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BEARING HAVING SPACED PADS AND METHODS

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(57) Claim

1. A one-piece bearing for supporting a shaft which rotates about an axis, the bearing comprising: a plurality of circumferentially spaced bearing pads arranged around the axis, each of the bearing pads including an arcuate pad surface, opposed longitudinally extending circumferentially spaced edges and a pair of axially spaced axial edges extending between the circumferentially spaced edges; at least one circular base ring having an axis which is substantially coincident with the axis of rotation of the shaft; and a plurality of longitudinal beams, each of the longitudinal beams extending between the circular base ring and one of the bearing pads.

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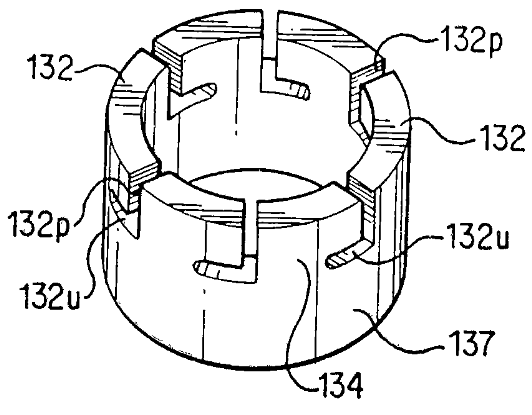


(57) Abstract

A hydrodynamic thrust, journal or combined radial and thrust bearing and methods of manufacturing the same. The bearing includes a bearing pad structure (132) that may change shape and move in any direction (six degrees of freedom) to optimize formation of a converging wedge for hydrodynamic operation, equalization of load on the bearing pads in thrust bearings and to adjust for any shaft misalignment. The bearing pad may be formed so as to contact the shaft in the installed state and to deflect under fluid film pressure. The pad may have undercut grooves (132u).

# BEARING HAVING SPACED PADS AND METHODS

## Background of the Invention

The present invention relates to hydrodynamic bearings. In such bearings, a rotating object such as a shaft is supported by a stationary bearing pad via a pressurized fluid such as oil, air or water. Hydrodynamic bearings take advantage of the fact that when the rotating object moves, it does not slide along the top of the fluid. Instead, the fluid in contact with the rotating object adheres tightly to the rotating object, and motion is accompanied by slip or shear between the fluid particles through the entire height of the fluid film. Thus, if the rotating object and the contacting layer of fluid move at a velocity which is known, the velocity at intermediate heights of the fluid thickness decreases at a known rate until the fluid in contact with the stationary bearing pad adheres to the bearing pad and is motionless. When, by virtue of the load resulting from its support of the rotating object, the bearing pad is deflected at a small angle to the rotating member, the fluid will be drawn into the wedge-shaped opening, and sufficient pressure will be generated in the fluid film to support the load. This fact is utilized in thrust bearings for hydraulic turbines and propeller shafts of ships as well as in the conventional hydrodynamic journal bearing.

Both thrust bearings and radial or journal bearings normally are characterized by shaft supporting pads spaced about an axis. The axis about which the pads are spaced generally corresponds to the longitudinal axis of the shaft to be supported for both thrust and journal bearings. This axis may be termed the major axis.

In an ideal hydrodynamic bearing, the hydrodynamic wedge extends across the entire bearing pad face, the fluid film is just thick enough to support the load, the major axis of the bearing and the axis of the shaft are aligned, leakage of fluid from the ends of the bearing pad surface which are adjacent the leading and trailing edges is minimized, the fluid film is developed as soon as the shaft begins to rotate, and, in the case of thrust bearings, the bearing pads are equally loaded. While an ideal hydrodynamic bearing has yet to be achieved, a bearing which substantially achieves each of these objectives is said to be designed so as to optimize hydrodynamic wedge formation.

The present invention relates to hydrodynamic bearings that are also sometimes known as movable pad bearings and methods of making the same. Generally these bearings are mounted in such a way that they can move to permit the formation of a wedge-shaped film of lubricant between the relatively moving parts. Since excess fluid causes undesirable friction and power losses, the fluid thickness is preferably just enough to support the maximum load. This is true when the formation of the wedge is optimized. Essentially the pad displaces with a pivoting or a swing-type motion about a center located in front of the pad surface, and bearing friction tends to open the wedge. When the formation of the wedge is optimized, the wedge extends across the entire pad face. Moreover, the wedge is formed at the lowest speed possible, ideally as soon as the shaft begins to rotate.

U.S. Patent No. 3,107,955 to Trumpler discloses one example of a bearing having beam mounted bearing pads that displaces with a pivoting or swing-type motion about a center located in front of the pad surface. This bearing, like many prior art bearings, is based only on a two dimensional model of pad deflection. Consequently, optimum wedge formation is not achieved.

In the Hall patent, U.S. No. 2,137,487, there is shown a hydrodynamic moveable pad bearing that develops its hydrodynamic wedge by sliding of its pad along spherical surfaces. In many cases the pad sticks and the corresponding wedge cannot be developed. In the Greene Patent, U.S. No. 3,930,691, the rocking is provided by elastomers that are subject to contamination and deterioration.

U.S. Patent 4,099,799 to Etsion discloses a non-unitary cantilever mounted resilient pad gas bearing. The disclosed bearing employs a pad mounted on a rectangular cantilever beam to produce a lubricating wedge between the pad face and the rotating shaft. Both thrust bearings and radial or journal bearings are disclosed.

There is shown in the Ide patent, U.S. No. 4,496,251 a pad which deflects with web-like ligaments so that a wedge shaped film of lubricant is formed between the relatively moving parts.

U.S. Patent 4,515,486 discloses hydrodynamic thrust and journal bearings comprising a number of bearing pads, each having a face member and a support member that are separated and bonded together by an elastomeric material.

U.S. Patent No. 4,526,482 discloses hydrodynamic bearings which are primarily intended for process lubricated applications, i.e., the bearing is designed to work in a fluid. The hydrodynamic bearings are formed with a central section of the load carrying surface that is more compliant than the remainder of the bearings such that they will deflect under load and form a pressure pocket of fluid to carry high loads.

It has also been noted, in Ide U.S. Patent No. 4,676,668, that bearing pads may be spaced from the support member by at least one leg which provides flexibility in three directions. To provide flexibility in the plane of motion, the legs are angled inward to form a conical shape with the apex of the cone or point of intersection in front of the pad surface. Each leg has a section modulus that is relatively small in the direction of desired motion to permit compensation for misalignments. These teachings are applicable to both journal and thrust bearings. While the disclosure of this patent represents a significant advance in the art, it has some shortcomings. One such shortcoming is the rigidity of the support structure and bearing pad which inhibits deformation of the pad surface. Further, the bearing construction is not unitary.

