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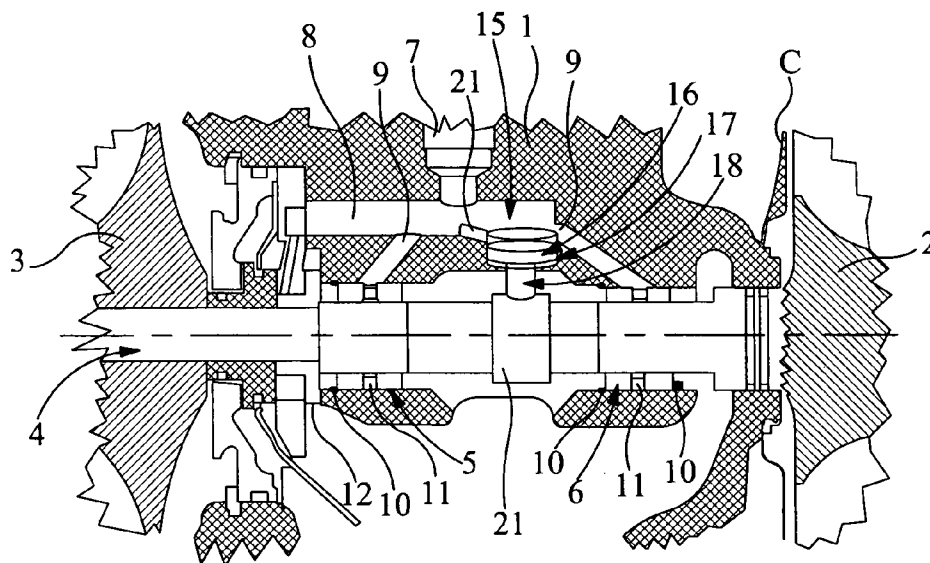


Fig 1

(57) Abstract: A turbocharger shaft (4) is supported for rotation about its axis in a bearing housing (1) by a pair of journal bearings (5,6) disposed between the turbine wheel (2) and the compressor impeller (3). A device (15) such as a hydraulic or magnetic actuator applies a non-axial force to the shaft at an axial location offset from the bearing so as to suppress non-axial sub-synchronous shaft vibrations that occur during operation of the turbocharger. This reduces turbocharger noise.

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TURBOCHARGER WITH VIBRATION SUPPRESSING DEVICE

The present invention relates to a turbocharger and, in particular, to an arrangement for suppressing vibrations in a turbocharger shaft.

Turbochargers are well known devices for supplying air to the intake of an internal combustion engine at pressures above atmospheric (boost pressures). A conventional turbocharger essentially comprises an exhaust gas driven turbine wheel mounted on a rotatable shaft within a turbine housing. Rotation of the turbine wheel rotates a compressor wheel mounted on the other end of the shaft within a compressor housing. The compressor wheel delivers compressed air to the intake manifold of the engine, thereby increasing engine power.

The turbocharger shaft is supported for rotation by journal bearings in a bearing housing that is intermediate the compressor and turbine housings. In addition one or more thrust bearings may provide axial support. In automotive heavy-duty diesel engine applications turbocharger shafts are typically supported for rotation in the bearing housing by two separate floating ring bearings which are retained in position by circlips or some other conventional mechanical configuration. In a floating ring bearing, the shaft rotates relative to an inner bearing surface defined by a bearing ring, which also defines an outer bearing surface which itself rotates relative to the surrounding housing. A supply of lubricant is delivered (e.g. through passages in the bearing housing) to the bearings so as to provide inner and outer hydrodynamic films of bearing lubricant between the shaft and inner bearing surface and between the outer bearing surface and the housing respectively. In a fully floating ring bearing the bearing ring rotates but at a rotational velocity less than that of the shaft, whereas in a semi-floating ring bearing the bearing body is stationary. The inner and outer film pressures of the films are dictated at least in part by the rotations. The film thicknesses can be optimised for damping performance depending on the conditions.

At turbocharger operating speeds the shaft and bearings often exhibit significant sub-synchronous vibrations (i.e. vibrations of a frequency less than the shaft rotation frequency) that propagate in a generally radial direction. This phenomenon is known as “oil whirl” or “oil whip” and the vibrations can result in an undesirable increase in turbocharger noise as a result of their transmission to the

bearing housing. This can lead to reduced reliability and durability of the bearings as in such conditions they operate with reduced clearances.

It is an object of the present invention, amongst others, to obviate or mitigate the aforementioned disadvantage. It is also an object of the present invention to provide for an improved bearing arrangement.

According to a first aspect of the present invention there is provided a turbocharger comprising a turbine wheel, a compressor impeller and an interconnecting shaft that extends through a housing of the turbocharger and is supported for rotation about an axis by at least one bearing disposed between the wheel and the impeller, and a device for applying a non-axial force to the shaft at an axial location offset from the at least one bearing so as to suppress non-axial shaft vibrations during operation of the turbocharger.

The suppression of vibrations by the device includes damping them, eliminating them and/or altering the vibration mode frequency such that the mode is not excited at the range of angular velocities of the shaft in the normal operating range of the turbocharger. The vibrations that are suppressed occur in the normal operating range of the turbocharger (or would otherwise occur in the absence of the non-axial force) and are preferably sub-synchronous i.e. they are of a frequency lower than the rotational frequency of the shaft.

The non-axial vibrations that are suppressed by the device are preferably those caused by oil whip and/or oil whirl in which the shaft axis describes a substantially cylindrical locus or one or more substantially conical loci. Such vibrations are sometimes referred to in the art as rotor conical and cylindrical lateral modes.

The device may apply a positive or negative force to the shaft i.e. the force may act in a direction towards or away from the shaft. The force is non-axial, that is it has a radial component; it may be applied in a radial direction, but not necessarily.

