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A SELF-ADJUSTING SPRING THRUST BEARING

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A careful study of the difficulties experienced with the types of thrust bearings used in supporting the heavy loads of water wheels and the electric generators driven by them, led the author to design the flexible bearing described in the paper.

This bearing consists of a runner of a special grade of cast iron resting on a thin steel ring with a babbitted surface. The babbitted stationary ring, in turn, rests on a large number of short helical springs (ordinarily wound of ½-in. wire, 2 in. in diameter and 1½ in. long) and is held against rotation by dowel pins. A saw cut through one side does away with any tendency to dish with a change in temperature. This construction prevents the possibility of undue pressure at any point and compels each element of the surface to carry its share of the load.

Advantages to be derived from the use of such bearings are set forth and figures are given showing a reduction in friction accompanying increased unit pressure. Two designs are described and illustrated.

It has been shown that the pressure on the babbitted surface of an ordinary journal bearing varies greatly in different parts of the circumference of the bearing, being greater at the center line of the resultant load than toward the sides, and varying approximately inversely with the thickness of the oil film. Whether the greater part of this variation in pressure is due to the difference in thickness of the film or to the dragging of the oil by the shaft to a point from which it cannot readily escape, is hard to determine. The thickness of the film depends, apparently, on the load per unit area, the viscosity of the oil, and the surface speed of the shaft. Several investigators have shown that with ordinary loads of 100 lb. average pressure per square inch, the thickness of the oil film at the bottom of the bearing is about 0.0002 to 0.0003 in. With this in mind, it will readily be understood why the surfaces of the bearing have to be fitted closely to the shaft, why the supporting shell must be made rigid, and, finally, why a soft metal, which may conform to the shaft, is much better for a bearing surface than a hard one. In

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5 Many of the problems found in the construction of journal bearings are met with in the design of thrust bearings. Small bearings, up to perhaps ten inches in diameter, may be readily fitted so that at ordinary speeds the surfaces are sufficiently accurate to allow a film thin enough to support the load without danger of dragging the babbitt. The parts must be made quite rigid and the seat is usually supported on a spherical surface to correct for slight inaccuracies of alignment.

6 Thrust bearings for supporting the heavy loads of water wheels and the electric generators driven by them, are now very widely used. The difficulties in the fitting and use of plate bearings are much aggravated as the weight is increased, on account of the large overall dimensions of the supporting plate. It is true that the surfaces can be fitted quite accurately by machining, but the writer has known cases where the surfaces were turned slightly conical so that they touched hard on the inner or the outer edge. The deflection of the supporting collar on the shaft may allow the runner to be slightly dished, or there may be a deflection of the supporting surface, thereby dishing the babbitted seat or causing one side to be lower than the other. The self-adjusting spherical seat provided to correct some of these difficulties is of doubtful value on large bearings on account of the great frictional resistance which must be overcome to make it shift.

7 Thrust-bearing surfaces are usually scraped to each other, or to a surface plate, to avoid dangerously high spots; but, since the oil film is of the order of 0.0002 in. to 0.0003 in. in thickness, the difference in level must be smaller than these values. This work must usually be done without load, and, no matter how carefully done, when the bearing is loaded the parts will probably not fit each other, due to deflection.

8 A careful study of the above difficulties led the writer to the design of a flexible bearing surface pressed against the runner by springs. It seemed that this would prevent the possibility of undue pressure at any point, and compel each element of the surface to carry its share of the load. On trial, this solution proved satisfactory.