The last two patents are of particular interest because they demonstrate that despite the inherent and significant differences between thrust and journal bearings, there is some conceptual similarity between hydrodynamic journal bearings and hydrodynamic thrust bearings.

This application relates in part to hydrodynamic thrust bearings. When the hydrodynamic wedge in such bearings is optimized, the load on each of the circumferentially spaced bearings is substantially equal.

## Summary of the Invention

The present invention discloses a pad type bearing and methods of making the same. The pad type bearing, which is preferably unitary, can be formed from a single piece of heavy walled tubing or a cylindrical journal that has been machined or formed with small grooves and slits, bores or cuts through or on the bearing wall to define a flexible journal or thrust pad and a support structure. The pads and support structure are designed to optimize the shape of the converging wedge formed between the pad surface and the shaft when the shaft rotates. This can be done by modifying the pad shape, the support structure or both. Specifically, the pad can be modified to include grooves, cuts, rails and recesses to achieve desired deformations under load. The support structure can be designed to support the pads for movement in the six degrees of freedom (i.e., translation or movement in the +x, -x, +y, -y, +z and -z directions) and rotation about the X, Y, and Z axes so as to optimize formation of the hydrodynamic wedge, but this is not necessary for all applications.

The bearings of the present invention can be designed in three dimensions to provide deflection with six degrees of freedom so as to ensure optimum wedge formation at all times. Specifically, it has been discovered that a hydrodynamic bearing operates most effectively when the hydrodynamic wedge has several characteristics. In particular, the wedge should extend across the entire pad surface; the wedge should have an appropriate thickness at all times; the wedge should be shaped so as to minimize fluid leakage; the wedge should accommodate misalignment such that the major axis of the bearing is colinear or substantially parallel to the axis of the shaft; and the wedge should be formed at the lowest speed possible to prevent damage to the wedge forming surface which generally occurs as a result of shaft to pad surface contact at low speeds. Moreover, with thrust bearings, the loading among the spaced bearing pads should be equal.

With regard to thickness of the fluid film, it should be understood that the optimum thickness varies with loading. Under high or heavy loading, a relatively thick fluid film is desirable to adequately support the load. However, thicker films increase friction and power loss. Thus, the bearings are preferably designed to provide the minimum thickness necessary to support the shaft at maximum load.

The support structure is preferably unitary (one-piece) and comprises support stubs, beams, and/or membranes connected to a housing which is sometimes defined by the radially outermost portion of the bearing in the case of a journal bearing or, in the case of thrust bearings, a housing into which the bearing is mounted.

The inventor has discovered that in many specific applications such as in high speed applications, it is necessary to examine and evaluate the dynamic flexibility of the entire system consisting of the shaft or rotor, the hydrodynamic lubricating film and the bearing. In computer analysis of this system using a finite element model, it has been determined that it is necessary to treat the entire bearing as a completely flexible member that changes shape under operating loads. By adding more or less flexibility via machining of the basic structure, bearing characteristics may be achieved that provide stable low friction operation over wide operating ranges. A number of variables have been found to substantially affect the bearing's performance characteristics. Among the most important variables are the shape, size, location and material characteristics (e.g. modulus of elasticity etc.) of the pad and support members defined by the bores, slits or cuts and grooves formed in the bearing. The shape of the support members has been found to be particularly important. Also by providing a fluid backing to the flexible members, a high degree of damping may be achieved that further adds to system stability. In some instances, this damping has replaced secondary squeeze film dampening that is present when the oil film is present between the casing of the bearing and the housing.

Specific applications of the bearings of the present invention include electric motors, fans, turbochargers, internal combustion engines, outboard motors, and compressors/expanders. Test speeds have exceeded 300,000 r.p.m. It is noted that the cuts, grooves and openings, in addition to allowing the bearing pad to move to form a converging wedge for hydrodynamic lubrication, allow the pad itself to deflect and change shape by, for example, flattening. This improves operating performance by, among other things, changing the eccentricity of the bearing.

The bearings may be formed of metals, powdered metals, plastics, ceramics or composites. When manufactured in small quantities, the bearings are typically machined by facing, turning, and milling the blanks to form larger grooves or openings; smaller grooves are formed by water-jet cutting, electrical discharge or laser machining methods and allow total design flexibility to tune the bearing to provide desired characteristics. Tuning will essentially change the stiffness that in turn eliminates vibration. Manufacture of larger quantities of a single type bearing is preferably accomplished through injection molding, extrusion, powdered metal die casting, investment casting or some similar manufacturing technique. In accordance with one aspect of the present invention, intermediate quantities of bearings are manufactured according to a novel method combining machining and investment casting techniques. The present invention also contemplates easily moldable bearings which include no hidden openings such that they can be molded in a simple two-piece mold. In general, the bearings of the present invention can be manufactured at a fraction of the cost of competitive bearings.

Unlike prior pad type bearings which have a support structure that is essentially oriented in the direction of load, the present invention provides an orientation that allows for comparable deflections within a smaller envelope (i.e., the difference between the radially inner journal surface and the radially outer journal surface in journal bearings) especially in journal bearings; allows for movement of the bearing pad in any direction (i.e., six degrees of freedom) to form a converging wedge shape; allows for the pad itself to change shape (e g., flatten) to improve performance; allows for development of a membrane damping system for improved stability; and allows the bearings to compensate for misalignment of the supported part or shaft and to equalize loading among the bearing pads in a thrust bearing. All of these characteristics contribute to formation of an optimum hydrodynamic wedge.

While there are numerous arrangements of bores, grooves, cuts, or slits there are primarily two modes of deflections: namely, one or more ligaments or membranes which deflect in the general direction of load in a bending mode and secondly, by torsional deflection in a beam or membrane in a direction extending away from the pad along the longitudinal axis of the shaft in journal bearings. The degree of deflection in the bending mode is, in part, a function of the stiffness of the support structure in the radial direction. The pad itself may be made to deflect under a load to form a different shape by providing internal cuts beneath the pad or by undercutting the edges of the pad. In either case, the cuts are specifically made to result in a predetermined shape under load. By surrounding or backing certain ligaments or membranes with lubricating fluid, a damping element may be added to the design.

Similar cuts are used for journal bearings and thrust bearings. The primary determinant is the deflections desired for optimum performance. However, since journal and thrust bearings perform significantly differently functions there, are inherent differences in desired performance requiring different desired deflections. Consequently, despite the general conceptual similarity between the journal bearings and thrust bearings of the present invention there are also significant conceptual differences and plainly evident structural dissimilarities.