The application of the force on the shaft causes a reaction force from the at least one journal bearings to the shaft on the basis that it is not balanced around the circumference of the shaft in the regions of application. This serves to restrict movement of the shaft in the plane occupied by the force direction and the shaft axis and thus prevents the non-axial vibrations. In particular it prevents at least two modes of sub-synchronous vibration, namely: vibrations that cause the shaft axis to describe

a locus that is substantially cylindrical and those that cause the shaft axis to describe a locus comprising two adjacent point-to-point cones.

There may be a pair of bearings in the form of spaced journal bearings, which may be floating ring bearings of the kind that comprise at least one bearing body that defines an outer bearing surface and an inner bearing surface around the shaft. Inner and outer hydraulic films of bearing lubricant are provided between the inner bearing surface and shaft, and the outer bearing surface and a housing in which the bearing is disposed. In a fully floating ring bearing the bearing body rotates but at a velocity less than that of the shaft. In a semi-floating ring bearing the bearing body is substantially stationary.

The force may be applied anywhere along the length of the shaft between the impeller and turbine wheels provided it is offset from the bearing or bearings i.e. it should not coincide with the position of the bearing(s) but can be applied at any position between them or axially outboard of them.

The device may comprise a hydrostatic bearing arrangement comprising a lubrication fluid delivery passage connected to a hydrostatic bearing cavity disposed around at least a portion of the shaft, the cavity retaining hydraulic fluid supplied from the delivery passage at a predetermined pressure so as to apply said non-axial force.

The applied force may be predetermined.

The device may be disposed wholly or partially within the bearing housing of the turbocharger.

The device may comprise an actuator which may be hydraulic or otherwise. In the case of a hydraulic actuator pressurised hydraulic fluid is supplied to move the actuator. The fluid may comprise oil from an oil supply to bearing housing of turbocharger. A separate oil feed passage may interconnect an oil passage or gallery of the bearing housing to the actuator.

The actuator may be connected to the shaft via a hydrodynamic bearing arrangement. The actuator may be in the form of a piston and cylinder with a rod of the piston defining or being connected to the hydrodynamic bearing arrangement. The piston may be penetrated by an oil passage for supplying oil to the bearing arrangement. The bearing arrangement may be provided by a bore that receives said

shaft with a clearance for the hydraulic fluid. The bore may be defined in the piston rod or by a member attached to the rod. The member may be a bearing sleeve.

In one embodiment the actuator may be magnetic so that it applies an attractive or repulsive force to the shaft. The actuator may comprise a permanent magnet or an electromagnet.

The actuator may be damped.

There may be a plurality of actuators disposed along the length of shaft.

There may be a pair of journal bearings. The force may be applied between the bearings or axially outboard of them.

According to a second aspect of the present invention there is provided a method for operating a turbocharger, the turbocharger comprising a turbine wheel, a compressor impeller and an interconnecting shaft that extends through a housing of the turbocharger and is supported for rotation about an axis by at least one bearing disposed between the wheel and the impeller, comprising the step of applying a non-axial circumferentially unbalanced force to the shaft to one side of the bearing so as to suppress non-axial shaft vibrations that occur during operation of the turbocharger.

The vibrations are preferably sub-synchronous as described above.

The non-axial vibrations suppressed may be non-circumferential, that is vibrations other than torsional vibrations about the axis of the shaft. These can typically be caused by twisting of the shaft.

Specific embodiments of the present invention will now be described, by way of example only, with reference to the accompanying drawings, in which:

Figure 1 is an axial sectioned view of a central part of a turbocharger including a central part of the bearing housing and part of the compressor and turbine, and incorporates a schematic representation of an actuator for vibration suppression in the turbocharger shaft;

Figure 2 is a perspective view of a turbocharger in accordance with the present invention shown partially cut away to reveal the interior of the bearing housing, the configuration being similar to that shown in figure 1;

Figure 3 is a schematic representation of a turbocharger shaft bearing arrangement with a hydraulic actuator for vibration suppression;

Figure 3a is an end view of a lower part of the actuator and the shaft of figure 3;

Figure 4 is a schematic representation of a turbocharger shaft bearing arrangement with a spring-loaded actuator and hydrodynamic bearing for vibration suppression;

Figure 5 is a schematic representation of a turbocharger shaft bearing arrangement with a magnetic device for vibration suppression;

Figure 6 is a schematic representation a turbocharger shaft bearing arrangement with a hydrostatic bearing for vibration suppression; and

Figure 6a is a cross section of part of the hydrostatic bearing of figure 6.

Referring to figures 1 and 2, the illustrated turbocharger comprises a turbine joined to a compressor via a central bearing housing 1. The turbine comprises a turbine wheel 2 (hidden behind a cover C in figure 2) rotating within a turbine housing (not shown). Similarly, the compressor comprises a compressor impeller 3 that rotates within a compressor housing (not shown). The turbine wheel 2 and compressor impeller 3 are mounted on opposite ends of a common turbocharger shaft 4 that extends through the central bearing housing 1.

In use, the turbine wheel 2 is rotated by the passage of exhaust gas passing over it from the internal combustion engine. This in turn rotates the compressor wheel 3 that draws intake air through a compressor inlet and delivers boost air to the inlet manifold of an internal combustion engine via an outlet volute (not shown).

The turbocharger shaft 4 rotates on fully floating journal bearings 5 and 6 housed towards the turbine end and compressor end respectively of the bearing housing 1. Oil is fed to the bearings under pressure from the oil system of the engine via an oil inlet 7, gallery 8 and passages 9. Each journal bearing 5, 6 is retained in place by circlips 10 and is provided with circumferentially spaced radial holes 11 for oil to pass to the turbocharger shaft 4.

