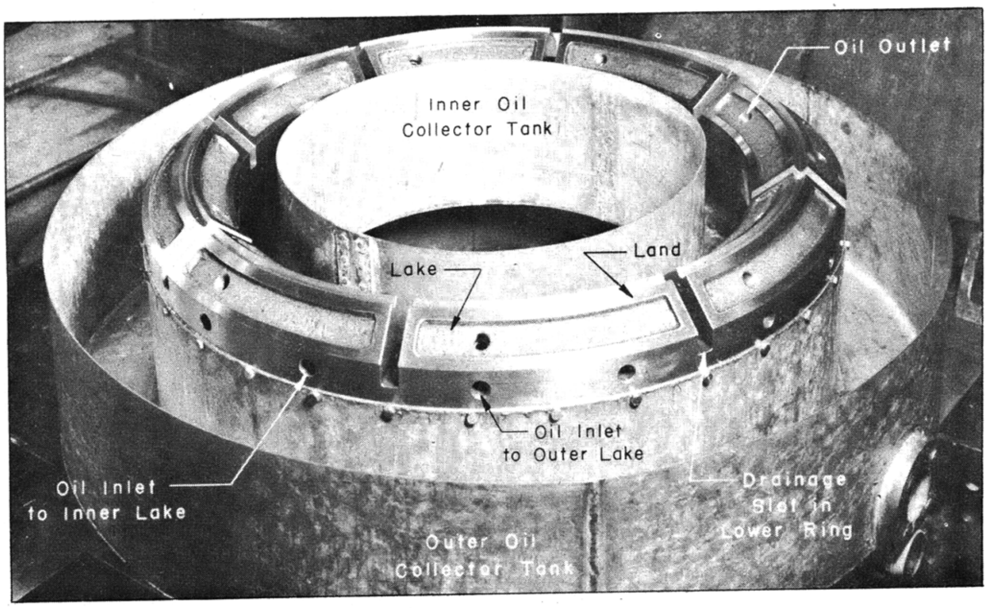

Figure 8 - Lower Bearing Ring of Third Model

The lower ring of the model is cast in one piece with transverse slots representing the spaces between the full-scale segments. There are only 16 lakes instead of 64. Note that the depressed surfaces of the lakes are not machined.

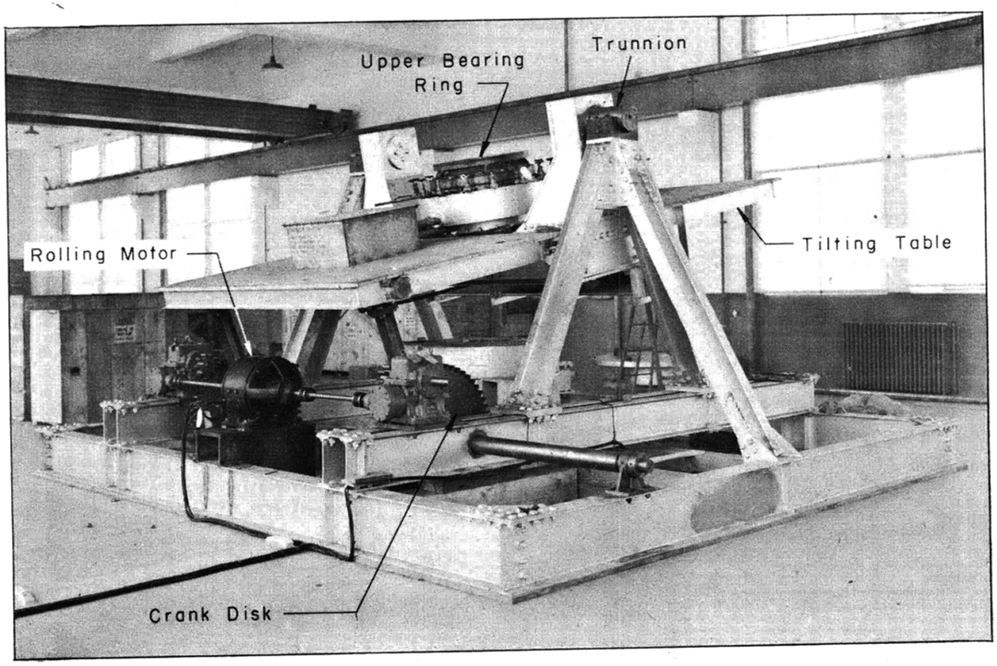

Figure 9 - Tilting Frame for Testing Third Model

The frame may be tilted 10 degrees to each side at a frequency 7 cycles per minute, or statically up to 30 degrees.

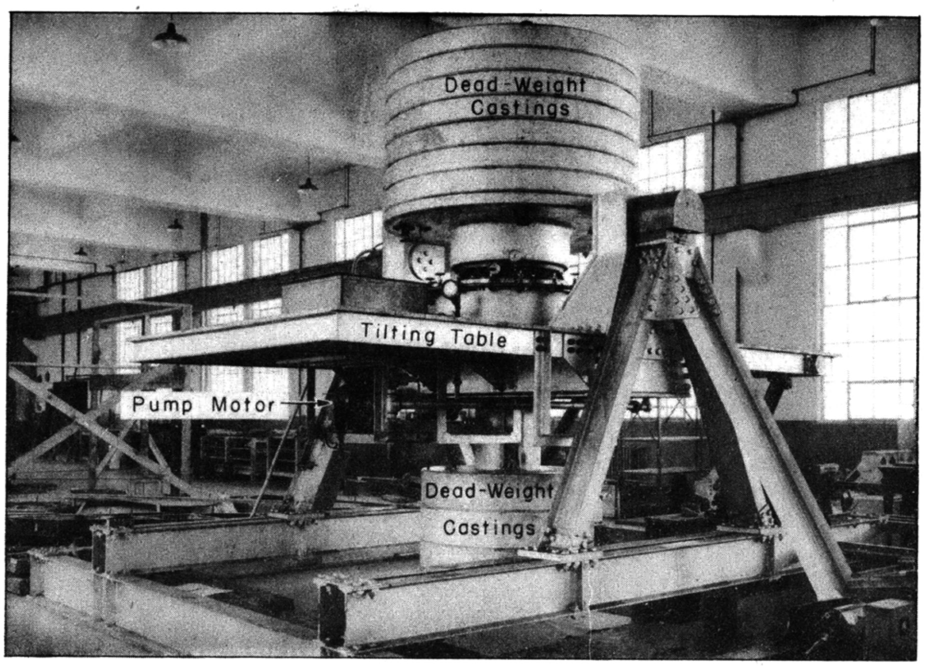

Figure 10 - Third Model Assembled Ready for Test

80,000 pounds in the form of castings bolted together is carried by the bearing to represent the dead weight of the turret. The weights are disposed vertically to place the center of gravity of the assembly in the proper position relative to the bearing.