The bearing of the present invention includes a pad that may change shape and move in any direction (i.e., is supported for movement with six degrees of freedom). The bearing also may have a built-in damping system and is preferably of unitary or single piece construction for high volume economical manufacture. The journal bearings of the present invention also fits in a relatively small envelope (i.e., spacing between the housing outer diameter and the pad inner diameter).

In accordance with the present invention, a number of methods of manufacturing the bearings of the present invention are also contemplated. The selection of a particular method of manufacturing depends largely on the volume of the particular bearing to be manufactured and the materials used. In low volume applications, or when it is desired to produce prototypes for testing and/or production of molds or the like, the bearings are preferably manufactured from metallic cylindrical blanks such as heavy wall tubing or other journals which are machined to provided radial and/or facing bores or grooves and formed with radial cuts or slits through either numerically controlled electrical discharge manufacturing techniques, numerically controlled laser cutting techniques, or numerically controlled water-jet cutting. In intermediate volumes, the bearings of the present invention are preferably manufactured using an investment casting method in accordance with the present invention. In high volume applications, the bearings of the present invention can be manufactured using a wide variety of materials such as plastics, ceramics, powdered and non-powdered metals, and composites. In high volume applications, a number of manufacturing methods, including injection molding, casting, powdered metal, die casting, and extrusion, can be economically employed. The bearings of the present invention can be formed in a shape which is easily moldable.

**Brief Description of the Drawings**

The details of the invention will be described in connection with the accompanying drawings in which:

Fig. 1 is a perspective view of a thrust bearing according to the present invention.

Fig. 1A is a top view of the thrust bearing of Fig. 1.

Fig. 1B is a side view of the thrust bearing of Fig. 1.

Fig. 2 is a side view of another radial or journal bearing according to the present invention.

Fig. 2A is a flattened interior view of the bearing of Fig. 2 along the lines indicated in Fig. 2.

Fig. 2B is a flattened side view of the bearing of

Fig. 2.

Fig. 2C is a cross-section of the bearing of Fig. 2 along the lines indicated in Fig. 2B.

Fig. 3 is a side view of another radial bearing according to the present invention.

Fig. 3A is a flattened interior view of a portion of the bearing of Fig. 3.

Fig. 3B is a flattened edge view of the bearing of Fig. 3.

Fig. 3C is a cross-section of the bearing of Fig. 3 along the lines indicated in Fig. 3B.

Fig. 4 is a side view of another radial bearing according to the present invention.

Fig. 4A is a flattened interior view of a portion of the bearing of Fig. 4 along the lines indicated in Fig. 4.

Fig. 4B is a flattened edge view of a portion of the bearing of Fig. 4.

Fig. 4C is a cross-section of the bearing of Fig. 4 along the lines indicated in Fig. 4A.

Fig. 4D is a cross-section of the bearing of Fig. 4 along the lines indicated in Fig. 4A.

Fig. 5 is a schematic view of a conventional prior art turbocharging system used in an internal combustion engine.

Fig. 5A is a schematic view of a portion of a turbocharger including one of the bearings of the present invention.

**Detailed Description**

In describing the bearings of the present invention in an understandable way, it is helpful to describe the bearing structures as being formed from a cylindrical blank by providing grooves, slits, bores and other openings in the cylindrical blank. As This is sometimes a useful technique for manufacturing a prototype bearing. However, the reference to the cylindrical blank is primarily intended to assist understanding of the present invention. It should be noted that although many of the bearings of the present invention could be manufactured from a cylindrical blank, it is not necessary that any of them be so manufactured. Indeed the bearings can be manufactured in numerous ways, some of which are discussed hereinafter.

Figs. 1, 1A and 1B illustrate a thrust bearing according to the present invention. The bearing shown in these figures is a relatively simple bearing formed from a pipe or tube like cylindrical piece of material by providing circumferentially spaced axially extending pad defining grooves 132p on one end of the cylindrical member and extending these grooves with circumferentially extending undercut grooves 132u so as to define a circumferentially spaced series of pads 132 each of which is supported on a base portion 137 by a relatively rigid support post 134.

As is readily apparent from Figs. 1 and 1B, the pads 132 each have two circumferential edges one of which is undercut by the undercut groove 132u and the other of which is substantially rigidly supported by the support post 134. By virtue of this construction, the edge of the pad 132 which is undercut is less rigidly supported than the other edge of the pad. Thus, under loading, the undercut edge of the pad 132 is free to deflect downward whereas deflection of the other edge of the pad 132 is inhibited by the support post 134. Consequently, the pads 132 can deflect downwardly at one end to permit formation of a hydrodynamic wedge.

Since the bearing illustrated in Figs. 1-1B is relatively rigid, it is to be expected that there may be some pad to shaft contact at start-up. To prevent unnecessary wear the surface of the pad 132 could be provided with a wear surface such that the shaft to pad contact will not result in undesirable wear. In accordance with the preferred embodiment of the present invention, however, the entire bearing can be formed of a material having a high PV limit. The currently preferred material is CELEDYNE™ a material known to have a high PV limit. Tests have shown that a bearing formed of CELEDYNE™ and constructed as shown in Figs. 1-1B exhibits excellent wear characteristics at start-up and hydrodynamic wedge operation after start-up. CELEDYNE™ is also less rigid than metal materials thus, giving the relatively rigid bearing construction more flexibility.

Although the bearing construction shown in Figs. 1-1B does not include a support structure for supporting the pads 132 for movement with six degrees of freedom, it does offer a number of advantages. Specifically, the construction is quite simple and easy to mass produce at low cost. Moreover, even without providing six degrees of freedom, the bearing pads 132 provide far superior performance to plain bearings of similar material. Thus, the simple L-shaped groove formations provided by the pad defining grooves 132p and the undercutting grooves 132u dramatically improve pad performance. This embodiment illustrates how the teachings of the present invention can be applied to achieve improved results by modifying the support for a pad 132 by simply undercutting a portion of that support. This demonstrates how through finite element analysis modelling the pads 132 as a piece of putty by removing the specific portions of the overall structure superior results can be achieved.

Figs. 2-2C illustrate a radial or journal bearing construction according to the present invention. In this particular embodiment, the bearing is constructed with a very small radial dimension to allow the bearing to provide support in an extremely small radial envelope. In part, this small radial dimension is achieved by using longitudinal support beams as described below. Specifically, as shown in Figs. 2-2C, the bearing includes a series of circumferentially spaced bearing pads 32. Each of the bearing pads 32 includes circumferentially spaced longitudinally extending edges and axially spaced axial edges 32e. Each of the pads 32 is also preferably formed with a taper 32t to increase inlet bending as discussed previously. The pads 32 are supported at both axial edges by a longitudinal beam 34L. The longitudinal beams 34L are, in turn, supported by continuous base rails 30.