A thrust bearing assembly 12 (figure 1 only) flanks the journal bearing 5 at the compressor end.

At a location along the shaft between the two bearings 5, 6 there is provided a device 15 for applying a non-axial force to the shaft 4. In the example illustrated by the embodiments of figures 1 to 3, the device 15 is in the form of an actuator

comprising a piston and cylinder arrangement in which a piston head 16 is reciprocally disposed in a cylinder 17 and a piston rod 18 extends towards the shaft 4 in a substantially radial direction. The rod 18 is coupled to the shaft 4 by virtue of a transverse bore 19 in which the shaft 4 is received with a radial clearance 20. In the embodiment of figure 1 there is provided a sleeve 21 at one end of the rod 18 that defines said transverse bore, said sleeve being removed in the embodiment of figure 2 for clarity. In the embodiment of figure 3 the sleeve is not shown and the bore 19 is depicted as penetrating the rod 18, although it is to be understood that lateral sleeve portions may be provided on each side of the rod to provide an enlarged bearing area. Oil is supplied to an upper surface 16a of the piston head 16 via a supply bore 21 that is connected to the oil supply 8,9 in the bearing housing 1 and is transmitted to the clearance 20 between the bore wall and shaft 4 via a central passage 22 that penetrates the piston head 16 and rod 18. In the figure 2 embodiment the actuator 15 is angularly offset from the oil gallery and passages 8, 9 with respect to the shaft. The supply bore in the bearing housing 1 (fig. 2) comprises a first portion 21a defined in a wall of the cylinder 17 which extends in parallel with the piston rod 18 and then a second portion 21b that extends through an upper edge of the wall and into the cylinder 17 at a position above the upper surface 16a of the piston head 16.

In operation, the piston 16, 18 is slidably supported in the cylinder 17 by a piston ring seal 25, moves up and down in response to the oil pressure P and the rod 18 thereby applies a substantially radial force to the shaft 4. The configuration of the interconnection between the shaft 4 and the piston rod 18 is configured to be a hydrodynamic bearing by virtue of the oil film which forms a lubricating wedge in the clearance 20 between the bore 19 in (or attached to) the piston rod 18 of the actuator and the turbocharger shaft 4. The central passage 22 through the piston has a relatively small diameter and may serve to damp reciprocal movement of the piston. In this embodiment the engine lubricating oil supplied to the bearing housing 1 serves as the hydraulic fluid but it is to be appreciated that a separate supply of pressurised hydraulic fluid may be delivered to the actuator instead. Using the engine oil supply ensures that the static load is not applied until the oil is delivered to the hydrodynamic bearing 19, 20 whereby the bearing is not over-loaded with the application of a force prior to delivery of the oil.

A modification to the above-described embodiment is illustrated in figure 4 in which the actuator 15 is supplemented with a coil spring 26 disposed between the upper surface 16a of the piston head 16 and a facing wall of the cylinder. Any other type of elastic or resilient member may be used to provide a desired biasing force. The purpose of the spring is to modify the force being applied to the shaft 4 at the hydrodynamic bearing 19, 20. In the configuration shown the spring 26 ensures there is a small force applied at all times in addition to that applied by the oil pressure. In an alternative configuration the spring 26 or other biasing member may be provided between the lower surface 16b of the piston head 16 and the facing wall of the cylinder so as to provide a small biasing force that acts in opposition to the force applied by the oil pressure on the piston. This latter arrangement ensures that the actuator does contact the shaft or otherwise apply a potentially damaging force during engine start-up and shut-down when there is little or no oil present in the bearing.

In an alternative embodiment illustrated in figure 5, the device 15 for applying a non-axial force to the shaft 4 takes the form of a magnetic actuator. The actuator is positioned in the bearing housing 1 at a predetermined radial distance from the shaft at a position substantially intermediate the two floating ring bearings 5, 6 so as to define an air gap 27 (in some applications this may be filled with oil). The actuator may take any suitable form and may include, for example, a simple a permanent magnet or an electromagnet 28. The poles of the magnetic actuator are arranged to apply an appropriate repulsive or attractive force to the turbocharger shaft 4 which is sufficient to suppress or eliminate lateral vibrations in the shaft. The magnetic air gap 27 between the actuator and the shaft is pre-selected to ensure that sufficient load is applied without contact between the actuator and the shaft. This arrangement has the advantage that the force between the magnet 28 and the shaft 4 will vary less (in comparison to other actuator arrangements) as the radial position of the shaft varies about its nominal axis as a result of design tolerances.

In a variation to this design an electromagnet actuator may be employed in the same manner as the piston actuator described above with a separate oil supply being directed to a hydrodynamic bearing connection between the actuator and the shaft.

In an alternative arrangement the magnetic actuator may be positioned in, on or around the shaft with adjacent ferromagnetic elements being disposed in the

bearing housing asymmetrically about the shaft axis. In general, if the magnet is aligned with the shaft axis the electromagnetic induction losses from the shaft's axial rotation can be reduced or minimised. Alternatively, induction losses may be avoided by using non-conducting materials such as carbon fibre.

In a further alternative embodiment, the actuator may take the form of an asymmetric ferrous metal element that surrounds the shaft and which is made from or supports thereon a ferromagnetic element. This may be in the form of a magnetic collar, which preferably has its magnetic axis aligned with the rotor. Such ferromagnetic elements, whether associated with the shaft or the bearing housing, preferably have a high Curie temperature such as SaCo or AlNiCo magnets, and are preferably located closer to the compressor impeller than the turbine wheel.

