**Title:** CAGE POSITIONED TILTING PAD BEARING

**Abstract:** A tilting pad bearing for a high speed rotary machine, such as a turbocharger for an internal combustion engine, comprises a cage and a plurality of tilting pads. The cage is formed by two annular rings separated by spacers machined from a single billet of high-temperature metal. The bearing pads are inserted between the spacers of the cage. One or more O-rings retain the bearing pads within the cage during assembly of the bearing but are not required after installation of the bearing. When retained in the final assembly and installation of the bearing, the O-rings allow tilting movement of the pads within the cage. The outer surfaces of the spacers have a plurality of grooves, and outer surfaces of each of the pads have matching grooves. One of the grooves is an oil groove, wherein the oil groove of the spacers communicating with and oil inlet holes in the spacers. At least one of the other grooves receives an O-ring that hold the pads in place in the cage during installation. When two O-rings are used on either side of the oil groove in the final assembly, the O-rings serve to direct oil to the oil groove and prevent or reduce oil leakage.
CAGE POSITIONED TILTING PAD BEARING

DESCRIPTION

BACKGROUND OF THE INVENTION

Field of the Invention

The present invention generally relates to bearings having limited radial space for the hydrodynamic bearing housing and, more particularly, to a novel cage positioned tilting pad bearing having particular application in turbochargers and other high rotary speed devices.

Background Description

High speed turbochargers are intended to increase the power of internal combustion engines. The first turbocharger was invented in the early twentieth century by the Swiss engineer Alfred Buchi who introduced a prototype to increase the power of a diesel engine. Turbocharging was not widely accepted at that time, but in the last few decades, turbocharging has become standard for most diesel engines and is used in many gasoline engines as well. Since the earliest turbocharger prototypes, researchers have attempted to improve turbocharger reliability and increase turbocharger life (Born, H. R., "Analytical and experimental investigation of the stability of the rotor-bearing system of new small turbocharger," in Proceedings of the Gas Turbine Conference and Exhibition, Anaheim, CA, May 31-June 4, 1987). Since vibration-induced stresses and bearing performance are major failure factors, rotordynamic analysis should have been an important part of the turbocharger design process. A thorough rotordynamic investigation was, however, very difficult and relatively few studies were published in the early years.

Advances in rotor dynamic analysis computer programs have now made the analysis of a turbocharger rotor-bearing system a reality (Gunter, E. J. and Chen, W. J.,
2001, DyRoBeS - Dynamics of Rotor Bearing Systems User's Manual, RODYN Vibration Analysis, Inc., Charlottesville, VA). Manufacturers have begun using these tools to better understand the dynamics of high speed turbochargers. Design improvements, however, cannot depend on computational analysis alone (Holmes, R., Brennan, M. J., and Gottrand, B., "Vibration of an automotive turbocharger - a case study," in Proceedings of the 8th International Conference on Vibrations in Rotating Machinery, Swansea, UK, September 7-9, 2004, pp. 445-450) and on-engine test data are still required for these still very difficult analytical predictions.

A turbocharger consists basically of a compressor and a turbine coupled on a common shaft. The turbocharger increases the power output of an engine by compressing excess air into the engine cylinder, which increases the amount of oxygen available for combustion. Since the output of reciprocating internal combustion engines is limited by the oxygen intake, this increases engine power (Ward, D. et al., U.S. Patent No. 6,709,160). Since the turbine is driven using energy from the exhaust, turbocharging has little effect on engine efficiency. By contrast, a supercharger using power from the engine shaft to drive a compressor also increases power, but with an efficiency penalty.

An important factor in the design of an automotive turbocharger is the initial cost. The same power increase provided by the turbocharger can be provided by simply building a larger engine. Since engine weight is not a major part of overall weight for a diesel truck, the turbocharger is only competitive if it is less expensive than increasing engine size. For passenger cars the turbocharged diesel must compete with lighter and less expensive gasoline engines. To keep costs down while maintaining reliability, the designs of automotive turbochargers are usually as simple as possible.