As best shown in Fig. 2C, the radially outer dimension of the pads 32 is flush with the radially outer dimension of the base rings 30. Consequently, the central region of the bearing pads 32 is rigidly supported. On the other hand, the circumferential edges of the bearing pad are free to deflect. By virtue of this support, the pads 32 are adapted to rock or pivot about the center of the pad so as to form a wedge with respect to the rotating shaft in the known manner.

Several other significant features and advantages of the bearing construction shown in Figs. 2-2C should be noted. One important advantage is that the bearing is constructed as a unitary piece. Among other things, this means that the position of the circumferentially spaced pads 32 with respect to one another is fixed with a single tolerance thus avoiding the problem of tolerance stack up which is inherent with any multi-piece design. Moreover, the bearing can be molded as a single piece at a fraction of the cost of multi-part assemblies particularly when both manufacturing and assembly costs are considered. Another important feature is that the pads 32 are supported only at their axial edges 32e such that the pads are easily supported for torsional twisting. I t should be noted that the degree of torsional flexibility depends on the dimensions of the longitudinal beams 34L. As best shown in Fig. 2C, the illustrated embodiment includes longitudinal beams 34L of a moderate dimension. Naturally, thin or thick beams could be used to vary torsional flexibility in the known manner. In the embodiment shown in Figs. 2-2C, the pad has little, if any, radial flexibility in its central region. In this particular embodiment, this is intended since it is only desired that the beam pivot to form a wedge. If, on the other hand, it were desirable to build some radial flexibility into the support for the pads 32 this could be done by providing rails radially extending the base rings 30 so as to provide rails as discussed elsewhere herein.

Another radial bearing construction is shown in Figs. 3-3C. This construction is similar to the construction shown in Figs. 2-2C and discussed above with several exceptions. One difference is that the pads 32 are more sharply tapered, as best shown in Fig. 3. As a consequence of this sharp taper, the edges of the pad are extremely flexible and would have a tendency to deflect a relatively great amount to allow inlet bending. Naturally, the degree of taper can be varied to suit desired performance characteristics.

The most significant functional difference, however, is that the longitudinal beams 34L have a triangular cross-section with the apex of the triangle providing a pivot point support for the pads as best shown in Fig. 3C. This pivot point support extends in a line across the back side of the pads. By virtue of this construction, the pads are in effect a series of triangular pads which have a tendency to readily deflect to form a wedge. The pads 32 have certain amount of stability, however, because of the support and torsional resistance of the longitudinal beams 34l. Further, the circumferential position and alignment of the individual pads 32 is assured by the integral base ring 30 which maintains the alignment of the various pads 32.

Another journal bearing construction is shown in Figs. 3-4D. This bearing construction is very similar to that shown in Figs. 3-3C except that it includes two sets of circumferentially spaced bearing pads spaced from each other by longitudinal connecting beams 34LC. In addition, the circular base rings 30 have radially extending rails 30r formed thereon so that the longitudinal beams 34L, 34LC and the pads 32 can move radially outward to give the bearing additional flexibility. As with the bearing of Figs. 2-2C, the longitudinal beams 34L, 34LC are connected to the pads at axial ends 32e of the pads 32. Thus, the beams act as torsional stabilizers and supports for the pads 32. Naturally, if desired the rails 30r provided on the base rings 30 can be eliminated so that the support acts as a simple tilt support as with the embodiment of Figs. 2-2C.

An advantage of the construction shown in Figs. 4-4D is that a shaft can be supported at two separate locations along the shaft by a single bearing. Moreover, the two locations of support are readily and permanently aligned because of the longitudinal connecting beams 34lc.

The bearing construction shown in Figs. 4-4D is particularly well suited for supporting a high speed shaft which must be supported in a very small radial envelope. One such application is a turbocharger of the type commonly used in connection with automobiles. When used in such applications, the bearing is preferably formed of a high temperature polymer such as CELEDYNE™. An example of such an application is discussed below in connection with Figs. 5 and 5A.

Fig. 5 schematically illustrates a conventional turbocharging arrangement for use in internal combustion engines. The simple system includes a turbine wheel 2 and a compressor wheel 3 both of which are supported o' a shaft 5 which is in turn supported on shaft bearings 5b. The shaft and wheels are mounted in the air passages of the engine which also includes a carburetor 4, a combustion chamber 6 which includes a piston, spark plug and valves and an actuator 7.

The operation of such a conventional system is well known. Specifically, the exhaust gas pressure and heat energy causes the turbine wheel 2 to rotate. This causes the compressor wheel 3 to rotate. Air is mixed with fuel by the carburetor 4. The rotating compressor wheel 3 compresses the air fuel mixture it receives from the carburetor and delivers it under pressure to the intake manifold 8a. As a result, a denser charge enters the combustion chamber 6. The denser charge and the combustion chamber 6 develops more horsepower during the combustion cycle. Exhaust gas from the exhaust manifold 8b flows into the turbine 2. When the manifold pressure reaches a set value, the actuator 7 opens the waste gate to bypass some exhaust gas. The cooled expanded exhaust gas is directed by the turbine housing to the exhaust system via a passage 8c.

One of the most common problems with conventional systems of this type is that the bearings fail as a result of the extreme temperatures and high speeds of the shaft rotation.

Fig. 5A schematically illustrates how a bearing of the type shown in Figs. 4-4D can be incorporated into an otherwise conventional turbocharging system. Specifically, the two pad bearing is mounted in the housing so as to support the shaft 5 which in turn carries the turbine wheel 2 and the compressor wheel 3. Tests have shown that the simple one-piece bearing of the type shown in Fig. 4-4D provides dramatically improved results over conventional rolling element or tilt pad bearing constructions.

When manufacturing low volumes or prototypes of the bearings of the present invention, the bearings are preferably constructed by electrical discharge machining or laser cutting methods. Such small volumes or prototypes are usually constructed of metal. However, when higher volume production of a particular bearing is contemplated, other methods of manufacture such as injection molding, casting, powdered metal die casting and extrusion are more economical. In connection with such manufacturing methods, it may be more economical to employ plastics, ceramics, powdered metals or composites to form the bearings of the present invention. It is believed that methods such as injection molding, casting, powdered metal die casting with sintering and extrusion are sufficiently well known that the processes need not be detailed herein. It is also believed that once a prototype bearing is constructed, the method of producing a mold or the like for mass production of the bearing is well known to those skilled in the molding and casting art.

