Abstract Title: Turbocharger rotor assembly

A turbocharger rotor assembly comprises a turbine wheel 5 and a compressor wheel 7 mounted to a rotor shaft 8 for rotation about a turbocharger axis. The turbine wheel 5 is mounted to a turbine shaft portion 8a and the compressor wheel 7 is mounted to a compressor shaft portion 8b. The turbine shaft 8a and compressor shaft 8b are connected to each other to define said rotor shaft 8. The turbine shaft 8a and compressor shaft 8b may be connected by a screw threaded connection 30, 32, a bore of which may be provided in either the turbine shaft or the compressor shaft. Alternatively a splined connection may be provided. A lock nut 25, oil slinger 33, thrust collar 33a, and thrust bearing 42 may be provided. In a method of assembling the turbocharger, a compressor wheel and compressor shaft sub-assembly, and a turbine wheel and turbine shaft sub-assembly are independently balanced.
**Turbomachine**

The present invention relates to a turbomachine, and in particular to a rotor of a turbomachine.

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 one end of 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 conventionally supported by journal and thrust bearings, including appropriate lubricating systems, located within a central bearing housing connected between the turbine and compressor wheel housing. The shaft, the attached compressor wheel and the turbine wheel (attached to opposite ends of the shaft) collectively form a rotor assembly. It is known that the balance of the rotor assembly is important so as to ensure the shaft rotates true within the bearings. Any imbalance may be detrimental to the operation and longevity of the turbocharger. For instance, an imbalance in the rotor assembly may lead to an increase in noise and vibration of the turbocharger as well as increased wear, particularly of the bearing surfaces. In extreme cases, an imbalance of the rotor assembly may for instance lead to the compressor wheel moving radially outward with respect to the axis of rotation of the shaft, potentially resulting in a catastrophic failure of the turbocharger.

The process of achieving acceptable rotor assembly balance is typically a multi-step process. First, the specific components which make up the rotor assembly are balanced individually before assembly. Next, the turbine wheel is friction welded to the shaft and then the shaft and attached turbine wheel are balanced. Subsequently, the shaft and attached turbine wheel are inserted into the turbocharger and the other components of the rotor assembly, including the compressor wheel, are attached to the
opposite end of the shaft to complete rotor assembly. Finally, the rotor assembly is then balanced in situ in the turbocharger. Balancing the rotor assembly involves adjusting the balance and/or position of the components of the rotor assembly so as to compensate for any imbalance. Adjusting the balance of a component may occur by either adding or removing material from a specific portion of that component. Alternatively, individual components of the rotor assembly can be swapped in order to try to obtain a combination of components which form a balanced rotor assembly. Since balancing of the rotor assembly, and in particular the adding or removal of material, is carried out whilst the rotor assembly is in situ, the procedure can be complex, as access to the rotor assembly is limited. In addition, as compressor wheels are often made of a softer material than the shaft, it is well-known that attachment, removal and replacement of the compressor wheel to and from the shaft during the balancing process may cause irreparable damage to the compressor wheel. Any such damaged compressor wheel may have to be discarded.

It is an object of the present invention to obviate or mitigate at least some of the above problems.

According to the present invention there is provided a turbocharger rotor assembly, comprising a turbine wheel and a compressor wheel mounted to a rotor shaft for rotation therewith about an axis, wherein:
the turbine wheel is mounted to a turbine shaft;
the compressor wheel is mounted to a compressor shaft; and
the turbine shaft and compressor shaft are connected to each other to define said rotor shaft.

The turbocharger rotor assembly according to the present invention may thus be constructed as two sub-assemblies, namely a turbine end sub-assembly comprising the turbine wheel mounted to the turbine shaft portion and a compressor end sub-assembly comprising a compressor wheel mounted to the compressor shaft portion. The two rotor shaft portions may then be connected together coaxially to produce the complete rotor shaft. One advantage of this construction is that each of the sub-
assemblies may be separately balanced away from the turbocharger and before the rotor shaft is assembled on the turbocharger.

The turbine shaft and compressor shaft may be releasably connected to one another. This will for instance allow this assembly of the rotor assembly to allow one or more components of the assembly to be replaced.

The turbine shaft and compressor shaft may be connected together by respective screw threaded surfaces which engage one another. For instance, one of the turbine shaft and compressor shaft may be provided with a screw thread on a portion of its outer surface, and the other shaft may be provided with an axially extending screw threaded bore to receive the screw threaded shaft. For instance in one embodiment of the invention the compressor shaft is externally screw threaded and screws into a bore defined along the axis of the turbine shaft. It will be appreciated that in this case the shaft with the external screw thread will have a smaller diameter, or at least a portion of its axial length will have a smaller diameter, than the outer diameter of the shaft defined in the internal bore (or socket) to receive the other shaft. The shaft provided with an external screw thread, and the threaded bore, may extend along a substantial portion of the length of the rotor shaft. Alternatively the bore may be relatively short as the externally threaded shaft may be a stub shaft.