In a yet further embodiment there is provided a first element at the compressor end of the shaft, attracted to an off-axis, or asymmetric second element in the compressor housing, at least one of the first or second elements being ferromagnetic. An example of this the first element may be a magnetized nut (preferably with aligned magnetic and rotation axes) used to fasten the compressor impeller to the shaft, where the nut is attracted to, or otherwise acted on by the second element extending across or through the compressor inlet to at least one side of the nut. Preferably the element is contoured to minimise air resistance.

In a further example of the preceding embodiment, the second element is concave and is of ferrous metal partially or fully surrounding the nut on one side of the rotation axis. In another arrangement, the second element is magnetised in a similar direction as the nut so as to repel it, whereas on the other side the element may attract the nut using ferromagnetic or ferrous metal means. As an alternative to a nut, the end of the shaft may be magnetised.

A further alternative embodiment of the device for applying a non-axial force to the shaft is illustrated in figures 6 and 6a. The device takes the form of a hydrostatic bearing arrangement 30 in which an oil delivery passage 31 supplies oil to a pocket or recess 32 defined in the bearing housing 1 immediately adjacent to and partially surrounding the shaft 4. Again the device is shown intermediate the two floating ring bearings 5, 6 but it is to be understood, as with all the other embodiments discussed herein, that the exact position of the device can be varied in relation to the

bearings 5, 6 as required, as discussed further below. The oil present (represented in dotted line in figure 6a) in the pocket 32 provides a hydrostatic bearing 30 that applies a static force to the shaft. As with the other hydraulic actuator embodiments the hydraulic fluid may be provided from the engine oil supply or a separate supply of pressurised hydraulic fluid may be provided.

The axial position of the device for applying the load, and therefore the point of application of the load along the shaft, need not necessarily be exactly intermediate the two floating ring bearings, indeed it could be applied at a position outboard of one or both of the bearings towards the end of the shaft, particularly as applying the force there can lead to greater reaction forces at the journal bearings 5, 6 which may be desirable.

The application of a non-axial force in any of the above-described ways serves to suppress, restrict or eliminate sub-synchronous shaft vibrations that occur in a non-axial direction and/or in a non-circumferential direction in the operating range of the turbocharger. This prevents propagation of the vibrations to other parts of the turbocharger such as, for example, the bearing housing where significant noise can be generated. In the process the life expectancy of the bearings is extended. It also enables them to be considered for use in harsher environments or conditions. The arrangement may simply have the effect of modifying the vibration mode frequency of the shaft and bearing arrangement such that the mode is not excited at the range of angular velocities of the shaft that occur during the normal operating range of the turbocharger. The vibrations are caused by the circumferential flow of the oil films in the floating ring bearings and this flow is modified by the non-axial force. The application of the force is reacted by the bearings and thus disturbs the oil film to the extent that the vibrations are suppressed. The vibrations that are suppressed are sub-synchronous i.e. they are of a frequency lower than the rotational frequency of the shaft.

The application of the force on the shaft, given that it is not applied equally around the entire circumference of the shaft but is rather unbalanced around the circumference, causes a reaction force from the journal bearings 5, 6 to the shaft 4. This serves to restrict movement of the shaft in the plane occupied by the force direction and the shaft axis and thus prevents at least two modes of sub-synchronous

vibration, namely: vibrations that cause the shaft axis to describe a locus that is substantially cylindrical and those that cause the shaft axis to describe a locus comprising two adjacent point-to-point cones.

The force may be applied by the device directly or indirectly on to the shaft and it may be static (constant) or dynamic (variable). In the case of a dynamic force the magnitude may be adjustable in response to a measured or inferred value of a parameter of the turbocharger system such as, for example, turbocharger rotational speed or another operating condition of the turbocharger (e.g. which may be determined by reference to a compressor map stored in a look-up table). Moreover, in more sophisticated systems, there may be provided one or more transducers for determining the amplitude and/or frequency of the shaft vibration that allow a dynamically variable force to be applied by the device at the appropriate frequency in opposite phase in order to cancel out the vibration. The magnitude of the force will depend on many circumstances including the size and nature of the turbocharger or shaft assembly. However, it will be appreciated that it should be sufficient to suppress the sub-synchronous vibrations but not sufficient to interfere with, or impair operation of, the shaft by overloading the bearings or producing significant eccentric shaft motion. For a given turbocharger system the magnitude of the force will typically be determined empirically. If the device is configured to apply a constant force over the entire operating range of the turbocharger the magnitude is set to be slightly greater than the minimum force required at the maximum intended angular velocity of the shaft, thereby providing a suitable safety margin. The magnitude will be a function of the position of the applied force, the configuration and dimensions of the bearing assembly, the maximum operating speed of the turbocharger or shaft assembly and in the case of a hydraulic actuator, the temperature and pressure of the hydraulic fluid supplied. Moreover, the required magnitude depends on factors such as the size of the turbocharger, the length of the shaft (i.e. the distance between the centres of mass of the impeller and turbine wheels) and the distance between the centres of the journal bearings. For most turbocharger applications it is envisaged that the force will not exceed 500N. The minimum load requirement is likely to be greater than 25N and more probably greater than 50N or even 75N.

By way of example, for smaller turbochargers where the distance between the wheels is typically around 60mm, the force required to suppress the vibrations is greater than around 50N and less than around 350N and typically around 200N. For larger turbochargers where the distance between the wheels is typically 200mm, the lower force limit is around 100N and the upper limit around 500N.

Based on a preliminary experimental data, a relationship to define the possible upper and lower force limits could be expressed as follows:

The lower limit may be:

$$F (N) > 0 + 0.17 \times RL(mm)$$

Where RL is the shaft (rotor) length as defined above, the distance between the bearings being dependent on the shaft length.