9 A typical design of a spring thrust bearing for vertical-shaft machines is shown in Fig. 1. The bearing consists of a runner of a special grade of cast iron resting on a thin steel ring with a babbitted surface. The babbitted stationary ring, in turn, rests on short helical springs and is held against rotation by dowel pins. A saw

cut through one side eliminates any tendency of the ring to dish with a change in temperature. The high base ring shown, on which the springs stand, is often used in connection with a deep housing to increase the amount of oil in the surrounding bath. The tube in the center forms a retaining wall around the shaft, for the oil. The springs ordinarily used are wound of $\frac{1}{2}$-in. round wire and have an

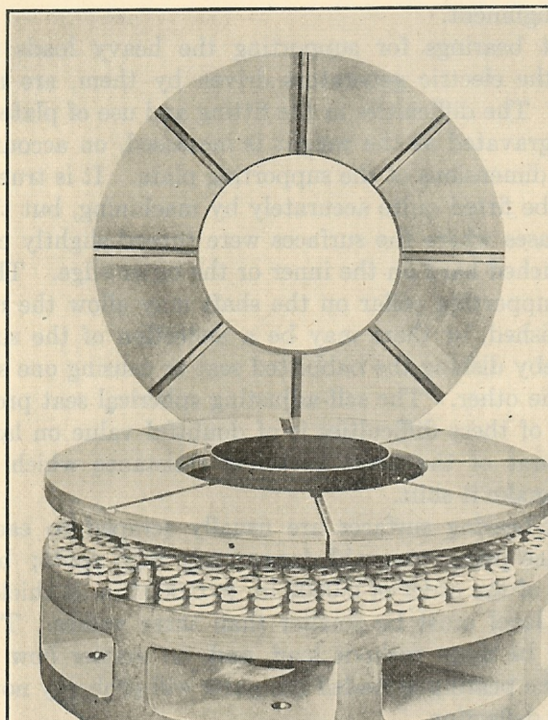

FIG. 1 SPRING THRUST BEARING, FOR VERTICAL WATER-WHEEL-DRIVEN GENERATOR, TO CARRY A LOAD OF 300,000 LB. AT 100 R.P.M.

(View shows rubbing surface of rotating ring; stationary ring is raised to show the arrangement of springs.)

outside diameter of 2 in. and a free length of $1\frac{1}{2}$ in. Under load the springs close about $\frac{1}{16}$ in., and the total pressure is well distributed. By this means it is possible to avoid excessive pressures at any point. Thus, it is safe to run with a much higher average pressure than when there is no definite limit to the pressure which may occur over a small area.

10 It will be seen that this type of bearing differs from the solid-ring thrust bearing in that one of the bearing surfaces is made to yield at any point by using a comparatively thin plate supported by a large number of springs. While solid bearings may be used successfully for small loads, a bearing which thus automatically adjusts itself to faults in finish and in alignment is preferable for carrying very heavy weights.

11 Oil grooves are provided in one of the members and sometimes in both. In order to insure proper circulation of the oil for cooling purposes, in the case of bearings operating at low speed it is necessary to have grooves in the rotor. On high speeds these grooves may sometimes be omitted, relying for circulation only on the friction of the rotor on the oil while passing the grooves in the stator. In many cases we have had very satisfactory results by placing radial grooves in both the rotating and the stationary surface. It is our practice to have different numbers of grooves in the two plates, for instance, six and eight. With grooves in each of the surfaces we have a continuous flooding of oil on all the bearing surfaces and a very effective means of cooling. Much of the heat would otherwise have to be transmitted through the metal of the stationary part of the bearing.

12 The pressure usually allowed on these bearings is from 300 to 400 lb. per sq. in., the design permitting a very thin oil film without metallic contact. It is necessary to have the runner very smooth and free from scratches, especially any at an angle to the direction of rotation, as these might cause injury to the babbitt. The babbitted surface does not need to be scraped but is turned with a tool as smooth as is convenient. Wearing sometimes occurs in minute spots all over the plates. When this happens there is no risk of dragging the metal. The bright spots that show themselves are produced while starting and slowing down, before a pressure film is formed. When in operation the weight is apparently entirely supported on the oil film.

13 It is desirable to run bearings at a high pressure if they can be designed to do this safely, as the parts then are smaller, the rubbing speed is less, and the friction very much reduced. With this design of bearing the tendency to excessive pressure at one point is automatically relieved by the springs yielding, and while there will be some uneven distribution, a variation in pressure of two or three

times the average is comparatively unimportant and does not cause bearing failures; it is pressures of twenty or more times the average that cause injury. These excessive pressures are prevented by the construction just described. For this reason it is safer to operate this bearing with high pressures than a more rigid bearing at lower pressures.