It will be appreciated that the screw thread may be arranged so that the rotational torque transferred from the turbine wheel to the compressor wheel when the rotor shaft assembly is installed in a working turbocharger will tend to tighten rather then loosen the screw threaded engagement between the turbine shaft and compressor shaft.

Other means may be used to connect the compressor shaft to the turbine shaft. For instance, the connection may be a splined connection rather than a screw-threaded connection.
According to the invention, the compressor shaft is mounted to the compressor wheel. In some embodiments a first end of the compressor shaft may extend through a bore provided along the rotational axis of the compressor wheel, and the compressor wheel may be retained on the shaft by a nut which threads on said first end of the compressor shaft and bears directly or indirectly against the compressor wheel to clamp the wheel against a radial abutment defined by a portion of the compressor shaft. The nut may, for instance bear directly against a nose portion of the compressor wheel, or a washer or the like may be positioned between the locking nut and the compressor wheel. Similarly, the compressor wheel may be clamped against the radial abutment provided by the shaft, or a washer or the like may be provided between the two. In alternative embodiments the bore may only extend part way through the compressor wheel and a first end of the compressor shaft may be received within the bore and be attached by way of co-operating screw threads or by any other suitable means of attachment. As a further alternative the compressor wheel may comprise a stub which extends into a corresponding bore which extends part way into the compressor shaft. Again, co-operating screw threads may be used to attach the compressor wheel stub and compressor shaft to one another, although any other suitable means of attachment may be used. Methods of mounting a compressor wheel to a turbocharger shaft are known and any conventional method may be used to mount a compressor wheel to the compressor shaft in accordance with the present invention.

The radial abutment may for instance be an annular shoulder defined by an enlarged diameter portion of the compressor shaft. Such an enlarged diameter portion of the compressor shaft may for instance define a seal boss adapted to be received with an aperture defined by a turbocharger housing wall separating the compressor wheel from a turbocharger bearing cavity, and wherein an annular groove is defined in a radially outer surface of the seal boss to receive a seal ring. The turbocharger housing wall may for instance be a wall of the bearing housing or the back plate of the compressor housing (this often referred to as the compressor back plate) or a combination of both. The seal ring may, for instance, be a conventional piston ring or other suitable seal member as for instance typically carried by an oil slinger in conventional compressor end seal/bearing arrangements.
Abutting surfaces of the compressor wheel and the enlarged diameter portion of the compressor shaft may extend substantially perpendicular to the axis or may for instance be generally frustoconical about the axis of the shaft such that they each lie on the same generally conical surface of revolution about said axis. In the latter case the abutting surfaces may then interact so as to tend to centre the compressor wheel about said axis as an axial clamping force is applied to the compressor wheel. The surfaces may for example lie on the surface of a cone defined by rotation of a straight line about said axis, or a generally conically shaped surface defined by rotation of a curved line about said axis. The abutting surface defined by the compressor wheel may be concave, the abutting surface defined by the enlarged diameter portion of the compressor shaft may be convex, or vice versa.

Abutting surfaces between the compressor wheel and the lock nut may be similarly profiled to assist in centering the compressor wheel on the compressor shaft. For example, a nose portion of the compressor wheel may have a concave or convex conical surface which mates with a convex or concave conical surface provided by the lock nut, or by any component (such as a washer or the like) located between the lock nut and the compressor wheel.

An oil slinger and/or thrust collar may be supported on the rotor shaft and be clamped between a radially extending surface defined by the turbine shaft and said radial abutment defined by a portion of the compressor shaft, or other radially extending surface defined by the turbine shaft or compressor shaft.

For instance, in embodiments of the invention in which the compressor shaft is received within an axial bore defined in the turbine shaft, the thrust collar/oil slinger may be clamped between a radial surface defined at the end of the turbine shaft and an annular surface defined by the seal boss provided on the compressor shaft. The thrust collar/oil slinger may be a single component or the thrust collar may be separate to the oil slinger. Where the two are formed as a single component, a thrust bearing
maybe provided with a radially extending slot to allow the thrust bearing to be fitted to the thrust collar.

The turbine wheel may be connected to the turbine shaft in any conventional manner. Similarly, the turbine shaft may be provided with a piston boss adjacent the turbine wheel which defines an external annual groove for receiving a seal ring in a conventional manner.