More preferably the lower limit might be expressed as:

$$F (N) > 30 + 0.4 \times RL(mm);$$

or more preferably still as:

$$F (N) > 113 + 0.63 \times RL(mm)$$

The upper limit could be expressed as:

$$F (N) < 363 + 1.33 \times RL(mm);$$

or more preferably:

$$F (N) < 280 + 1.1 \times RL(mm);$$

or more preferably still as:

$$F(N) < 196 + 0.86 \times RL(mm)$$

In applications where more than one force is applied by the same device or separate devices it is desirable that the forces act in generally the same direction particularly when the forces act between the bearings. The forces should not be in opposition unless they are axially separated and it is not absolutely necessary that the forces act in parallel directions.

The devices described above may be configured to apply the force with resilient flexibility. The resilience may be engineered such that there is negligible variation in the force in response to non-axial movement of the shaft owing to design tolerances. Any of the devices may be provided with appropriate damping mechanisms to assist in damping the shaft vibrations.

The device is suitable for application to a normal heavy-duty diesel engine turbochargers but the principle may be extended to any other rotating shaft assembly.

It is to be appreciated that numerous modifications to the above-described embodiments may be made without departing from the scope of the invention as defined in the appended claims. For example, it will be understood that the precise shape and configuration of the components that make up the bearing assembly may vary. Moreover, the device may be housed wholly or partially within the bearing housing as depicted above or may be disposed wholly outside the bearing housing. In the hydraulic device embodiments, the standard oil feed galleries and passages in the turbocharger bearing housing may be moved in a circumferential direction relative to the shaft to accommodate the device in an axially central position at the top of the bearing housing.

CLAIMS

1. A turbocharger comprising a turbine wheel, a compressor impeller and an interconnecting shaft that extends through a housing of the turbocharger and is supported for rotation about an axis by at least one journal bearing disposed between the wheel and the impeller, and a device for applying a non-axial circumferentially unbalanced force to the shaft at an axial location offset from the at least one journal bearing so as to suppress non-axial shaft vibrations during operation of the turbocharger.
2. A turbocharger according to claim 1, wherein there is provided a pair of spaced journal bearings.
3. A turbocharger according to claim 1 or 2, wherein the at least one journal bearing is, or the pair of spaced journal bearings are, floating ring bearings.
4. A turbocharger according to claim 2 or 3, wherein the device is arranged so as to apply the force to the shaft at a position between the pair of bearings.
5. A turbocharger according to any preceding claim, wherein the device is disposed wholly or partially within the housing of the turbocharger.
6. A turbocharger according to any one of claims 1 to 4, wherein the device is disposed outside of the housing.
7. A turbocharger according to any preceding claim, wherein the device is a hydraulic actuator.
8. A turbocharger according to claim 7, wherein the actuator is connected to a source of pressurised hydraulic fluid.

9. A turbocharger according to claim 8, further comprising a hydraulic fluid supply passage extending between the actuator and an oil passage or gallery defined in the housing.
10. A turbocharger according to claim 8 or 9, wherein the actuator is a piston that is reciprocally disposed in a chamber.
11. A turbocharger according to claim 10, wherein the piston comprises a head disposed in the chamber and a rod that extends toward the shaft.
12. A turbocharger according to claim 10 or 11, the chamber being connectable to a source of pressurised hydraulic fluid so as to drive said piston and apply said force to said shaft.
13. A turbocharger according to claim 10, 11 or 12, wherein a biasing element acts on said piston so as to bias it towards or away from the shaft.
14. A turbocharger according to any one of claims 7 to 13, wherein the actuator is connected to the shaft by means of a hydrodynamic bearing arrangement.
15. A turbocharger according to claim 13, wherein the hydrodynamic bearing arrangement comprises a member defining a substantially annular bearing surface that receives said shaft with a radial clearance for the hydraulic fluid.
16. A turbocharger according to claim 15, when dependent from claim 11, wherein said member is the rod which has a transverse bore for receipt of said shaft with a radial clearance for hydraulic fluid.

17. A turbocharger according to claim 15, wherein the member is in the form of a sleeve disposed around the shaft.
18. A turbocharger according to any one of claims 14 to 17, wherein the actuator is penetrated by a passage for delivery of hydraulic fluid to said hydrodynamic bearing arrangement.
19. A turbocharger according to any preceding claim, wherein the device comprises a hydrostatic bearing arrangement comprising a lubrication fluid delivery passage connected to a hydrostatic bearing cavity disposed around at least a portion of the shaft, the cavity retaining hydraulic fluid supplied from the delivery passage at a pressure sufficient so as to apply said non-axial force.
20. A turbocharger according to any preceding claim, wherein the device is a magnetic actuator.
21. A turbocharger according to claim 20, wherein the actuator comprises a permanent magnet or electromagnet disposed at a predetermined distance from said shaft.
22. A turbocharger according to claim 20, wherein the actuator comprises a permanent magnet or electromagnet disposed in, on or around said shaft and at least one ferromagnetic element disposed adjacent to said shaft.
23. A turbocharger according to any preceding claim wherein there is a plurality of devices disposed along the length of shaft.
24. A turbocharger according to any preceding claim, wherein the non-axial force is applied to suppress non-axial sub-synchronous vibrations of the shaft.

25. A turbocharger according to any preceding claim, wherein the non-axial force is applied to suppress non-axial and non-circumferential vibrations of the shaft
26. A method for operating a turbocharger comprising a turbine wheel, a compressor impeller and an interconnecting shaft that extends through a housing of the turbocharger and is supported for rotation about an axis by at least one bearing disposed between the wheel and the impeller, comprising the step of applying a non-axial circumferentially unbalanced force to the shaft at an axial location offset from the bearing so as to suppress non-axial shaft vibrations during operation of the turbocharger.
27. A method according to claim 26, wherein the non-axial force is applied to suppress sub-synchronous non-axial shaft vibrations
28. A method according to claim 26 or 27, wherein the non-axial force is applied to suppress non-circumferential vibrations in the shaft.