14 The loss of alignment due to settling of foundations or other causes does not affect the bearing adversely. In one water-wheel-

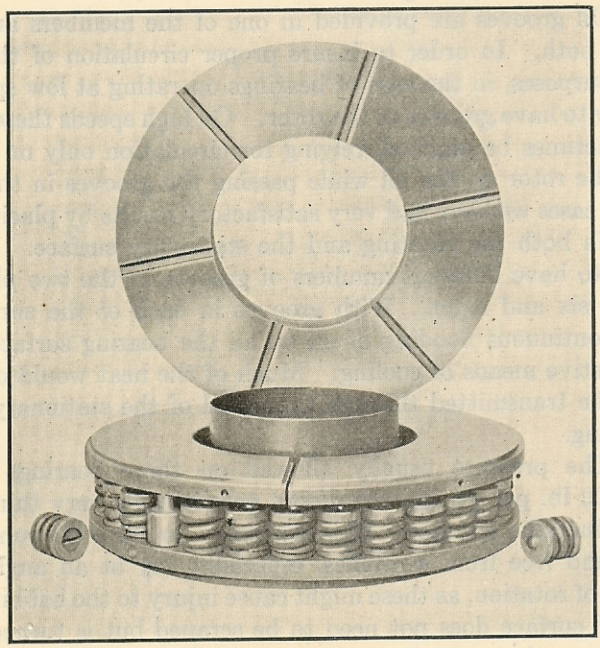

Fig. 2 Spring Thrust Bearing with "Compressed Springs" for Machines Having Small Clearances

(Stationary babbitted ring is raised to show springs and dowel pins.)

driven alternator installation the striking of the field against the armature led to the discovery that the coupling between the two units had loosened, which allowed the shaft to "run out." The bearing operated without injury with over 0.03 in. vertical movement of the outer edge of the rubbing surface. This caused an uneven distribution of load on the bearing, to the extent of reducing the load on one side of the bearing about 16 per cent and increasing the pressure a similar percentage on the extreme opposite side.

15 An advantage of increased pressure in the reduction of friction is shown by the following table of comparison:

Bearing number	1	2
Revolutions per min.	200	200
Total load, lb.	300,000	300,000
Outside diameter of bearing, in.	35	46
Inside diameter of bearing, in.	17.5	17.5
Net area, sq. in.	600	1200
Pressure, lb. per sq. in.	500	250
Average rubbing speed, ft. per min.	1370	1670
Coefficient of friction	0.0018	0.0033
Kilowatt loss	16.7	38
Horsepower loss	22.5	51

16 In some designs the vertical clearance between the water wheel and the casing is very small, so that the displacement caused by a free spring under the variation of hydraulic suction is objectionable. In such cases an initial compression equal to full load, or to an overload, is put on the springs. The load will still distribute, since an overload at any point will cause the spring to close beyond the initial compression. Such a bearing is shown in Fig. 2. However, this bearing was designed to replace a roller bearing, and was so made that the parts of the water wheel would occupy the same relative positions as before. The principles used in the construction of these thrust bearings are applicable also to journal bearings and to bearing surfaces having a reciprocating motion, like the crosshead of a steam or gas engine.

17 Referring to Par. 2, it may be said that the oil film is maintained by not allowing any part of the surface to carry abnormal pressure. The writer does not know whether the front edge of a section between oil grooves is further from the runner than the leaving edge. If there is a difference, it must be exceedingly small, since the babbitted steel plates used have a material thickness. Possibly there is no such difference in the thickness of the oil film but that the oil film moves with different speeds at different places on the bearing surface, faster at the entrance from an oil groove than at the exit, and very much faster than on the side where the flow is produced by the pressure of the oil and not by adherence to the runner.

18 These bearings have been used on shafts running at speeds of 3600 r.p.m. The natural period of the springs when unloaded is very much higher than this value, and, in the writer's opinion, the bearings will undoubtedly follow irregularities at this speed. But the higher the speed, the more important it is to have the parts run true.