What is Claimed is:

1. A one-piece bearing for supporting a shaft which rotates about an axis, the bearing comprising: a plurality of circumferentially spaced bearing pads arranged around the axis, each of the bearing pads including an arcuate pad surface, opposed longitudinally extending circumferentially spaced edges and a pair of axially spaced axial edges extending between the circumferentially spaced edges; at least one circular base ring having an axis which is substantially coincident with the axis of rotation of the shaft; and a plurality of longitudinal beams, each of the longitudinal beams extending between the circular base ring and one of the bearing pads.

2. The bearing of claim 1, wherein each longitudinal beam extends between the circular base ring and one of the axial edges of one of the bearing pads.

3. The bearing of claim 1, further comprising a second circular base ring and a second set of longitudinal beams extending between the second circular base ring and the bearing pads.

4. The bearing of claim 1, wherein the longitudinal beams have a triangular cross-section with the apex of the triangle directed away from the pad surface.

5. The bearing of claim 4, wherein the pad has a triangular cross-section with the apex of the triangle being a aligned with the apex of the longitudinal beams such that the pads and beams rest on the apex for tilting movement of the apex.

6. The bearing of claim 1, wherein the longitudinal beams have a substantially rectangular cross-section.

7. The bearing of claim 1, wherein the circular base rings, the longitudinal beams and the pads each have a maximum radial dimension and wherein the maximum radial dimension of the circular base rings exceeds the maximum radial dimension of both the longitudinal beams and the pads such that the base rings support the longitudinal beams and the pads for radially outward deflection.

8. The bearing of claim 1, further comprising:

a second set of circumferentially spaced bearing pads, the second set of bearing pads being axially spaced from the first set of bearing pads;

means for maintaining the axial spacing between the first and second sets of bearing pads; and

means for supporting both the first and the second set of spaced bearing pads for tilting movement while simultaneously maintaining the circumferential spacing and axial alignment of the bearing pads.

9. The bearing of claim 8, in combination with a turbocharger.

10. A one-piece bearing for supporting a rotating shaft, the bearing comprising:

a first set of circumferentially spaced bearing pads, each of the bearing pads including an arcuate pad surface, opposed longitudinally extending circumferentially spaced edges and a pair of axially spaced axial edges extending between the circumferentially spaced edges;

a second set of circumferentially spaced bearing pads, each of the bearing pads including an arcuate pad surface, opposed longitudinally extending circumferentially spaced edges and a pair of axially spaced axial edges extending between the circumferentially spaced edges, the second set of circumferentially spaced bearing pads being axially spaced from the first set of circumferentially spaced bearing pads;

a plurality of longitudinal connecting beams, each longitudinal connecting beam extending between an axial edge of one of the bearing pads of the first set of bearing pads and an axial edge of one of the bearing pads of the second set of bearing pads;

a first circular base ring located adjacent to but spaced from the first set of circumferentially spaced bearing pads; a first set of longitudinal support beams, each longitudinal support beam extending between the first circular base ring and the axial edge of each of the bearing pads in the first set of bearing pads which is opposite to the axial edge from which the longitudinal connecting beam extends;

a second circular base ring located adjacent to but spaced from the second set of circumferentially spaced bearing pads; and

a second set of longitudinal support beams, each longitudinal support beam extending between the circular base ring and the axial edge of one of the bearing pads of the second set of circumferentially spaced bearing pads which is opposite the axial edge from which the longitudinal connecting beam extends.

11. The bearing of claim 10, wherein the longitudinal beams have a triangular cross-section with the apex of the triangle directed away from the pad surface.

12. The bearing of claim 11, wherein the pad has a triangular cross-section with the apex of the triangle being a aligned with the apex of the longitudinal beams such that the pads and beams rest on the apex for tilting movement of the apex.

13. The bearing of claim 10, wherein the longitudinal beams have a substantially rectangular cross-section.

14. The bearing of claim 10, wherein the circular base rings, the longitudinal beams and the pads each have a maximum radial dimension and wherein the maximum radial dimension of both the circular base rings exceeds the maximum radial dimension of both the longitudinal beams and the pads such that the base rings support the longitudinal beams and the pads for radially outward deflection.

15. A unitary thrust bearing comprising:

a one-piece substantially cylindrical body having an axis and a cylindrical radially inner surface, a cylindrical radially outer surface spaced radially outward from the cylindrical inner surface, and two axially spaced edge surfaces at least one of which is substantially planar; the cylindrical body being formed with a plurality of circumferentially spaced axially extending grooves extending between the inner and outer surfaces from the planar edge surface so as to define a plurality of circumferentially spaced substantially planar bearing pads; a plurality of circumferentially extending grooves, each circumferentially extending groove extending between the inner and outer radial surfaces in continuance of an axially extending groove so as to undercut a portion of the bearing pad defined by the axially extending groove.

16. The thrust bearing of claim 15, wherein the bearing is formed of CELEDYNE™.

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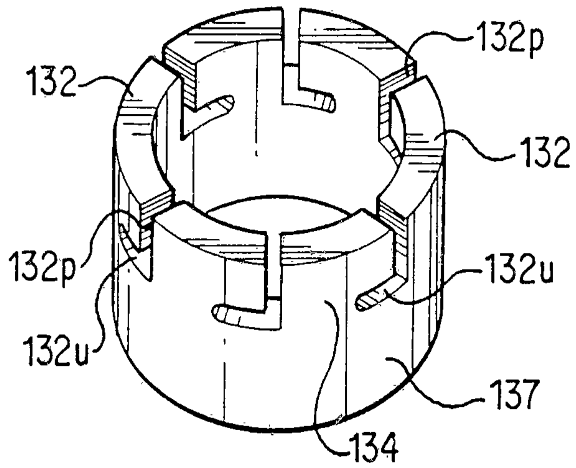