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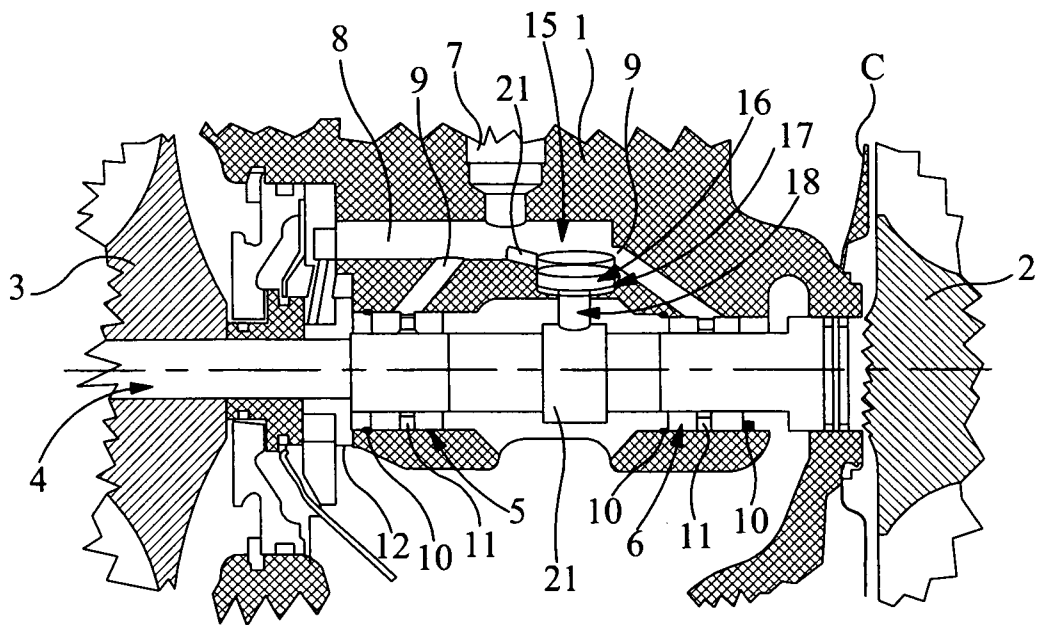