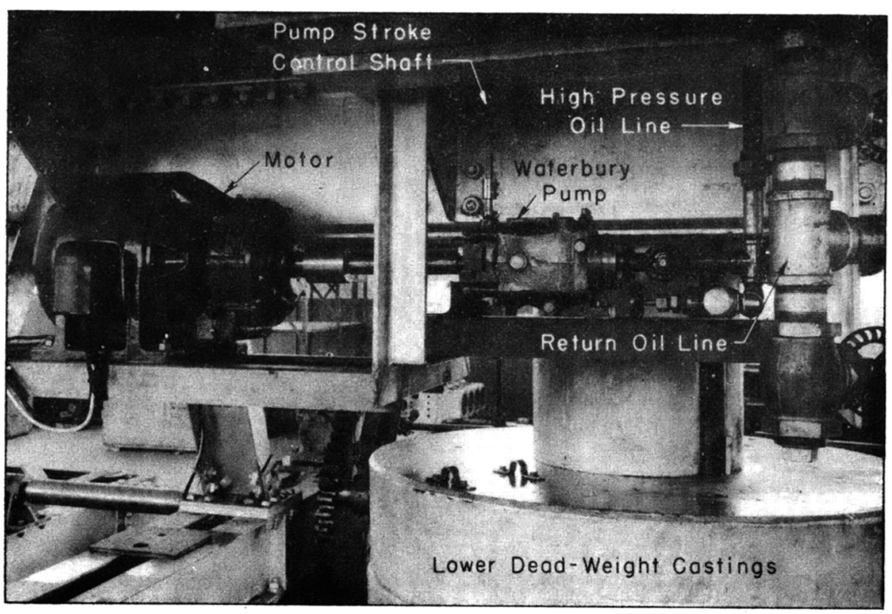

Figure 11 - Variable-Stroke Oil Pump and Motor for Third Model

Continuous circulation of oil to and from the bearing is maintained by a small variable-stroke oil pump, the delivery of which may be controlled from the platform above, as shown in Figure 12. The entire oil supply system is carried by the tilting table.

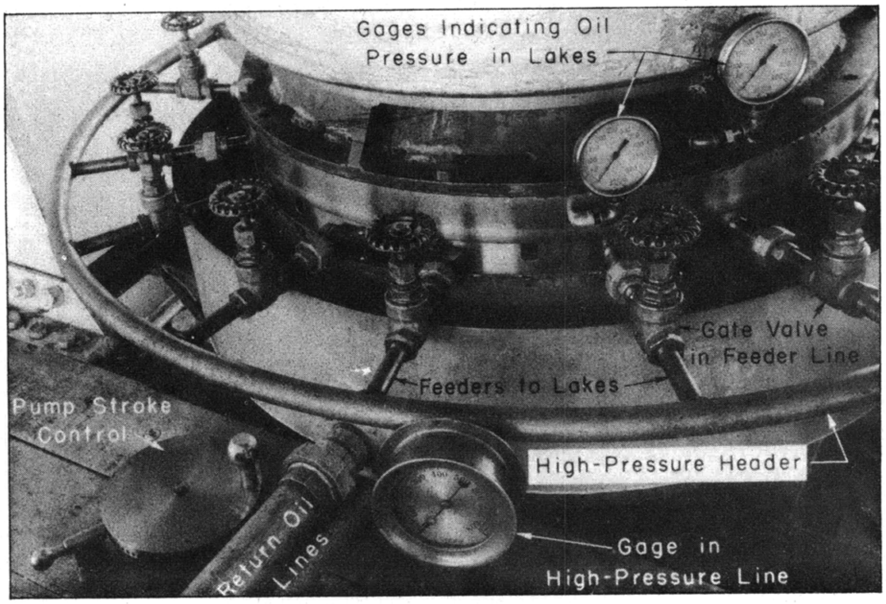

Figure 12 - High-Pressure Oil Supply System of Third Model

The discharge line from the pump leads to a ring-shaped header. There are 16 branches from this header to each of the 16 lakes in the lower bearing ring. Each branch has a gate valve, so that the supply of oil to any lake or combination of lakes can be cut off to simulate a casualty. Large oil return lines lead from the two circular collector tanks to a sump tank on the tilting platform.

Two pressure gages are installed in the upper ring, by which the pressure in any of the lakes can be observed as the model turret is rotated.

However, when the turret was listed, it could not be rotated at an angle with the vertical greater than 5 degrees without increasing the pressure. The reason for this difficulty may be seen from an examination of the diagram in Figure 13, and from its explanatory footnote. At a header pressure of 600 pounds per square inch, it could be rotated with a constant list of about 10 degrees. This is probably close to the maximum list at which major-caliber guns on a capital ship could be kept firing.

It was found possible to train this model by hand for an oscillation of 10 degrees to each side at a frequency of 7 cycles per minute, corresponding to a rolling period of about 8.6 seconds. This gave approximately the same angular acceleration as would be developed by a 30-degree roll at a 15-second period on a battleship. The average coefficient of friction for this condition was about 0.003, which is normal for an oil-lubricated plain bearing of orthodox design.

This fact, combined with a measured lateral movement of the rotating assembly during the roll of about 0.006 inch, indicated that full flotation was not being maintained in the rolling condition, although there was sufficient lubrication to permit continuous training of the turret. When rolling, the pressure in the inner lakes was found to fluctuate between 50 and 450 pounds per square inch. The pressure in the outer lakes remained practically constant at 300 pounds per square inch, corresponding to the pressure when the turret axis was vertical. The explanation of this appears to be that although the rotating member, with the upper bearing ring, slides downhill when the bearing ring is inclined, the fact that the center of gravity of the rotating member is 8 inches above the top of the bearing increases the normal load on the downhill half of the bearing and decreases it on the uphill half. The combined effects of side slip and change in normal load are additive on the inner bearing surfaces but cancel one another on the outer surfaces. Hence the pressure fluctuations are great in the inner lakes and small in the outer lakes.

The thickness of the oil film, when it was maintained, varied from about 0.003 to 0.008 inch. It is expected that in the full-scale installation, using the same oil, and with a land width of the order of 1 inch, the oil film thickness would be approximately the same.

Although both upper and lower bearing surfaces of the third model are of cast steel, and might be expected to show some galling if the bearing were vulnerable in this respect, there has been no difficulty of this kind. Experience with the second model indicated, however, that the bearing must be kept free of all foreign matter and