FIG. 1

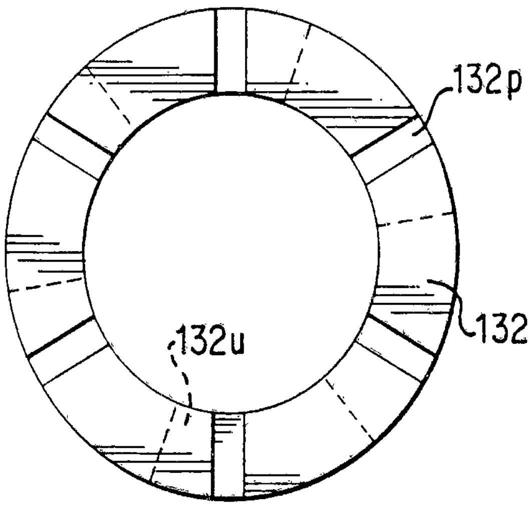


FIG. 1A

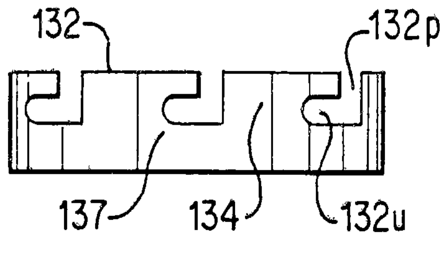


FIG. 1B

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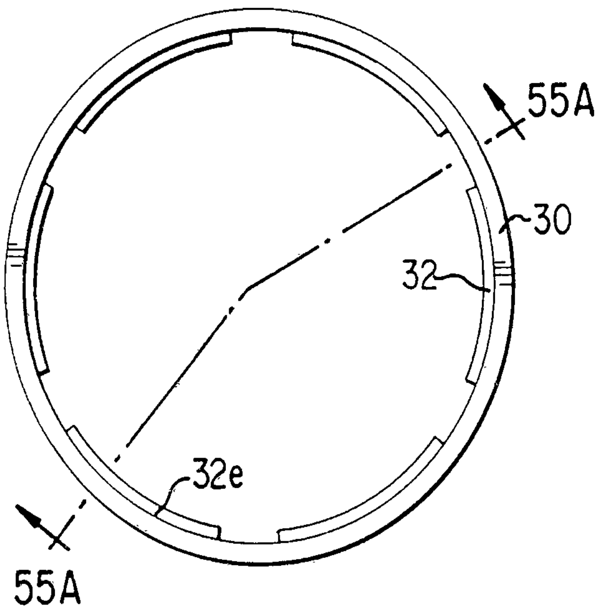



FIG. 2

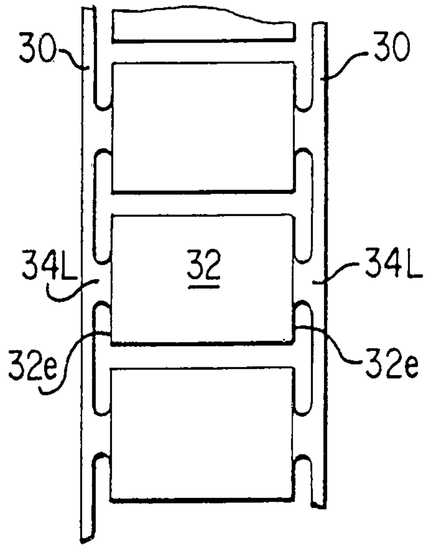



FIG. 2A

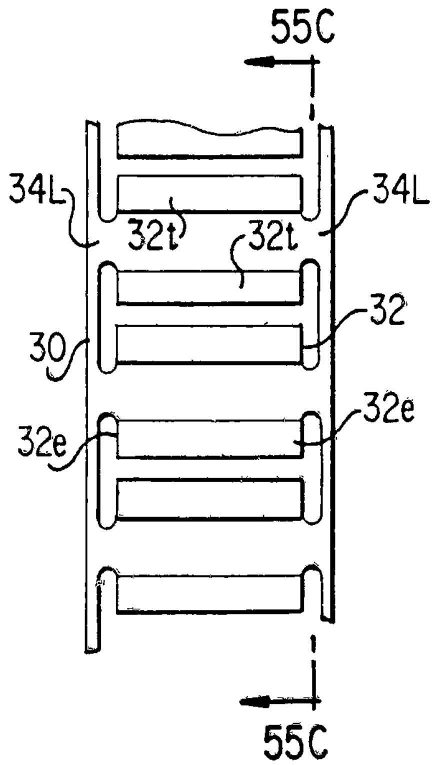



FIG. 2B

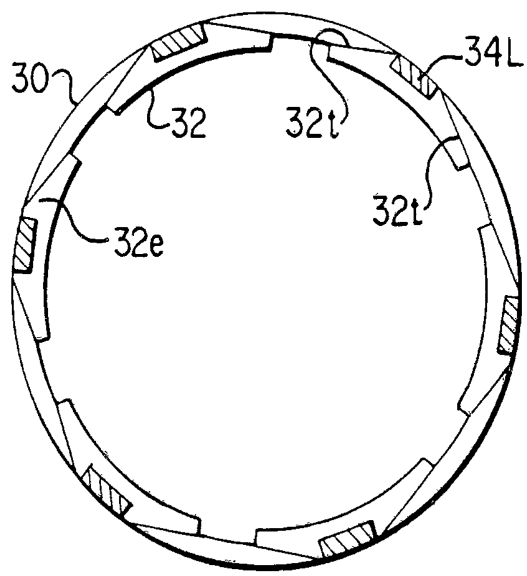
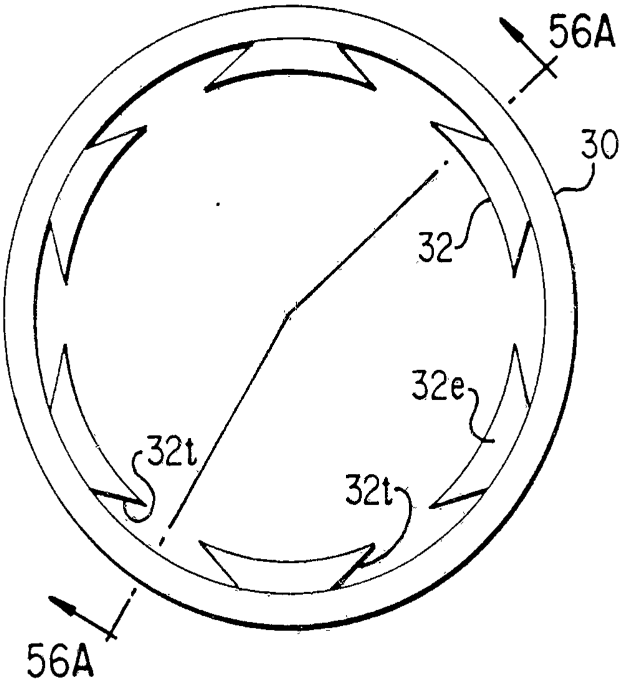