The present invention provides a turbocharger comprising a rotor assembly according to any preceding claim.

The present invention further provides a method of assembling a turbocharger rotor assembly in a turbocharger comprising a turbine and a compressor located at opposite ends of a bearing cavity, a first housing wall separating the turbine from the bearing cavity and a second housing wall separating the compressor from the bearing cavity, each housing wall being provided with an aperture for receiving a turbocharger rotor shaft which extends from the turbine to the compressor through said bearing cavity, the method comprising:

providing a first rotor sub-assembly comprising a turbine wheel mounted to a turbine shaft;

providing a second rotor sub-assembly comprising a compressor wheel mounted to a compressor shaft;

independently balancing the first and second rotor sub-assemblies;

inserting the turbine shaft into the bearing cavity through the aperture provided in the first housing wall and inserting the compressor shaft into the bearing cavity through the aperture provided in the second housing wall;

and coaxially connecting the turbine shaft and compressor shaft together to complete the turbocharger rotor assembly.
According to another aspect of the present invention there is also provided a method of servicing a turbocharger including a turbocharger rotor assembly as described above, the method comprising disconnecting the turbine shaft from the compressor shaft and replacing or repairing one or more components of the rotor assembly.

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 cross-section through a known turbocharger with a fixed geometry turbine which illustrates the basic components of a turbocharger and turbocharger rotor assembly;

Figure 2 is an axial cross-section through a known rotor assembly which may form part of a turbocharger such as that illustrated in Figure 1;

Figure 3 is an axial cross-section through a rotor assembly in accordance with a first embodiment of the invention;

Figure 4 is an axial cross-section through a rotor assembly in accordance with a second embodiment of the present invention.

Figure 5 is a side elevation of a thrust bearing of the rotor assembly of Figure 4; and

Figure 6 is an axial cross-section through a rotor assembly in accordance with a third embodiment of the present invention.

Figure 1 is an axial cross-section through a typical turbocharger with a fixed geometry turbine which illustrates the basic components of a turbocharger. The turbocharger comprises a turbine 1 joined to a compressor 2 via a central bearing housing 3. The turbine 1 comprises a turbine housing 4 which houses a turbine wheel
5. Similarly, the compressor 2 comprises a compressor housing 6 which houses a compressor wheel 7. The turbine wheel 5 and compressor wheel 7 are mounted on opposite ends of a common turbo shaft 8 which is supported on bearing assemblies 9 within the bearing housing 3.

The turbine housing 4 is provided with an exhaust gas inlet 10 and an exhaust gas outlet 11. The inlet 10 directs incoming exhaust gas to an annular inlet chamber, i.e. volute 12, surrounding the turbine wheel 5 and communicating therewith via a radially extending annular inlet passageway 13. Rotation of the turbine wheel 5 rotates the compressor wheel 7 which draws in air through an axial inlet 14 and delivers compressed air to the engine intake (not shown) via an annular outlet volute 15. The assembly including the turbine wheel 5 and compressor wheel 6 mounted on the shaft 8 is referred to as a rotor assembly 16.

Referring to both Figure 1 and Figure 2 (which shows an axial cross section through a rotor assembly) the turbocharger rotor assembly 16 comprises a turbine wheel 5, a compressor wheel 7, a thrust collar 21a and an oil slinger 21 all mounted to the turbo shaft 8 for rotation therewith. The turbine wheel 5 is friction welded to a seal boss 17 (sometimes referred to as a piston boss) provided at the turbine end of the shaft 8. The seal boss 17 is an enlarged diameter portion of the shaft 8, which has an annular groove 18 in its outer surface. A seal ring 19 (e.g. piston ring) sits in the groove 18 and seals the shaft 8 with respect to an aperture 19a in a bearing housing wall 19b which separates the turbine 1 from the bearing housing 3.

At the compressor end of the shaft 8 there is a reduced radius portion (indicated generally by 20). The reduced radius portion 20 receives a thrust collar 21a, an oil slinger 21 and the compressor wheel 7, via appropriately sized axial bores (22 and 23 respectively) therein. The oil slinger 21 and compressor wheel 7 are generally co-axial and are situated adjacent one another, the oil slinger 21 being the closer of the two to the turbine wheel 5. A thrust bearing 21b is received between the oil slinger 21 and the thrust collar 21a. The oils slinger 21, thrust collar 21a and compressor wheel 7 are clamped to the shaft 8 intermediate an annular shoulder portion 24 of the
shaft 8 and a locknut 25 which is received by a thread 26 at the end of the shaft 8. The oil slinger 21 has a portion which extends through an aperture 28a in a wall of the bearing housing 3 and is provided with a circumferential groove 27 within which a seal ring 28 sits (e.g. piston ring). The seal ring 28 seals the rotor assembly with respect to the bearing housing 3.