Fig 1

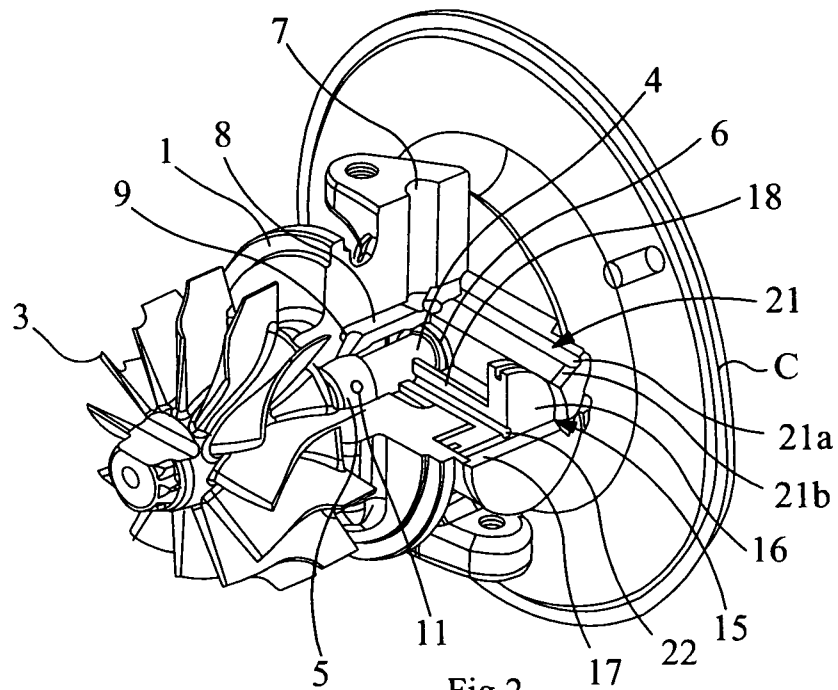


Fig 2

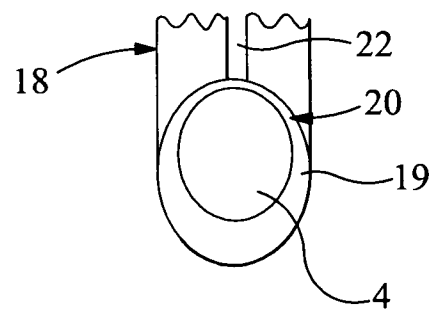
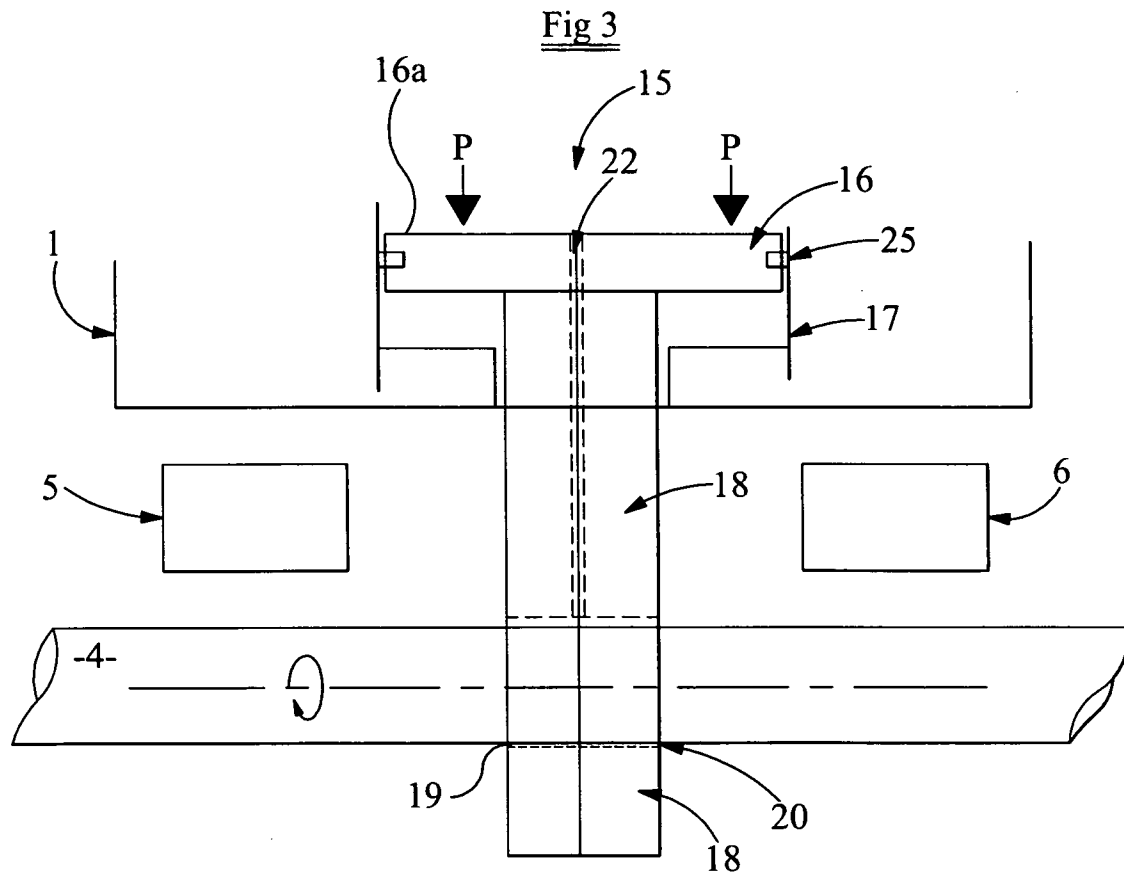


Fig 3a

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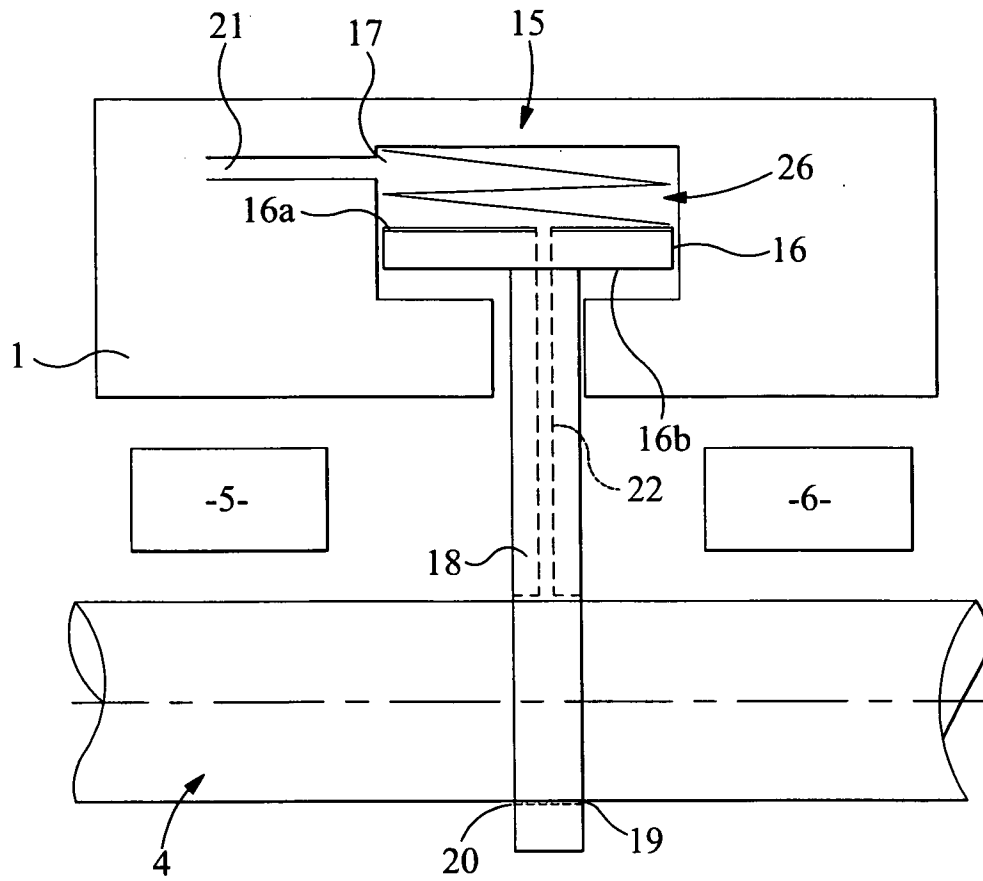