FIG. 2C



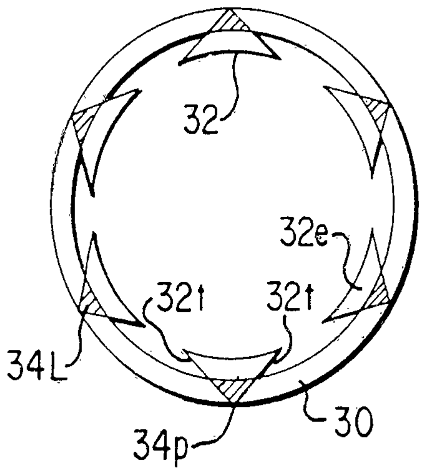
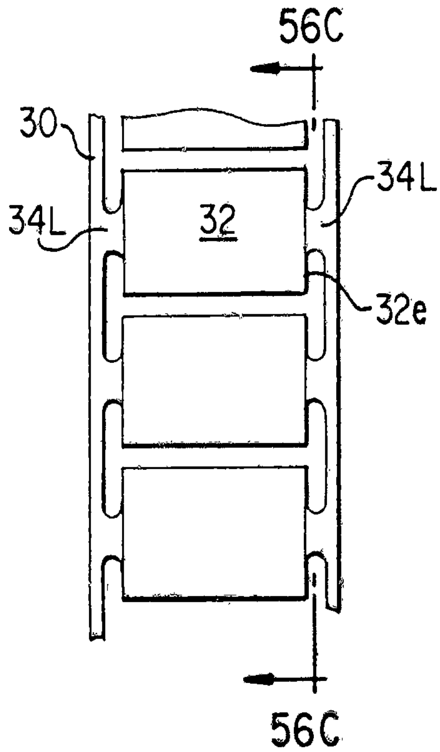