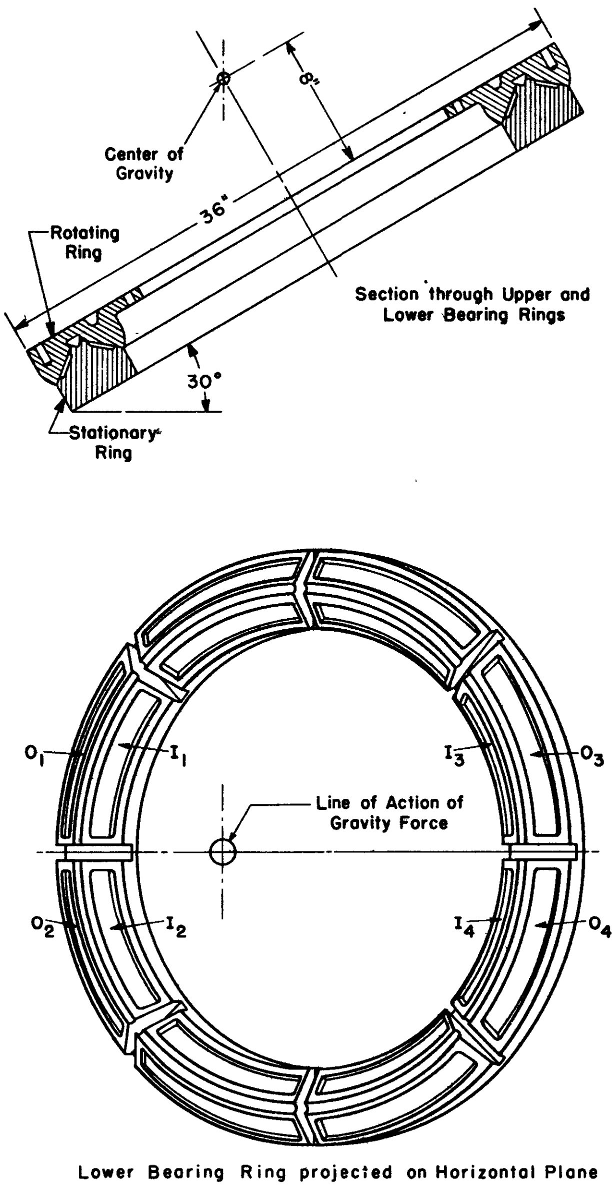

Figure 13 - Diagram of Roof Type Oil-Film Bearing Model in Tilted or Listed Position

Note here that the gravity load is not symmetrical on the lower bearing ring because of the list. Further, the projected area of the outer lakes O_1 and O_2 on the low side is rather small, and they cannot carry their proportionate share of the increased vertical load. This must be compensated for by an increased pressure on the principal load-carrying lakes I_1 and I_2.

that the oil must be carefully strained. Once properly assembled, there should be no difficulty in maintaining both these requirements.

No model experiments have been conducted with bearing surfaces of dissimilar materials, as it has been considered that no special problems were involved here, and that this work should preferably be done in full scale. Experience with equipment now in service indicates that the upper bearing can be made of any type of steel which is weldable and which can be readily machined to a smooth surface. Grinding is indicated for the final machining operation, and this requirement might limit to some extent the usable types of steels. The lower bearing segments could be forged or cast steel blocks, with contact surfaces of bronze* welded in place.

The valve and piping arrangement on the third model was devised to permit a test of the ability of the oil-film bearing to function with the oil supply to alternate lakes cut off, as explained previously on page 12. Theoretically, this involves only an increase of oil pressure of 100 per cent in the remaining lakes, and a stiffness of the structures which will prevent the surfaces adjacent to the no-pressure lakes from coming into contact. As the oil-film thickness may be as small as 0.003 inch, the deformations of the bearing members under this unequal load distribution must be quite small.

Although it is proposed to make the lower bearing ring in entirely separate segments, a reference to Figure 6 on page 9 will indicate that the bending strength of the lower supporting ring in a horizontal plane is not impaired. The box section just below the lower bearing ring segments should give adequate torsional and circumferential rigidity, especially with the assistance of the solid upper bearing ring and the turret pan plate.

PRESENT STATUS OF THE PROBLEM

The development so far has shown that this type of bearing has possibilities as a support for turrets of the heaviest type. Naturally such a radical change from present design practice would require much further experimentation than has been carried out so far, a program scarcely feasible during an emergency unless the need for a better solution is very urgent.

The chief questions yet to be answered are:

1. whether machining such a bearing after installation on the ship is feasible,

2. whether a bearing of this type can withstand the most severe conditions of transverse loading on a combatant vessel,

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* These are similar to the rubbing surfaces on the steel valve plates of hydraulic pumps and motors now in successful use.

3. whether the fire hazard can be overcome, and
4. whether the lubrication system and the bearing surfaces can be adequately protected against damage in battle.

It is believed that the 10-degree rolling at the faster rate used with the third model demonstrates that training could be accomplished under the most severe rolling conditions that would be encountered on a battleship in action. However, to clarify this point a long-range program should include the installation of a training mechanism on the present model to eliminate the danger of hand operation while the model is rolling and to permit a life test.

Some method should also be devised for subjecting the turret model to a heavy shock load while rotating, thus simulating the recoil load. It is quite possible that under these conditions the bearing surfaces would come momentarily into metal-to-metal contact, but if the metals were such that neither of them scarred or dragged, no damage would be done. In fact, some metal-to-metal contact, with its attendant increase in coefficient of friction, might be beneficial in reducing the rotating moment to be carried by the training gear when a wing gun is fired, and in reducing the whip under these conditions. As the recoil loads last only for about 1/4 second on the ship, the complete destruction of lubrication and consequent damage to the bearing is not to be expected.

It seems quite possible that some compromise design between the full-flotation bearing and the ordinary plain bearing would make an oil-film bearing thoroughly reliable, provided that bearing materials and lubricants were used which would guarantee operation during the brief intervals of extra heavy loads.

The question of machinability in the full scale can be settled only by an experimental installation on a combatant vessel. However, some forecasts can be made in the light of the experience gained so far. It has been demonstrated that elaborate grooving or baffling of the lands in the bearing surfaces is not required. Hence the surfaces to be machined are not much more complicated than the present roller paths. It is estimated that the tolerances required to prevent excessive escape of oil from an oil-film bearing might be somewhat closer than those specified for the present methods of machining roller tracks. This objection appears not to be a major one, however, as the subdivision of the lower ring into segments should facilitate the procedure of aligning the lower with the upper ring.

Examination of the upper ring surface could be made between the lower ring segments as the turret is rotated, but no procedure has as yet been devised to remove and to replace any lower ring segment at will.

Further study of the rather complicated oil supply system developed several years ago and shown on Figure 6, page 9, in which there are two separate oil supply mains running completely around the turret stool, and in which there is a correspondingly long oil return line, indicated clearly that it was much too vulnerable. Captain McBride, a former Director of the David Taylor Model Basin, proposed that the segments of the lower bearing ring each be supplied with oil by one or more small, self-contained, motor-driven pumps, so that each of these pumps could draw from a sump surrounding that segment only, and so that the vulnerable supply and return oil mains could be eliminated.

As a development of this method, it appears that all the oil passages in any one segment can be drilled or cored through it, and that the oil pump can be bolted to the segment in such fashion as to eliminate all oil piping whatsoever, by having the pump suction low in the sump, and by making up the pressure joint between the pump discharge and the internal oil passages when the pump is bolted to the segment. A logical continuation of this development is to attach the driving motor to the oil pump housing, or vice versa, so that each pumping unit is a single assembly.

However, inquiries circulated among all the pump manufacturers in the United States, and correspondence with them covering the better part of two years, has failed to bring to light an oil pump which meets all requirements for this service. It may be necessary to design and build a special one.

In the course of development on this project, it has several times been proposed that no effort be made to take the combination rolling, listing and recoil loads on the oil-film bearing, but that the side loads be taken by vertical rollers without flanges, leaving the vertical load only to be taken by the oil.*

To be sure, this modification would in some respects immensely simplify the problem, in that the oil bearing would represent a relatively easy task in design and construction. However, the resulting turret support would be of composite construction, with possibly all the disadvantages of both but certainly not all their advantages. An additional side structure would have to be incorporated to take the horizontal roller thrust and transmit it to the foundation stool, and there would be vertical as well as horizontal bearing surfaces to machine on both the fixed and rotating structures.