Many automotive-size turbochargers incorporate floating bushing journal bearings. These bearings are designed for fully hydrodynamic lubrication at normal operating speeds. For low cost and simple maintenance, turbochargers use the engine oil system for lubrication instead of having a separate system.

The primary consideration in the rotodynamic design of high-speed machinery is to control and minimize vibration. Large-amplitude vibration is undesirable in that it generates noise and can have large amplitudes that cause rotor-stator rub. In most rotating machinery, the dominant vibration is a forced response to rotor imbalance. There exists, however, another class of vibration termed rotodynamic instability or self-excited vibration. Vibration of this type requires a different design approach. Almost all rotors of automotive turbochargers exhibit both forced vibrations and self-excited vibrations (Choudhury, Pranabesh De, "Rotodynamic stability case studies", *International Journal of Rotating Machinery*, 2004, pp. 203-211).

Forced vibrations from imbalance are harmonic and occur at the turbo shaft
speed. They are generally driven by either mass eccentricity in the rotor or shaft bow. Mass eccentricity is a result of manufacturing tolerances, while shaft bow can be due to manufacturing tolerances or thermal effects. Unbalance vibrations can usually be minimized by designing the rotating element so that no natural frequencies are close to the desired operating speed range. Thermal bowing is the only exception to this previous statement.

Self-excited vibrations usually occur at frequencies that are a fraction, rather than a multiple, of shaft speed. The sub-synchronous vibrations do not require a driving imbalance in the rotating element, but are due to the interaction between the inertia and elasticity of the rotating elements, the aerodynamic forces on the rotor and the hydrodynamic forces in the bearings.


The desire would be to have a small synchronous vibration that would allow the bearing to have less dynamic loading and better performance. It is also desirable to provide a turbocharger capable of higher top speeds with less oil leakage and hence
lower emissions.

SUMMARY OF THE INVENTION

It is therefore an object of the present invention to improve bearings having limited radial space for the hydrodynamic bearing housing.

It is another more specific object of the invention to improve turbocharger performance through a new and innovative design feature in the bearing.

According to the invention, a tilting pad bearing for a high speed rotary machines comprises a cage and a plurality tilting pads, typically four in number. The cage is formed by two annular rings separated by spacers machined from a single billet of metal.

The bearing pads are inserted between the spacers of the cage. One or more O-rings retain the bearing pads within the cage. The O-rings allow tilting movement of the pads within the cage. The outer surfaces of the spacers have grooves, and outer surfaces of each of the pads have matching grooves. One of the grooves is an oil groove. The oil groove of the spacers communicates with oil inlet holes in the spacers. The other groove or grooves receive O-rings that hold the pads in place in the cage during assembly of the bearing. When two O-rings are used on either side of the oil groove in the final assembly, the O-rings serve to direct oil to the oil groove and prevent or reduce oil leakage.

The new bearing design can be used in turbochargers that have the existing standard floating bearing design of the turbocharger industry. The size of the bearing would be determined by the current bearing housing inside diameter in addition to the shaft diameter. The design can also be utilized on new turbochargers with essentially no modification to the basic bearing housing design. The novel bearing design consists of a positioning cage that isolates each pad one from the other, and allows each pad to load itself owing to the offset and preloaded pad design. The cage-and-pad assembly is further held by small diameter O-rings that prevents the pads from coming out of the cage during assembly. In addition, the small preload O-ring assists in directing the majority of the oil flow around the bearing assembly and into the oil inlet holes of the cage.

The new bearing according to the present invention has the potential to eliminate
the nonlinear jump behavior as well as the large amplitude sub-synchronous instability frequencies.