FIG. 3C

FIG. 3



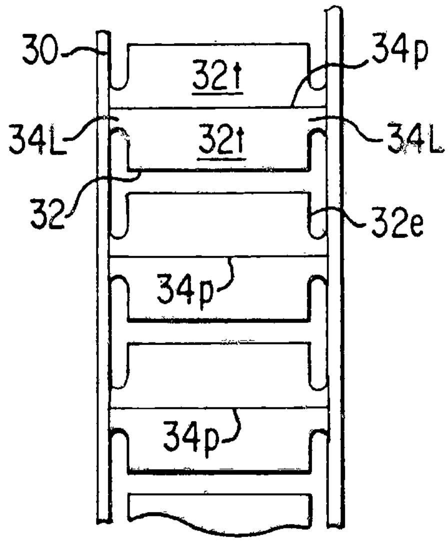




FIG. 3B

FIG. 3A

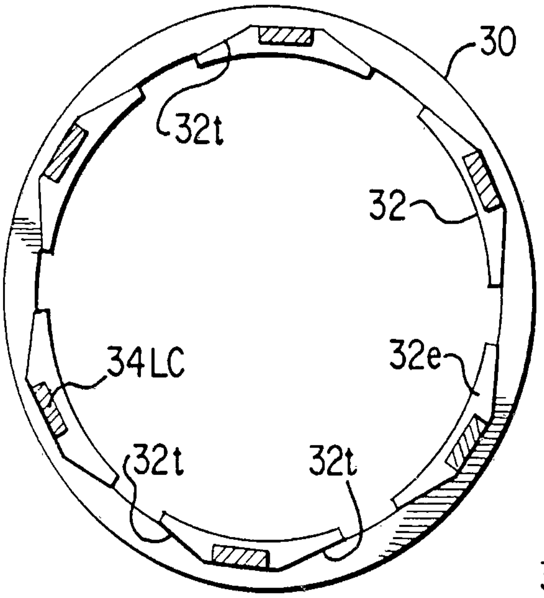



FIG. 4C

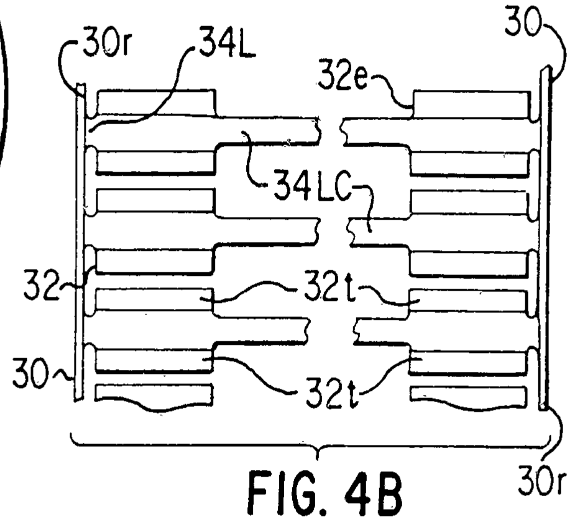



FIG. 4B

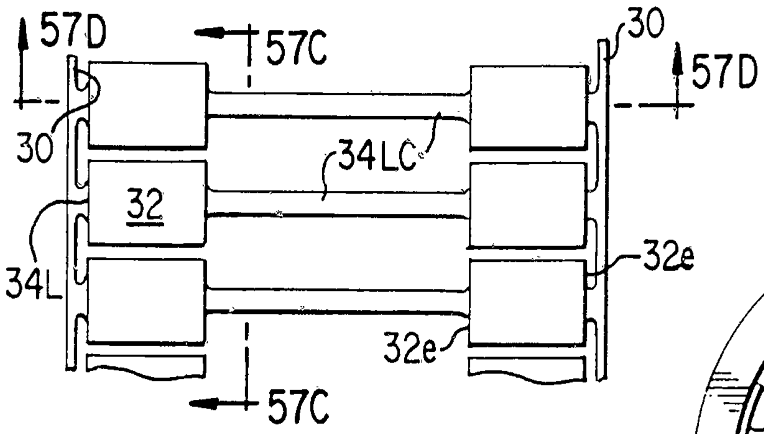



FIG. 4A

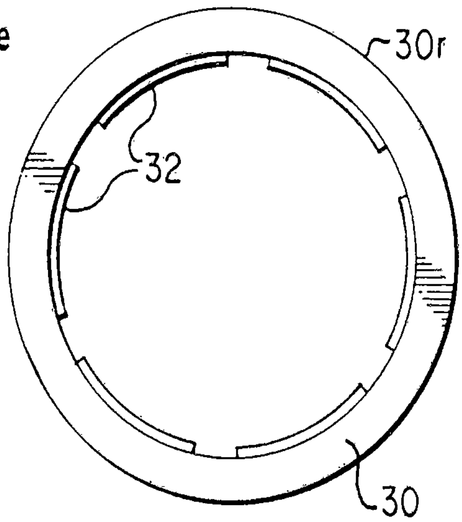



FIG. 4

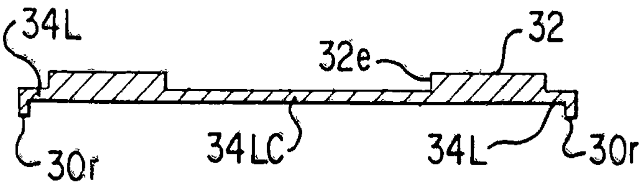



FIG. 4D

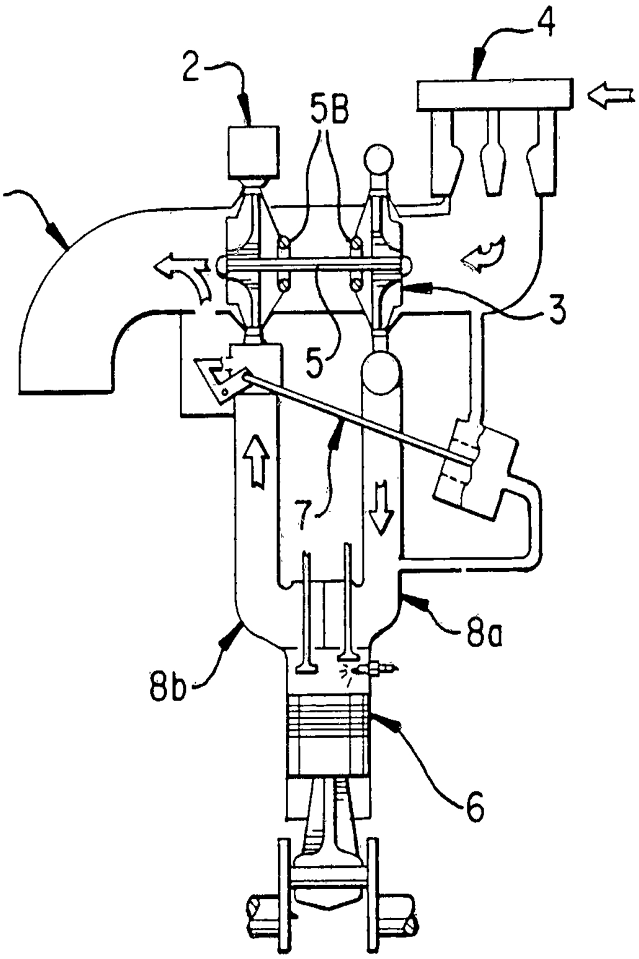



FIG. 5 PRIOR ART

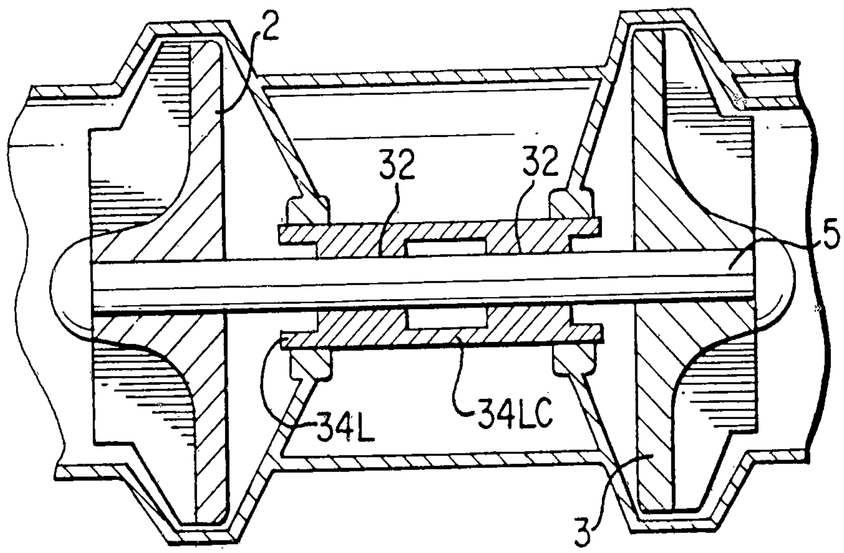



FIG. 5A

# INTERNATIONAL SEARCH REPORT

International application No.
PCT/US93/09372

**A. CLASSIFICATION OF SUBJECT MATTER**

IPC(5) :F16C 17/03, 06
US CL :384/107
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)

U.S. : 384/107, 117, 119, 122, 124, 308, 312

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

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

**C. DOCUMENTS CONSIDERED TO BE RELEVANT**

| Category* | Citation of document, with indication, where appropriate, of the relevant passages | Relevant to claim No. |
|---|---|---|
| Y | US, A, 4,676,668 (IDE) 30 June 1987 (See Fig. 6.) | 1-14 |
| Y | US, A, 2,424,028 (HAEBERLEIN) 15 July 1947 (See entire document.) | 15-16 |

☐ 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 part of particular relevance

"E" earlier document published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

| Date of the actual completion of the international search | Date of mailing of the international search report |
|---|---|
| 26 November 1993 | 16 DEC 1993 |

| Name and mailing address of the ISA/US | Authorized officer |
|---|---|
| Commissioner of Patents and Trademarks<br>Box PCT<br>Washington, D.C. 20231 | Sheila Veney for<br>LENARD A. FOOTLAND |
| Facsimile No. NOT APPLICABLE | Telephone No. (703) 308-2683 |

## INTERNATIONAL SEARCH REPORT

International application No.
PCT/US93/09372

**Box I Observations where certain claims were found unsearchable (Continuation of item 1 of first sheet)**

This international report has not been established in respect of certain claims under Article 17(2)(a) for the following reasons:

1. ☐ Claims Nos.:
because they relate to subject matter not required to be searched by this Authority, namely:

2. ☐ Claims Nos.:
because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out, specifically:

3. ☐ Claims Nos.:
because they are dependent claims and are not drafted in accordance with the second and third sentences of Rule 6.4(a).

**Box II Observations where unity of invention is lacking (Continuation of item 2 of first sheet)**

This International Searching Authority found multiple inventions in this international application, as follows:

GROUP I. Claims 1-9 drawn to a radial bearing classified in class 384, subclass 119.
GROUP II. Claims 10-14 drawn to pivot bearing classified in class 384, subclass 117.
GROUP III. Claims 15-16 draw a thrust bearing classified in class 384, subclass 122.

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims.

2. ☒ As all searchable claims could be searched without effort justifying an additional fee, this Authority did not invite payment of any additional fee.

3. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims for which fees were paid, specifically claims Nos.:

4. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claims Nos.:

**Remark on Protest**
☐ The additional search fees were accompanied by the applicant's protest.
☐ No protest accompanied the payment of additional search fees.