For optimum operation of a turbocharger it is important that the rotor assembly 16 is balanced, so that the centre of mass of the rotor assembly substantially falls on the rotor assembly’s axis of rotation, to thereby minimise the total moment of the rotor assembly 16 about a point on the rotational axis of the rotor assembly 16. A reduction in the total moment of the rotor assembly 16 about a point on the rotational axis results in a reduction in the centrifugal force experienced by the rotor assembly 16 as it rotates, which in turn increases the rotor assembly’s stability. Significant centrifugal force acting on the rotor assembly 16 may lead to vibration of the rotor assembly 16 and hence an increase in turbocharger noise and a reduction in its durability. Furthermore, the balance of the rotor assembly 16 is of particular importance due to the fact that the turbocharger of which the rotor assembly 16 forms part may operate at very high rotational speeds (<140,000rpm) and high temperatures (<200 °C). A large force on the rotor assembly 16, due to any imbalance combined with the high rotational speed; and in conjunction with compromised material properties due to the high operating temperature, may result in the compressor wheel 7 slipping radially with respect to the shaft 8, oil slinger 21 and locknut 26. Should slippage of the compressor wheel 7 occur, the overall imbalance of the rotor assembly 16 will deteriorate, thus resulting in an increase in the turbocharger vibration and hence noise. In extreme cases catastrophic turbocharger failure may occur.

An imbalance in the rotor assembly 16 can result from a number of causes, including, for example: inaccuracies in component squareness at mating interfaces; inaccuracies in component manufacture leading to radial clearances between the compressor wheel 7 and/or oil slinger 21 and the shaft 8; and the exertion of a force with a radial component on the compressor wheel 7 and/or oil slinger 21 due to tightening of the locknut 26. Additionally, the rotor assembly 16 may be deformed as
a result of any torque which is applied in the course of tightening the locknut 26. If a component of the rotor assembly is not square, both to other components and relative to the clamping axis of the locknut 26 (which in this case is also the rotor assembly 16 axis) then any clamping force applied to such a non-square component, by the tightening of the locknut 26, will have a component which is perpendicular to the rotor assembly 16 axis, which may result in radial movement of the component and hence cause an imbalance in the rotor assembly 16.

A known process used to achieve acceptable rotor assembly 16 balance is as follows: once the shaft 8 has been attached to the turbine wheel 5, the balance of the turbine wheel 5 and attached shaft 8 is checked. This may be done by purpose built automated machines. In one such machine the shaft 8 is supported in an air bearing as the shaft 8 and wheel 5 are rotated to about 3000rpm. Corrections to any imbalance may then be made by grinding material off the wheel 5. The compressor wheel 7 is temporarily attached to a mandrel and is balanced in the same manner as the shaft 8 and turbine 5. Once balanced, the turbine wheel 5 and attached shaft 8 is located in the bearing housing 3 and turbine 1 respectively so that the compressor end of the shaft 8 extends through the housing wall aperture 28a into the compressor housing 6. The shaft 8 receives the thrust collar 21a, thrust bearing 21b and oil slinger 21 by means of them being slid on to the compressor end of the shaft 8. The compressor wheel 7 is then assembled to the shaft 8 such that the compressor wheel 7 locates within the compressor. The locknut 26 is then tightened so as to clamp the wheel 7, slinger 21 and thrust collar 21 against the annular shoulder 24. Once assembled the entire rotor assembly 16 is balanced whilst in situ in the turbocharger. Any imbalance in the rotor assembly may be corrected by removing material from at least one of the components. Due to the fact that the rotor assembly 16 is balanced whilst it is in situ in the turbocharger, any removal of material from part of it, such as the back face of the compressor wheel is severely restricted by limited access as a result of the turbocharger housing.

Lack of access to the rotor assembly 16 whilst it is in situ means that the known balancing process often requires removal of the locknut 26, compressor wheel
7, thrust collar 21a and oil slinger 21 from the shaft 8 and replacing at least one of them. As all the components of the same type are substantially identical, except for having slightly different balances, the replacement of at least one of the rotor assembly components with an alternative one may result, upon reassembly of the rotor assembly 16, in the rotor assembly 16 being more balanced. The component replacement process is repeated as many times as is required in order to achieve acceptable rotor assembly 16 balance. In addition, replacement of components of the rotor assembly 16 may be impeded by galling of, say the compressor wheel 7 to the shaft 8.