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* Gun mounts of this general type, in which the horizontal and vertical loads are taken by two separate sets of rollers, are fitted on the earlier U.S. heavy cruisers of the PENSACOLA and AUGUSTA classes (5).

An alternate design is that represented by some early types of turrets in the French Navy (4), in which the entire vertical rotating turret load was taken by a step bearing (frequently hydraulic) on one of the platform decks low in the ship, while the horizontal load was taken by a set of vertical cylindrical rollers, without flanges, placed at the level of the uppermost deck, just below the turret. In fact, the first turreted vessel, John Ericsson's MONITOR, had in substance the same mechanical method of support and restraint (1) (2).

Theoretically, this arrangement has much to be said for it, but practically it is not so suitable because the foundation between the turret and the step bearing must be essentially conical, and a structure of this type does not lend itself to the utilization of this space for shell platforms, electric decks, ammunition handling spaces, etc., as in modern turret designs.

CONCLUSIONS AND RECOMMENDATIONS

That there is a very definite and real need for a type of heavy gun turret support superior to the double-flanged rollers will be admitted by all who have to do with the design, construction and operation of the vessels carrying these turrets. It is believed that the model tests and the research work described in the foregoing indicate the possibility of developing a simple, sturdy and reliable method of oil-film support, but this end result has not yet been attained.

Work on this project has practically been at a standstill for a year past, because of lack of trained personnel to carry it on, and the low priority assigned to this item. This report has been prepared to describe the progress to date, and to bring forth comment and suggestions which it is hoped will aid in a final solution to the problem. It is planned to continue the work as opportunity affords, and in the not distant future to design an installation which can be tested on a medium caliber mount on an active naval vessel.

REFERENCES

(1) "The Modern System of Naval Architecture," by J. Scott Russell, about 1864, Vol. III.

(2) "Navies of the World," by Lt. E.W. Very, USN, New York, 1880, pages 357 to 360.

(3) "Oil-Pad Bearings and Driving Gears of 200-inch Telescope," by M.B. Karelitz, Mechanical Engineering, July, 1938, pp. 541 to 544. This article contains a general and detail description of the telescope mount bearings, and theoretical and practical explanations of the behavior of the oil-film bearings.

(4) "Der Aufbau Schwerer Geschütztürme an Bord von Schiffen" (Construction of Heavy Gun Turrets on Shipboard), by K. Thorbecke, Jahrbuch der Schiffbautechnischen Gesellschaft, vol. 12, 1911, pp. 133 to 144.

(5) Naval Gun Factory Plan 144301 - 8-inch Training Gear, General Arrangement, Foundation Section.

BIBLIOGRAPHY

For convenient reference there are listed here a few additional sources of information on this subject.

Letter reports, memoranda, notes and correspondence on this project will be found on TMB Confidential File C-S72-1-(3).

There will be found on this file a confidential letter S72-3(1), dated 9 January 1939, from the Director of the U.S. Experimental Model Basin to the former Bureau of Construction and Repair, forwarding a memorandum report entitled "Turret Support by Flotation in Oil," dated September 1938. The latter contained preliminary reports on the first and second models built and tested on this project.

General Specifications for Building Vessels of the United States Navy, 1936, pp. T-1-4, 5 and 6, contain the specifications for turret rollers, roller paths, and the like.

"Studies in Boundary Lubrication," by W.E. Campbell, Transactions of the American Society of Mechanical Engineers, October 1939, pp. 633-641. With this article there is a rather extensive list of references on this subject.

The shop plans that have been drawn for the second and third models are listed here:

TMB Number	Title	Date
A-1344	TURRET FRICTION TEST, Model No. 2, Assembly of	21 April 1938
A-1345 Alt. III	TURRET FRICTION TEST, Model No. 2, Detail of	22 April 1938
A-1346	TURRET FRICTION TEST, Model No. 2, Detail of	25 April 1938

TMB Number	Title	Date
A-1347	TURRET FRICTION TEST, Model No. 2, Detail of	26 April 1938
A-1384	TURRET FRICTION, Model No. 3, Bearing Rings	16 March 1939
A-1385 Alt. II	TURRET FRICTION, Model No. 3, Bearing Rings	16 March 1939
A-1387	PROPOSED OIL FLOTATION BEARING, Full Scale Ship Installation	17 March 1939
A-2038	TURNTABLE FRICTION TEST, Turret Test Assembly	30 August 1939
A-2038a	TURNTABLE FRICTION MODEL, Turret Test Assembly (Preliminary)	18 September 1939
A-2039	TURNTABLE FRICTION MODEL, Turret Test, Half Section and Piping, Elevation Assembly	18 September 1939
A-2040	TURNTABLE FRICTION MODEL, Turret Test, Piping Layout, Plan Assembly	18 September 1939
A-2041 Alt. II	TURNTABLE FRICTION MODEL, Turret Test, Details of	26 September 1939
A-2042 Alt. II	TURNTABLE FRICTION MODEL, Turret Test, Details of	26 September 1939
A-2043	TURNTABLE FRICTION MODEL, Turret Test, Details of	26 September 1939

APPENDIX 1

DESCRIPTION AND TESTS OF THE FIRST PRELIMINARY MODEL

A cast-iron block or slider, 36 inches by 7 inches by 3 inches, with a system of grooves cut at the edges of its upper and lower faces, was fitted between two fixed cast-iron guide blocks which in turn rested between two heavy loading beams. Figure 2 shows the arrangement schematically, and Figure 14 the details of the sliding and guide blocks. The slider is sandwiched between the two fixed guides, which makes two sliding surfaces.

Light machine oil was pumped under pressure directly to the inlet holes of the guide blocks. This oil passed through the spaces between the

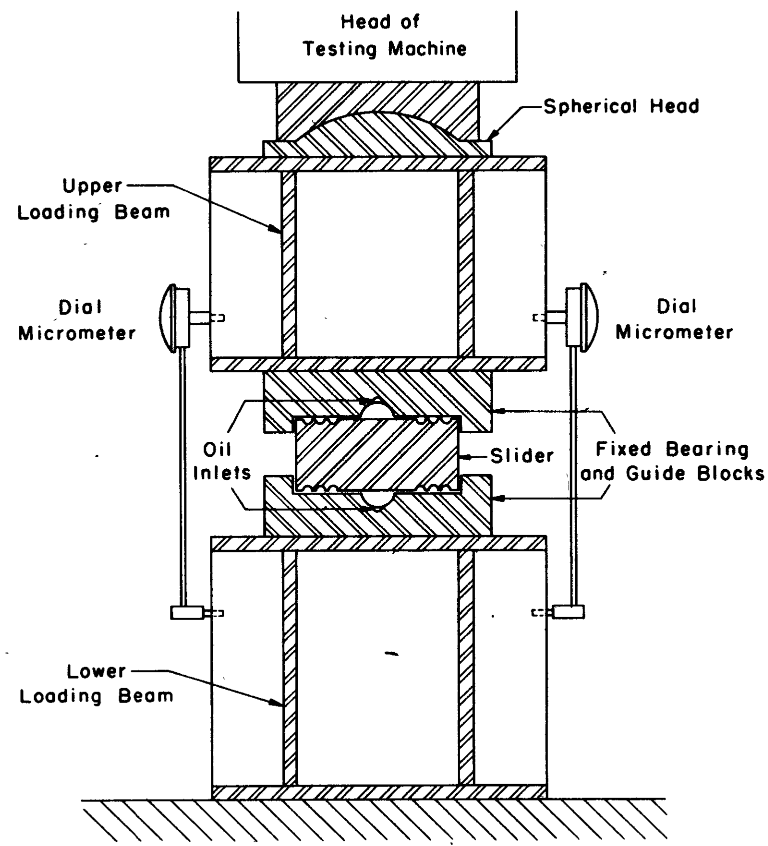

Figure 2 - Sectional View through Slider Type Bearing used on First Preliminary Model

In this design, it is intended that the oil be delivered to the recesses near the inlets, thence that it be forced out through the narrow clearance spaces shown. The grooves are intended to restrict the quantity of oil passing through.