**BRIEF DESCRIPTION OF THE DRAWINGS**

The foregoing and other objects, aspects and advantages will be better understood from the following detailed description of a preferred embodiment of the invention with reference to the drawings, in which:

Figure 1 is a partially cross-sectional and exploded view of a turbocharger housing and assembled turbocharger shaft with stock bearings shown above the housing;

Figure 2 is a perspective view of the basic cage for the tilting pad bearing according to the invention;

Figure 3 is a perspective view of the basic cage shown in Figure 2 with the addition of projections to restrain the cage from spinning the in the bearing journal;

Figure 4 is a perspective view of one of the pads for the tilting pad bearing according to the invention;

Figure 5 is a perspective view of a pad ring from which four pads can be made;

Figure 6 is a perspective view of the basic cage of Figure 2 and pads for the four tilting pad bearing according to one embodiment of the invention;

Figure 7 is a perspective view of the cage of Figure 3 and pads for the four tilting pad bearing according to a second embodiment of the invention;

Figure 8 is a perspective view of the second embodiment of the invention assembled with the spin-restrained cage of Figure 3 and with O-rings in place;

Figure 9 is a graph showing typical spectrum content on stock bearing where the turbocharger is shown to be unstable in the first mode and second mode with a frequency in 12-20 kcpm frequency range; and

Figure 10 is a graph showing spectrum content of the new cage design according to the invention.
DETAILED DESCRIPTION THE INVENTION

The design of rotating machinery is concerned with the placement of the critical speeds, the response sensitivity to imbalance and the predicted stability of the rotor bearing system. The program DyRoBeS© (Gunter and Chen, 2001, *ibid.*, and Chen, Wen J., and Edgar J. Gunter, *introduction to Dynamics of Rotor-Bearing Systems*, Victoria: Trafford, 2007) has been used to compute the analysis results in the following discussion. The most basic analysis is the undamped critical speed analysis that concerns only the rotor shaft and a range of possible stiffness values for die bearings. To make that analysis meaningful, it is necessary to have a range of expected stiffness for the bearings. The first two modes are nearly rigid body modes while the third mode is a bending mode and operation should be below this critical speed. The turbocharger that is used on the available diesel engine has been designed to operate below this bending mode. The re-excitation of the first mode has been in the range from 10,000 to 20,000 cpm, the second mode was has been from 26,000 to 35,000 cpm (Kirk, R. Gordon., et al., 2010, *ibid.*).

A common design assembly for a turbocharger consists of a radial outflow compressor and a radial inflow turbine on a single shaft. Bearings are mounted inboard, with the compressor and turbine overhung. Figure 1 is a partially cut-away and exploded view of a conventional turbocharger. As shown in Figure 1, the housing 10 is cut away to show the locations of oil feed paths 11 to the bearings 12. The turbine rotor 13 in most common automotive turbochargers is connected to its shaft 14 by a friction or electron beam welding method. The compressor wheel 15, or impeller, is usually a clearance or very light interference fit on the other end of the shaft 14. A locknut 16 is used to hold the impeller against a shoulder on the shaft. Friction from the interference fit and/or nut clamping pressure is generally sufficient to transmit the torque, therefore splines or keys are not required. The bearings 12 have a single groove about their circumferences with oil passage holes to provide lubrication to the shaft 14.

The new bearing is a multi-pad tilting design. As shown in Figure 2, an embodiment of the new bearing consists of a positioning cage 20 composed of two annular rings 21 and 22 separated by four spacers 23, 24, 25, and 26. Each of the spacers
is machined with three grooves 26, 28 and 29 in their outer surfaces. The innermost
groove 28 is an oil groove which communicates with an oil hole or passage in the spacer. 
The two outermost grooves receive O-rings. The O-rings may be replaced with metallic
split ring seals for very high temperature applications. The cage 20 is preferably
machined from a single billet of high temperature resistant steel, such as stainless steel.
Other methods of producing the cage are possible, such as casting with small finish
machining if required. The pads may be made of a special bearing bronze, but could be
made from other bearing materials.

In the embodiment shown in Figure 2, the "pinch" of the O-rings is relied on to
prevent unwanted spinning of the cage within the bearing journal. The depth of the O-	ring grooves can be different (smaller) for the cage so as to cause pinch of the O-rings to
prevent spin. A modification of the basic cage design of Figure 2 is shown in Figure 3. In
this modification, one of the annular rings 21 is provided with projections or "ears" 31,
32 and 33. These projections are designed to mate with corresponding indents or reliefs
in the bearing journal to resist spinning of the bearing in the bearing journal. In both of
the embodiments of Figures 2 and 3, four tilting pads are used, but three pads or five
pads could be used depending on the shaft journal diameter and also the magnitude of
the loading and available axial space for the bearing.