It is also known for the compressor wheel 7 to be made of a different material to that of the shaft 8. For example, the compressor wheel 7 may be made of aluminium whilst the shaft 8 may be made of steel. The difference in the material of the shaft 8 and wheel 7 and the high forces used to secure the shaft 8 and wheel 7 together can result in galling or deformation of the material. This problem is exacerbated by the component replacement process described above. Once this galling or deformation has occurred it is generally not possible to correct any imbalance in the rotor assembly 16 and the rotor assembly 16 must be scrapped causing unnecessary waste and expense.

Figure 3 shows a turbocharger rotor assembly in accordance with an embodiment of the present invention. Although the construction of the shaft differs in detail from that of the rotor assembly of Figure 2, where appropriate the same reference numbers are used to identify corresponding features. The turbo shaft (indicated generally as 8) has two parts: a turbine shaft 8a, to which the turbine wheel 5 is mounted, and a compressor shaft 8b, to which the compressor wheel 7 is mounted. The turbine shaft 8a and turbine wheel 5 form a turbine sub-assembly 38a. As before, the turbine shaft 8a comprises a seal boss portion 17 which is friction welded to the turbine wheel 5. Collectively, the compressor shaft 8b, compressor wheel 7, and their means of attachment (locknut 25 and thread 26 in this case) constitute a compressor wheel sub-assembly 38b. The compressor wheel 7 is clamped between an annular shoulder defined by an enlarged radius portion 35 of the shaft 8b
and the locknut 25, which co-operates with a threaded portion 26 of the compressor shaft 8b, in a similar manner to that described above.

In order to prevent the shaft 8b from rotating whilst the locknut 25 is tightened, it may for instance be held in a chuck, or be provided with flat faces (not shown) for engagement by a suitable tool.

In accordance with the present invention the turbine and compressor sub-assemblies are connected together to complete the rotor assembly. In the example shown, the turbine shaft 8a comprises a socket, or axial bore 29, having a threaded portion 30. The compressor shaft 8b has a smaller diameter (indicated generally as 31), which is shaped and sized so as to be received in a close fit manner within the bore 29. The end of the compressor shaft 8b additionally comprises a screw portion 32 which co-operates with threaded portion 30 so as to enable the turbine shaft 8a and compressor shaft 8b to be screwed together. The screw and thread portions 32 and 30 also co-operate such that, in use, any torque applied by the turbine wheel 5 on the shaft 8a acts so as to tighten rather than loosen the joint between the turbine shaft 8a and compressor shaft 8b.

An oil slinger 33 and thrust collar 33a, which have axial bores 22 and 22a are received by the smaller diameter portion 31 of the compressor shaft 8b, such that when the two shafts 8a and 8b are screwed together, the oil slinger 33 and thrust collar 33a are clamped between the shoulder portion 34 of the turbine shaft 8a and the enlarged radius portion 35 of the compressor shaft 8b. The thrust bearing 42 may be fed onto the shaft 8b intermediate the oil slinger 33 and thrust collar 33a as the components are assembled. The enlarged radius portion 35 comprises a circumferential groove 36 within which a seal ring 37 sits (e.g. conventional piston type seal ring).

In accordance with the present invention, a different process is used in the assembly and balancing of the rotor assembly 16 shown in figure 3 compared to the
known process discussed above. The turbine shaft 8a is attached to the turbine wheel 5 to form the turbine subassembly 38a the balance of which is checked and corrected in the same manner as previously considered. The compressor wheel sub-assembly 38b is then assembled by securing the compressor wheel 7 to the compressor shaft 8b, intermediate the enlarged portion 35 thereof and the locknut 26. The compressor wheel sub-assembly 38b is then balanced. The balancing may be carried out using the balancing machine as described above, wherein the compressor wheel sub-assembly 38b is supported by an air bearing via the diameter of shaft 8b. Alternatively, the screw 32 on the end of the shaft 8b may be used to connect the compressor wheel sub-assembly 38b to the balancing machine. Alternatively, the balancing machine may support shoulder portion 39, also known as a location face. Other methods of mounting the compressor sub-assembly for balancing will be apparent to the skilled person. Once again, acceptable balance of the compressor wheel sub-assembly 38b is achieved by adding material, or preferably removing material, from a potion of the compressor wheel 7. In contrast to the previously described conventional rotor assembly 16 assembly method, the majority of the balancing, i.e. that of the turbine wheel sub-assembly 38a and also that of the compressor wheel sub-assembly 38b, can be carried out without the rotor assembly 16 being in situ in the turbocharger. This makes the balancing process much more straightforward as there is no limitation on access to the rotor assembly 16 components. For instance there should not generally be any need to remove and replace individual components to achieve balance.