The separation between the upper and the lower bearing blocks is an indication of the double oil-film thickness. This is measured by four dial micrometers, reading to 0.0001 inch, at the four corners of the guide blocks.

The limit of longitudinal travel of the slider in this setup is 4 inches.

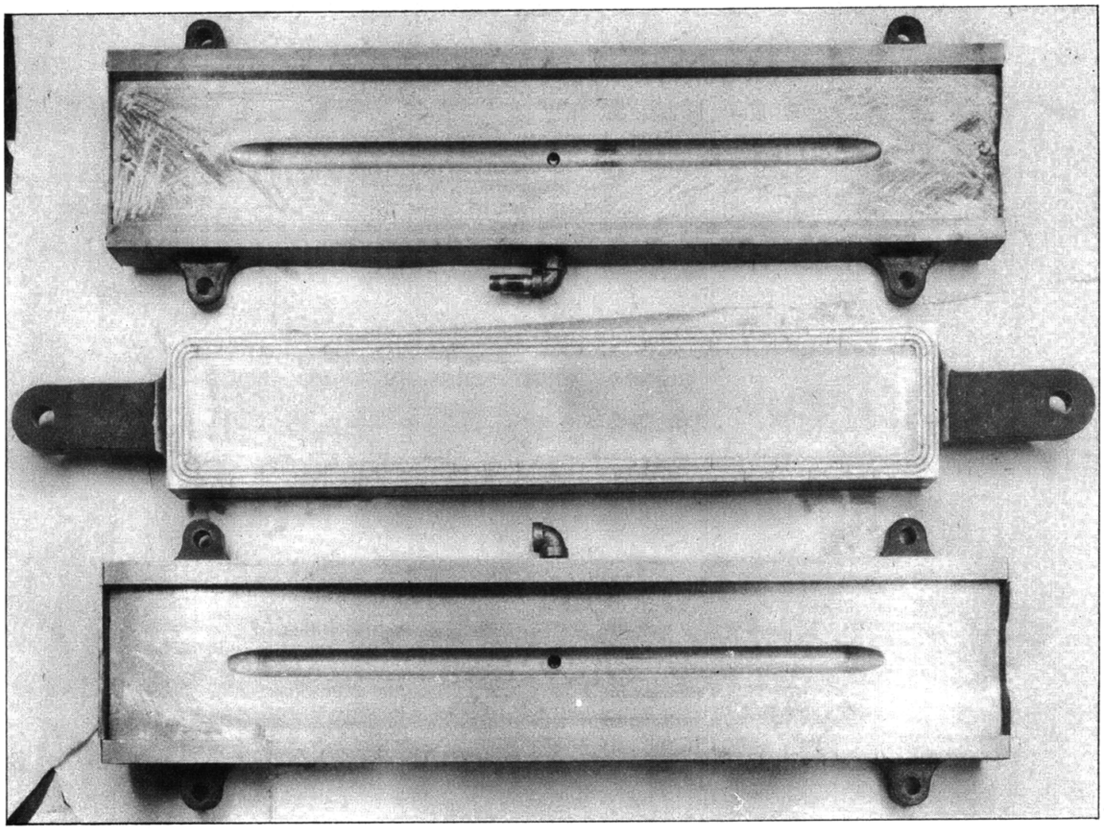

Figure 14 - Guide Blocks and Slider of First Preliminary Model

The slider, in the middle, was fitted between the guide blocks shown above and below it. The long recesses in the guide blocks were to distribute the oil and the multiple grooves around the slider were to reduce leakage from the bearing surface.

slider and the guide blocks and finally leaked off on all sides into a tray and flowed back to the sump.

A load of 50,000 pounds was applied to the loading beams through the spherical loading head of the testing machine. This subjected the slider to a unit load approximately the same as that placed on a track 32 feet in diameter and 15 inches wide by a 1700-ton turret. Oil was delivered from an oil pump through the inlet holes of the guide blocks until the load on the testing machine read higher than the 50,000 pounds originally applied. The slider was then moved longitudinally by a screw jack and the frictional resistance was read on a spring dynamometer.*

The tests showed this model to be quite unsatisfactory. The average coefficient of friction found was 0.005, a value far greater than would have been obtained from oil viscosity alone. Test runs with oils of different viscosities did not reveal anything conclusive. The lighter oil in one

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* Details of the test apparatus are shown on TMB photographs 1232 and 1233, not reproduced here.

case actually gave a higher frictional force than the heavy oil. Any effect due to a change in the viscosity of the oil was masked by imperfect lubrication or incomplete surface separation.

It was agreed that the difficulty of maintaining an oil film over the entire surface on each side of the slider made this type of model quite unsatisfactory. Crude as the model was, however, the minimum value of the coefficient of friction obtained in all tests was 0.0019, which was of the order existing for ball bearings. If full flotation had been maintained the coefficient should have been of the order of 1×10^{-6}.

In view of the scant similarity between this model and the type of bearing that would be required for turret support it was considered not warranted to spend further time on the development of the "sandwich."

APPENDIX 2
DESCRIPTION AND TESTS OF THE SECOND PRELIMINARY MODEL

In planning the second series of experiments an attempt was made to simulate the actual form of bearing that might be used on a full-scale turret. Among others, answers were desired to the following questions: (1) Could the oil film be built up and maintained if the load were first applied with the bearing dry? (2) What pump capacity would be required? (3) What thickness of oil film would exist? (4) What would be the effect of eccentric loading? (5) What grade of oil would be best suited to this type of bearing? (6) What would be the most satisfactory oil supply arrangement? and (7) What tolerances would be permitted in machining?