Figure 4 illustrates one of the pads 40 that is placed between adjacent spacers of
the cage. As shown in Figure 4, the outer surface is machined with corresponding
grooves 41, 42 and 43 that align with the grooves in the outer surfaces of the spacers. In
the embodiment shown, four pads are required, one for each of the four spaces between
the spacers in the cage. Figure 5 shows a pad ring from which the four pads can be made.
The pad ring may be, for example, cast bearing bronze. The four pads are cut from this
pad ring.

Figure 6 shows a cage 60 of the type shown in Figure 2 and four pads 61, 62, 63,
and 64 of the type shown in Figure 4 prior to bearing assembly. The cage 60 isolates
each pad 61, 62, 63, and 64 from one another and allows each pad to load itself owing to
the offset and preloaded pad design. The cage and pad assembly is further held by one or
more small diameter O-rings 65 and 66 that prevent the pads from coming out of the
cage during assembly. In the final assembly, the O-rings are optional and can be removed
as the bearing assembly is inserted into the bearing journal. If used only for pre-assembly retention of the pads, a single O-ring would suffice with a single O-ring groove in the outer surfaces of the pads and spacers of the cage. The use of two O-rings in the final assembly, however, has the additional advantage of assisting in directing the majority of the oil flow around the bearing assembly and into the oil groove and the oil inlet holes of the cage. The O-rings do not resist the tilting action of each pad. As shown in Figure 6, outer surfaces of the spacers and the outer surfaces of each of the pads 61-64 have matching grooves. Figure 7 shows a cage 70 of the type shown in Figure 3, four pads 71, 72, 73, and 74 of the type shown in Figure 4 prior to bearing assembly and O-rings 75 and 76. Figure 8 shows a pair of assembled tilt pad bearings using the cage shown in Figure 3.

The novel concept in the design is the positioning of the pads by the cage to keep the pads independent and prevent the pads from dropping out when the bearing assembly is removed from the housing.

The typical spectrum content of the stock bearings shown in Figure 1 is shown in Figure 9 where the turbocharger is shown to be unstable in the first and second modes with a frequency in the 12-20 kepm frequency range, even at the engine idle speed. The second mode instability comes in at a shaft speed near 80 krpm in the 35-38 kepm frequency range. This is typical of all stock bearings in this turbocharger (Kirk, R. Gordon., et al., 2010, ibid.), with past grooved design bearings producing lower frequencies with larger amplitudes. The engine was at design load for this result.

The new tilting pad bearing design according to the invention eliminates the instabilities exhibited with the stock bearings. Figure 10 shows the spectrum content of the tilting pad bearings according to the present invention used in the same turbocharger. As shown in Figure 10, first and second mode instabilities are eliminated. The new design provides totally synchronous frequency content without the instabilities evident in the stock bearings.

While the invention has been described in terms of a single preferred embodiment, those skilled in the art will recognize that the invention can be practiced with modification within the spirit and scope of the appended claims.
CLAIMS

What is claimed is as follows:

1. A tilting pad bearing for a high speed rotary machines comprising:
   a cage formed by two annular rings separated by a plurality of spacers as an integral metal structure; and
   a plurality of bearing pads of bearing metal inserted between the spacers of the cage.

2. The tilting pad bearing of claim 1, wherein the two annular rings separated by spacers of the cage are machined from a single billet of metal.

3. The tilting pad bearing of claim 1, further comprising one or more O-rings retaining the bearing pads in the cage during assembly.

4. The tilting pad bearing of claim 3 wherein the O-rings are included in final assembly of the bearing, the O-rings allowing tilting movement of the pads within the cage.

5. The tilting pad bearing of claim 4, wherein outer surfaces of the spacers have a plurality of grooves, and outer surfaces of each of the pads have matching grooves, a first one of the grooves is an oil groove, the oil groove of the spacers communicating with oil inlet holes in the spacers, while at least a second one of the grooves receives an O-ring that hold the pads in place in the cage.