Once the compressor wheel sub-assembly 38b has been balanced, the oil slinger 33 and thrust collar 33a are fed on to the compressor shaft 8b. The turbine wheel assembly 38a is inserted into the turbine end of the bearing housing 3. The compressor wheel sub-assembly 38b is inserted into the compressor end of the bearing housing before the smaller diameter portion 31 is inserted into the socket 29 and rotated so as to secure both shafts 8a and 8b to one another via the screw and thread arrangement 30, 32. As previously mentioned, the direction of relative rotation between the turbine wheel sub-assembly 38a and compressor wheel sub-assembly 38b required to secure them together is such that any torque on the turbine 5 whilst the turbocharger is in use acts so as to tighten the link between the turbine wheel sub-
assembly 38a and compressor wheel sub-assembly 38b. In this way the rotor assembly 16 is received by the turbocharger via the bearing assemblies 9. In order to rotate the compressor shaft 8b relative to the turbine shaft 8a, so as to utilise the screw and thread arrangement 30, 32, a tool may be used to engage the hexagonal, cross-sectional form of the locking nut 25. A significantly lower torque may be used to screw the shafts 8a, 8b together compared to the torque required to secure the locknut 25 adjacent the compressor wheel 7. This ensures that whilst attaching the two shafts 8a and 8b together, no slippage occurs between the compressor 7 and the locknut 25 which would be detrimental to the balance of the compressor sub-assembly 38b and hence any completed rotor assembly 16.

The shafts 8a, 8b are such that they have a tight tolerance fit when they are screwed together, in order to achieve good concentricity. In addition, the longitudinal axes of both the bore 29 and smaller diameter portion 31 are such when the shafts 8a, 8b are secured together, the shafts 8a, 8b are co-axial. This helps to ensure that when the pre-balanced turbine assembly 38a is attached to the compressor wheel sub-assembly 38b, the resulting rotor assembly 16 is substantially balanced. In addition, it is preferable that there is a good squareness between the mating edge (or shoulder) 34 and the longitudinal axis of the shaft 8a, as well as between the mating edge 39 and longitudinal axis of the shaft 8b. This again helps to ensure the shafts 8a and 8b are co-axial when secured together. Furthermore, the oil slinger 33 and thrust collar 33a are such that may have two substantially parallel faces, which engage the mating edges (34 and 39) whilst the shafts (8a and 8b) are connected to one another, and cooperate therewith to ensure the shafts 8a and 8b are co-axial.

The proposed assembly method is advantageous in that there is a reduction in the number of component interfaces when assembling the rotor assembly 16 on the turbocharger, compared to known assembly methods. Any such component interface may have a contribution to the rotor assembly 16 imbalance and as such, a reduction in component interfaces results in a minimisation of any possible rotor assembly 16 imbalance. The compressor wheel sub-assembly 38b can be assembled and balanced before construction of the rotor assembly 16. Balancing of the compressor wheel sub-
assembly 38b may compensate for various factors contributing to any imbalance of
the compressor wheel sub-assembly 38b and hence eventual rotor assembly 16,
including: component squareness inaccuracies in any of the mating surfaces between
the components of the compressor assembly; any imbalance which results from
inaccuracies in the fit between the compressor wheel 7 and the shaft 8b; any
individual component imbalance in either the shaft 8b, the compressor wheel 7 or the
locknut 25; and any imbalance that may occur as a result of a deformation of the
compressor wheel sub-assembly 38b due to the tightening of the locknut 25. As such,
balancing of the individual components that form part of the compressor wheel sub-
assembly 38b is not necessarily required before the construction of the compressor
wheel sub-assembly 38b. Once the compressor wheel sub-assembly 38b has been
balanced, any imbalance, which may have been caused by inaccuracies in any of its
constituent components, will no longer contribute to the imbalance of the rotor
assembly 16. As such, it is only the concentricity of the smaller diameter portion 31 of
the shaft 8b and socket 29, in combination with the mating surfaces of the oil slinger
33 and thrust collar 33a which may contribute to rotor assembly 16 imbalance. It is
expected that reducing the number of possible sources of rotor assembly 16 imbalance
will mean that any inaccuracies in the manufacture of the oil slinger and thrust
bearing 33a alone will not be significant enough to cause an unacceptable imbalance
of the rotor assembly 16.

Furthermore, due to the fact that the torque required to secure the turbine
wheel sub-assembly 38a and compressor wheel sub-assembly 38b together is
significantly less than the torque required to secure all of the components together in
the prior art rotor assembly 16 by use of the locknut 25, there is less chance that any
force on the rotor assembly 16 resulting from securing the sub-assemblies 38a, 38b
together will lead to a deformation and hence rotor assembly 16 imbalance.