The design of the second model is illustrated by the drawings, Figures 3 and 4, and the photographs, Figures 5 and 15. It was built to a scale of about 1 to 8. Except for the relatively great width of the bearing rings

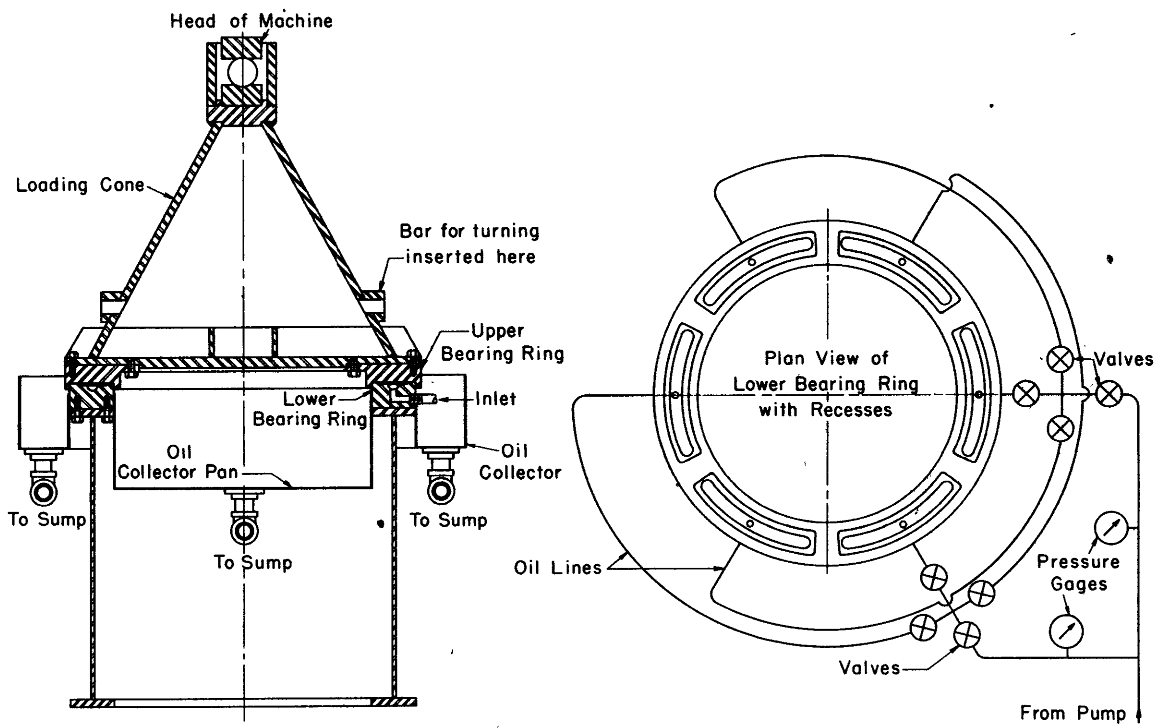
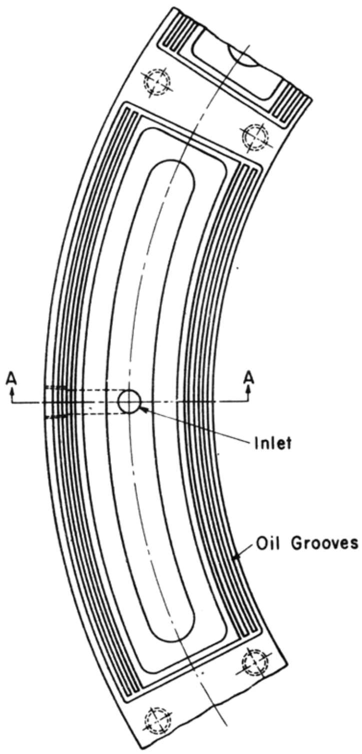


Figure 3 - Sectional Elevation and Oil Supply Arrangement of Second Preliminary Model

The lower fixed bearing ring is supported by a cylindrical shell of 1/4-inch steel plate, which simulates a turret foundation. It has 6 recesses or lakes in its upper surface.

The upper rotating bearing ring has a lower bearing surface which is perfectly flat, and it carries projections which fit over the edges of the lower bearing ring. It is loaded through a steel cone with a hardened circular plate at the top. Load is applied from the testing machine through a 3-inch steel ball which permits rotation of the upper bearing member when a downward force is exerted upon it.



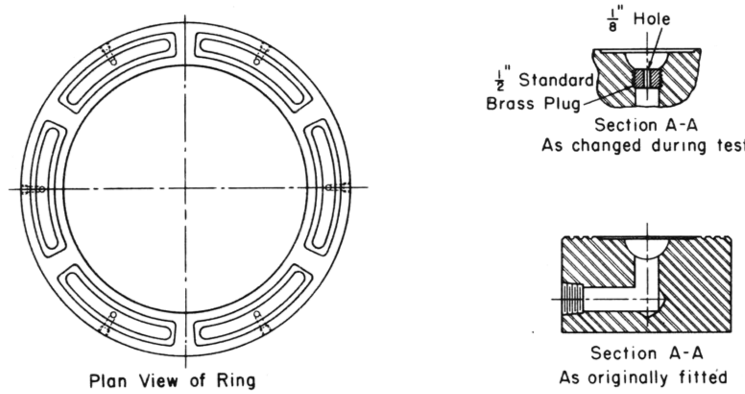


Figure 4 - Details of Lower Bearing Ring of Second Preliminary Model

A general view of this ring is shown in the photograph, Figure 5.

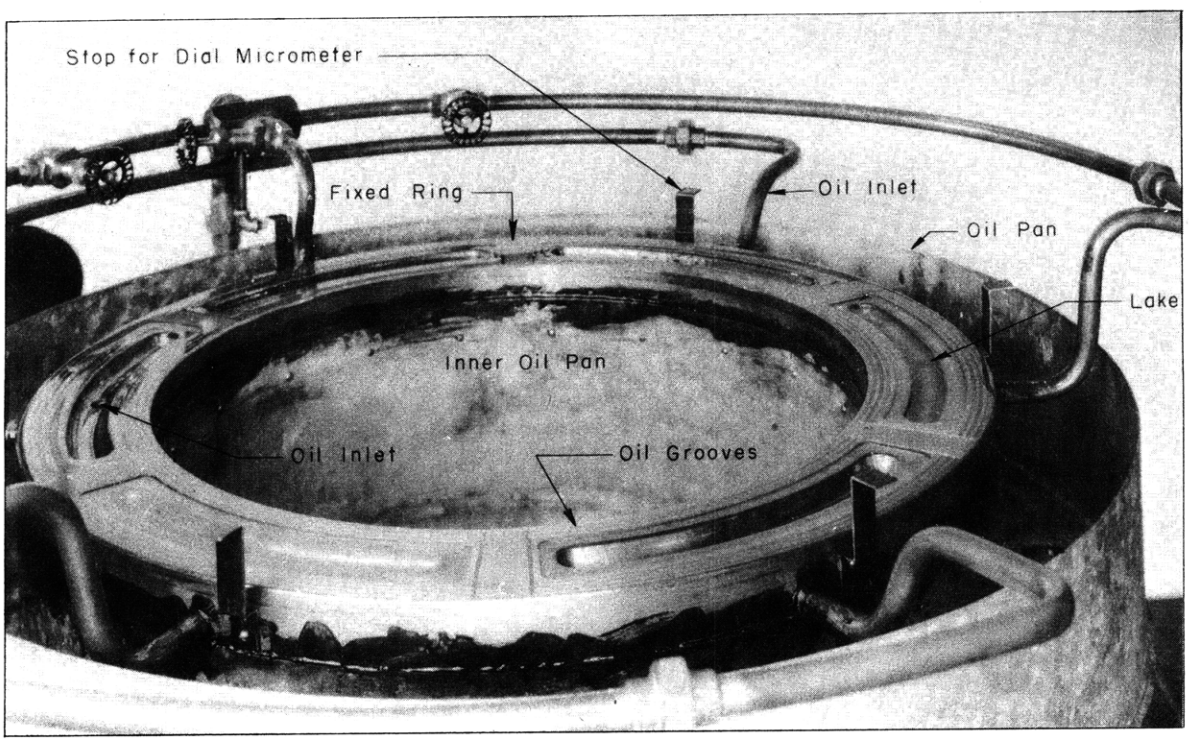


Figure 5 - Lower Bearing Ring of Second Preliminary Model

This view shows the recesses or lakes, the oil grooves surrounding them, and the high pressure valves and piping used to supply oil to the six segments of the bearing.