6. The tilting pad bearing of claim 1, wherein the cage is formed of four spacers separating the two annular rings and four bearing pads are inserted between the spacers of the cage.

7. The tilting pad bearing of claim 6, wherein three grooves are machined into the outer surfaces of the spacers and pads, the innermost groove being an oil groove and the two outermost grooves receiving O-rings.
8. The tilting pad bearing of claim 7, wherein the two outermost grooves are relatively shallow so as to pinch the O-rings within a bearing journal to resist spinning of the bearing within the bearing journal.

9. The tilting pad bearing of claim 6, wherein at least one of the annular rings is provided with at least one projection to restrain spinning of the cage within a bearing journal.

10. The tilting pad bearing of claim 1, further comprising one or more split ring metallic seals retaining the bearing pads in the cage.

11. The tilting pad bearing of claim 1, wherein the bearing is designed to be used on a shaft of a turbocharger.

12. A turbocharger for use with an internal combustion engine, comprising:
   a housing;
   a turbine rotor and an impeller mounted on a common shaft within the housing; and
   bearings supporting the common shaft adjacent each of the turbine rotor and the impeller within bearing journals in the housing, each of the bearings comprising:
   a cage formed by two annular rings separated by spacers machined from a single billet of metal; and
   a plurality of bearing pads inserted between the spacers of the cage.

13. The turbocharger of claim 12, further comprising one or more O-rings retaining the bearing pads in the cage during assembly of the bearings in the turbocharger housing.

14. The turbocharger of claim 13, wherein the O-rings are included in final assembly of the bearing in the turbocharger housing, the O-rings allowing tilting movement of the pads within the cage.

15. The turbocharger of claim 14, wherein outer surfaces of the spacers have a plurality
of grooves, and outer surfaces of each of the pads have matching grooves, a first one of the grooves is an oil groove, the oil groove of the spacers communicating with oil inlet holes in the spacers, while at least a second one of the grooves receives an O-ring that hold the pads in place in the cage.

16. The turbocharger of claim 12, wherein the cage is formed of four spacers separating the two annular rings and four bearing pads are inserted between the spacers of the cage.

17. The turbocharger of claim 16, wherein three grooves are machined into the outer surfaces of the spacers and pads of each of the bearings, the innermost groove being an oil groove and the two outermost grooves receiving O-rings.

18. The turbocharger of claim 17, wherein the two outermost grooves are relatively shallow so as to pinch the O-rings within a bearing journal to resist spinning of the bearing within the bearing journal.

19. The turbocharger of claim 16, wherein at least one of the annular rings of the cage of each bearing is provided with at least one projection to restrain spinning of the cage within a bearing journal.
FIG. 10

- True frequency
- Keyphase freq. / 3
- Synchronous speed
INTERNATIONAL SEARCH REPORT

A. CLASSIFICATION OF SUBJECT MATTER

F02B 39/00(2006.01)i, F02B 39/14(2006.01)i, FOID 25/16(2006.01)i, F16C 27/02(2006.01)i, F16C 33/00(2006.01)i

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

F02B 39/00; F02B 37/10; F16C 23/04; F16C 17/03

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Korean utility models and applications for utility models

Japanese utility models and applications for utility models

Electronic database consulted during the international search (name of database and, where practicable, search terms used)

eKOMPASS(KIPO internal) & Keywords: turbine, compressor, bearing, tilt, pad, curve, housing

C. DOCUMENTS CONSIDERED TO BE RELEVANT

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☐ Further documents are listed in the continuation of Box C. ☑ See patent family annex.

* Special categories of cited documents:
  "A" document defining the general state of the art which is not considered to be of particular relevance
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Date of the actual completion of the international search

28 JANUARY 2013 (28.01.2013)

Date of mailing of the international search report

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Name and mailing address of the ISA/KR

Korean Intellectual Property Office
189 Cheongsa-ro, Seo-gu, Daejeon Metropolitan City, 302-701, Republic of Korea

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Authorized officer

HAN Joong Sub

Telephone No. 82-42-481-5606

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**INTERNATIONAL SEARCH REPORT**
Information on patent family members

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