As previously described, the rotor assembly 16 is supported, in use, on bearing
assemblies 9 within the turbocharger. The rotor assembly may be arranged so that any
portion 40 of the shaft 8a which may potentially deform in response to the tightening
of the threaded portions 30, is not surrounded by a bearing assembly 9. This is so that
any deformation of the shaft 8a does not affect the clearances between the bearing assemblies 9 and the shaft 8a. Any decrease in clearance may increase the friction between the shaft 8a and the bearing assembly 9 or may cause the shaft 8a to become jammed within the bearing assembly 9.

In contrast to known rotor assemblies, in which the compressor end piston seal ring 28 is located in a groove 27 in the oil slinger 21, in the rotor assembly 16 shown in figure 2, the seal ring 37 is located in a groove 36 in the enlarged radius portion 35. This allows the axial length of the oil slinger to be shortened and thereby avoids an increase in length of the proposed rotor assembly 16 compared with a conventional turbocharger rotor assembly (were a standard oil slinger to be used in combination with the increased radius portion 35). This allows the proposed rotor assembly 16 and conventional rotor assemblies to be interchangeable.

Figure 4 shows an alternative embodiment according to the present invention. Its general form is very similar to that of the previous embodiment, and as such like features have been given the same reference numerals. A difference between this and the previous embodiment is that the smaller diameter shaft portion 31 is formed as part of the turbine wheel sub-assembly 38a as opposed to the compressor wheel sub-assembly 38b. Accordingly, the socket 29 of the embodiment of the invention shown in Figure 4 is formed as part of the compressor wheel sub-assembly 38b as opposed to the turbine wheel sub-assembly 38a.

The embodiment shown in figure 4 also comprises an oil slinger 41a with integrated thrust collar. The unitary nature of the thrust collar and oil slinger results in elimination of the interface between the separate oil slinger and thrust collar of the previous embodiments. Once more, as each component interface may have a contribution to the rotor assembly 16 imbalance, a reduction in the number of component interfaces within the rotor assembly 16 results in a minimisation of any possible rotor assembly 16 imbalance. It will however be appreciated that the two part oil slinger and thrust collar of the embodiments described above could take the place of the combined oil slinger 41a. Similarly, it will be appreciated that the two part oil
slinger and thrust collar in the previous embodiments could be replaced by the combined oil slinger of the embodiment of the invention in figure 4.

The thrust bearing 42 of the figure 4 embodiment is shown in Figure 5 and comprises a radial slot 44, which is sized so as to fit over the thrust collar portion of the combined oil slinger/thrust collar. The thrust bearing 42 is also provided with oil channels 43 as is known in the art.

Figure 6 shows a further embodiment in accordance with the present invention. Where appropriate, the same reference numerals have been used in figure 6 as used in Figures 3 to 5. The complimentary mating surfaces 47 between the enlarged portion 35 of the shaft 8b and the compressor wheel 7 and the complimentary mating surfaces 48 between the locknut 25 and the compressor wheel 7 are both frusto-conical. The frusto-conical mating surfaces 47, 48 co-operate, whilst the compressor 7 is being fixed to the shaft 8b by the tightening of the locknut 25, so as to aid the centralisation of the compressor wheel 7 on the shaft 8b. In addition, whilst the rotor assembly 16 is in use, the mating surfaces 47, 48 co-operate so as to help prevent the compressor 7 from moving radially relative to the shaft 8b. As such, the balance of the rotor assembly 16 may be less likely to deteriorate over time whilst the rotor assembly 16 is in use. Furthermore, the nature of the mating surfaces may reduce any galling of a soft compressor wheel 7 which may occur due to securing it to a hard shaft 8b of different geometry.

In the embodiment shown, the mating surface of the compressor wheel 7 is concave and have a slightly curved profile with respect to the rotor assembly axis. It will be appreciated that as an alternative the compressor wheel mating surface may be convex and the mating surface on the shaft could be concave. Also the surfaces need to be curved but could lie on a conical surface In addition, alternative features may be used to resist radial movement of the compressor wheel 7 relative to the shaft 8b.
In addition to simplifying assembly and balancing of the turbocharger rotor shaft, embodiments of the present invention may also allow separation of the turbine and compressor wheel sub-assemblies to aid in servicing or repair of a turbocharger.

Other modifications to the embodiments of the invention described above, and applications of the invention, will be readily apparent to the appropriately skilled person.
CLAIMS

1. A turbocharger rotor assembly, comprising a turbine wheel and a compressor wheel mounted to a rotor shaft for rotation therewith about an axis, wherein:
   - the turbine wheel is mounted to a turbine shaft;
   - the compressor wheel is mounted to a compressor shaft; and
   - the turbine shaft and compressor shaft are connected to each other to define said rotor shaft.