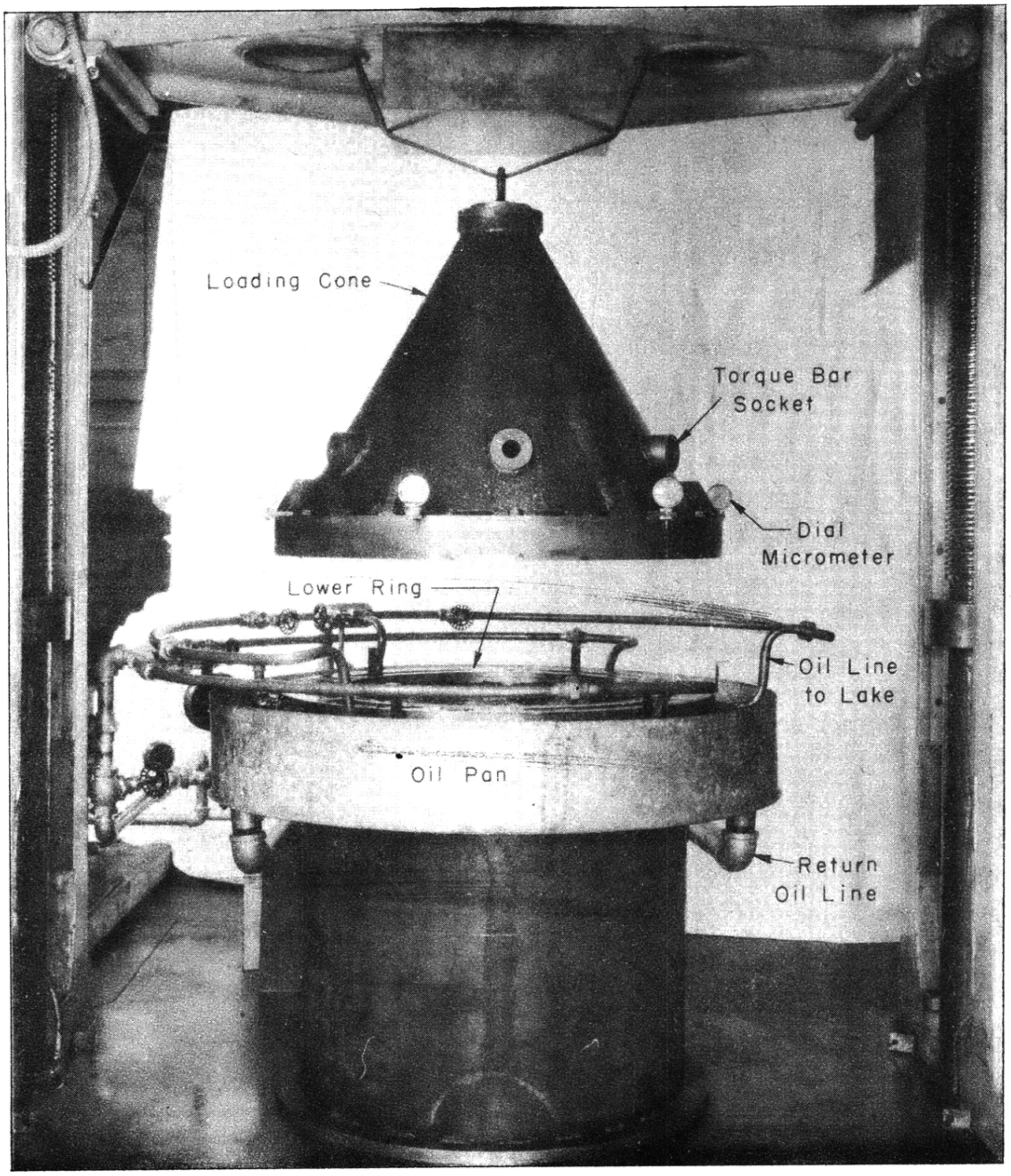


Figure 15 - Second Preliminary Model under Testing Machine, Partially Disassembled

The loading cone carries the upper bearing ring and takes the load of the testing machine. The cone is rotated by a torque bar inserted in one of the holes in the cone.

as compared with their diameter the arrangement was not unlike that which might be used in an actual turret.

The lower ring is a steel casting of rectangular section. Oil enters this ring through six cored passages which supply six shallow lakes or recesses in its upper surface. The surface outside the lakes is grooved as shown in Figure 4. The small staggered radial grooves connecting the main grooves as shown in the drawing were omitted for the first test and the results indicated no necessity for them. The edges of the lakes and grooves were originally made quite sharp, but as the preliminary tests showed that under heavy loads these edges acted as oil scrapers they were rounded off slightly with a stone.

The upper ring as shown in Figure 3 has simply a flat bearing surface. The projecting sides keep the two rings centered; it was at first proposed that these would resist the horizontal component of the recoil force in an actual installation. When the two rings are exactly centered there is a clearance of 0.014 inch all around between the sides of the upper and the lower rings.

The horizontal bearing area of the ring is 429 square inches, and on the tentative assumption that oil pressures less than 200 pounds per square inch would be used in an actual installation, the load for the model was fixed at 80,000 pounds. This corresponds to an oil pressure of 186 pounds per square inch over the whole bearing surface.

A variable-stroke (Waterbury) pump, size 2 1/2, was used in this installation. As shown in Figures 4 and 5, the delivery line of the pump was divided into two separate branches, each supplying three alternate lakes. Valves were installed in the lines leading to the individual lakes as well as at the origin of each branch. Pressure gages were provided in both branches. The oil escaping between the rings was caught in pans and returned by gravity to a sump after passing through a strainer.

As the type of oil-film bearing under test was expected to have an extremely low coefficient of friction if metal-to-metal contact were avoided, it was obvious that any appreciable friction introduced by the loading system might entirely mask the results. The load was therefore transmitted from the testing machine to the model through a 3-inch steel ball, as shown in Figure 16. Experiment showed that the torque required to rotate a 3-inch steel ball with a Rockwell hardness of C-60 between two steel plates of the same hardness was a fairly constant function of the load up to 80,000 pounds. The arrangement for measuring the ball friction is shown in Figure 17. It is easily shown that the frictional torque introduced by the ball when the cone is rotated is one-half that required to rotate the ball when both plates are held fixed, provided the load is the same in both cases. The correction to be applied for ball friction when the cone is rotated is thus readily obtained.