Fig 4

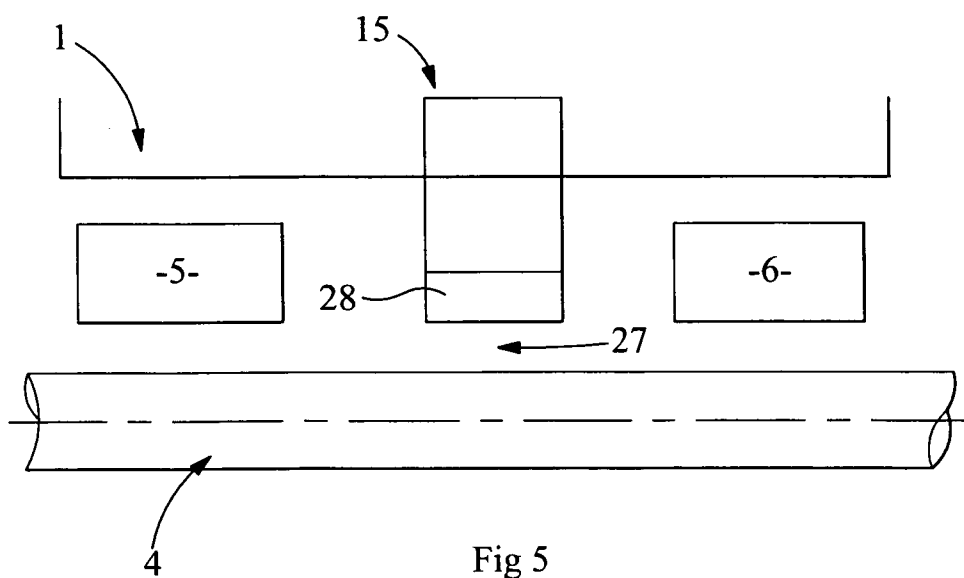


Fig 5

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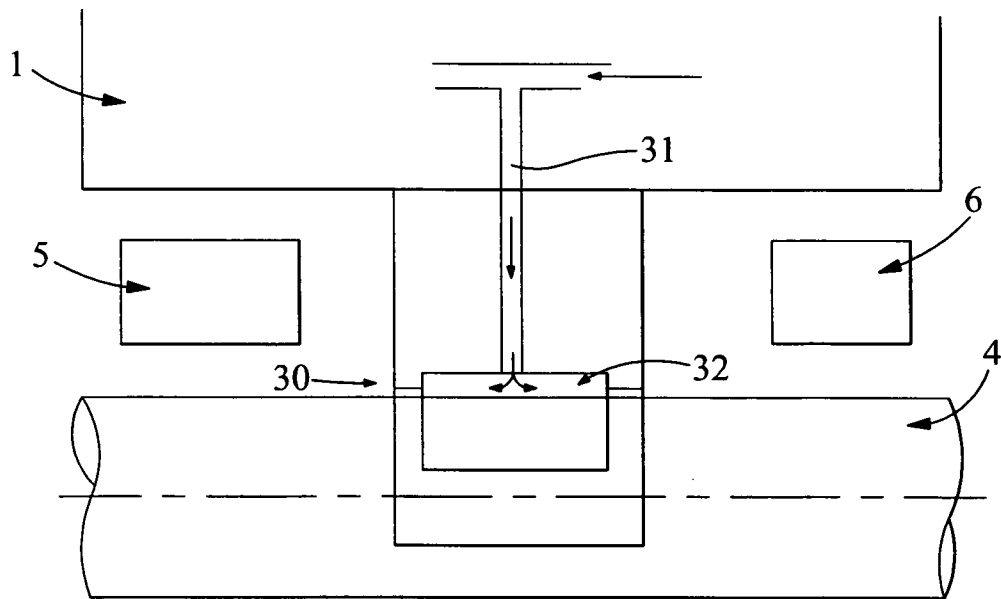


Fig 6

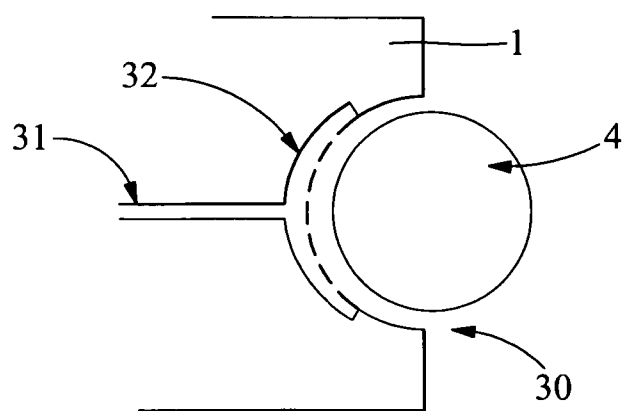


Fig 6a

INTERNATIONAL SEARCH REPORT

International application No.
PCT/GB2008/002240

A. CLASSIFICATION OF SUBJECT MATTER

INV. F01D5/10 F01D25/16 F02C6/12

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)

F01D F02C

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 practical, search terms used)

EPO-Internal, PAJ, WPI Data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 2004/011340 A1 (HOSNY DIAA [US] ET AL) 22 January 2004 (2004-01-22) paragraphs [0003], [0028], [0041], [0046], [0054]; figures 5,9	1-3,5, 23-28
X	US 6 418 722 B1 (ARNOLD STEVEN DON [US]) 16 July 2002 (2002-07-16) column 1, lines 34,35; figure 1 column 1, line 66 - column 2, line 12 column 2, line 63 - column 3, line 8	1,2,4,5, 7-9,14, 15, 17-19,26
X	JP 2003 307103 A (EBARA CORP) 31 October 2003 (2003-10-31) paragraphs [0014], [0017], [0018], [0023]; figures 2,3	1,2,4,5, 20-22,26
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☒ Further documents are listed in the continuation of Box C.

☒ See patent family annex.

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* & * document member of the same patent family

Date of the actual completion of the international search

13 October 2008

Date of mailing of the international search report

27/10/2008

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INTERNATIONAL SEARCH REPORT

International application No

PCT/GB2008/002240

C(Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
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A	US 5 248 239 A (ANDREWS FRANK X [US]) 28 September 1993 (1993-09-28) figure 3 -----	1-28

INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No

PCT/GB2008/002240

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