2. A turbocharger rotor assembly according to claim 1, wherein the turbine shaft and compressor shaft are releasably connected to one another.

3. A turbocharger rotor assembly according to claim 1 or claim 2, wherein the turbine shaft and compressor shaft are connected together by respective screw threaded surfaces which engage one another.

4. A turbocharger rotor assembly according to claim 3, wherein one of the turbine shaft and compressor shaft is provided with a screw thread on a portion of its outer surface, and the other shaft is provided with an axially extending screw threaded bore to receive the screw threaded shaft.

5. A turbocharger rotor assembly according to any preceding claim, wherein a first end of the compressor shaft extends through a bore provided along the rotational axis of the compressor wheel, and wherein the compressor wheel is retained on the shaft by a nut which threads on said first end of the compressor shaft and bears directly or indirectly against the compressor wheel to clamp the wheel against a radial abutment defined by a portion of the compressor shaft.

6. A turbocharger rotor assembly according to claim 5, wherein the radial abutment is an annular shoulder defined by an enlarged diameter portion of the compressor shaft.
7. A turbocharger rotor assembly according to claim 6, wherein said enlarged diameter portion of the compressor shaft defines a seal boss adapted to be received with an aperture defined by a turbocharger housing wall separating the compressor wheel from a turbocharger bearing cavity, and wherein an annular groove is defined in a radially outer surface of the seal boss to receive a seal ring.

8. A turbocharger rotor assembly according to claim 6 or 7, wherein abutting surfaces of the compressor wheel and the enlarged diameter portion of the compressor shaft are generally frustoconical about the axis of the shaft such that they each lie on the same generally conical surface of revolution about said axis.

9. A turbocharger rotor assembly according to any one of claims 5 to 8, wherein an oil slinger and/or thrust collar is supported on the rotor shaft and is clamped between a radially extending surface defined by the turbine shaft and said radial abutment defined by a portion of the compressor shaft.

10. A turbocharger rotor assembly according to any preceding claim, wherein an oil slinger and/or thrust collar is supported on the rotor shaft and is clamped between radially extending surfaces defined by the turbine shaft and the compressor shaft respectively.

11. A turbocharger comprising a rotor assembly according to any preceding claim.

12. A method of assembling a turbocharger rotor assembly in a turbocharger comprising a turbine and a compressor at opposite ends of a bearing cavity, a first housing wall separating the turbine from the bearing cavity and a second housing wall separating the compressor from the bearing cavity, each housing wall being provided with an aperture for receiving a turbocharger rotor shaft which extends from the turbine to the compressor through said bearing cavity the method comprising:

   providing a first rotor sub-assembly comprising a turbine wheel mounted to a turbine shaft;
providing a second rotor sub-assembly comprising a compressor wheel mounted to a compressor shaft;

independently balancing the first and second rotor sub-assemblies;

inserting the turbine shaft into the bearing cavity through the aperture provided in the first housing wall and inserting the compressor shaft into the bearing cavity through the aperture provided in the second housing wall;

and coaxially connecting the turbine shaft and compressor shaft together to define the turbocharger rotor assembly.

13. A method according to claim 12, wherein the turbocharger rotor assembly is a rotor assembly according to any one of claims 1 to 10.

14. A method of servicing a turbocharger including a turbocharger rotor assembly according to any one of claims 1 to 10, the method comprising disconnecting the turbine shaft from the compressor shaft and replacing or repairing one or more components of the rotor assembly
Application No: GB0816393.3
Examiner: Chris Vosper
Claims searched: 1-14
Date of search: 23 January 2009

Patents Act 1977: Search Report under Section 17

Documents considered to be relevant:

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<th>Category</th>
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<td>GB 0595669 A POWER (figs. 1 and 2, noting screw and spline connections between shafts 11,20,26B)</td>
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<td>US 2004/0202556 A1 SVIHLA (fig. 1, noting turbine shaft 32 coupled to compressor shaft 52.)</td>
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<td>DE 3625996 A1 KUEHNLE (figs. 1, 2, noting compressor shaft 120 coupled to turbine shaft 10.)</td>
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<td>US 6499969 B1 GENERAL (fig. 2, noting stub-shafts on turbine and rotor discs 18,20 coupled together by components 52,64.)</td>
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<td>KR 200219261 A HYUNDAI (fig. 1, noting coupling 160 between turbine shaft 145 and compressor shaft 150)</td>
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<td>US 4719075 A NGK (col. 6, lines 24 to 30)</td>
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F02B; F02C
The following online and other databases have been used in the preparation of this search report:
ONLINE: EPODOC, WPI, OPTICS

International Classification:

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