CONDUCT OF TESTS

A light oil, SAE 10, was tried first to find the oil of minimum viscosity that could be used. With this oil it was easily possible to float

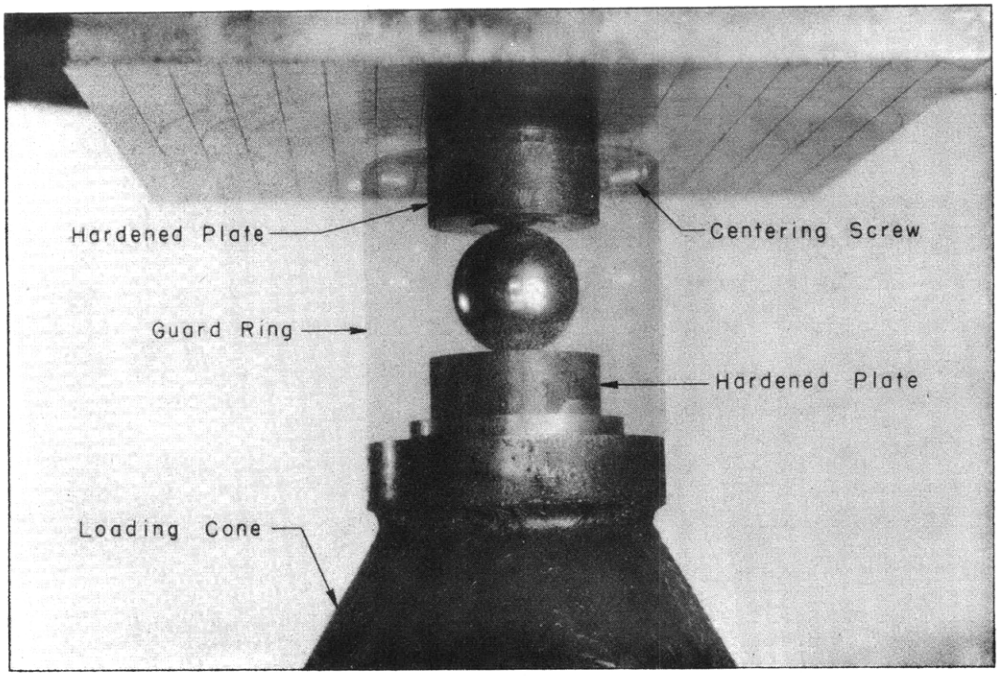


Figure 16 - Position of 3-inch Ball within Guard Ring during Test

During test of the bearing, the guard ring shown here in phantom form prevents accidents due to fracture or slipping of the steel ball.

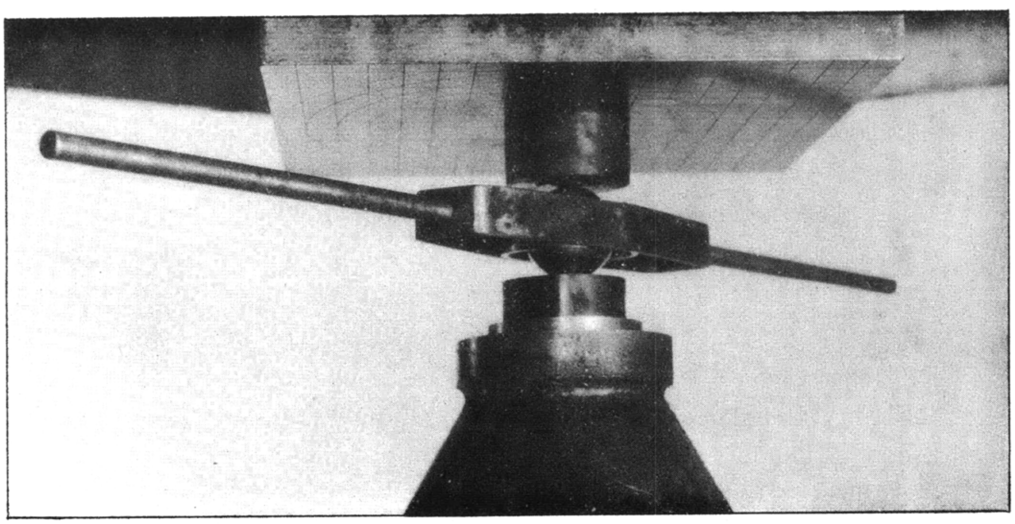


Figure 17 - Arrangement for Measuring Ball Friction

The 3-inch steel ball was set between two hardened plates, and the cone was kept from rotating by draining the oil from the bearing. The torque required to rotate the ball was measured by a spring balance attached to the torque bar. In all cases the surfaces of contact between the ball and the hardened plates were lubricated with machine oil.

the cone when no additional load was applied by the testing machine. However, when load was applied the film thinned out rapidly and at a load of 60,000 pounds seizure of the upper and lower rings occurred. The average oil-film thickness indicated by the dial micrometers was 0.067 inch at 10,000 pounds load; at 50,000 pounds it was reduced to 0.006 inch. The film thickness was far from uniform around the circumference under the various loads and it was impossible to obtain uniformity by manipulation of the cut-out valves. The estimated torque values and coefficients of friction, measured by a spring balance attached to a torque bar in the loading cone, were of the same order of magnitude as those found with the "sandwich," namely about 0.005. This was considered much too high a value.

It appeared that a heavier oil would be required to maintain the film, and accordingly the oil was changed to SAE 50. With the oil supply valves wide open the cone could be turned at a load of 80,000 pounds but only when the oil film was first established before the application of the load. As it was considered essential to build up the oil film under full load in the actual turret, these results were considered inadequate. Observation indicated that due to slight eccentricity of loading the cone would tilt slightly to one side, whereupon most of the oil would escape on the high side and the bearing would seize on the low side.

This condition was remedied by restricting the flow in the individual inlets to the lakes so that sufficient oil could not escape through any one lake to cause a great loss of line pressure. Brass plugs drilled with 1/8-inch holes were inserted in the supply inlets leading to the lakes. With these orifices in place it was possible to establish and maintain an adequate oil film at the full load of 80,000 pounds. The friction was so slight that the torque required to turn the cone was less than the estimated correction for ball friction. This indicates that the coefficient of friction was practically zero, or that it was limited to the viscous friction of the oil. With the 3-inch steel ball set 1/4 inch off center, which was the maximum eccentricity permitted within the guard ring, there was still no appreciable friction.

RESULTS OF TESTS WITH SECOND PRELIMINARY MODEL

Data for the final tests of this model are given in Table 2.

TABLE 2

Grade of oil used	SAE 50
Orifices	1/8 inch diameter, 3/8 inch long
Total load on bearing	80,000 pounds
Unit bearing load on upper bearing ring	186 pounds per square inch
Pressure measured at pump	250 pounds per square inch
Oil delivery rate	9.8 gallons per minute
Torque required to rotate cone	1056 inch-pounds
Frictional torque due to ball, estimated	1200 inch-pounds
Coefficient of bearing friction	Less than 1×10^{-6}
Oil temperature	90 degrees Fahrenheit; 5 degrees above room temperature

Lift of bearing at 80,000-pound load (thickness of oil film) in inches

Station 1	0.010	Station 4	0.002
2	0.010	5	0.003
3	0.009	6	0.006